Effect of blade profile on the performance of the Wells self-rectifying air turbine

S. Raghunathan and C. P. Tan*

The effect of blade profile and thickness on the starting and running performance of a 0.2 m diameter Wells turbine is reported. The starting torque can be increased considerably by using thicker NACA aerofoils and modified NACA aerofoil blades and by increasing the turbine solidity. Thicker and modified NACA aerofoil blades also improved the running performance of the turbine. Artificially roughened blades showed degraded performance

Keywords: *turbomachinery, turbine blades, wave energy*

The Wells turbine is an axial flow self-rectifying air turbine suitable for the energy conversion from oscillating air flows, conditions encountered in the oscillating water column of wave energy converters. The turbine rotor consists of several symmetrical aerofoil blades set around a central hub (Fig 1). For axial air flow velocity V_x and a tangential blade velocity of U at a radius r from the axis of the rotor, the relative velocity W is at an angle of incidence α to the blade chord. This would result in a lift force L and drag force D normal and parallel to the direction of the velocity vector W, respectively. The lift and drag forces can be resolved into tangential and axial components F_t and F_x respectively:

$$F_t = L \sin\alpha - D\cos\alpha$$

$$F_x = L\cos\alpha + D\sin\alpha.$$

For a symmetrical aerofoil, the direction of tangential force, F_t, is the same for both positive and negative values of α. Therefore the rotor will rotate in the direction of F_t, regardless of the direction of air flow. In an oscillating air flow the values of F_t and F_x will vary with both the radius of the rotor and time. However, the turbine can be made to rotate at a constant speed by using a rotor of high inertia. The oscillating axial forces F_x have to be borne by suitable bearings.

A practical problem encountered with this turbine is the difficulty in running up to operational speed when starting from rest. This phenomena, termed cranking, where the turbine runs up to a certain speed which is much lower than the operational speed, can be explained by the observation of a typical F_t–α curve for a symmetrical aerofoil (Fig 2). When the turbine starts from rest, the value of α is 90%. Since C_t is positive at this value of α, the turbine is able to rotate. Assuming a constant axial inlet velocity, α will decrease with the increase in rotor speed. The value of F_t is positive in regime IV, therefore enabling the rotor to keep on rotating. The operating regimes for the rotor are in regimes I and II and, in order to move into this regime, the rotor must move through the regime III where the values of F_t are negative. The inability of the rotor to pass through III results in crawling. The magnitudes of C_t and the range of α in the regime III can be decreased by increasing the solidity of the blades. By a suitable combination of the solidity and the hub to tip ratio, the rotor can be made to self start.

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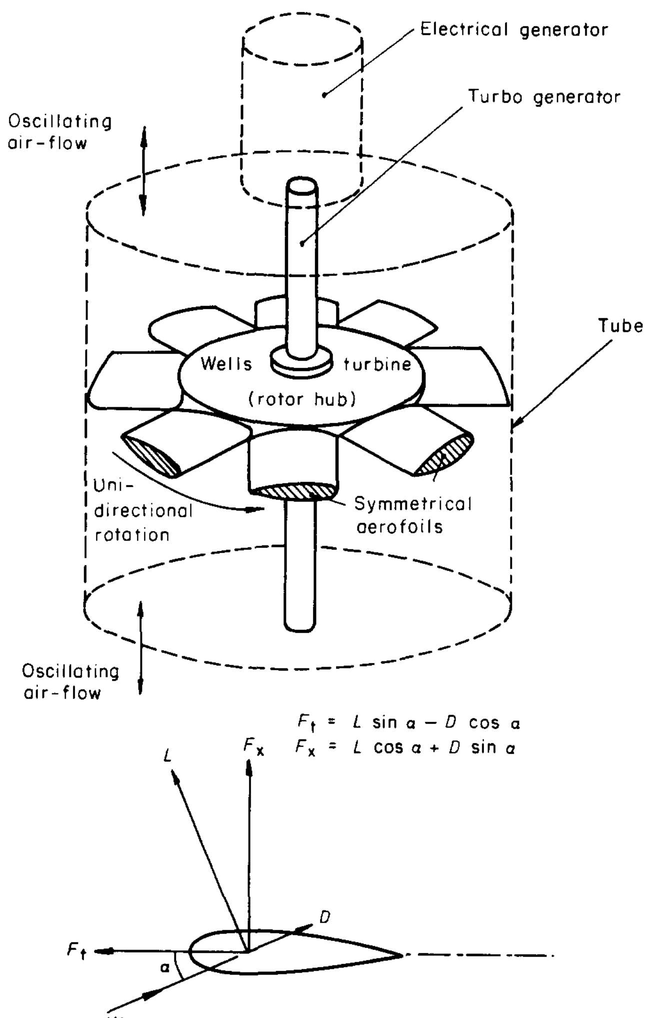

Fig 1 Wells turbine

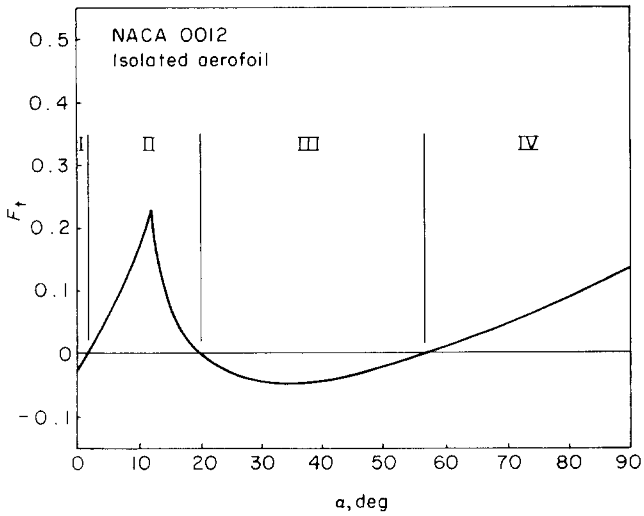

Fig 2 Variation of F_t *with* α

There has been some experimental investigation on the Wells turbine in unidirectional air flow rigs[1-4] which demonstrated that the Wells turbine is an efficient energy converter. Previous work dealt with the influence of several aerodynamic variables on the performance of the turbine[4] and prediction methods for the turbine performance and the starting characteristics of the turbine[5,6]. The objective of this report is to evaluate the effect of blade profile shape and roughness on the starting and running characteristics of a small scale Wells turbine.

Test facility, model and instrumentation

Experiments were performed in a unidirectional air flow rig. It is generally accepted that the performance of the Wells turbine in low frequency oscillating air flows of less than 1 Hz can be predicted from unidirectional air flow tests; this is under investigation at Queen's University of Belfast. The test rig consisted of a centrifugal fan, settling chamber, contraction section and a 0.2 m diameter test section in which the turbine was mounted.

The wall of the rotor hub was made of steel tube in which an Electro-Craft permanent magnet type dc motor and an optical rotary torque transducer ASL DORT6 were located. The servo motor also served to run the turbine to operational speed when the turbine was not self starting. The control gear used for loading this turbine consisted of a few rheostats connected in series. Pressure tappings were arranged circumferentially around the casing in groups of 4, 0.02 m upstream and downstream of this rotor. The flow rate was measured using a calibrated orifice plate mounted 0.7 m downstream of the rotor. Low differential pressure transducers, type FC40, were used for the pressure measurements. The turbine had a hub to tip ratio, $h = 0.62$. The solidity S was varied from 0.25 to 0.75 by varying the number of blades.

Five symmetrical aerofoil sections of constant chord $c = 0.065$ m but different thickness ratios were investigated. Four were of the standard NACA 4-digit symmetrical sections with the position of maximum thickness at 0.30c. They were NACA 0012, 0015, 0021, and 0024. The other section tested was a modified profile with a maximum thickness at 0.225C (NACA 0015H). The power output was calculated from the torque and speed measurements.

The power input to the rotor was calculated from the flow rate and the pressure drop across the rotor. Tests were performed with pressure drops in the range 200 to 1500 N/m^2, and flow rate 0.05 to 0.25 m^3/s, for the various turbine profiles and solidities. The Reynolds number based on the blade chord, Re, was approximately equal to 2.0×10^5 during the tests.

It was observed during the earlier tests that it is necessary to have a high solidity for the turbine to accelerate to the operational speed when starting from rest. Otherwise the turbine tended to crawl[6]. In order to further understanding of the starting behaviour of the turbine, the rotor was locked (prevented from rotating) and the variation in torque at zero speed with pressure drop and flow rate measured. In this investigation, one end of the shaft of the torque transducer was restrained from moving while the other end was coupled to the rotor. Any twisting or torque was measured by the torque transducer.

Results and discussions

The variation of the pressure drop across the turbine at zero speed of rotation with flow rate, for a range of solidity, is shown in Fig 3. The pressure drop was proportional to the square of the flow rate, indicating that

Notation

c	Blade chord length, m
D_t	Tip diameter of rotor, m
D	Drag force, N
$F_{t,x}$	Tangential, axial force on the rotor
h	Hub to tip ratio, r_h/r_t
l	Blade length, $r_t - r_h$
L	Lift force, N
n	Number of blades
P^*	Pressure coefficient, $(\bar{\Delta}P/\rho\omega^2 D_t^2)$
$\bar{\Delta}P$	Pressure drop across the rotor, N m^2
$r_{h,t}$	Hub, tip radius of rotor, m
r_m	Mean radius of rotor, $(r_h + r_t)/2$
Re	Reynolds number, $W_\infty c/v$
S	Solidity, $nc/\{\pi r_t(1+h)\}$
U_t	Tangential velocity at the tip, m/s
V_x, W_∞	Axial, relative flow velocity, m/s
$W_{I,O}$	Power input, output, watts
W^*	Power coefficient, $W_0/(\rho\omega^3 D_t^5)$
α	Angle of incidence, degrees
η_{max}	Efficiency, maximum efficiency, W_0/W_I
v	Kinematic viscosity of air, m^2/s
ρ	Density of air, kg m^3
τ	Torque at zero speed, Nm
τ^*	Torque coefficient $\tau/(\frac{1}{2}\rho V_x^2 c l \eta r_m)$
ϕ	Flow coefficient V_x/U_t
ω	Angular velocity, rad/s

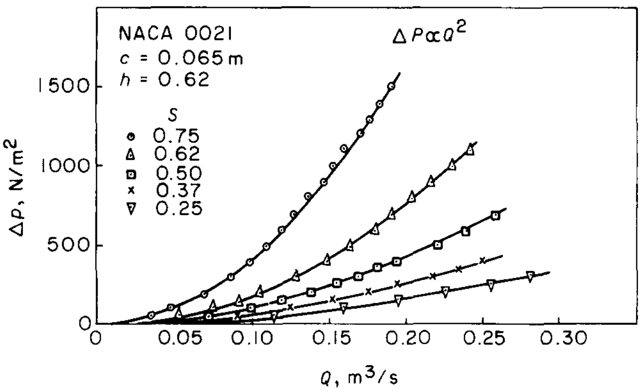

Fig 3 The variation of pressure drop with flow rate

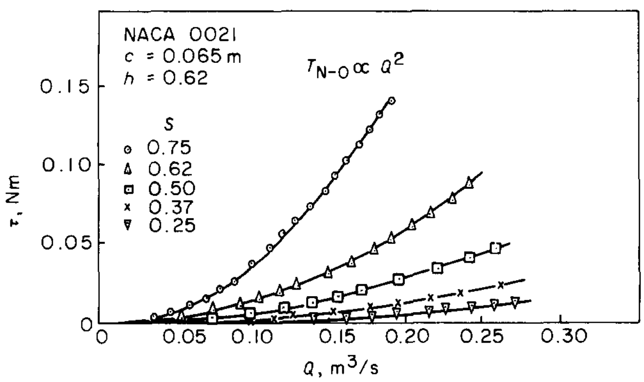

Fig 4 Torque at zero speed against flow rate

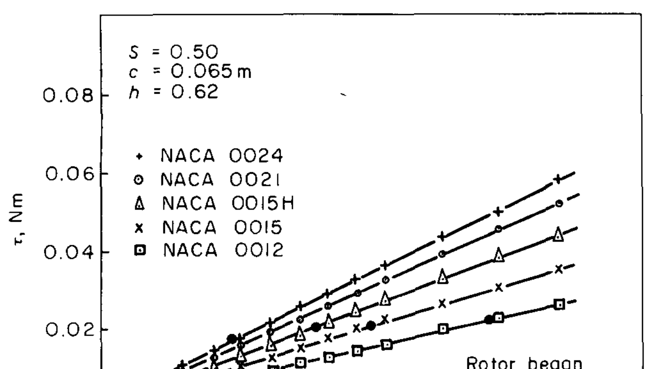

Fig 5 Torque at zero speed against pressure drop

when the turbine was not rotating it had the same pressure drop versus flow rate characteristic as that of an orifice plate.

The variation of the torque at zero speed with flow rate, Q, for a range of solidities, is shown in Fig 4. These torque values increase with the increase in solidity for a constant flow rate. It was found that for a given solidity the torque at zero speed was proportional to the square of the flow rate.

Figs 5 and 6 show the effect of thickness/chord ratio, t/c, on the torque at zero speed against pressure drop for $S=0.5$ and 0.7 respectively. At constant pressure drop, an increase in t/c results in an increase in the torque values. Although the results given here are for turbine solidities of 0.75 and 0.50, the trends of the results should be similar for other solidity values. Referring to Fig 5, the torque at zero speed for NACA 0021 blade section is slightly more than twice the value obtained for NACA 0012 blade section at the same pressure drop. It was observed that the rotor with NACA 0021 blades began to rotate when the pressure drops across the rotor were about 25 N/m^2, whereas for the rotor with NACA 0012 blades, the pressure drop values were about 50 N/m^2. Both these values corresponded to almost the same value of torque. Also illustrated in Figs 5 and 6 are the pressure drop values taken just before the rotor began to rotate. It is interesting to note here that all the values of pressure drop recorded correspond to almost the same torque value, ie about 0.02 Nm. This torque value must therefore correspond to the frictional torque of the system for this model Wells turbine and depends on the type of bearings used and the starting friction of the motor.

A unique correlation between torque and pressure drop can be obtained if plotted in a nondimensional form shown in Fig 7. This correlation is independent of pressure drop, flow rate and solidity and only a function of blade profile.

Figs 8 and 9 show the effect of thickness/chord

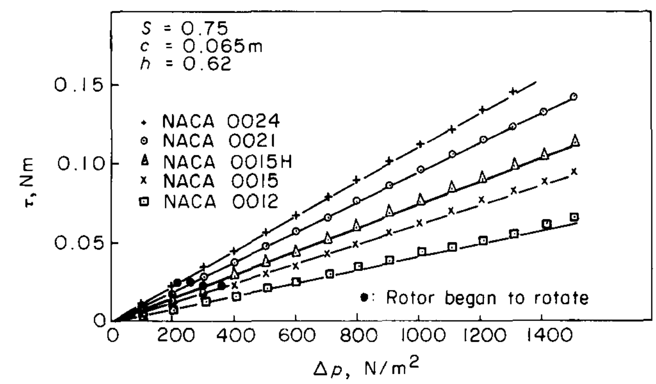

Fig 6 Torque at zero speed against pressure drop

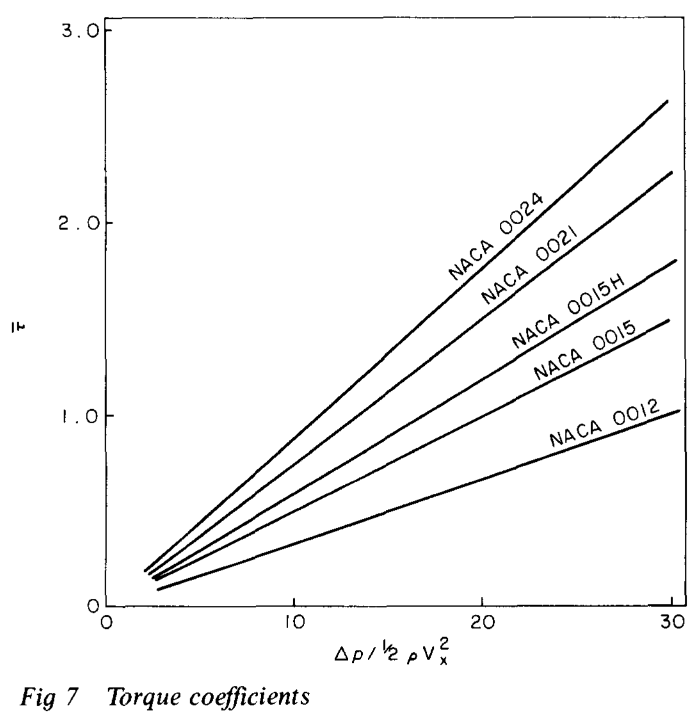

Fig 7 Torque coefficients

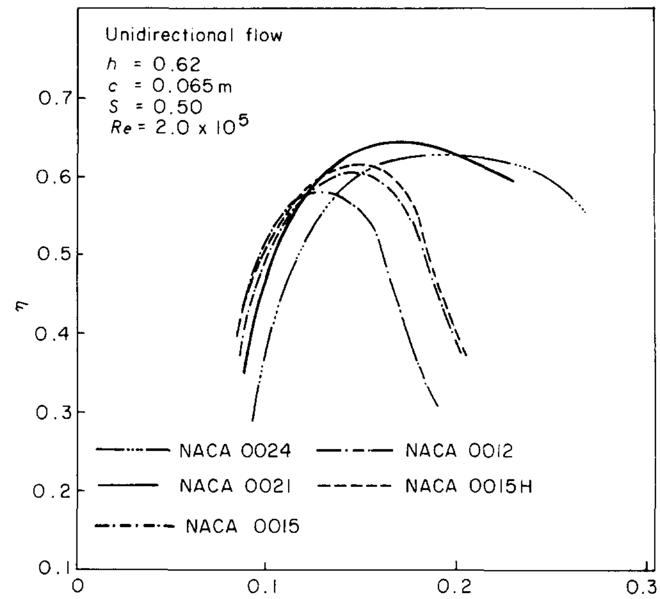

Fig 8 Effect of blade thickness on efficiency

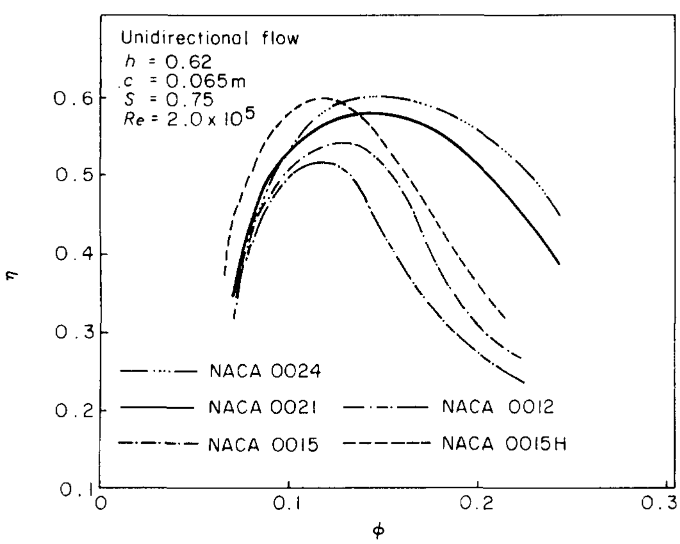

Fig 9 Effect of blade thickness on efficiency

ratio on the efficiency of the turbine for $S=0.50$ and 0.75 respectively. For clarity, only the best fitted lines through the experimental points are shown for comparison. For $S=0.50$, both the peak efficiency, η_{max}, and stalling incidence, ϕ_{max}, increase with the increase in thickness/chord ratio, t/c. This can be explained by the fact that the stall associated with a thicker aerofoil is more gradual and the reduction in lift is less sudden. This would suggest an improvement in the average cyclic performance of the turbine with thicker turbine blades. NACA 0015H profile shows a slight improvement in η_{max} compared with NACA 0015 profile.

The η against ϕ curves for $S=0.75$ show that the stall are more gradual and the reduction in efficiency are less sudden than those for $S=0.50$. This implies that the average cyclic performance improves with solidity. It is also important to note that the η_{max} values obtained for $S=0.75$ are lower than those for $S=0.50$. The effect of t/c on η is more pronounced at $S=0.75$. Modification to the profile shape (from NACA 0015 to NACA 0015H) for $S=0.75$ improves the η_{max} by 10%, which is a considerable increase.

Fig 10 is a plot of power coefficient, W^*, against flow coefficient, ϕ, for NACA 0012, 0015, 0021, 0024 and 0015H profiles, for solidities of 0.50 and 0.75. For both solidities, it was observed that the effect of thickness/chord ratio on the power coefficient was negligible before stall. For $S=0.5$, both W^*_{max} and ϕ_{max} increased with the increase in t/c. Therefore the use of thicker profiles would enable the turbine to be operated over a broader range of flow rates. Within the range of Reynolds numbers tested, the ϕ_{max} values increase from 0.16 to 0.18 for an increase in t/c from 0.12 to 0.15. The corresponding values of W^*_{max} are 11.2×10^{-4} and 15.2×10^{-4} respectively. Further increase in W^*_{max} and ϕ_{max} can be obtained by the use of modified profile as shown by comparing the NACA 0015H results with NACA 0015. For the NACA 0024 or 0021 blade profile, it appears that the stall is too gradual for detection from this plot. Comparisons of the curves for $S=0.75$ with those of $S=0.50$ show that an increase in solidity generally results in an increase in W^*_{max}. Even for a thin profile, the stall appears to be more gradual at a high solidity. If these curves were to be extrapolated to zero values of W^*, it would show that an increased solidity reduces the range of ϕ for which the value of W^* is negative. This would be beneficial for a turbine operating in an oscillating airflow.

The corresponding curves for pressure coefficient, P^*, plotted against the flow coefficient, ϕ, are shown in Fig 11. At $S=0.5$, the effect of thickness or modification to its shape has a small effect on the pressure drop. It is

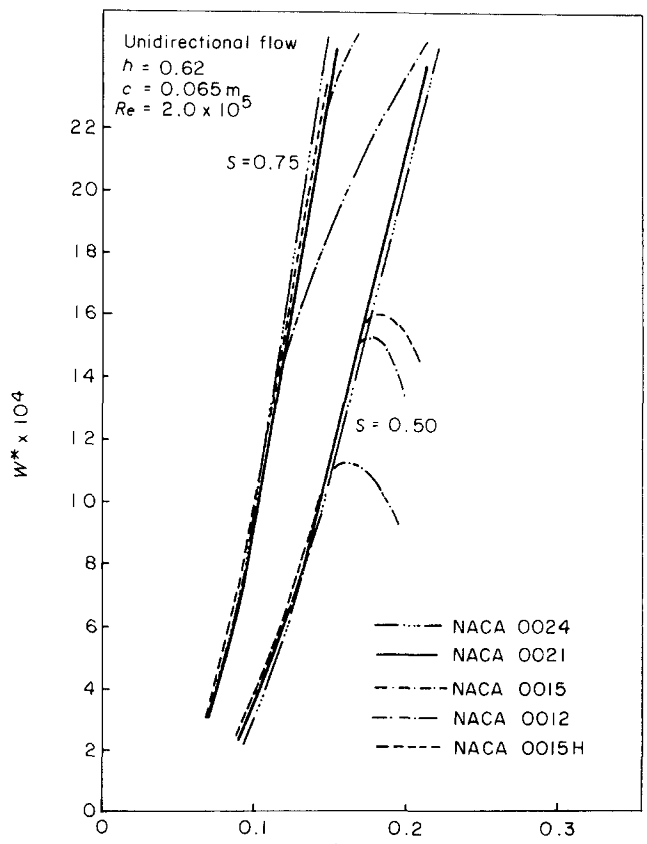

Fig 10 Power coefficients

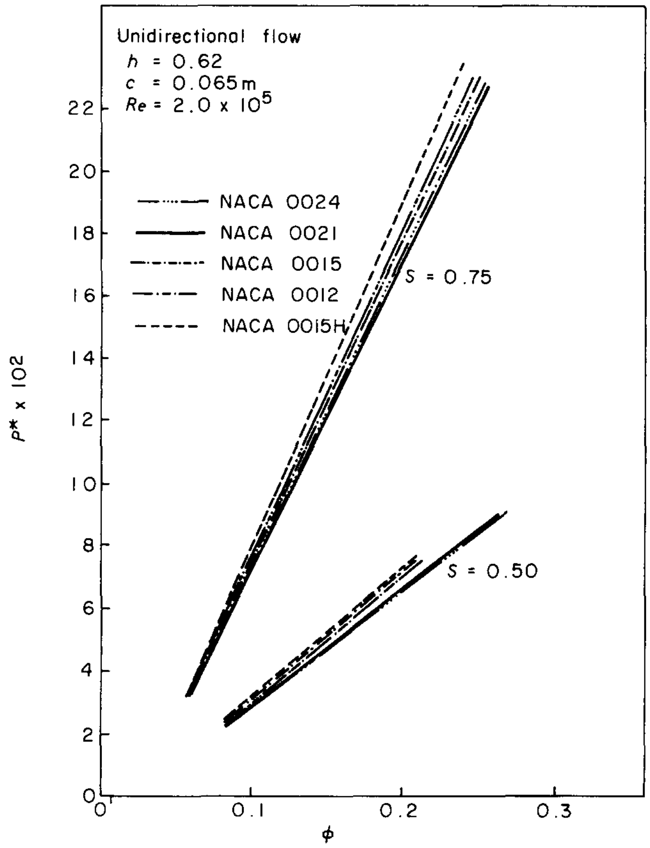

Fig 11 Pressure coefficients

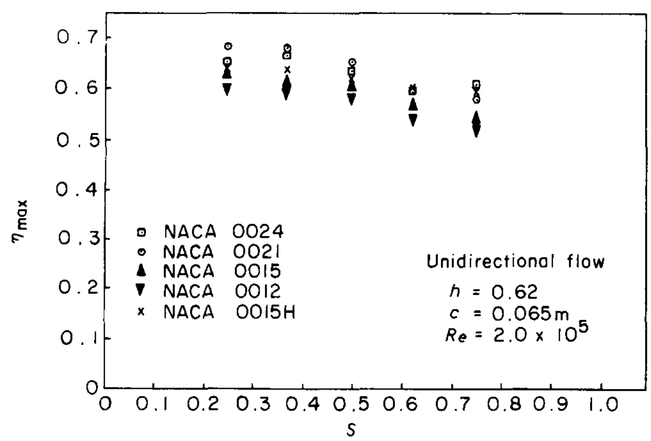

Fig 12 Effect of solidity on peak efficiency

interesting to note that a linear relationship exists between P^* and ϕ even in the stalled regime, a condition which is well suited for matching the turbine to an oscillating water column type wave energy device. The effect of t/c on the pressure coefficient, P^*, is slightly more noticeable at $S=0.75$. Generally there is a slight reduction in the pressure drop with the increase in t/c. NACA 0015H produces a larger pressure drop than NACA 0015. Typically at a value of $\phi=0.15$, the modified profile produces an increase in pressure drop of slightly less than 6%. This effect could be beneficial when the available pressure drop in the water column of the wave energy device is large.

The peak values of efficiency, η_{max}, are plotted against solidity, S, for various values of t/c in Fig 12. Generally, a drop in the performance is observed when S increases. On average, an increase in solidity from 0.5 to 0.75 results in a decrease in η_{max} of about 10%.

The performance of the Wells turbine is highly dependent on the tangential force coefficient, C_t, which in turn is a function of lift and drag coefficients. Any reduction in the lift to drag ratio as a result of surface roughness, will obviously affect the performance of the Wells turbine.

To investigate the effect of surface roughness, NACA 0021 aerofoil blades of 0.065 m chord length were chosen and two types of roughness tests performed. The first type was with artificial roughness applied to the leading edge of all the aerofoil blades and the second, with artificial roughness applied over the whole surfaces of all the blades. It should be mentioned that no attempt has been made here to simulate ice formation or salt deposition on the blades, but merely to study the sensitivity of the performance to any of these effects described. In both types of tests, the artificial roughness applied to the surfaces of the blades consisted of a carborundum layer with a grain size of 0.06 mm. The carborundum particles were glued to the surfaces using shellac and uniformly packed against each other. The roughness layer was about 0.15 mm thick. For the leading edge roughness test, the roughness strip covered about 8% of the surfaces in the region of $0 \leqslant x \leqslant 0.08c$ measured from the leading edge.

The effect of roughness on the efficiency is shown in Fig 13 for solidity of 0.50. Generally, a slight degradation in performance over the whole range of flow coefficient, ϕ, was observed for blades with leading edge roughness. Both the peak and cyclic efficiencies of the turbine were drastically reduced when the artificial roughness layers were applied completely over the surfaces of all the blades. The reduction in peak efficiency is about 10%. It appears from these figures that the peak efficiency occurs at a lower flow coefficient for blades with smooth surfaces. The effect of roughness on P^* was negligible.

Conclusions

The main conclusion which can be drawn from this study of the effect of blade profile and thickness on the

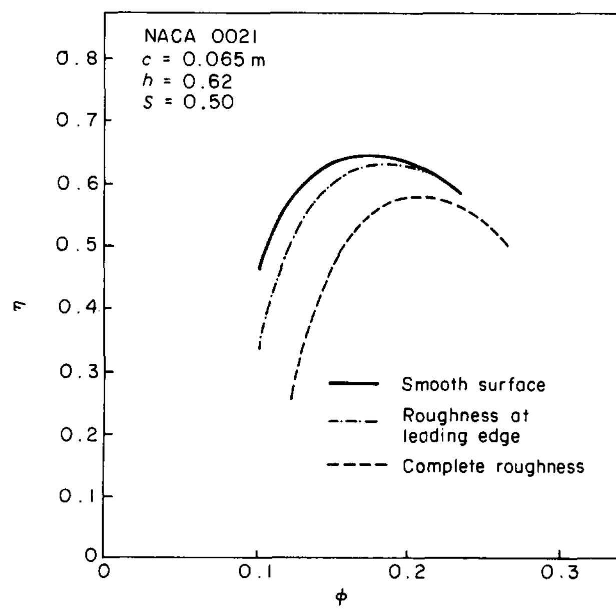

Fig 13 Effect of roughness

starting and running performance of a small scale Wells turbine are:

(1) The starting torque can be considerably increased by using thick aerofoil sections for the rotor. Modified aerofoil blades with the maximum thickness nearer to the leading edge also produce similar effects.
(2) The NACA 0021 blades produced the maximum peak efficiency and this could be further improved by modifying the profile shape.
(3) The effect of leading edge roughness is a slight degradation in the turbine performance over the whole range of operation. A reduction in the peak efficiency of about 10% can result with the blades completely covered with roughness.

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BOOK REVIEW

Heat Flow Through Extended Surface Heat Exchangers

M. Manzoor

One notes with gratification that the publisher of this book has established a procedure where new developments in fields of engineering emphasising the application of methods in engineering analysis and design can be published quickly, informally and at a high quality level. Thus, lectures in a new field, material requiring timely publication, research reports, software developments and proceedings of meetings of exceptional interest (or devoted to a topic of wide interest) can be published within six months of receipt of manuscript in a series entitled 'Lecture Notes in Engineering'.

The subject work by Manzoor is at a level where the mathematics, if not of overpowering difficulty, is excruciatingly detailed. The development is based on what he claims is a more sophisticated model; one that treats a finned assembly consisting of both finned and prime surfaces as a combined entity. In addition, he provides a systematic attack on some of the limiting assumptions that prevail in the analysis of extended surface; two-dimensional effects, the analysis of radiation from extended surface and the applicability of the perfect contact assumption.

Most certainly, however, these lecture notes are on a graduate level and the use of the boundary integral equation analysis is proposed. It is felt that this rather new procedure, which provides an alternative to finite difference and finite element analysis, must be presented in conjunction with some rather concise examples so that the reader (user) can get into the swing of the method without the worry of whether he is proceeding correctly. The author does provide an example here and there, but there is fault to be found with both the number and scope of the examples.

The author deserves a great deal of credit in his attempt to bury the fin efficiency as a design tool (chapter 2). In this attempt, he comes up with an augmentation factor and then an enhancement factor which strongly resembles the fin effectiveness (usually attributed to Gardner). But the claim that 'the fin efficiency gives no indication whatsoever regarding the effect of the fins on the overall heat transfer' is felt to be somewhat ill-conceived. To be sure, there are better methods of looking at the effect of the fins (for example, by an electrothermal and by a transmission line analogy which Manzoor advocates and discusses), but as a design tool the fin efficiency may be better for the engineer designing the equipment than the somewhat formidable boundary integral equation approach.

In conclusion, the attack on the limiting assumptions in this work is commendable but we must never, as design people, lose sight of the fact that 'with all the assumptions, if it works twice as well in an analysis, then it might work twice as well in service'. The analyses in this book are works of art but much guidance is needed on how they may be applied to the proper fabrication of hardware.

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