High-performance ocean energy harvesting turbine design – A strategy of compound leaning

B. Ranjith1, P. Madhan Kumar1, Manabu Takao2, Abdus Samad1*

1Wave Energy and Fluids Engineering Lab, Department of Ocean Engineering, IIT Madras Chennai-600036, India
2Department of Mechanical Engineering, National Institute of Technology, Matsue College, Matsue 690-8518, Japan

*samad@iitm.ac.in

Abstract. Blade leaning works well for steam, hydro and gas turbines, where the flow is unidirectional. In contrary, the turbine used in an oscillating water column wave energy device has a unique design which can work only in bidirectional flow and gives a unidirectional torque. In this article, a compound leaning concept is explored through numerical modelling. An impulse turbine with 0.3 m diameter and two rows of guide vanes mirroring each other was considered as a case study. Reynolds-averaged Navier-Stokes equation with k-ε turbulence closure model was solved in ANSYS-CFX code. The present numerical result matches well with the existing experimental results. The modified rotor blade gave 8.8% higher efficiency as it reduced the intensity of trailing edge vortex shedding and the downstream recirculation region.

1. Introduction

In turbomachinery, most of the researches were done to get better performance by changing the geometrical parameters. The application area such as gas turbine, steam turbine or hydro turbine has already implemented the concept of leaning [1-4]. Leaning is the bending of a blade perpendicular to the chord line [3]. Lean towards pressure surface is called positive lean, while the opposite one is called negative lean.

On the other hand, the bidirectional flow turbine which is a specially designed turbine used in the wave energy system is relatively newer and very few researchers are working on it. Commonly used bidirectional turbines in this system are Wells turbine and impulse turbine. Compared to the Wells turbine, the impulse turbine operates under wider wave conditions. Among different types of impulse turbines, fixed guide vanes have simple construction, which gives low maintenance and high robustness because of only one moving component rotor [5]. The maximum efficiency with a turbine of 0.3m having fixed guide vane (GV) turbine goes up to ~40% [5]. The performance of this turbine depends on various geometric parameters of the rotor blade and guide vanes. Ample research has been done on parameters modifications to increase performance (Table.1). Among those parameters, the lean is an effective parameter to reduce incidence losses [1,3-4]. The rotor lean concept was studied by Kim et al. [16] and found that leaning blade on running direction increases the performance of self-pitched guide vane type turbine. Ranjith et al. [15] studied that the optimum lean angle is 10° for 0.3m diameter turbine. However, the study on other turbines shows that compound leaning of the rotor blade in certain case is better than simple rotor lean. Effect of compound leaning of rotor blades termed as bowing by some
authors and helps to reduce 3D losses. This effect was studied on other types of axial turbines [1-4], but not for the bidirectional turbines.

Table 1. Design modifications performed on impulse turbine

| Author and Dhanasekaran [6] | Design modification | Effect on performance |
|-----------------------------|---------------------|-----------------------|
| Tip clearance               | Tip clearance of 1% performs almost similar to 0% up to flow coefficient = 1.0. |

| Hyun and Moon [7]           | End plates at rotor blade (RB) tip | Efficiency increased from 3 to 5% is noticed. |
| Thakker and Hourigan [8]    | Hub to tip ratio                   | Optimum value is 0.6 and 0.7 for Reynolds number 5.0E4 and 4.0E4, respectively. |
| Thakker [9]                 | 3D GV                             | Efficiency improved by 4.5%. |
| Gomes et al. [10]           | Multi-point aerodynamic optimization | Optimized rotor showed enhanced efficiency by 4.4%. |
| Ying and Liu [11]           | Solidity ratio                    | Optimum solidity ratio = 0.63. |
| Badhurshah and Samad [12]  | Optimized the number of RB and GV | Increase in relative average efficiency up to 13%. |
| Badhurshah and Samad [13]  | Optimized the number of RB and GV with operating characteristics | Optimized rotor improved efficiency by 16%. |
| Liu et al. [14]             | Blade setting angle               | Optimum blade setting angle 5° enhanced mean efficiency by 9.2%. |
| Ranjith et al. [16]         | RB and GV lean                    | Optimum RB and GV lean angle is 10°. Efficiency increased by 3 to 4% for RB lean and 4 to 5 % for GV lean. |

The objective of the paper is to find the effect of compound lean on rotor blades on a bidirectional flow impulse turbine and finding optimum rotor design. The blade was leaned along the spanwise direction, various configurations of the leaned blade are numerically evaluated, and their performance was compared with the reference blade. The fluid dynamics behind the performance improvement is explained.

2. Computational fluid dynamic simulation approach
The computational fluid dynamics (CFD) simulations follow basic approach such as the generation of geometry, meshing, setting initial and boundary conditions, solving CFD equations. Finally, the obtained data is analyzed to form information that can be used by other researchers. For this study, an impulse turbine of diameter 0.3m provided in the study of Maeda et al. [5] was chosen as the reference geometry (Figure 1). The reference geometry specifications are given in Table 2. A CAD model is prepared for the flow domain. A single blade with two rows of upstream and downstream GVs was taken as flow domain to exploit the periodic symmetry of impulse turbine thereby reducing the computational time. The upstream and downstream length of flow domain was fixed as eight times of their chord length, and the GVs domain was stationary.
2.1 Governing equations
The governing equations of the steady, incompressible flow are provided in Equation 1 and 2.

Continuity equation
\[
\frac{\partial \bar{u}_i}{\partial x_i} = 0
\]  
(1)

Reynolds averaged momentum equation
\[
\rho \left[ \frac{\partial \bar{u}_i}{\partial t} + \frac{\partial (\bar{u}_j \bar{u}_i)}{\partial x_j} \right] = -\frac{\partial \rho}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ (\mu + \mu_t) \left( \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \right] 
\]  
(2)

k and \( \varepsilon \) equations
\[
\rho \left[ \frac{\partial (k)}{\partial t} + \frac{\partial (\bar{u}_j k)}{\partial x_j} \right] = \frac{\partial}{\partial x_j} \left[ (\mu + \mu_t) \frac{\partial k}{\partial x_j} \right] + TP - \varepsilon 
\]  
(3)
\[
\rho = \frac{\partial}{\partial x_j} \left[ (\mu + \mu_t) \frac{\partial \varepsilon}{\partial x_j} \right] + \frac{\varepsilon}{k} (c_{\varepsilon 1} TP - c_{\varepsilon 2} \rho \varepsilon) 
\]  
(4)

2.2 Meshing and Turbulence model
The flow domain was discretized with structured hexahedral elements using ANSYS-TurboGrid (Figure 2). The meshing process was automated, and the global size factor was used to control the grid refinement. The meshing of tip clearance region was performed by specifying the value of tip clearance to generate structured elements along the tip gap. The performance of turbine was computed by solving Reynolds averaged Navier Stokes equations with \( k-\varepsilon \) as turbulence closure. Previous researchers [9,11,14] found that \( k-\varepsilon \) predicts the performance better for this type of impulse turbine; hence the \( k-\varepsilon \) turbulence model was adopted in this study. For current simulation, combination of velocity inlet and static pressure outlet boundary condition was used for the most robust result as per CFX theory guide [17]. The current simulation technique can be extended to other axial turbines with the limitation turbulence model. \( k-\varepsilon \) which suits well for current case may not predict well for other turbines with airfoil type RBs (e.g. Wells turbine); \( k-\omega \) SST is proven to be a better model for such turbines.

Periodic boundary condition was used for the entire flow domain (Figure 2) to reduce the computational cost and time. No slip wall is applied at the hub, blade, and shroud. The counter-rotating wall option was
adopted for shroud. Velocity inlet and pressure outlet were employed for maximum robustness as per the CFX solver theory guide. A wide range of flow coefficients was achieved by varying inlet velocity from 4 to 20 m/s with constant rpm (as 600). 5% turbulence intensity was used at the inlet. The residual convergence value was set at 10E-05, and double precision was used for all simulations.

### Table 2. Reference geometry specifications [5].

| Specification            | Value  |
|--------------------------|--------|
| RB chord length          | 54 mm  |
| Number of RB             | 30     |
| Rotor profile            | Elliptic-circular |
| Rotator blade inlet angle| 60°    |
| RB max thickness         | 16.1 mm|
| Number of GVs            | 26     |
| GV profile               | Plate type |
| GV chord length          | 70 mm  |
| GV blade thickness       | 0.5 mm |
| GV setting angle         | 30°    |
| Hub diameter             | 210 mm |
| Tip diameter             | 298 mm |
| RB tip clearance         | 1 mm   |
| Rotor stator axial distance (G) | 20 mm |

**Figure 2.** Meshed computational domain

### 3. Compound leaning

To study the effect of compound leaning, the rotor blade is shown with three different sections. The bottom cross-sectional plane which is attached to the hub, the top cross-sectional plane which is in tip and midplane in between hub and tip (shown in Figure 3). Mid and tip cross-section of the blade was moved from a mean position (Figure 3b) to achieve compound leaned design. The mid and tip section moved to a distance of $\Psi_1$ and as $\Psi_2$ from the centreline. Ten variants of compound leaning obtained by different combinations of $\Psi_1$ and as $\Psi_2$.

**Figure 3.** Composite leaned blades
4. Results and discussion

4.1 Grid sensitivity and numerical validation
Grid sensitivity test to choose the optimum number of mesh elements and different grid resolutions is done by modifying the global mesh size parameter in Ansys-Turbogrid. The grid convergence was achieved faster with 0.5 million elements in structured mesh compared to 3 million elements in the case of unstructured meshing [15]. The performance parameters of the reference turbine were evaluated with three different grid resolution such as coarse (499,788), medium (784,458) and fine (984,144) to obtain the optimum grid size. Figure 4 shows that there is no significant difference in performance parameters for the different resolution of grids. Hence, the grid with 499,788 elements was chosen.

Figure 4. Grid sensitivity study

(a) Input coefficient
(b) Torque coefficient

Figure 5. Validation with experimental results

Figure 5 shows the numerical validation, and it can be noticed that the numerical result obtained follows the same trend with experimental result for the performance parameters such as input coefficient and torque coefficient. There is a slight mismatch between experimental and numerical validation because of the non-
inclusion of some experimental effects in the numerical simulation such as mechanical losses in friction etc. It confirms that the governing equations, discretization technique, boundary conditions used for simulation are near to experimental condition.

4.2 Effect of compound blade leaning
Ten different designs were obtained from the reference by changing two parameters $\Psi_1$ and $\Psi_2$ as shown in Figure 3. Different designs with a combination of $\Psi_1$ and $\Psi_2$ is given in Table 3. For all designs, the CAD model was generated, and meshing was done with approximately the same number of elements obtained by grid independence study. After meshing, the flow was simulated for peak flow coefficient as per reference model.

The results attained are compared with the reference geometry results (1 in Table.3). The design 8 gives 8.8% increase in efficiency by reducing the pressure loss, which is measured using a dimensionless parameter called input coefficient for almost same torque output. The optimum design in Table.3 performs better at peak flow coefficient (where the efficiency is maximum). Hence design 8 (optimum design) is chosen as the best combination of $\Psi_1$ and $\Psi_2$, and its performance is checked for a wider flow coefficient (Figure 6b). It performs better for flow coefficients from 0.75 to 2.0. The increase in efficiency is due to the decrease in input coefficient (Figure 6a), which is a function of pressure drop across the turbine. This indicates that pressure energy loss was reduced across the turbine by compound leaning while the shaft power remained constant (Figure 6c).

Table 3. Performance of different designs at peak flow coefficient ($\phi$=1.25).

| SI No. | $\Psi_1$ (mm) | $\Psi_2$ (mm) | $\eta$ | $Ct$ | $Ca$ | $\Delta\eta$ (%) | $\Delta\text{C}_t$ (%) | $\Delta\text{C}_a$ (%) |
|--------|---------------|---------------|-------|-----|-----|----------------|----------------|----------------|
| 0      | 0             | 0             | 0.419 | 1.721 | 3.271 | 0               | 0               | 0               |
| 1      | -2.5          | 2.5           | 0.430 | 1.717 | 3.185 | 2.6             | -0.2            | -2.6            |
| 2      | -2.5          | -5            | 0.414 | 1.693 | 3.268 | -1.1            | -1.6            | -0.9            |
| 3      | 5             | 7.5           | 0.424 | 1.752 | 3.301 | 1.2             | 1.8             | 0.9             |
| 4      | -5            | 7.5           | 0.446 | 1.728 | 3.094 | 6.4             | 0.4             | -6.4            |
| 5      | -5            | -7.5          | 0.421 | 1.687 | 3.196 | 0.4             | -1.9            | -2.2            |
| 6      | 0             | -6            | 0.402 | 1.667 | 3.311 | -4.0            | -3.1            | 1.2             |
| 7      | 5             | 0             | 0.436 | 1.706 | 3.113 | 4.0             | -0.8            | -4.8            |
| 8      | -5            | 10            | 0.456 | 1.721 | 3.013 | 8.8             | 0.0             | -8.8            |
| 9      | -7.5          | 10            | 0.452 | 1.724 | 2.999 | 7.8             | 0.1             | -7.8            |
| 10     | -10           | 12.5          | 0.440 | 1.680 | 3.051 | 5.0             | -2.3            | -6.7            |

4.3 Flow field analyses
The major pressure loss occurs in downstream GV, and considerable pressure loss occurs in rotor domain due to the diverging path in second half symmetry of RB path as well as downstream GV. Loss across upstream GV is negligible due to the accelerating flow domain. Figure 7 shows that compound leaning reduced the formation of trailing edge vortex shedding, which arise due to interaction of tip leakage across tip clearance provided between pressure side (PS) and suction side (SS) and end wall casing boundary layer formation which contributes to significant losses as analyzed in the last study [15].
The tip leakage vortex mentioned above enters the downstream guide vane where it gets diffused again and grows in size. In compound leaned design, the downstream recirculation region also get suppressed and slightly pushed towards the hub (Figure 8). These recirculation regions absorb energy from the fluid and result in low efficiency of the turbine. By suppressing this recirculation, the losses can be reduced to a greater extent.

Figure 6. Performance comparison of reference and optimum design

(a) Input coefficient

(b) Torque coefficient

(c) Efficiency

Figure 7. Reduction in trailing edge vortex shedding after compound leaning (at peak flow coefficient)

Figure 8. Reduction in the recirculation region in the downstream guide vane (at peak flow coefficient)
5. Conclusions
This paper investigates the effect of compound leaning on the rotor blade of 0.3m impulse turbine by solving the Reynolds-averaged Navier-Stokes equation with $k$-$\varepsilon$ turbulence closure model including grid sensitivity study and experimental validation. The important conclusions drawn from this study are listed below.

- The compound leaning process significantly affected the aerodynamic performance of the impulse turbine.
- The optimum design of compound leaning enhanced the relative efficiency by 8.8%, whereas the torque exerted on the rotor blade remains constant for a wide range of flow coefficients.
- The compound leaning increased the performance of the turbine by reducing various losses occurs across the turbine such as trailing edge vortex shedding.
- Additionally, the compound leaning suppresses the downstream guide vane recirculation and enhances efficiency.

References
1. Wanjin H, Zhongqi W, Chunqing T, Hong S and Mochun Z 1994 Journal of Turbomachinery, vol 116 pp 417-424
2. Deshpande S, Thern M and Genrup M 2015 In ASME 2015 Gas Turbine India Conference, Hyderabad, India
3. Karrabi H and Rezasoltani M 2011 In ASME International Mechanical Engineering Congress and Exposition, Denver, Colorado, USA pp 965-972
4. Asgarshamsi A, Benisi AH, Assempour A and Pourfarzaneh H 2015 Proceedings of the Institution of Mechanical Engineers, Part G: Journal of Aerospace Engineering, vol 229 (5) pp 906–916
5. Maeda H, Santhakumar S, Setoguchi T, Takao M, Kinoue Y and Kaneko K 1999 Renewable Energy vol 17 (4) pp 533–547
6. Thakker A and Dhanasekaran TS 2004 Renewable Energy vol 29 (4) pp 529–547
7. Hyun BS, Moon JS, Hong SW and Lee YY 2004 In Fourteenth International Offshore and Polar Engineering Conference, Toulon, France pp 253–259
8. Thakker A, Hourigan F, Dhanasekaran TS, Henry ME, Usmani Z and Ryan J 2005 International Journal of Energy Research vol 29 (1) pp 13–36
9. Thakker A and Dhanasekaran TS 2005 International Journal of Energy Research vol 29 (13) pp 1245–1260.
10. Gomes RPF, Henriques JCC, Gato LMC and Falcão AFO 2012 Energy vol 45 (1) pp 570-580
11. Cui Y and Liu Z 2015 Advances in Mechanical Engineering vol 7 (1) pp 1-10
12. Badhurshah R and Samad A 2014 International Journal of Rotating Machinery
13. Badhurshah R and Samad A 2015 Renewable Energy vol 74 pp 749-760
14. Liu Z, Cui Y, Kim KW and Shi HD 2016 Numerical study on a modified impulse turbine for OWC wave energy conversion Ocean Engineering vol 111 pp 533-542
15. Ranjith B, Halder P and Samad A 2019 Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy vol 233 (3) pp 379-396
16. Kim TW, Kaneko K, Setoguchi T, Matsuki E and Inoue M 1990 In KSME/JSME Thermal and Fluid Engineering Conference pp 277–281
17. CFX A Solver theory guide Canonsburg PA Ansys Inc