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PERFORMANCE OF A DARRIEUS WATER TURBINE AT VARIOUS SOLIDITIES

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SUMMARY A 3m diameter straight-bladed water turbine of the Darrieus type was tested experimentally at stream speeds in the range of 0.5-0.7 m/s. Tests were carried out using 2.4m blades of modified NACA 0015 section and with three different rotor and blade configurations giving solidities of 0.18, 0.24 and 0.3. Performance at the two lower solidities came close to that predicted by a multiple streamtube model, while results for the highest solidity were significantly better than predicted. Only the highest solidity rotor was self-starting and it produced a starting torque of about 25% of the maximum. The results suggest that by employing a medium solidity design the turbine's start-up problem can be solved while sacrificing little in steady state performance at these low stream speeds.

1 NOTATION

a	interference factor
A_s	streamtube cross section area
C_d	blade drag coefficient
C_ℓ	blade lift coefficient
C_n	normal force coefficient
C_p	power coefficient
C_{pb}	power coefficient based on Betz maximum
C_t	tangential force coefficient
C_x	thrust coefficient based on V
f_s	non-dimensional streamwise force
F_s	thrust in the streamtube
$\overline{F}_s$	average thrust in the streamtube
ℓ	blade length
N	number of blades
N_s	number of streamtubes
r	rotor radius
T_s	Torque produced by streamtube
$\overline{T}_\omega$	Mean torque
V	Local fluid velocity
V_r	Fluid approach velocity
V_∞	Velocity upstream of turbine
α	blade angle of attack
β	ratio of blade speed to upstream speed
ρ	density of fluid
ω	angular speed of rotor
σ	solidity ($= \frac{Nc}{2r}$)

2 INTRODUCTION

Interest in the Darrieus turbine over the last 10 years has been centred on its application to harvesting wind energy. Its use as a 'zero head' water turbine is equally feasible from an engineering point of view. Energy extraction from tides using this type of turbine is proposed by Musgrove and Fraenkel (1979) and Davis (1982) as well as the Author, while Thomson (1980) has proposed its use in tapping the power in ocean currents. Its application as a river turbine for Third World countries has been proposed by Fraenkel (1980) and Hilton (1980). The design of the turbine is essentially very simple and may consist of straight, constant chord blades. (The familiar troposkien or "egg-beater" shape of many wind turbines used to minimise centrifugal bending stresses is not necessary due to the low speed of rotation).

In an earlier paper by Hilton (1982) results were presented for a 3 metre diameter machine operating in stream speeds of around 0.6 m/s. The 3-blade and 4-blade configurations corresponded to solidities of .18 and .24 respectively. Although the steady state performance was satisfactory, the turbine proved difficult to start, an experience frequently reported by researchers of wind turbines of the Darrieus type.

In tests on one of these wind turbines, however, Mays and Musgrove (1979) reported that at high solidity their machine became readily self-starting. This tended to be consistent with the Author's own experience insofar as the 4-blade machine was not as difficult to start as the 3-blade version. It was therefore decided to carry out further tests with the 3-blade rotor but with blades of increased chordal width (solidity 0.3)

(a) to determine the extent to which self-starting was possible

(b) to determine the price that has to be paid for this in terms of power coefficient.

3 TURBINE PERFORMANCE PREDICTION

3.1 Theory

Computer prediction was based on the multiple streamtube model of Strickland (1976). The width of each individual streamtube is $r\delta\phi \sin\phi$ for an increment of azimuth angle $\delta\phi$ (see fig.1). The continuity, momentum and energy equations for an individual streamtube yield the following equation for local fluid velocity

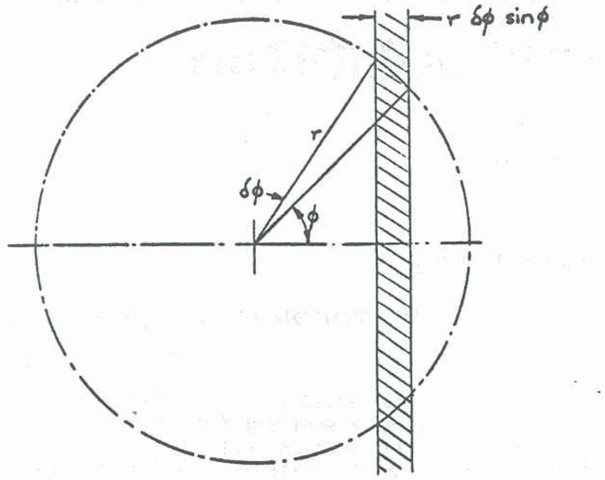

Figure 1 Streamtube geometry

to upstream velocity ratio:

$$\frac{V}{V_\infty} = \frac{1}{(1 + \frac{1}{2}C_x)} \qquad (1)$$

where C_x is the thrust coefficient based on the local velocity V and average thrust in the stream tube $\overline{F}_s$ and given by:

$$C_x = \frac{\overline{F}_s}{\frac{1}{2}\rho A V^2} \qquad (2)$$

The fluid approach velocity relative to the blade V_r is given by:

$$V_r^{\,2} = (r\omega + V\cos\phi)^2 + (V\sin\phi)^2 \qquad (3)$$

where ϕ is the azimuthal position of the blade. The angle of attack from which lift and drag forces are calculated is then given by:

$$\tan\alpha = \frac{V\sin\phi}{r\omega + V\cos\phi} \qquad (4)$$

The average thrust in the streamtube is given by:

$$\overline{F}_s = 2\rho\, A_s\, V(V_\infty - V) \qquad (5)$$

If each blade exerts a thrust F_s as it passes through the streamtube, then the average thrust will depend on the number of blades and the fraction of the total time that each blade spends passing through. Since each blade passes through the streamtube twice per revolution this fraction is $\delta\phi/\pi$. Thus the time-averaged force is:

$$\overline{F}_s = N\, F_s \frac{\delta\phi}{\pi} \qquad (6)$$

Putting in the expression for A_s and eliminating $\overline{F}_s$ from (5) and (6) yields:

$$\frac{NF_s}{2\,\pi\rho r l \sin\phi V_\infty^{\,2}} = \frac{V}{V_\infty}(1 - \frac{V}{V_\infty}) \qquad (7)$$

Strickland refers to the left hand side of equation (7) as the non-dimensional streamwise force, which we shall call f_s. The thrust in the direction of the stream is:

$$F_x = \frac{1}{2}\rho V_r^{\,2}\, lc(C_n \sin\phi - C_t \cos\phi) \qquad (8)$$

where the tangential and normal force coefficients are derived from the blade's lift and drag coefficients as follows:

$$C_t = C_l \sin\alpha - C_d \cos\alpha \qquad (9)$$

$$C_n = C_l \cos\alpha + C_d \sin\alpha \qquad (10)$$

The non-dimensional streamwise force is found by combining (7) and (8):

$$f_s = \frac{NC}{4\pi r}\left(\frac{V_r}{V_\infty}\right)^2 (C_n - C_t \cot\phi) \qquad (11)$$

Using Strickland's interference factor 'a' defined by:

$$a = 1 - \frac{V}{V_\infty} \qquad (12)$$

the streamwise momentum equation can be written in the form:

$$a = f_s + a^2 \qquad (13)$$

The computational procedure commences by setting 'a' to zero. For each given increment of ϕ, the angle of attack and Reynolds number are computed and values of C_l and C_d determined. f_s is then computed from equation (11), following which a new value of 'a' found from equation (13). When the desired accuracy is reached V is found for the streamtube from equation (12). (In the computation ϕ was taken through increments of 5^o). Now the torque produced by the blade element as it passes through the streamtube is:

$$T_s = \frac{1}{2}\rho r c l\, C_t\, V_r^{\,2} \qquad (14)$$

The mean rotor torque is now found by time averaging the torque for each blade and summing for all blades. The power coefficient relative to power in the stream then becomes:

$$C_p = \frac{\overline{T}\omega}{\rho r l V_\infty^{\,3}} = \frac{Nc\beta}{2rN_s}\sum_{1}^{N_s}\left(\frac{V_r}{V_\infty}\right)^2 C_t \qquad (15)$$

and relative to the Betz maximum (16/27 or 0.593):

$$C_{pb} = \frac{27Nc\beta}{32rN_s}\sum_{1}^{N_s}\left(\frac{V_r}{V_\infty}\right)^2 C_t \qquad (16)$$

3.2 Predicted Performance Curves

Performance was predicted using blade lift and drag coefficient data largely taken from Willmer (1979) relating to Reynolds numbers in the range $10^5 - 10^6$. The exception was that for angles of attack within $\pm 8^o$ drag coefficient was taken from measured data resulting from experiments on a 185mm chord turbine blade of symmetrical section with rotor arm fixed in position. (Reynolds number 0.6×10^6). In this way it was possible to take account of rotor arm drag and blade junction losses.

Figure 2 shows predicted performance expressed in the usual form of power coefficient C_{pb} versus β, the blade speed/stream speed ratio, for a stream speed of 0.6m/s and a water temperature of 14^oC. Curves are presented for various solidities. Since lift and drag are sensitive to Reynolds number similar predictions were made for a stream speed of 1.0m/s. The peak power coefficient for both speeds is shown against solidity in Figure 3.

The curves indicate that the best steady state performance would be produced by employing a solidity of between 0.1 and 0.2.

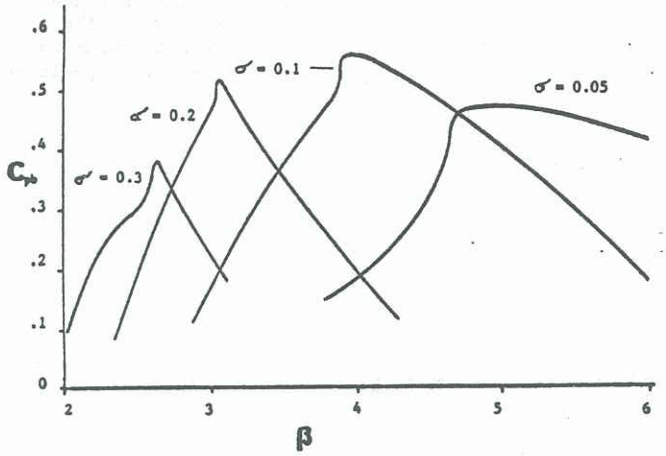

Figure 2 Plot showing predicted power coefficient curves for various solidities
(V_∞ = 0.6 m/s)

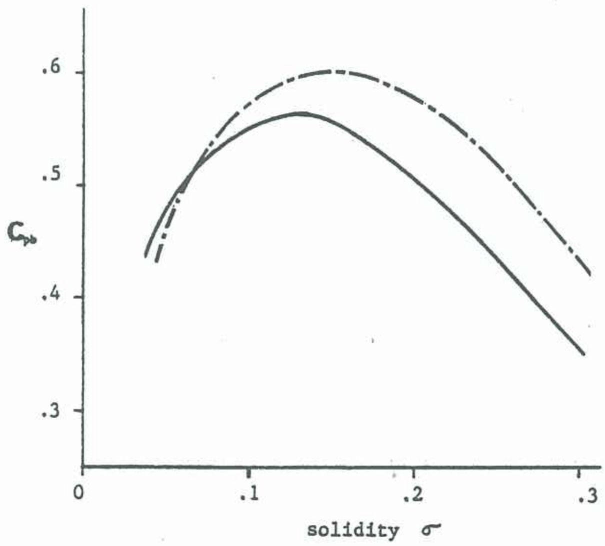

Figure 3 Plot showing predicted peak power coefficient against solidity at two stream speeds
——— V_∞ = 0.6 m/s -·-·- V_∞ = 1 m/s

4 EXPERIMENTAL WORK

4.1 Description Of Test Rig

The 3m experimental turbine rotor was mounted on a vertical cantilever shaft beneath a flat-bottomed vessel of 5m length and 2.5m beam. The 2.4m long blades were made from GRP with tubular steel reinforcement for added stiffness. The rotor arms were made from tubular steel clad with fairings of NACA 0020 section made from GRP. The rotor shaft was fitted with self-aligning detachable bearings which enabled it to be lifted out of the water and the whole rig transported to and from the test site by road trailer.

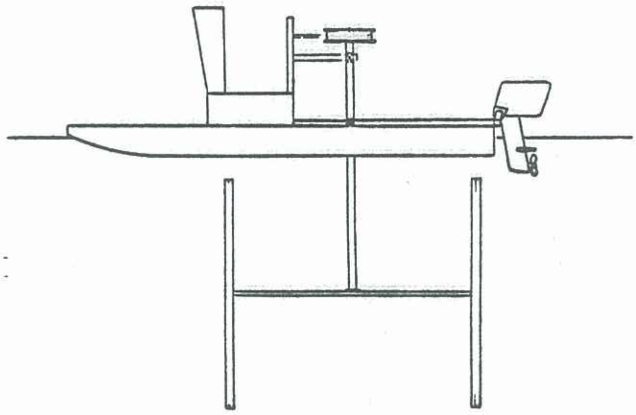

Figure 4 Sketch of experimental rig

Figure 4 shows a sketch of the rig in the working position. The rig was propelled by an outboard engine, and the forward speed measured by using marker posts and a stop watch. The tips of the rotor blades were below and well in front of the outboard's propeller, thereby minimising the effect of the propeller's own streamtube on the rotor performance.

Torque was measured using a simple rope brake and torque arm in conjunction with a load cell with digital readout. Rotor speed was measured by magnetically operated reed switches, producing audible signals which were recorded onto magnetic tape.

4.2 Rotor Configuration

3-bladed and 4-bladed rotors had been used previously in conjunction with blades of 185mm chordal width giving solidities of 0.18 and 0.24 respectively. The same 3-bladed rotor was used in conjunction with blades of 300mm chord to give a solidity of 0.3

4.3 Blade Forms Used

Though cambered in appearance, all the blades in the experiments had a pseudo-symmetrical profile. The actual profile was determined by first taking the symmetrical shape of a NACA 0015 section and modifying it by a transformation which had the effect of wrapping it around a circle of the same diameter as that of the rotor.

This largely compensated for the 'flow curvature' effect inherent in this type of turbine, and was intended to present to the relative fluid stream a profile which, though varying cyclically to a degree, was essentially symmetrical. In the case of the 185mm chord blades a small amount of shrinkage took place during production, resulting in a mean thickness/chord ratio of 14% rather than the 15% intended.

4.4 Stream Speed

All tests were carried out at a nominal speed of 0.6 m/s, though in most test runs variations in rotor thrust at different blade speeds resulted in the stream speed varying between 0.55 and 0.62 m/s.

4.5 Results

Figure 5 shows the power coefficient curves for solidities of 0.18 and 0.24 which were shown in an earlier paper, together with the curve for the higher solidity. Theoretically predicted curves are also plotted for each case, taking account of actual water temperature in the calculation of Reynolds number.

Start-up proved difficult with the lower solidities. With solidity at 0.18 the rotor needed to be spun and a resisting torque was encountered at certain azimuthal positions. With solidity at 0.24 start-up was easier and in certain azimuthal positions the rotor became self-starting at stream speeds of around 0.5 m/s. By contrast the turbine started up easily with a solidity of 0.3. Under no-load conditions, it self-started with stream speeds as low as 0.2 m/s. With greater stream speeds it self-started under load and at 0.8 m/s self-starting was achieved with a brake torque up to 60 Nm (i.e. approximately 25% of the maximum output torque). Start-up did not appear to depend significantly upon the rotor azimuthal position.

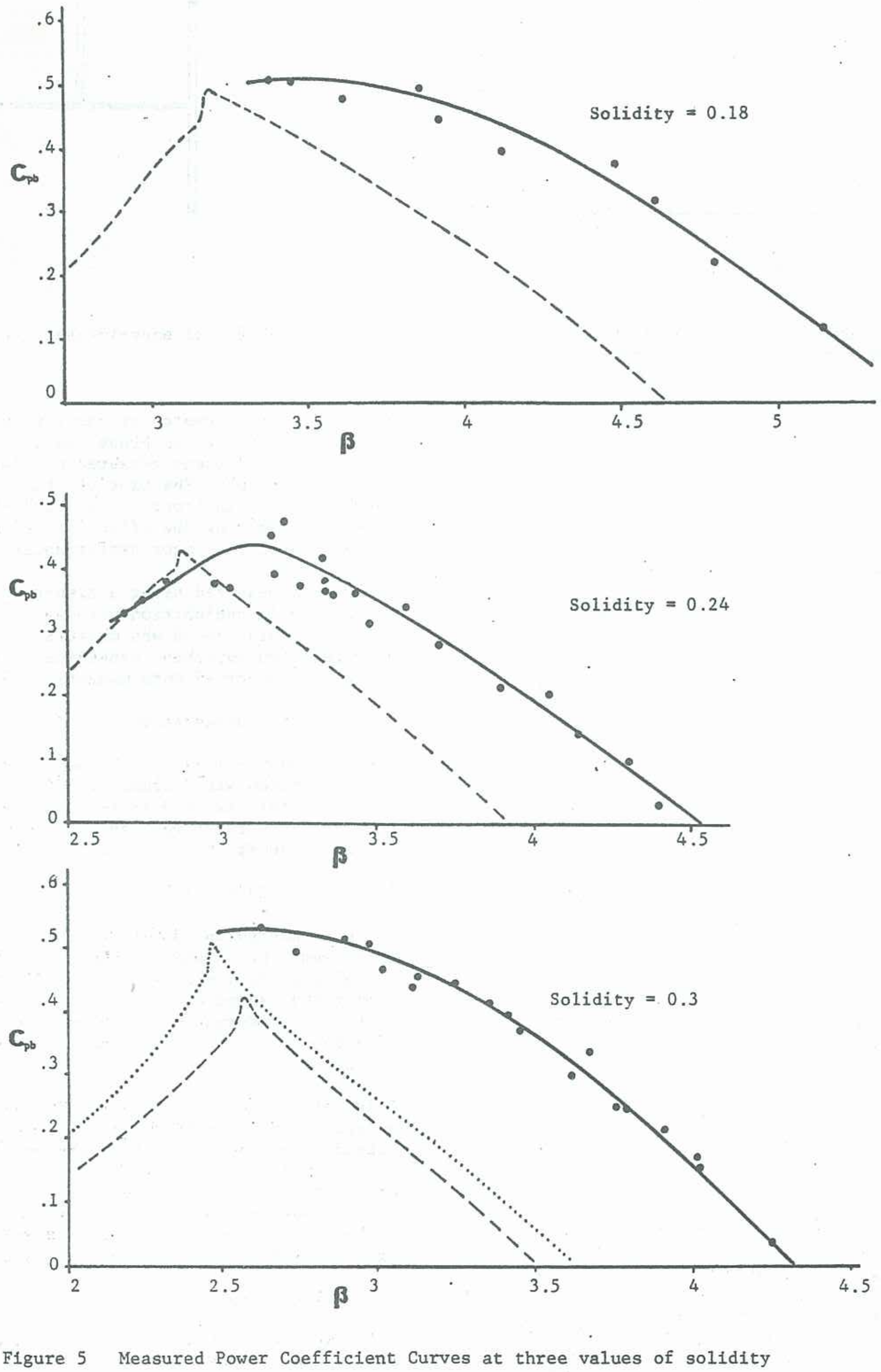

Figure 5 Measured Power Coefficient Curves at three values of solidity

– – – – theoretical prediction

………… prediction with Reynolds number doubled

5 DISCUSSION

5.1 Self-starting Characteristics

The tests confirmed beyond doubt that a moderate increase in solidity greatly improves the starting characteristics of the turbine. The improvement in ease of start-up was much more marked in going from a solidity of 0.24 to 0.3 than in going from 0.18 to 0.24, suggesting the possibility of a critical threshold in the former region. As to the mechanism responsible for self-starting, this is due to the "downwash" produced by the trailing vortices as discussed by Mays & Musgrove (1979). Further tests would be needed, however, to determine the importance of aspect ratio as a parameter as distinct from solidity.

5.2 Steady State Performance

In the case of the two lower solidities, the actual peak power coefficients are very close to those predicted, as are the non-dimensional speeds at which they occur.

However, the measured power coefficient does not fall away with increasing speed as rapidly as theory would suggest. In general, the multiple streamtube theory appears to give a slight underestimate of the turbine's performance. A feature of figure 5 is that experimental scatter is far less than that associated with similar wind turbine test data. (Owing to the relative ease with which stream speed can be measured).

An unexpected outcome of the experiments was that the high solidity rotor produced a peak power coefficient at least as high as that of the lowest solidity rotor, a result which was not predicted by the theory. The former's performance was to an extent enhanced by the high water temperature (24°C) which prevailed during the test, resulting in an increased Reynolds number. However, figure 5 shows that the measured peak performance is still 25% higher than predicted. The reason for this disparity is not certain but there were a number of possible factors. Firstly, the larger size of the 300mm chord blades meant that relatively they were more dimensionally accurate, particularly around the leading edge. This, together with the fact that the 185mm blades with slightly under the design thickness, could have given the 300mm blade section a superior performance. Secondly, the same rotor arms (185mm chord) were used in all experiments, and therefore the losses at the junction would be somewhat lower in the case of the 300mm blades.

Another factor which affected all theoretical prediction was the difficulty in applying lift/drag data taken from steady-state tests to transient conditions. Since stall conditions require a finite time to become established, it is possible that in a situation of rapidly changing angle of incidence, stall will be delayed and the stall angle will therefore be greater than under steady conditions. It could therefore be argued that in predicting turbine performance some upward correction to the calculated stall angle (or alternatively Reynolds number) might be appropriate.

For instance, figure 5 shows an additional predicted curve derived as a result of doubling the calculated Reynolds number, and it can be seen that this second curve comes much closer to the experimental results than the first. Clearly there is a need for lift/drag data to be obtained at low Reynolds number under oscillating conditions before the reliability of theoretical prediction can be improved.

6 APPLICATION AND CHOICE OF SOLIDITY

The straight-bladed Darrieus turbine concept appears to offer promise as a water turbine for converting the energy of stream flows in rivers, tides and ocean currents. Its economic feasibility in various applications has yet to be established, but it is likely that the most favourable situation for medium-sized units might be in tidal turbines serving small coastal communities remote from the electrical grid. Two immediate small scale applications are in powering navigation buoys as described by Grant (1981) and in irrigation of riverside plots in Third World countries.

In the context of tidal and ocean power, large turbines would have the advantage not only of economy of size, but would operate at higher Reynolds number for a given stream speed and so should produce peak power coefficients higher than those measured in these experiments. (0.7 is predicted from theory).

The choice of solidity would depend on the application. With small and medium size machines self-starting is a highly desirable feature, and this is likely to dictate the choice. Although there is still need for more experimental work, it has been shown that a solidity of around 0.3 is adequate for self-starting with most types of load. Where the load is a positive displacement pump, however, a centrifugal clutching system might be deemed necessary to allow the turbine to get up to speed. With large machines the cost of a starter gear might not be prohibitive, and in any case self-starting may not be desirable from the point of view of machine maintenance; hence a lower solidity of 0.1 to 0.2 might be preferable.

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