

A parameter study of the influence of struts on the performance of a vertical-axis marine current turbine

Anders Goude¹, Staffan Lundin¹ and Mats Leijon¹

¹Swedish Centre for Renewable Electric Energy Conversion,
The Ångström Laboratory,
Uppsala University,
Box 534, SE-751 21 Uppsala, Sweden

Abstract

Marine currents are an important offshore source of renewable energy. A lot of effort is spent on the development of technology for, for example, electricity generation from tidal currents. In the present paper, the performance of a vertical axis marine current turbine is examined numerically under the variation of certain parameters.

The turbine is modelled with an in-house code, based on the double multiple streamtube model. Corrections are made due to a finite aspect ratio and tip losses for the blades. Published experimental data for the lift and drag coefficients of the blades for different Reynolds numbers are used in the model.

Structural integrity is a major concern of any underwater machinery due to the considerable hydrodynamic forces involved. Special attention is paid to the importance of struts and related supporting structure for the turbine blades. As a rule of thumb, the efficiency of the turbine may be expected to rise with increased aspect ratio of individual blades. However, with leaner blades more support structure is required, which carries a cost in terms of a negative effect on efficiency. We study how the level of acceptable stress on a turbine blade influences the total turbine efficiency depending on the number of struts required to support the blade.

Keywords: Turbine, blades, struts, simulation

Nomenclature

N_b	= number of blades
N_s	= number of struts
c	= chord for the blade
c_s	= chord for the strut
c_0	= reference chord length
σ	= stress
D	= stress proportionality constant
h	= turbine height
R	= turbine radius

L	= distance between struts
C_D	= drag coefficient
C_L	= lift coefficient
F_D	= drag force
F_L	= lift force
F_{NH}	= normal (radial) force on blade
F_T	= tangential force on blade
F_{NC}	= centrifugal force on blade
F_{NI}	= total normal force on blade per unit length
T	= torque
ω	= rotational speed of the turbine
V	= flow velocity
V_b	= flow velocity at the blade
V_e	= flow velocity at the centre
V_c	= relative velocity at a point on the strut
V_1	= free-stream velocity
ρ	= water density
x, y	= cartesian coordinates in the turbine plane
r	= coordinate along the direction of the strut
θ	= angle in the turbine plane
x_{blade}	= x coordinate for a given blade
y_{blade}	= y coordinate for a given blade
θ_{blade}	= θ coordinate for a given blade
P_{loss}	= losses due to drag of struts

1 Introduction

Turbines for marine current power generation come in many shapes and sizes. In a recent review, Khan et al. [1] considered several ‘turbine’ and ‘non-turbine’ systems for hydrokinetic energy conversion. From an engineering point of view, different concepts entail various pros and cons, and what technological solution is objectively best in a given situation is rarely very clear.

One main category of turbine is the *cross-stream axis turbine*, characterized by having its axis of rotation perpendicular to the main direction of fluid flow. There are sub-groups within this category, and for wind power applications several variants of the Darrieus and Savonius type turbines have been used [2]. For applications in water, however, it is mainly the straight-bladed Darrieus turbine – either in a ‘squirrel-wheel’ configuration [3] or in an ‘open-ended’ configuration [4] – and the Gorlov

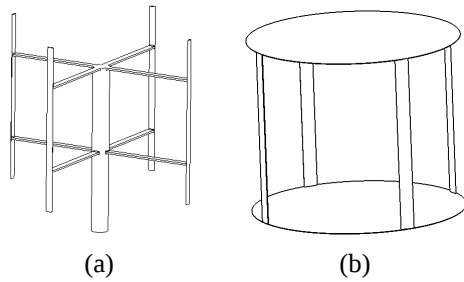


Figure 1: Two types of cross-stream axis turbine. (a) Open-ended Darrieus. (b) Squirrel-wheel Darrieus.

turbine – which resembles the Darrieus turbine but has helical blades [5] – which have been used (Fig. 1). Due to the orientation of the axis, these turbines are often referred to as vertical-axis turbines, but obviously the axis may be cross-stream regardless of its orientation relative to the horizon.

In the case of the open-ended Darrieus turbine, the turbine blades are connected to the turbine axis by means of struts. Depending on the geometry of the turbine – length, radius, solidity, number of blades –, the number of struts required to fully support the blades will vary, as will the geometric specifics of individual struts. The struts carry a penalty in terms of drag, which will adversely affect the power coefficient of the turbine. It is therefore of some interest to study the impact of blade struts on the performance of a cross-stream axis turbine.

1.1 Problem considered in present paper

The purpose of the present investigation was to study the impact in terms of drag loss of adding support struts to turbine blades as the blades become leaner due to the number of blades being increased.

Consider a straight-bladed, open-ended Darrieus turbine with vertically oriented axis of rotation. For simplicity, we assume that the blades are not tapered (i.e. they have a constant chord length and hydrofoil thickness). For a given solidity, the blade chord will be inversely proportional to the number of blades, and, consequently, the aspect ratio of a single blade will be proportional to the number of blades. Higher aspect ratio of the blade gives a lower induced drag, which will increase the lift-to-drag ratio. For the same total blade surface area, it might then be expected to see lower drag losses for the same amount of lift when the surface area is divided among a greater number of blades. A smaller chord does however decrease the Reynolds number of the blade, which usually gives a lower lift-to-drag ratio. This means that the hydrodynamical benefits of increasing the number of blades are lower for turbines which have a large height compared to their radius.

A leaner blade also possesses less structural strength. Depending on the material from which it was made, a blade can only withstand a certain amount of stress. The stress will depend on the forces experienced by the blade and its geometrical properties, especially the distances between support points. As the aspect ratio of a blade is increased, thresholds will be passed where additional

support struts for the blade will be required. These supports cause drag without adding to the lift, and this drag penalty may or may not exceed the gain in terms of induced drag reduction from the increase in aspect ratio.

The limiting factor for the number of struts to a blade is the stress on the blade due to the radial hydrodynamic force. Depending on the thickness of the blade, the properties of the material it is made of in combination with the construction method will allow a maximum distance between support points. Given the blade length, this translates to a minimum number of struts to support the blade.

The dominating force for sizing the struts is the weight of the blade. The stresses due to gravitational forces dominate over those due to hydrodynamic forces for the struts. This is again in part due to the choice of material and method of construction, which highly influence the blade weight.

In the following section, the expressions for calculating the maximum distance between struts and the required minimum number of struts per blade will be derived. These equations are then used to illustrate the impact of strut losses on the performance of two example turbines designed for different flow cases.

2 Theory

2.1 Hydrodynamic model

The simulation model chosen for the hydrodynamical simulations was the double multiple streamtube model developed by Paraschivoiu [6], which has shown good results for simulating Darrieus wind turbines. The model separates the turbine into one upstream part and one downstream part, where both parts are solved separately using momentum conservation and the velocity is calculated at blade positions and at the centre of the turbine only. Further details about the method may be found in references [7] and [8]. The present model is based on experimental data for lift and drag coefficients obtained from [9]. For dynamic stall modelling, the method developed by Gormont [10] and later modified by Massé [11] and Berg [12] was chosen. For the calculation of forces on the blades to be used in structural mechanical calculations, corrections due to a finite aspect ratio were neglected in order to get a small overestimation on the forces. For power coefficient calculations, tip corrections were applied according to Paraschivoiu [8] with the modification that the lift and drag coefficients were recalculated with the dynamic stall model using the reduced angle of attack, and induced drag was applied after the dynamic stall calculations. This modification was made to prevent overestimations of the power coefficient at high tip speed ratios.

2.1.1 Calculation of loss due to struts

The losses due to struts can either be determined by calculating the force due to drag over the strut and integrating to obtain the losses for the whole turbine, or

by using an equivalent drag coefficient for the entire turbine as suggested by Moran [13], where the losses are proportional to the cube of the rotational speed. Here, the force will be integrated for higher flexibility and because the drag coefficients for the selected strut profile are known. It is assumed that the changes in flow velocity due to the struts can be neglected, and that the strut is approximately horizontal giving zero angle of attack.

To determine the drag forces, it is necessary to know the local velocity at the strut. Consider a blade position on the upwind disc. Symbols are illustrated in Fig. 2. To determine the velocity, start by noting that when the blade is located at angle θ_{blade} , the y coordinate of the blade is given by

$$y_{\text{blade}} = R \sin \theta_{\text{blade}}. \quad (1)$$

Now, since we are interested in a point on the strut, let the coordinate r denote the position on the strut. We have

$$0 < r < R. \quad (2)$$

This gives the local coordinates as

$$y = r \sin \theta_{\text{blade}} \quad \text{and} \quad x = r \cos \theta_{\text{blade}}. \quad (3)$$

We want to find out in which streamtube this point is located. If streamtube expansion is neglected as done in [7], this will correspond to the streamtube with θ value

$$\theta = \arcsin \left(\frac{r}{R} \sin \theta_{\text{blade}} \right). \quad (4)$$

In the double multiple streamtube model, the velocities at the blade and at the centre are known. To approximate the velocity at the strut, it is assumed that the velocity varies linearly between the boundary of the turbine and the centre. By using the velocities V_b at the blade and V_e at the centre for the streamtube obtained from Eq. (4), linear interpolation gives the velocity as

$$V = V_e + \frac{V_b - V_e}{\sqrt{R^2 - r^2 \sin^2 \theta_{\text{blade}}}} r \cos \theta_{\text{blade}}. \quad (5)$$

Following the streamtube model, it is assumed that the velocity only has x -components. Therefore, in the local frame of reference of the blade, the velocity in the direction of the chord is given by

$$V_c = -\omega r + V \sin \theta_{\text{blade}}. \quad (6)$$

The corresponding expression can be derived for the downwind part.

When the velocity is known, the force can be calculated according to

$$dF_D = C_D \rho c_s \frac{V_c^2}{2} dr. \quad (7)$$

The torque is given by

$$T(\theta_{\text{blade}}) = \int_0^R C_D(r, \theta_{\text{blade}}) \rho c_s(r) \frac{V_c(r, \theta_{\text{blade}})^2}{2} r dr \quad (8)$$

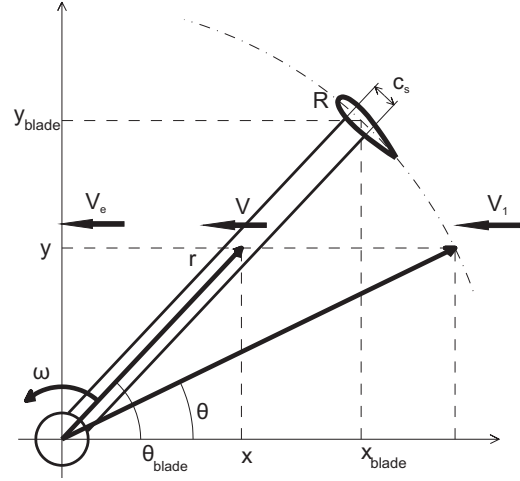


Figure 2: Symbols used in derivations.

and the power loss by

$$P_{\text{loss}} = N_b N_s \omega \frac{1}{2\pi} \int_0^{2\pi} T(\theta_{\text{blade}}) d\theta_{\text{blade}}, \quad (9)$$

where N_s is the number of struts and N_b is the number of blades.

2.2 Support struts geometry

If the normal forces on the blades are balanced to give a low torque on the joint between the blade and the strut, the bending moment in the blade will mainly depend on the distance between the strut and the force applied. For a given profile, the bending resistance will increase with the cube of the chord, giving the expression for the tension in the blade as

$$\sigma \approx D \frac{L^2}{c^3} F_{NI} \quad (10)$$

where F_{NI} is the total normal force per unit length, and D is a constant (see Sec. 2.2.3). F_{NI} is given by

$$F_{NI} = \frac{F_{NH} + F_{NC}}{h}. \quad (11)$$

Eq. (10) assumes that the distance $L \gg c$ and that the displacements are small enough for linear theory to hold.

2.2.1 Distance between struts as a function of number of blades

For a turbine, the tip speed ratio that gives the highest power coefficient is primarily a function of the solidity of the turbine. This means that for a turbine to have the same optimum tip speed ratio, the solidity of the turbine should remain constant. By denoting the chord and number of blades of one turbine as c_1 and N_{b1} and of a second turbine as c_2 and N_{b2} , a constant solidity gives

$$c_1 N_{b1} = c_2 N_{b2}. \quad (12)$$

If it is assumed that the power coefficient is approximately the same for both turbines, the torque has to be

the same (due to the assumption of same tip speed ratio), giving the relation between the tangential forces of the blades as

$$F_{T1}N_{s1} \approx F_{T2}N_{s2} \quad (13)$$

and, since the power coefficient is assumed to be the same, the flow field through the turbine should be similar, giving approximately equal angles of attack for the blades, which means that the relations between the hydrodynamical normal forces should be similar to Eq. (13), i.e.

$$F_{NH1}N_{s1} \approx F_{NH2}N_{s2}. \quad (14)$$

The centrifugal force will depend on the mass of the blade. For a solid blade, the mass is proportional to the square of the blade chord, and the relation between the centrifugal forces is

$$F_{NC1}c_2^2 = F_{NC2}c_1^2 \quad (15)$$

for a constant rotational speed. By combining Eqs. (10), (11), (12), (14) and (15) and assuming that the maximum allowed stress is the same, one obtains the expression

$$L_2 \approx L_1 \frac{N_{b1}}{N_{b2}} \sqrt{\frac{F_{NH1} + F_{NC1}}{F_{NH1} + F_{NC1} \frac{N_{b1}}{N_{b2}}}}. \quad (16)$$

It should be noted that by changing the chord, both the Reynolds number and the aspect ratio of the blade will change, effects of flow curvature will be different and the behaviour in dynamic stall will change. Therefore, the assumption of constant power coefficient, and hence Eqs. (13) and (14), can be questioned.

2.2.2 Distance between struts as a function of design flow velocity

Using the expression for the hydrodynamic lift force,

$$F_L = \rho c h C_L \frac{V^2}{2}, \quad (17)$$

and the assumptions that the variation in lift coefficient due to the change in velocity is small and that the angle of attack remains constant, one will obtain the relation between the forces at different velocities as

$$F_2 \approx F_1 \left(\frac{V_2}{V_1} \right)^2. \quad (18)$$

By combining Eqs. (10) and (18) and assuming constant chord, one obtains the relation between the maximum distance between the struts and the design flow velocity as

$$L_2 \approx L_1 \frac{V_1}{V_2}. \quad (19)$$

2.2.3 Number of struts per blade

The constant D in Eq. (10) can be determined from FEM calculations. This was carried out assuming that the force was evenly distributed over the entire blade, which is the force distribution obtained due to the assumptions outlined in Sec. 2.1. The joints between the

struts and the blades were considered as fixed surfaces. For these boundary conditions, the maximum stress on a blade is found close to the joints. For the NACA0021 blade, the constant was determined to be $D \approx 18$. For an evenly distributed force and a constant chord, the distance between two struts should be approximately 2.5 times the distance between the outer strut and the blade tip in order to obtain the same stress on both sides of the junction of the outermost strut. (In the real case, the force is smaller close to the blade tip due to the tip vortices; hence the struts should be a little closer to each other in the real case to create the same stress on both sides of the outermost junction.) By using this result, once the distance between the struts has been calculated, it is possible to determine the necessary number of struts as

$$N_s = \frac{h}{L} + 0.2 \quad (20)$$

where N_s should be rounded to the nearest larger integer and should never be smaller than 2.

2.2.4 Calculation of the size of the struts

The forces on the struts can be separated into three parts: the normal (or radial) force, the tangential force and the gravitational force. The normal force has two contributions, the hydrodynamic force and the centrifugal force. Since its direction is parallel to the strut, the maximum tension will be obtained at the point where the strut chord is smallest. At the strut root, this force will be negligible compared to the other forces, which cause bending moments. The tangential force is an oscillating force, and will cause a bending moment that is largest at the strut root.

The last force is gravity, which also causes a bending moment. One difference between the tangential force and gravity is that in the tangential direction, the blade can move freely, while in the vertical direction, the blade cannot move freely since there are several struts connected to the blade, preventing it from rotating. This can cause additional bending moments in the strut tip, and also in the blades. For the case of blades with high density, the contribution from gravity will be the dominating term at the strut tip.

Based on FEM calculations, the strut chord was chosen as 0.8 times the blade chord at the tip and 1.2 times the blade chord at the root for a blade with chord 0.2 m, which causes the tension in the blade to remain within the chosen limits. Since gravity gives a constant load, fatigue will not be a major concern here, since the point where the stress from gravity is located is where strut is at its maximum thickness, while the tangential force will give its largest contribution at the trailing edge. However, it is desired that the deflection of the struts is limited, since it can affect the performance of the turbine and can cause additional tension in the blades. The deflection increases approximately with the chord of the strut to the power of four, while it decreases linearly with the force, which for gravity decreases with the square of the blade chord. Considering this, the chord of the strut

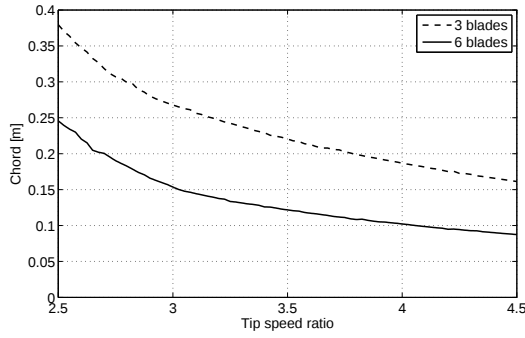


Figure 3: Chord lengths of the 2.5 m/s turbines.

was chosen as

$$c_s = 0.8\sqrt{c_0 c} \quad (21)$$

for the strut tip and

$$c_s = 1.2\sqrt{c_0 c} \quad (22)$$

for the strut root, where $c_0 = 0.2$ m as in the FEM calculation.

Since the bending resistance decreases with the cube of the strut chord, while the drag on the strut only decreases linearly with the chord, small optimisations of the strut chord will not lead to any major reduction in the drag from the struts.

Different thicknesses of the hydrofoil were investigated, but the possible reduction in strut chord obtained by choosing a thicker profile were basically compensated by the higher drag coefficient of the thicker chord, giving approximately the same drag losses. The same was seen for a thinner profile. Therefore, the same profile was chosen for the struts as for the blades.

3 Application

In order to illustrate the effect of the equations derived in Sec. 2, two theoretical turbines were studied; one with 3 blades, the other with 6 blades, but besides that essentially equivalent.

The size of the turbine was taken as 5 m in length with a radius of 2.5 m, and the maximum flow speed was assumed to be 2.5 m/s. These admittedly arbitrary figures

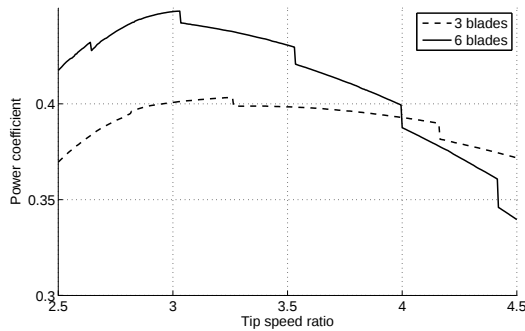


Figure 4: Power coefficients of the turbines optimised for flow speed 2.5 m/s. The stair-step shape of the curves due to the inclusion of more struts with increasing tip speed ratio is clearly visible.

would give an expected maximum power output on the order of 60 kW depending on the power coefficient of the turbine and the efficiency of the system as a whole; not a very large station but quite sufficient for our present purposes. Both the blades and the struts were given the NACA0021 profile, and all components were assumed to be constructed from solid steel with a mass density of 8 times that of water. Using data from ship propeller design [14], the allowable stress in the blades was taken as 100 MPa, which gives reasonable safety factors.

Neglecting for the time being the strut losses, the optimum turbine solidities for tip speed ratios from 2.5 through 4.5 were calculated (Fig. 3). Given these solidities, the maximum normal (radial) force on the blades was calculated, neglecting three-dimensional blade tip effects in order not to underestimate the force, but including centrifugal effects due to the mass of each blade. Based on the normal force, the minimum number of struts for operation at each speed was computed, and finally the losses due to the struts as well as the blade tip effects were included. This process was carried out for two flow speeds, 1.5 m/s and 2.5 m/s.

The power coefficient curves of the turbines optimised for 2.5 m/s are plotted in Fig. 4. The curves assume a stair-step character, due to the discrete addition of strut losses. As predicted by Eq. (16), for the 6 bladed turbine with the thinner, leaner blades, the steps occur more frequently than for the 3 bladed turbine, and as a consequence the power coefficient drops off considerably more quickly as the tip speed ratio increases.

Since the turbines were optimised for 2.5 m/s, they might be expected to perform less well at 1.5 m/s. In Fig. 5, the power coefficients of all turbines are plotted at the lower speed. The 6-bladed turbines again exhibit more steps in the curves and a quicker drop-off with increasing tip speed ratio. Not surprisingly, the turbine designed for 1.5 m/s performs consistently better than its higher-speed counterpart. For the 3 bladed turbines the difference is less pronounced, and the two curves behave almost identically until the slightly leaner-bladed 2.5-m/s turbine takes a step down due to extra struts being added at an approximate tip speed ratio of 3.2.

At almost any site considered for establishment of a

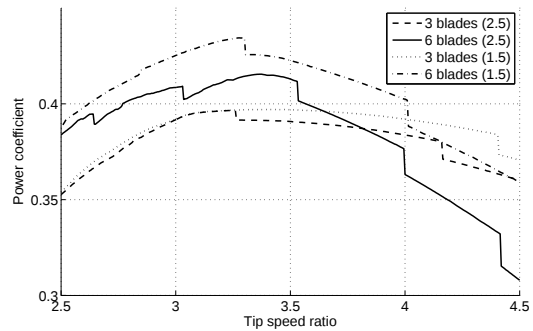


Figure 5: Power coefficients of turbines optimised for flow speeds of 1.5 m/s and 2.5 m/s, all run at 1.5 m/s.

marine current power station – be it a tidal channel, a river reach, or some other location – the strength of the water current will vary with time. The main point illustrated by Fig. 5 is that it is far from obvious that a turbine should be optimised for the peak flow speed. The details all depend on the specific flow characteristics of the particular site, but on the whole it can be expected that the peak flow speed will occur during a rather small fraction of the operating life of the power station. Looking at the 6-bladed turbines of the present example, it is clear that the turbine designed for 2.5 m/s will not be the best choice if it spends most of its service life operating at 1.5 m/s. The other turbine, on the other hand, might not be structurally strong enough to handle the hydrodynamic forces at 2.5 m/s, and so would have to be shut down when the flow speed exceeds some predetermined value. The choice of design speed would have to be based on statistics of the flow at the site under consideration, so that it might be determined which option gives the best output over time.

4 Discussion

It is hardly controversial to claim that strut losses are, in some sense, significant for the performance evaluation of an open-ended Darrieus turbine. The mechanisms governing the losses are however quite complex, and the fact that strut losses are added in “steps” with additional struts makes it far from obvious what will be the optimal design choice for a given case.

In theory, more blades means leaner blades, which means higher blade aspect ratio, which means lower induced drag. However, as is clearly seen in Fig. 4, leaner blades also entails higher sensitivity to hydrodynamic forces and thus an increased requirement for struts – which brings drag. Fig. 5 indicates that the 3-bladed design has a significantly smoother strut requirement with varying tip speed ratio, which clearly ought to be desirable in many cases.

That being said, there are other factors to take into account. As mentioned previously, gravitational effects on the struts were largely disregarded in this study. In reality, the struts have to carry the weight of the blades, which is proportional to the square of the blade chord length (assuming solid blades). This means that the total weight of the turbine blades on the 3 bladed turbine is twice that of the 6 bladed turbine, since the turbine solidity is approximately the same and the blade chords of the two turbines consequently vary by a factor of 2. The weight of the struts will somewhat even out the difference, but on the whole a turbine may be expected to be heavier the lower the blade number.

The turbines were assumed to be constructed out of solid steel. This is simple and not unreasonable. Many other materials and construction techniques might however be considered. It could, for instance, be an option to make the larger blades (such as on the 3 bladed turbine) hollow, in order to save weight and material. Various composite materials might also be an option. Such

changes would influence the mechanical properties of the blades and struts, and might serve to mitigate the effects of strut losses. These considerations are however outside the scope of the present study.

Acknowledgments

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