The effect of rectangular Gurney flap on Wells turbine performance

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Abstract. Wells turbine is very adept at converting pneumatic power from ocean waves into mechanical energy. However, such turbines incur from low performance, limited operating range and low efficiency. The present study introduces a method to enhance the power produced from Wells turbine by implementing rectangular Gurney flap (GF). The Gurney flap is placed on both pressure and suction sides of the trailing edge (TE) and perpendicular to the chord line without change chord length. GF increases the lift coefficient by altering the Kutta condition at the TE. The turbine performance is herein evaluated by solving numerically 3D incompressible Reynolds Averaged Navier–Stokes equations (RANS) by using ANSYS Fluent 19.0. The performance is demonstrated according to the flow coefficient, torque coefficient, total pressure loss coefficient, and the turbine efficiency, as well as the velocity and pressure fields around the turbine blades. The validation of the present work was achieved using previous experimental work and CFD work by using SST k – ω turbulence model. The present work showed that the Wells turbine augmented with GF increases the torque coefficient by 41.98%. Besides, the stall has been delayed compared with the conventional Wells turbine.

1. Introduction
Wells turbine is utilized to convert ocean wave energy into a mechanical power. It is environmentally device friendly so researchers around the world are investigating on how to boost its performance. Different designs were experimentally studied by Gato et al. [1] and Curran and Gato [2] to investigate the effect of theirs designs on performance of turbine. Kim et al. [3] made a numerical simulation to study the blade sweep and its effect on the turbine performance. Setoguchi et al. [4,5] introduced influence of self-rectifying turbines and utilizing of inlet guide vanes on design parameters of turbine. Brito-Melo et al. [6] studied numerically different turbine design parameters. Dhanasekaran and Govardhan [7] introduced a numerical study for turbine performance. The effect of inlet conditions on the Wells turbine performance was introduced by Raghunathan et al. [8]. The previous studies confirmed that geometrical parameters such as blade geometry [9], blade setting angle [10], the use of end plates [11] and tip clearance [12] affect Wells turbine performance.

The previous studies showed that the optimum blade profile was found to be NACA0020 for small-scale Wells turbine [13], while the optimum blade profile was NACA0015 for large-scale turbines [14]. The effect of guide vanes on the constant chord Wells turbine was showed by Thakker et al. [15]. Besides, Setoguchi and Takao [16] introduced its effect on the variable chord. A study by [14,17] had
been demonstrated that Wells turbine with guide vanes stalled before those without guide vanes. The effect of tip leakage flow on the Wells turbine performance was showed by Taha et al. [18,19] with uniform tip clearance [18] and also with non-uniform tip clearance [19]. Both studies showed that tip leakage flow effect on the performance of wells turbine and turbine stalling. The influence of geometry of turbine duct on its performance was numerically studied by Shaaban and Hafiz [20]. Halder and Samad [21] showed that the casing treatment influence the vortex pattern which result in stall limit change.

The effect of a number of rotor blades on the wells turbine performance had been studied through many researchers. Some of the previous studies recommended the use of eight rotor blades [1,2,12,22] and others recommended the use of six rotor blades [22,13]. On the other side, using air turbines to be immersed in water gave a promising performance. Hamed et al. [23] proved that operating wells turbine as a hydraulic turbine achieves better performance. The circumferential casing groove (rectangular shape) effects on the turbine performance at different groove depth and width were demonstrated by Halder et al [24]. Moreover, two different cases, with and without a tip groove, were considered to guess the optimal turbine speed for the different flow velocities [25].

The design with grooved-casing performs better than that of the without grooved casing. Using variable blade sweep angles at the tip and mid sections enhanced Wells turbine performance and provided higher range of operating [26, 27]. Torres FR, et al [28] demonstrated a methodology to determine the optimal size of a Wells turbine taking into account hydro-aerodynamic model. Moreover, Nazeryan M, et al [29] investigated numerically the influence of blade thickness on both aerodynamic and entropy generation analysis of a Wells turbine. Besides, Gratton T, et al [30] performed for optimizing the blade profile in the Wells turbine for better performance, by the torque coefficient magnification. Kumar and Samad [31] showed the effect of rectangular flap and flap height on wells turbine performance.

In accordance to previous researches, still much effort is needed to boost Wells turbine performance that incurs from low performance, limited operating range and low efficiency. Accordingly, the goal of the present paper is to overcome these disadvantages. The present study introduces a method to enhance the power produced from Wells turbine by implementing different geometries of Gurney flap (GF). The Gurney flap is placed on both pressure and suction sides of the trailing edge (TE) and perpendicular to the chord line without change chord length.

Wells turbine is investigated while having symmetrical NACA 0015 blades. The turbine performance is here in evaluated by solving numerically 3D incompressible Reynolds Averaged Navier–Stokes equations (RANS) by using ANSYS Fluent 19.0. The performance is demonstrated according to the flow coefficient, torque coefficient, total pressure loss coefficient, and the turbine efficiency, as well as the velocity and pressure fields around the turbine blades. The choice of the number of blades used is rationalized from previous studies.

2. Computational methodology
In this section, Wells turbine theory of operation, its gurney flap and its computational methodology are showed.

2.1. Wells turbine
Wells turbine is one of the most important Ocean-wave energyconverters that convert Ocean-wave energy into mechanical power. Actually, in oscillation water column (OWC) wave energy converters, pressure fluctuations are generated due to motion of wave, inside a plenum, and therefore producing an oscillating air-flow, so the self-rectifying Wells turbine can efficiently convert whose energy into a unidirectional rotational motion; see figure 1a. In its basic arrangement, this is an axial-flow air turbine, consisting of a rotor with a number of symmetrical and untwisted airfoil blades, usually coding to the NACA00XX series, usually chord line is perpendicular to rotational axis (90°stagger angle); see figure 1b, [12].
2.2. Gurney flap scheme

The present study introduces a method to enhance the power produced from Wells turbine by implementing different geometries of Gurney flap (GF). The Gurney flap is placed on both pressure and suction sides of the trailing edge (TE) and perpendicular to the chord line without change chord length. In the present study, a rectangular Gurney flap (GF) is studied, see figure 2. In this paper, a rectangular Gurney flap (GF) height was varied from 0.5-2% C and the width from 0.4-0.6% C.

2.3. Computational equations

In figure 3, the flow having an angle of incidence (α) will produce a lift force (L) and a drag force (D). The components of these forces in both tangential and axial directions are denoted by $F_t$ and $F_a$, respectively. Accordingly, $F_t$ generates torque on the turbine rotor, while $F_a$ produces axial stress on the rotor as a result of thrust force on the turbine rotor axis [7]. Accordingly, for Wells turbine in an oscillating airflow, the angle of incidence (α) changes from positive to negative values in accordance with the upward and downward directions of flow, respectively; see figure 1. The tangential force $F_t$ is acting in the same direction regardless of the positive or negative value of an angle of incidence (α) because of symmetry of the rotor blade.
In accordance with, torque has a unidirectional feature that is created from an oscillating flow [7]. The flow coefficient, φ, can be introduced as:

$$\phi = \frac{C_x}{U}$$

The flow coefficient is changed in accordance with the variation of the axial flow velocity $C_x$ while keeping the rotor tip speed $U$ constant. Thus, the torque coefficient $C_T$ can be introduced as:

$$C_T = \frac{T}{\rho \omega^2 R_T^5}$$

where $T$ the torque on rotor is, $\rho$ is the working fluid density, $\omega$ is the angular velocity of turbine and $R_T$ is the radius of rotor tip. Total pressure drop coefficient $C_P$ is expressed by the following formula:

$$C_P = \frac{\Delta P_o}{\rho \omega^2 R_T^2}$$

Where $\Delta P_o$ is the total pressure drop.
The Turbine efficiency ($\eta$) is introduced as:

$$\eta = \frac{T \omega}{\Delta P_o Q}$$

where $Q$ is the volume flow rate through the turbine.

Rotor solidity is an important parameter that affects the performance of Wells turbines. The rotor solidity $\sigma$ is defined as:

$$\sigma = \frac{Z \cdot c \cdot H}{\pi \cdot (R_T^2 - R_h^2)} = \frac{Zc}{2\pi R_m}$$

where $Z$ is the number of blades, $c$ is the chord length; $H$ is the blade height, $R_T$ is the radius of rotor tip, $R_h$ is the radius of rotor hub and $R_m$ is the turbine mean radius. Eq. (5), shows that the rotor solidity $\sigma$ can be varied by change the number of turbine blades.

In the present study, the influence of using different geometries of Gurney flap (GF) on performance of Wells turbine is numerically investigated by solving the 3D incompressible Reynolds-averaged Navier-Stokes equations RANS for wells turbine with Gurney flap (GF). The basic purpose of using any turbulence model is to bring results closer to the experimental results. In this study, the SST $k-\omega$ Turbulence model is used. In addition, Non-dimensional wall distance $y^+ < 1$ to guess flow pattern on buffer and sub-layer zones. The good prediction of the aerodynamic performance was achieved by the combination of the SST $k-\omega$ Turbulence model with non-dimensional wall distance $y^+ < 1$. Besides, the Moving Reference Frame method (MRF) is applied to simulate the
turbine rotor, but the main governing equations are solved in the absolute frame. The finite volume method is applied to appreciate the absolute frame using the computational fluid dynamics (CFD) code “Fluent” V19.0. The present simulation uses different discretization methods. These methods are shown in the following table; see Table 1:

| Solution methods                      | SIMPLEC               |
|---------------------------------------|-----------------------|
| Pressure-velocity coupling            | SIMPLEC               |
| Gradient                              | Least squares cell based |
| Pressure                              | Standard              |
| Momentum                              | The second order upwind |
| Turbulent kinetic energy(k)           | The second order upwind |
| Specific dissipation rate(ω)          | The second order upwind |

The SIMPLEC algorithm applied for pressure-velocity coupling eliminates the influence of mesh skewness on the final solution. The maximum residual values for momentum, continuity, velocity x, velocity y, velocity z and turbulence equations of the solutions are $10^{-4}$. Moreover, the valve of turbulence intensity at inlet and outlet is 5% (The turbulence intensity is defined as the ratio of the root-mean-square of the velocity fluctuations to the mean flow velocity).

3. Results and discussions
In this part four parts are presented and they are a mesh size independence study, model validation, effect of different geometries of Gurney flap on turbine performance, and the field of flow around the blades.

3.1. Mesh independent study
Curran and Gato [2] wells turbine dimensions are used in this investigation. The specifications of the Wells turbine that is under study are shown in Table 1. The full cylindrical domain of eight-blade rotor is reduced to slice of domain angle $\theta = 45^\circ$ of the full domain, see Figure 4. This minimized was made on account of the rotor of turbine symmetry around the axis of rotational and as a result, the computational time is reduced [20]. Besides, Table 2 presents domain meshing and domain boundary conditions. Figure 4 and Figure 5 describe the domain of computational with boundary conditions and the mesh of computational, respectively.

The large cylindrical domain extended 4C and 8C upstream and downstream the rotor, respectively [20] to confirm that the boundaries don’t influence on the flow at the turbine, Here, refinement grids with inflation are used near the blade surface to accurately capture the flow field inside the boundary layer. The grid is made up of unstructured tetrahedral mesh. The total number of cells in the computational domain is 2,614,335. The inlet and outlet conditions of computational domain are uniform velocity and outlet pressure, respectively. For the blades, hub and casing surfaces no slip wall conditions are used. Flow coefficient $\phi$ is changed from 0.1 to 0.325 and the fixed rotational speed of turbine (2000 rpm). Henceforth, the blade tip speed was kept constant.

Figure 6 shows the influence of number of cells in computational domain on the CFD results of turbine efficiency $\eta$ at flow coefficient $\phi = 0.15$. The number of cells was gradually increased in five steps from 1,015,551 cells to 3,953,000 cells, with constant non-dimensional wall distance $y^+ < 1$ at the same $\phi$. This study showed that the maximum percentage of the diffractions of torque coefficient $C_T$, total pressure loss coefficient $C_p$, and efficiency $\eta$ are ±0.99%, ±0.66% and ±0.33%, respectively. Cells number of 2,614,335 is the best solution quality, where CFD results are independent of the number of cells in computational domain. Thus, this meshes number was utilized during the present study.
Figure 4. The computational domain with boundary conditions.

Figure 5. Unstructured/tetrahedral computational mesh.
Table 2. Specifications of turbine model.

| Parameter                  | Dimension               |
|----------------------------|-------------------------|
| Blade Profile              | NACA0015                |
| Number of blades, Z        | 8                       |
| Domain angle, $\theta$     | $45^\circ$              |
| Blade chord length, $c$    | 0.125 m                 |
| Blade thickness, $t$       | 0.01875 m (15% of $c$)  |
| Hub diameter, $D_t$        | 0.4 m                   |
| Casing diameter, $D_c$     | 0.59 m                  |
| Tip clearance              | 0.001 m                 |
| Blade height, $H$          | 0.094 m                 |
| Solidity, $\sigma$         | 0.644                   |
| Stagger angle              | $90^\circ$              |
| Turbine rotational speed, N| 2000 rpm                |

Table 3. Meshing and boundary conditions.

| Parameter                   | Description               |
|-----------------------------|---------------------------|
| Flow domain                 | Single blade              |
| Interface                   | Periodic                  |
| Mesh/Nature                 | Unstructured/tetrahedral   |
| Elements                    | 2,614,335                 |
| Fluid                       | Air                       |
| Turbulence model            | SST $k - \omega$          |
| Inlet                       | Velocity                  |
| Outlet                      | Pressure                  |
| Hub                         | Wall                      |
| Casing                      | Wall                      |
| Blade                       | Wall                      |
| Maximum Residual criteria   | $1 \times 10^{-4}$        |

Figure 6. Influence of cells number on turbine efficiency.
3.2. Model validation

A validation curve of torque coefficient $C_T$, total pressure loss coefficient $C_p$, and efficiency $\eta$ has been determined at different values of flow coefficient $\phi$ as shown in figures 7a, 7b, and 7c, respectively. The validation results were compared with both the experimental results [2] and CFD results [12, 21, 23, and 31]. The results of the present work show good agreement with the results of Curran and Gato [2]. Accordingly, the present CFD model has been great admitted as a computational tool for the performance of Wells turbine.

![Figure 7](image1)

**Figure 7.** Validation with experimental and CFD results for:
(a) $C_T$, (b) $C_p$, and (c) $\eta$.

Figure 8 presents the error percentage of the present work compared with the experimental results of Curran and Gato [2]. As it is depicted from the figure that the maximum error percentage for torque coefficient, the total pressure drop coefficient, and efficiency are 9.9%, 7.3%, and 11.7%, respectively. It is notably to mention that the error percentage concerned the efficiency is dependent upon the errors of the torque and pressure drop coefficients.
Validation of the rectangular flap at FW = 0.5 %C & FH = 1.5 %C is also achieved with Kumar and Samad [31] for more satisfaction. Figure 9 presents that the present CFD results have a good agreement with Kumar and Samad [31]. However, Kumar and Samad [31] studied only the effect of varying the rectangular flap height (FH) on the Wells turbine performance. So, in this paper, the effect of varying flap height (FH) and flap width (FW) on the Wells turbine performance has been considered.

Figure 8. Error percentage with experimental results.

Figure 9. Verification of the present rectangular flap results. For:
(a) CT, (b) Cp, and (c) η.
3.3. Effect of rectangular flap height

Figure 10 demonstrates the effect of rectangular flap height on the variations of $C_T$, $C_p$, and $\eta$ at different values of $\phi$. The results show that the rectangular flap of $FW = 0.5\%C$ & $FH = 1.5\%C$ gives a maximum value of $C_T = 0.148$ at $\phi = 0.225$. While, the maximum $C_T = 0.113$ at $\Phi = 0.22$ for blade without flap ($FH = 0 \%C$). Hence, the percentage increase in $C_T$ is 41.98% at $\phi = 0.225$, see figure 10(a). On the other hand, $C_p$ is increased as the flap height increases especially at the values of $\phi > 0.2$ as shown in figure 10(b). Despite the decrease in the efficiency compared with the blade without flap (reference case), the amount of power generated had been enhanced, figure 10(c).

![Figure 10](image)

(a)  
(b)  
(c)  

Figure 10. Effect of rectangular flap height (FH) on: (a) $C_T$, (b) $C_p$, and (c) $\eta$.

The velocity streamlines are presented for Wells turbine blade without flap ($FH = 0 \%C$) and with flaps of different heights as shown in figure 11. The comparison of velocity streamlines between blades with flap is established at $FW = 0.5\%C$, $H = 50\%$, $\phi = 0.2$ and $FH = (0\%, 0.5\%, 1\%, 1.5\%, \text{and} 2\%)$. It is noticed that, at the blade without flap ($FH = 0 \%C$), a vortex is formed at the end of the trailing edge. This vortex separates the flow from the surface of the blade, which reduces the power generated, see figure 11 (a). The flap position at the end of the blades generates to vortices regions; a forward and backward. The forward vortices size is very small compared with the backward vortices, which diminish their effect. Hence, the backward vortices, that is located downstream the blade, made a low-pressure zone. This reduces the chance of the boundary layer separation and give the opportunity to entrain more streamlines that in turn enhance the power generated. Accordingly, the size and location of
the vortices have a very vital effect on the performance of the blades. The effect of flap of FH = 0.5 and 1 is very small compared with the flap of FH = 1.5 and 2 as shown in figures 11 (b, c, d, and e) respectively.

3.4. Effect of rectangular flap width

Figure 12 illustrate the effect of rectangular flap width on the variations of $C_T$, $C_P$, and $\eta$ at different values of $\varphi$. The rectangular flap of FW = 0.5 %C gives a maximum value of $C_T$ at $\varphi = 0.225$ compared with the other geometries as shown in figure 12(a). On the other hand, the rectangular flap of width

![Figure 11. Velocity streamlines of rectangular flaps of different heights at FW=0.5%C, H=50% and $\Phi=0.2$.](image-url)
(FW = 0.4, 0.5, and 0.6 %C) approximately the same values of Cₚ through the range of φ as shown in figure 12(b). Despite the decrease in the efficiency compared with the blade without flap (FH = 0%), the amount of power generated had been enhanced, figure 12(c). Therefore, the optimum rectangular flap dimensions are FH = 1.5 %C and FW = 0.5 %C.

(FW = 0.4, 0.5, and 0.6 %C) approximately the same values of Cₚ through the range of φ as shown in figure 12(b). Despite the decrease in the efficiency compared with the blade without flap (FH = 0%), the amount of power generated had been enhanced, figure 12(c). Therefore, the optimum rectangular flap dimensions are FH = 1.5 %C and FW = 0.5 %C.

The velocity streamlines are presented for Wells turbine blade without flap (FH = 0 %C) and with flaps of different widths as shown in Figure 13. The comparison of velocity streamlines between blades with flap is accomplished at FH = 1.5 %C, H = 50 %, φ = 0.2 and FW = (0%, 0.4%, 0.5%, and 0.6%). The backward vortices that is formed at FW = 0.5%C is very dense compared to others, see figure 13 (C).
4. Conclusions

The aim of the present work is enhancing the Wells turbine performance. Rectangular Gurney flap has been elaborated herein. The turbine performance is investigated under steady unidirectional flow conditions using SST k − ω model where its numerical results were validated. The model is based on 3D incompressible RANS equation in the single rotating reference frame. Accordingly, the following conclusions are taken:

1. The Gurney flap generates two types of vortices; forward and backward vortices. The forward vortices size is very small compared with the backward vortices that diminish their effect.
2. The blade without flap achieves maximum torque coefficient equals to 0.113 at Φ = 0.22. Furthermore, the stall inception appears at flow coefficient equals to 0.22.
3. Compared with the blade without flap, the blade with rectangular flap (FH = 1.5 %C and FW = 0.5 %C) achieves maximum torque coefficient equals to 0.148. Moreover, the stall point is delayed to flow coefficient equals to 0.225. Furthermore, the maximum percentage of increasing in the torque coefficient is 41.98% at such flow coefficient.
4. The Gurney flap width (FW) does not affect on total pressure drop coefficient through the whole range of operation for rectangle flap.

![Figure 13](image_url). Velocity streamlines of rectangular flaps of different width at FH=1.5%C, H=50% and Φ=0.2.
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NOMENCLATURE

Abbreviations

| Symbol | Definition |
|--------|------------|
| LE     | Leading edge |
| GF     | Gurney flap |
| FW     | Flap width |
| SS     | Suction side |
| OWC    | Oscillating water column |
| MRF    | Moving Reference Frame |
| RANS   | Reynolds-averaged Navier-Stokes |
| SST    | Shear stress transport |
| SIMPLEC| Semi-Implicit Method for Pressure-Linked Equations-Consistent |

Symbols

| Symbol | Definition |
|--------|------------|
| C      | Chord length (m) |
| H      | Blade height (m) |
| y*     | Non-dimensional wall distance(–) |
| F_t    | Tangential force (N) |
| F_a    | Axial force (N) |
| T      | Torque (Nm) |
| Q      | Volume flow rate (m³/s) |
| U      | Rotor tip speed (m/s) |
| C_c    | Axial velocity (m/s) |
| W      | Relative velocity (m/s) |
| Δp_o   | Total pressure drop (Pa) |
| C_T    | Torque coefficient (–) |
| C_P    | Total pressure drop coefficient (–) |
| Ø      | Flow coefficient (–) |
| η      | Efficiency (–) |
| α      | Incidence angle (°) |
| L      | Lift force (N) |
| *      | Non-dimensional parameter |