Numerical Analysis of a Non-Profiled Plate for Flapping Wing Turbines

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Abstract

A non-profiled (rectangular) plate section was investigated for its effect on the performance of a flapping wing turbine for power generation. The plate was subjected to laminar flow conditions having a steady free stream flow (Reynolds number $Re$ 1100); while flapping with prescribed sinusoidal pitching and plunging kinematics. Results of numerical simulation give a power efficiency as high as 34% for a non-profiled plate with a pitching amplitude of 75°, a plunge amplitude of 1 chord, a phase difference of 90° between pitch and plunge, a reduced frequency 0.8, at pivot location 1/3 chord from the leading edge. This performance is 5.35% over that of the profiled counterpart (NACA0012). Results also show that similar to the profiled plate, the plate thickness has an insignificant effect on the turbine performance.

Introduction

Water and wind are major sources of renewable energy which properly harnessed can significantly contribute towards meeting the increasing demand for clean energy. Both sources are currently underutilised; however the growth rate in utilising wind energy is very prominent when compared to other sources [1]. Oscillating plate/wing turbines are quite new compared to the ir rotary turbines. The foil is yet to be ascertained.

The design and manufacture of profiled plates are quite complex and costly compared to non-profiled plates. On the other hand, the extent of degradation or improvement on the performance when a non-profiled plate/wing is deployed in an oscillating plate turbine is yet to be ascertained.

The purpose of this study is therefore, to investigate the performance of a non-profiled (rectangular) plate section (Figure 1b) in an oscillating plate turbine. The rectangular plate has the same chord length ($c = 1.0m$) and cross-sectional area (0.08082$m^2$) as the profiled plate (Figure 1a). Its relative thickness ($b = 0.08082$ chord) will be varied to determine its effect on the overall performance. Numerical simulations are conducted using the commercial software package ANSYS 14 FLUENT to solve the unsteady 2D Navier-Stokes equations with prescribed motion kinematics for both pitch and plunge motions, at a low Reynolds number ($Re$) of 1100 for a steady free-stream flow. The $Re$ chosen allows comparison with the results in the literature; and for this flow, the flow can be assumed to be laminar.

Motion Kinematics

The oscillating non-profiled plate is set to undergo a sinusoidal motion in both plunging ($z$-direction) and pitching (about $z$-direction) directions. The plate was pivoted at $X_p$ and restricted to pitch about an angle $\theta_0$ while plunging about a defined vertical distance $h_0$. The phase difference $\varphi$ of 90° was implemented between both motions with the plunge motion $h(t)$ leading the pitch motion $\theta(t)$ (equation 1&2 ). Figure 3 shows the details of a combined pitch and plunge motion for power extraction over one
periodic cycle when a phase difference $\phi$ of 90° is implemented between the pitch and plunge.

\[
\theta(t) = \theta_0 \sin(\omega t) \quad \text{and} \quad \dot{\theta}(t) = \theta_0 \omega \cos(\omega t) \quad (1)
\]

\[
h(t) = h_0 \cos(\omega t) \quad \text{and} \quad V_p(t) = -h_0 \omega \sin(\omega t) \quad (2)
\]

where $\dot{\theta}(t)$ represents the pitching velocity, $V_p(t)$ the plunging velocity in the y-direction, $\omega$ the angular frequency of the oscillation.

\[\begin{align*}
\dot{\theta}(t) &= \theta_0 \sin(\omega t) \\
h(t) &= h_0 \cos(\omega t)
\end{align*}\]

Figure 3. Kinematic motion of a non-profile plate over one cycle

When the non-profiled plate is subjected to pitch and plunge motions defined in equations (1) & (2), the instantaneous effective angle of attack of the plate $\alpha_{eff}(t)$ is given by

\[
\alpha_{eff}(t) = \alpha_0 + \theta(t) + \tan^{-1}\left(\frac{V_y(t)}{U_\infty}\right) \quad (3)
\]

where $U_\infty$ is the free stream velocity. The non-dimensional reduced frequency $k$ of the oscillating plate is given by

\[
k = \frac{2\pi f_c}{U_\infty} = \frac{\omega c}{U_\infty} \quad (4)
\]

Energy Extraction and Efficiency

The amount of power $P$ extracted by the oscillating plate turbine is measured by its power coefficient $C_p$, given in terms of equations (5) and (6) respectively. Its efficiency $\eta$ is defined in equation (7) as the ratio of the extracted power to the available power within the swept area.

\[
P(t) = P_h(t) + P_\theta(t) = \frac{L}{2} V_p^2(t) + \frac{M}{2} \dot{\theta}(t) \quad (5)
\]

\[
C_p = \frac{P}{\frac{1}{2} \rho U_\infty^3 \omega} = \left[ C_{p_\theta}(\theta) + C_{p_h}(h) \right](t) = \left[ C_{M^\theta} \frac{\dot{\theta}}{U_\infty} + C_{L^\theta} \frac{V_p}{U_\infty} \right](t) \quad (6)
\]

\[
\eta = \frac{P}{\frac{1}{2} \rho U_\infty^2 \omega \theta_0} = \frac{C_p}{d} \quad (7)
\]

\[
d = \max \left( \frac{h_\theta}{h_0}, \frac{(h_\theta + LE \sin \theta_0)}{h_0} \right) \quad \text{or} \quad \left( \frac{h_\theta - TE \sin \theta_0}{h_0} \right) \quad (8)
\]

Here the span $s$ has a unit value for 2D calculations; $\rho$, $L$ and $M$ represent density, lift and moment forces respectively; $C_{p_\theta}(\theta)$ and $C_{p_h}(h)$ represent the power coefficient for the pitch and plunge motions, while $C_{M}$ and $C_{L}$ represent the coefficient of moment and lift respectively. The overall displacement of the plate $d$ is taken as the maximum displacement from either the leading edge $LE$, trailing edge $TE$ or plunge amplitude $h_\theta$ of the oscillating plate over a cycle.

Unsteady Flow Solver

All numerical simulations were undertaken using ANSYS 14 FLUENT CFD commercial software to solve the unsteady incompressible 2D Navier-Stokes equations for laminar free stream flow. The Pressure based solver was selected, adopting the SIMPLE pressure-velocity coupling solution method and Green-Gauss node based gradient. The residual level was set at $10^{-5}$; second order scheme was set for pressure while second order upwind scheme set for the momentum to achieve higher-order accuracy.

Two fluid domains centred at the plate’s pivot point were modelled to enclose the plate, with a circular domain positioned in-between the square domain (Figure 4). The circular domain bounded by a sliding interface has a radius of 10, while the dimension of each side of the square domain is 70. As employed in [5, 7], a source term method was adopted enabling the use of second order implicit central differencing scheme in time rather than the default first order implicit method deployed in the dynamic mesh strategy whose accuracy is quite limited to fine mesh and very small time-steps. In the source term method adopted, the circular domain containing the plate was made a moving frame thus revolving about the pivot location while the square domain was made a fixed frame. At the inlet boundary, the free stream velocity was applied in the x-direction while the transient plunge velocity was applied in the y-direction. The source term in form of the product of the plunging acceleration and density was added to the Navier-Stokes equation, and applied to both domains in the y-direction using the y-momentum option.

Description of Grid

A structured quad-grid (Figure 4) was used to discretize both domains. Three different sets of structured grids were used for grid independence tests: a coarse-grid of 65,000 cells & 450 nodes on the plate, a medium-grid of 121,000 cells & 550 nodes on the plate and a fine-grid of 190,400 cells & 700 nodes on the plate; while time-steps of 500, 2000 and 4000 per cycle were used for the time-step independence study. These tests were conducted for $x_\theta = 0.333c$, $h_0 = 1.0$, $\theta_0 = 76.33^\circ$, $k = 0.88$ and $Re = 1100$. For each test, at least 12 cycles were calculated until the ratio of $C_{pmean}$ for the current cycle to that of the previous cycle is less than 0.14%. Table 1 and Figure 5 show the effect of grid resolution and the number of time-steps on the calculated $C_{pmean}$ and $\eta$. Based on these results, the medium-grid with 2000 time-steps per cycle were used for simulations here.
Results and Discussion

Efficiency
time-step per cycle. Figure 6 shows that the results obtained (52,000 cells and 400 nodes around the airfoil was used, with 2000 X p airfoil at Dumas [5] (1100. Based on a grid and time-step independence study, a grid of h0 (0.6 to 1.2) with mean = 0.8685 & h0 = 0.69 (0.75 to 1.2) and Re = 1100. The predicted numerical results of Kinsey and Dumas [5] for a NACA0015 plate at Re = 1100. Based on a grid and time-step independence study, a grid of 52,000 cells and 400 nodes around the airfoil was used, with 2000 time-step per cycle. Figure 6 shows that the results obtained (Cp mean = 0.8685 & η = 33.89%) agree very well with Kinsey and Dumas [5] (Cp mean = 0.860 & η = 33.70%) to within 1%

The code and methodology adopted here was validated using the numerical results of Kinsey and Dumas [5] for a NACA0015 airfoil at Xp = 0.333c, h0 = 1.0c, θ0 = 76.33, k = 0.88 & Re = 1100. Based on a grid and time-step independence study, a grid of 52,000 cells and 400 nodes around the airfoil was used, with 2000 time-step per cycle. Figure 6 shows that the results obtained (Cp mean = 0.8685 & η = 33.89%) agree very well with Kinsey and Dumas [5] (Cp mean = 0.860 & η = 33.70%) to within 1%

Results and Discussion

Calculations were made for a range of values for θ0 (50° to 90°), k (0.6 to 1.2) with h0 = 1.0c, Xp = 0.333c at Re = 1100. The predicted efficiency η contours of the non-profiled plate at different θ0 and TSR are given in Figure 7. For the oscillating plate turbine, the TSR is related to the instantaneous plunging velocity Vp(t) by

\[ TSR = \frac{\omega c}{V_p(t)} = \frac{V_p(t)_{\text{rms}}}{U_{\infty}} \]  

Efficiencies greater than 33% can be found for TSR between 0.53 and 0.66 (i.e. k between 0.75 & 0.93) at θ0 = 75° which is similar to what Kinsey and Dumas [5] reported (k between 0.75 & 1.13 at θ0 = 75° to 80°) for their NACA0015 analysis.

\[ \eta = \frac{\text{Mean}}{\text{Kinsey and Dumas [5]}} \times 100 \]  

Calculations were conducted on the observed crests of the performance i.e. TSR 0.57 (k = 0.80) at θ0 = 75° for the non-profiled plate section and profiled plate (NACA0012) with results (Table 2) showing an improvement in performance as high as 5.35% over the NACA0012 plate. Also a sample of some rotary turbines shows high performance when TSR is between 6 & 10; and comparing with Figure 7 where high performance of oscillating turbine is between TSR of 0.53 to 0.66, oscillating turbine appears more efficient at low speed flows and low TSR. To examine the improvement over the NACA0012 plate, the flow field and pressure coefficient at V/T = 0.2, 0.35 and 0.45 were selected for further analysis.

The Cp predicted for the non-profiled plate section was lower than that for NACA0012 between t/T = 0.15 & 0.30 and 0.65 & 0.80 but higher between t/T = 0.30 & 0.45 and 0.80 & 0.95. As shown in Figure 9(a) at V/T = 0.20, the plates are pitching in the counter-clockwise direction, the LEV for the non-profiled plate develops earlier than for the NACA0012 plate. This because the sharp edge at the leading edge of the non-profiled plate causes the flow to separate earlier than that in the NACA0012 plate.

Vortex formation in an oscillating plate creates a suction effect due to low pressure on the occurring surface either on the upper or lower surface during the upward/clockwise or downward/counter-clockwise motion of the plate. A well timed LEV is considered to be important for achieving a better synchronisation between the motions and the forces [5, 7, 8]. The pressure coefficient at the leading edge of the NACA0012 plate in Figure 9(c) is lower than the non-profiled plate thus producing a higher adverse pressure gradient in the y-direction. The adverse pressure gradient plays a crucial role in determining the direction of the y-forces; whether a dynamic stall or lift will be experienced by the body. Thus with the dynamic stall synchronising with the direction of the motions of both plates, there is a reduction in the amount of power required to turn the plate in the counter-clockwise pitching direction in and

\[ V_p(t_{\text{rms}}) = \frac{V_p(t)_{\text{rms}}}{U_{\infty}} \]  

<table>
<thead>
<tr>
<th>Re = 1100, Xp = 0.333c &amp; θ0 = 75°</th>
<th>Peak condition k = 0.80, TSR= 0.57</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cp mean</td>
<td>η %</td>
</tr>
<tr>
<td>Non-profiled plate</td>
<td>0.8728</td>
</tr>
<tr>
<td>NACA0012</td>
<td>0.8285</td>
</tr>
</tbody>
</table>

Table 2. Comparison of Cp mean and η %of NACA0012 and non-profiled plate at k = 0.80 & θ0 = 75°

Figure 8. Comparison of Cp of NACA0012 & non-profiled plate at Xp = 0.333c, h0 = 1.0c, θ0 = 76.33, k = 0.88 & Re = 1100

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in the plunge direction (y direction). Because the pressure gradient is higher at t/T = 0.2 & 0.7 for the NACA0012 plate, the power input demanded to move the plate in the pitch and plunge direction is smaller; resulting to a higher power output and efficiency at this point when compared with the non-profiled plate.

**Effect of Thickness on Performance**

Calculations for the non-profiled plate were made for different thicknesses at k = 0.90 & b = 75°; 0.5b (Cp mean = 0.8492 & η = 33.30%), b (Cp mean = 0.8621 & η = 33.81%) and 2b (Cp mean = 0.8622 & η = 33.81%). These results show that at k = 0.90 & b = 75°, the effect of thickness on performance is insignificant, consistent with the study of Kinsey and Dumas [5] for NACA0015 plate. As shown in Figure 9, the instantaneous vortex contours for the 3 non-profiled plates with different thicknesses (2b, b & 0.5b) indicate a close similarity in the formation and shedding of vortices. However, the 2b thick plate appears to have slightly larger and fewer numbers of dynamic vortices. Conversely more investigations will be carried out to precisely ascertain this view.

![Figure 9](image)

**Conclusion**

A non-profiled plate has been shown by 2D CFD simulations at Re = 1100 to be 5.35% better in performance for power extraction than its profiled counterpart with a peak Cp of 0.8729 and η = 34.23% at k = 0.80 & θ = 75°. Similar to profiled plates, preliminary results show that thickness has a weak effect on the performance of non-profiled plates. These results demonstrate the potential of employing non-profiled plates for flapping wing turbines because of lower manufacturing costs than profiled plates while enhancing the efficiency and extracted power.

As oscillating plate turbines are more efficient than rotary turbines at low speed flows and small TSR, they are viable alternatives for energy extraction under these conditions.

**References**