

NUMERICAL SIMULATION OF THE LOADING CHARACTERISTICS OF STRAIGHT AND HELICAL-BLADED VERTICAL AXIS TIDAL TURBINES

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Abstract: The stress and deflection of straight and helical-bladed vertical axis turbines was investigated using hydrodynamic and structural analysis models. Using Double Multiple Streamtube (DMS) and Computational Fluid Dynamics (CFD) models, the hydrodynamic forces and pressures on the turbines were modeled for three rotational rates from startup to over speed conditions. The results from these hydrodynamic models were then used to determine stress and total deflection levels using beam theory and Finite Element Analysis (FEA) methods. Maximum stress and deflection levels were found when the blades were in the furthest upstream region, with the highest stresses found at the blade-strut joints for the turbines studied. The helical turbine exhibited on average 13% lower maximum stress levels than the straight-bladed turbine, due to the helical distribution of the blades around the rotational axis. All simulation models offered similar accuracy when predicting maximum blade stress and deflection levels; however for detailed analysis of the blade-strut joints the more computationally demanding CFD-FEA models were required. Straight-bladed, rather than helical turbines, are suggested to be more suited for tidal installations, as for the same turbine frontal area they produce higher power output with only 13% greater structural stress loading.

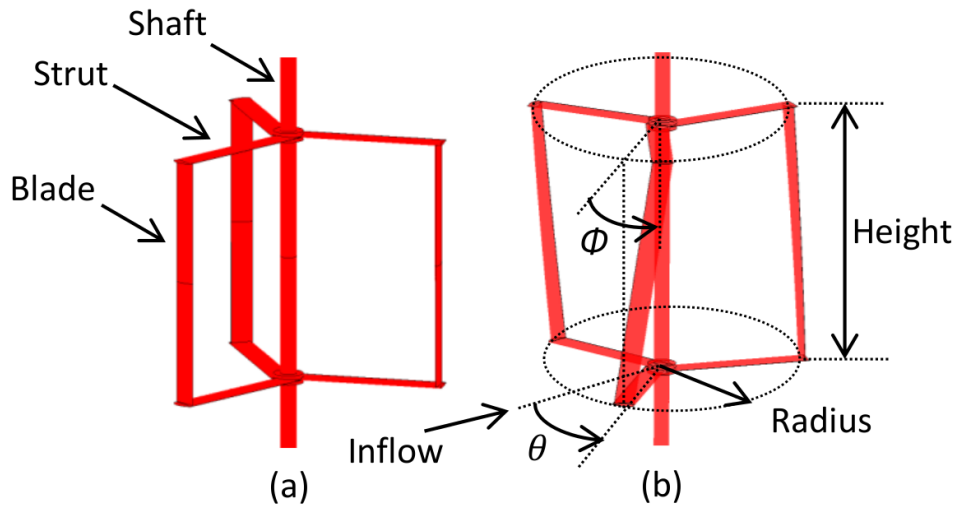
Keywords: Vertical Axis Turbine, Structural Loading, Stress and Deflection Computational Fluid Dynamics, Finite Element Analysis

1. Introduction

Existing studies of vertical axis turbines used for ocean power generation have concentrated primarily on hydrodynamics rather than structural analysis, as researchers have sought to maximise power output. To ensure longevity in marine environments however, detailed knowledge of turbine structural loading characteristics must be established. Although possible using strain gauges, Experimental Fluid Dynamics (EFD) studies to obtain loading are rarely performed. This fact, when combined with a general lack of turbine development over the last 15 years for both wind [1] and tidal turbines, has limited turbine usage. However, knowledge of turbine hydrodynamics and structural characteristics can be obtained by numerical simulation using methods such as coupled Computational Fluid Dynamics (CFD) and Finite Element Analysis (FEA) codes. Additional research into both hydrodynamics and structural characteristics using numerical techniques will further understanding of turbine operational characteristics.

Both straight and helical-bladed designs, as shown in Figure 1, are proposed by various researchers to generate power from the ocean's kinetic energy [2-5]. The designs differ in blade helicity, defined by the blade overlap angle Φ shown in Figure 1. Straight-bladed turbines have 0° blade overlap, whereas helical

17 turbines use blades that are distributed around the rotational axis at a defined overlap angle of Φ . Previous
18 research by the authors indicated that straight-bladed designs generated higher power output when
19 compared to helical turbines of the same frontal area and blade section as a result of the inclination of the
20 helical turbines blades to the inflow [2]. Conversely, helical turbine torque oscillation levels and mounting
21 forces were reduced when compared to straight-bladed turbines, due to the distribution of the turbine
22 blades around the rotational axis [2]. Comparisons of the influence of these factors on the structural loading
23 characteristics of the two designs is currently unknown, as previous research into loading characteristics has
24 concentrated primarily on straight-bladed turbine designs.
25



26
27
28 Figure 1: Straight (a) and helical-bladed (b) vertical axis turbines, showing definitions of azimuth rotational
29 angle θ , and blade overlap angle Φ
30

31 Characterization of vertical axis turbine loading characteristics can be performed numerically by
32 coupling Double Multiple Streamtube and CFD models with beam theory or FEA analysis methods [3-6].
33 However, considerable knowledge gaps exist in the characterisation of structural loading. Previous numerical
34 studies have often been limited to either helical or straight-bladed designs [3-6], with no comparison between
35 loading characteristics of the two designs performed. These works have often concentrated on blade loading,
36 with no determination of the loading of the struts and blade-strut joints performed [3,5,6]. Additionally, previous
37 simulations have concentrated on evaluating loading characteristics at a single rotational rate [3-6]. Research
38 extending numerical simulation models to investigate straight and helical-bladed turbines using models with all
39 geometrical features including struts at multiple rotation rates will give greater insight into turbine characteristics,
40 and allow for the evaluation of any advantages between the differing geometrical layouts.
41

42 In this current study, the blade loading of a straight and a helical vertical axis turbine was determined
43 to characterise blade and strut loading. The hydrodynamic inputs were generated using DMS and CFD
44 models, which were combined with the application of centrifugal and gravitational forces to form structural
45 analysis models using beam theory and FEA. Characterization of maximum stresses and deflection levels
46 and their relationships with blade azimuth angle were performed. This work also sought to determine
47 whether straight or helical turbines are more suited to generate ocean power from both hydrodynamics and
48 structural perspectives.
49

50 2. Turbine Geometry

51
52 Two vertical axis turbine designs were simulated to evaluate the influence of variations of blade
53 helicity on turbine structural loading characteristics. These models differed only in blade helicity as shown
54 in Figure 1, with all common geometrical dimensions outlined in Table 1. Only two designs were considered:
55 a straight-bladed turbine and a helical turbine with 15° of blade overlap. These were chosen as previous
56 studies demonstrated that power output reduced significantly as blade overlap increased above 15° [2],
57 reducing turbine utility for power generation. The geometrical layout of the straight-bladed turbine was
58 based on an EFD turbine from literature to allow for validation of the numerical simulation techniques
59 utilised [2,7]. The helical turbine used the same frontal area, strut geometry, blade chord, and blade section
60 to allow comparisons between the two designs. Both turbines had two struts per blade located at the blade
61 tips.

62
63 Table 1: Shared Geometry of the Straight and Helical Turbines

Geometry	Dimensions
Number of blades	3
Turbine height	0.685m
Blade section	NACA634021
Blade chord	0.065m
Blade overlap	0°
Radius	0.457m
Strut section	NACA0012
Strut chord	0.065m
Number of struts per blade	2
Shaft diameter	0.048m

64 3. Numerical Simulation Methods

65
66 Three loading simulation models were developed allowing for comparisons of the respective benefits
67 of each numerical simulation technique. The simulation models were performed in two steps, first the
68 hydrodynamics followed by the structural simulations. The models developed were the:

- 69
- 70 • DMS-Beam, DMS blade forces combined with a beam theory model;
 - 71 • CFD-Beam, CFD blade forces combined with a beam theory model; and
 - 72 • CFD-FEA, CFD model coupled to the FEA model using pressure mapping techniques.
- 73

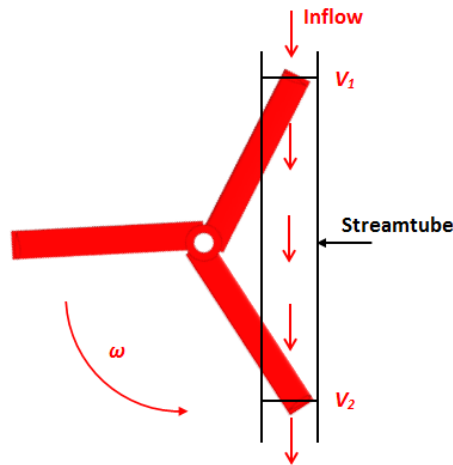
74 3.1 Hydrodynamic Simulations

75
76 Numerical simulations of the hydrodynamic forces were performed using DMS and CFD simulation
77 models. For both models, force coefficients normal to the blade chord were determined, with the forces
78 non-dimensionalised by dynamic pressure and blade chord. The CFD model was also used to output surface
79 pressure data for use with the coupled CFD-FEA model.

80 3.1.1 Double Multiple Streamtube (DMS) Model

81
82 The normal blade force coefficients were modeled using a DMS model previously developed by the
83 authors based on the methods outlined in literature [9]. The turbine was modeled using a double actuator

84 disk method to account for reductions in flow velocity through the streamtube from V_1 to V_2 as shown in
85 Figure 2, with no streamtube expansion modeled for simplicity. Using iterative methods upstream and
86 downstream, induction factors were calculated from which blade angles of attack were determined. Once
87 the latter were known, the forces normal to the blade chord were determined using lift and drag data
88 obtained using the viscous airfoil analysis tool Xfoil [9]. As NACA634021 data was not readily available from
89 literature at suitable Reynolds numbers, NACA634221 data was used as it was similar in profile, with a 2%
90 difference in blade camber. The DMS model included dynamic stall modeling using the Gormont method to
91 simulate the influence of the variations in blade angles of attack generated by the rotation of the blades
92 [10]. Currently the DMS model developed by the authors cannot model helical turbines, as the
93 hydrodynamic influence of the blade inclination has not been adequately accounted for.
94



95
96
97 Figure 2: DMS model showing an example of the streamtube method for calculation of upstream and
98 downstream flow velocity values V_1 and V_2

99 3.1.2 Computational Fluid Dynamics (CFD) Models

100
101 Turbine blade forces were simulated using transient time-accurate 3D CFD models using ANSYS CFX
102 [11], which solved the incompressible fully turbulent URANS equations using an element-based finite
103 volume method. All turbine models were meshed using unstructured tetrahedral elements using ANSYS CFX
104 13.0 [12-15]. Mesh resolution was set by specifying the mesh size and growth rates to allow for local
105 refinement of mesh zones, with inflation layers used on all surfaces to fully resolve the surface boundary
106 layer flow [12-15]. Turbine rotation was simulated by enclosing the turbine in an inner domain as shown in
107 Figure 3 that was rotated using the CFX transient rotor-stator model at the desired rotational rate. The
108 interface between the stationary and rotating domains was modeled using a General Grid Interface (GGI)
109 over which flow values are calculated using an intersection algorithm [11].
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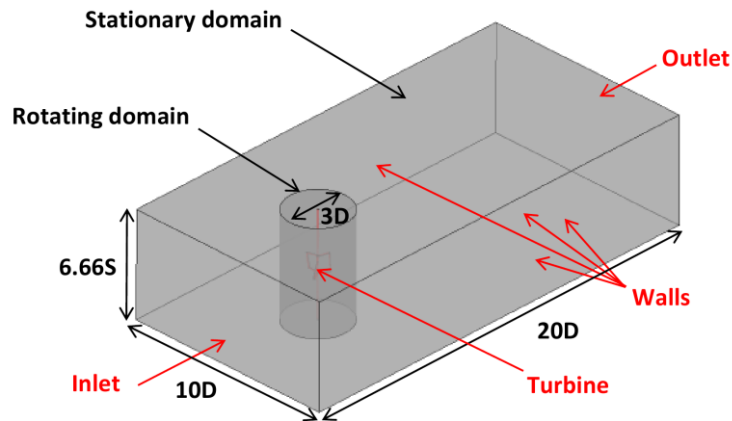


Figure 3: Simulation domain boundary nomenclature and sizing used for straight and helical CFD models. Dimensions in relation to turbine diameter, D , and height, S , as shared by the two turbine designs

The computational domains shown in Figure 3 were generated to simulate free stream conditions, with all corresponding boundary conditions outlined in Table 2. To ensure that the turbines were isolated from any domain wall effects and to allow for full wake development, systematic domain size studies were performed [2,12-15]. All turbines were assumed to operate at sufficient depth to minimise any free surface interaction effects, and thus only the water phase was modeled.

Table 2: Boundary Conditions for the Straight and Helical Turbines

Boundary	Condition
Inlet	Uniform flow: 1.5 ms^{-1}
Inlet turbulence level	5% turbulence
Outlet	Relative pressure: 0 Pa
Walls	Free slip walls
Turbine	No slip walls

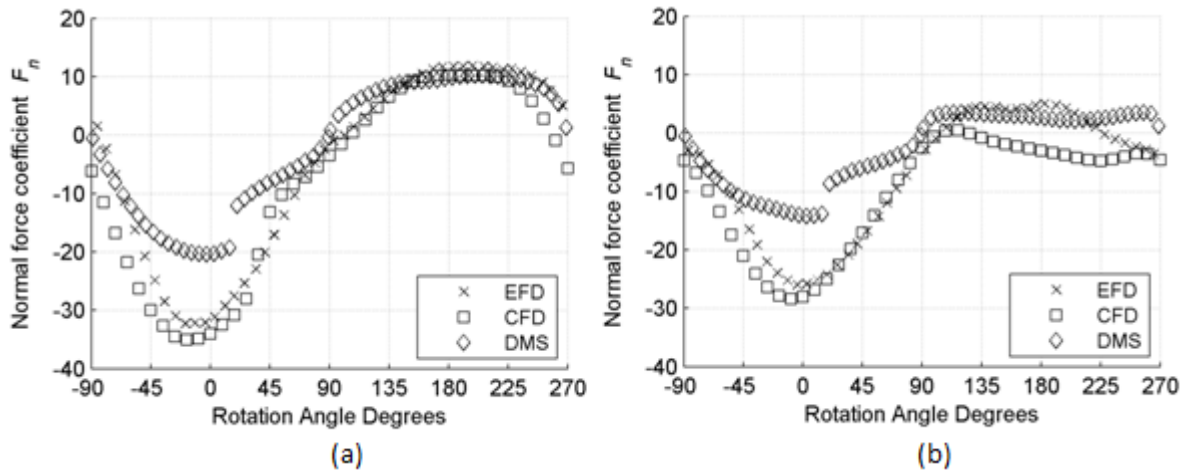
The $k-\omega$ SST turbulence model was utilised for turbulence closure due to its ability to accurately model both free stream and boundary layer regions as well as offering improved prediction of flow separation and adverse pressure gradients by the inclusion of transport effects into the formulation of the eddy-viscosity [16], with the $k-\omega$ SST CFD turbulence model commonly used for vertical axis turbine simulations [2,12-15,17-21]. To ensure numerical accuracy and stability, all simulations were performed using a high order advection and second order transient scheme [12-15]. Convergence was deemed achieved when solution residuals reduced to below 10^{-4} and reduced by more than three orders of magnitude.

Studies of the influence of factors including mesh density, time step size, y^+ , domain length, width and height were conducted. Independence was deemed satisfactory when significant increases in these parameters resulted in C_p differences between successive refinements trending to less than 5%. This resulted in a suitable balance between solution accuracy and computational effort. Full mesh convergence studies were conducted by the authors for the straight and helical-bladed turbine simulated in this work and were presented previously in [2,12-15].

3.1.3 Hydrodynamic Model Validation

Validation of CFD methods against EFD testing of a one and three-bladed turbine from literature revealed good agreement for normal force coefficient predictions [22,23]. The CFD maximum normal force

144 coefficients were predicted on average to within 5.7% of EFD [22,23], with the relationship with rotational
 145 angle replicated accurately. The DMS model was able to accurately predict the location of the maximum
 146 normal force as shown in Figure 4, however it under-predicted the normal force on average by 40% as a
 147 result of severe dynamic stall effects that the Gormont dynamic stall model was unable to satisfactorily
 148 capture.
 149



150
 151 Figure 4: Normal force coefficients for the (a) one-bladed and (b) three-bladed turbines compared to EFD
 152 results [22,23] at a rotational rate of 0.746 rads^{-1} and an inflow velocity of 0.091 ms^{-1}

153 3.2 Structural Simulations

154
 155 Two numerical simulation models were utilised to characterise turbine loading characteristics; beam
 156 theory and FEA models. These models used either force or pressure field results from the DMS and CFD
 157 models outlined in Section 3.1. The beam theory model simulated the structural loading using a simply
 158 supported model, whereas the FEA model simulated the entire turbine structure including the rigid
 159 blade-strut joints. The influence and limitations of these differing structural simulation approaches was
 160 investigated as part of this work.
 161

162 3.2.1 Beam Theory Model

163
 164 A beam theory model was developed using code scripted in Matlab. Three key assumptions were
 165 made to allow the use of this approach. The normal force was assumed to be uniformly distributed to
 166 simplify the coupling between the hydrodynamic and structural models, although the actual force
 167 distribution may be reduce near the tips of the blades due to blade end and blade-strut interaction effects.
 168 The normal force was also assumed to contribute the most to blade stress and deflection, as normal forces
 169 are on average an order of magnitude greater than the tangential forces [8]. The normal force also acts in
 170 the direction normal to the blade chord line, resulting in large bending moments when compared to the
 171 small bending moments caused by the tangential forces. The blades were also assumed to be simply
 172 supported at each end, resulting in the assumption that the stress at the blade ends was zero as beam
 173 models were unable to model the stress at the blade-strut joints due to the geometrical layout of vertical
 174 axis turbines. The beam theory models were developed to establish their accuracy when compared to
 175 CFD-FEA models in the simulation of blade stress and deflection as they require considerably less
 176 computational requirements and solutions times.

177 To calculate the blade stress and deflection, first the normal force coefficients are determined using
 178 the DMS or the CFD models. The forces determined are then transformed into a uniformly distributed load
 179 across the span of the blade. The centrifugal force F_c caused by the turbine rotation is found as,
 180

$$F_c = m\omega^2 r \quad (1)$$

181 where m is the blade mass, ω is the rotational rate, and r is the turbine radius. The distributed load, w ,
 182 acting on the blade span is the sum of hydrodynamic and centrifugal forces calculated. Using this total load,
 183 the bending moment, M is calculated using simple beam theory, where the bending moment is obtained as,
 184
 185

$$M = \frac{wl_e^2}{8} \quad (2)$$

186 where l_e is the blade span. The maximum stress, σ , is determined using,
 187
 188

$$\sigma = \frac{My}{I} \quad (3)$$

189 where y is half the maximum blade thickness, and I is the area moment of inertia determined using a simple
 190 approximation for hydrofoil sections [24] given by,
 191
 192

$$I = K_1 c^4 t(t^2 + \varepsilon^2) \quad (4)$$

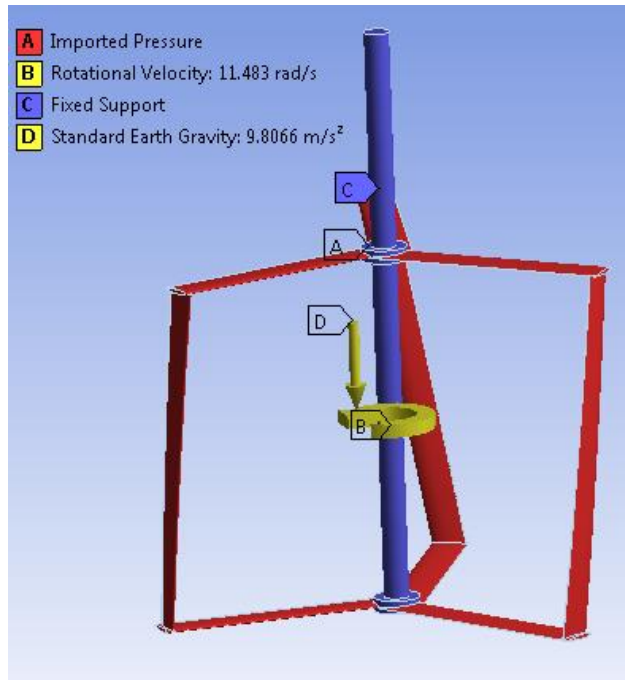
193 where K_1 is a derived proportional coefficient, c is the blade chord, t is the blade thickness, and ε is the
 194 camber percentage. The blade deflection is calculated using,
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 196

$$Deflection = \frac{5wl_e^4}{384EI} \quad (5)$$

197 where E is the material modulus of elasticity.
 198
 199

200 3.2.2 Structural Finite Element Analysis (FEA) Model

201
 202 The stress and deflection on turbine blades and struts were evaluated using the ANSYS FEA linear
 203 Static Structural analysis module [25]. The hydrodynamic pressures on the blades were calculated by the
 204 CFD models and mapped on to the structural model surfaces using Octree mapping [25], as shown in Figure
 205 5. Additionally, inertia and gravitational loads were included to model the steady inertial loads. The FEA
 206 model was constrained at the shaft and hubs to allow for evaluation of the blade and strut forces, reducing
 207 computational effort. Unlike the DMS-Beam model, the CFD-FEA model allowed for the determination of
 208 stress and deflection levels in both the blades and struts. The von Mises stress and total blade deflections
 209 were calculated at each turbine azimuth angle using a custom Python script written by the authors. This
 210 script loaded the surface pressure fields from the CFD transient analysis for each time step, enabling a
 211 one-way Fluid Structure Interaction (FSI) simulation, as any deflections calculated were not reverted back to
 212 the CFD model. Two-way FSI techniques were examined, however due to their excessive simulation time
 213 they were not considered feasible, unless mesh element count was reduced which would adversely affect
 214 the accuracy of the hydrodynamic simulations. The simulated turbines were constructed from steel with all
 215 material properties shown in Table 3.
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219 Figure 5: ANSYS structural model of helical turbine showing loading conditions including imported pressures,
220 rotational velocity, gravity, and the fixed supports

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Table 3: Material properties used for straight-bladed and helical turbine structural analysis

Material	Steel
Density (kg/m ³)	7850
Tensile Yield Strength (MPa)	250
Compressive Yield Strength (MPa)	250
Ultimate Tensile Strength (MPa)	460
Young's Modulus <i>E</i> (GPa)	200

223

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225 The geometry of the FEA turbines was identical to that used in the CFD models, except for the
226 addition of fillets at the blade-strut joints. Fillets of 0.0025m radius were added to avoid infinite or singular
227 stress concentrations at the re-entrant corners of the joints. These can occur as forces applied to mesh cells
228 of reducing size at the fillets will result in ever-increasing stress predictions as the mesh area reduces. To
229 ensure that the addition of fillets did not influence simulation accuracy, maximum von Mises stress
230 magnitudes were determined using CFD models with and without fillets. Variations of maximum stress of
231 less than 1.5% were determined, allowing the use of de-featured CFD models to increase computational
232 efficiency.

233

234 Mesh convergence studies were performed to verify all FEA meshing techniques utilised, with
235 independence studies for maximum and minimum mesh sizing, face sizing refinement, growth rate, and
236 curvature angle performed. Mesh convergence found to be highly dependent on the face sizing of the fillets
237 between the blades and strut joints where the maximum stress magnitudes were located. Successive mesh
238 refinement demonstrated mesh element count independence at 143,000 elements.

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240 4.0 Results and Discussion

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The loading characteristics of straight and helical-bladed turbines were investigated using the DMS-Beam, CFD-Beam, and CFD-FEA models. For each model, stress levels and total blade deflections were recorded over one rotation. All results were simulated at an inflow velocity of 1.5 ms^{-1} . Simulations of turbine loading characteristics were performed for three rotational rates representative of common turbine operational ranges corresponding to a rotational rate of:

- $\lambda=1.5$ similar to that found when starting the turbine;
- $\lambda=2.75$ corresponding to the maximum power output; and
- $\lambda=3.5$ representing an over speed condition.

where λ is the tip speed ratio defined as,

$$\lambda = \frac{r\omega}{V} \quad (6)$$

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and V is the inflow velocity.

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257 4.1 Normal and Tangential Force Coefficients

Using the DMS and CFD models, the normal force coefficients for the straight three-bladed turbine shown in Figure 1 were obtained at $\lambda=1.5$, 2.75, and 3.5 as shown in Figure 6. For $\lambda=1.5$ agreement between the two numerical methods was very good, with both the relationships with azimuth angle and the normal force coefficient magnitudes for each model agreeing closely. The maximum force coefficients were found to occur at approximately -22.5° by both numerical models, with the definition of rotational angle shown in Figure 1. This was due to peaks in the lift generated by the favorable angle of attack over the blades and dynamic stall effects at this azimuth angle. Differences in maximum force of 8.5% were determined between the two models, which may be attributed to differences in dynamic stall modeling, as these differences were found around the force coefficient peaks. The normal force coefficients in the downstream region from 90° to 270° were not fully reversed when compared to the upstream region, as a result of reductions in the flow velocity over the downstream blades caused by the preceding blade's wake. Large reductions in force in the downstream region were previously found in EFD and CFD studies, with force magnitudes of less than $1/3$ found when comparing peak values with average values in the downstream region [22,23].

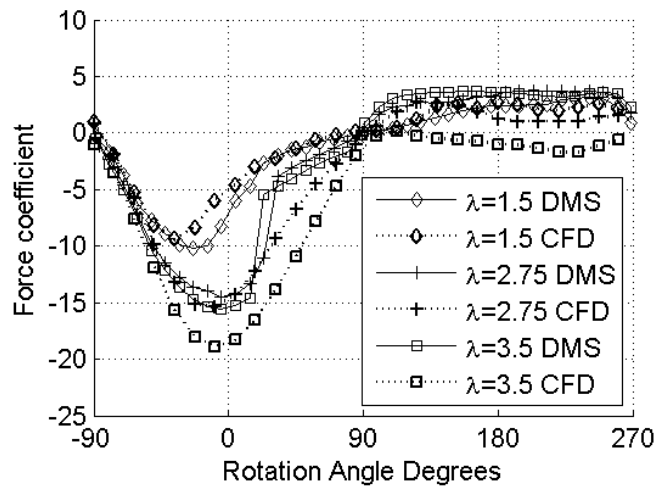


Figure 6: Normal force coefficient simulations for one revolution using the DMS and CFD models at $\lambda=1.5$, 2.75, and 3.5

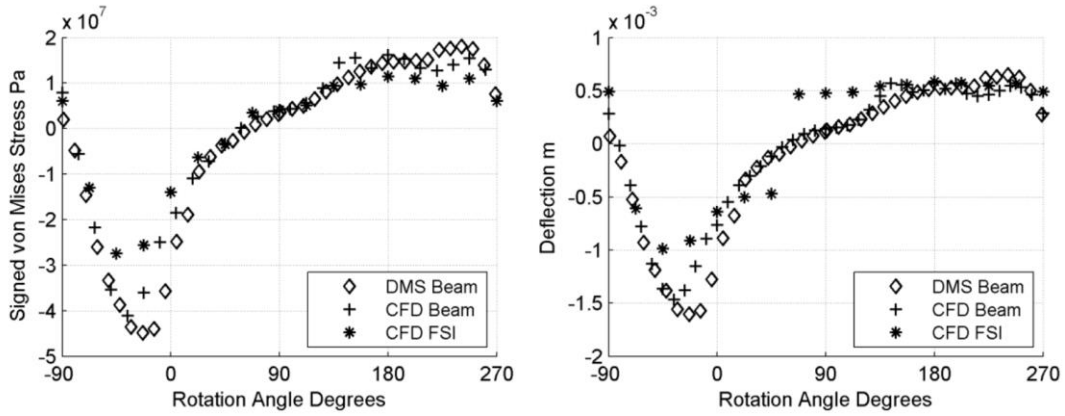
Figure 6 also compares simulations of normal force coefficients using the DMS and CFD models at $\lambda=2.75$. Maximum force coefficient predictions for both models at $\lambda=2.75$ were within 7.3%, with the location of the maximum force predicted at the same azimuth angle for both models. Although the shape of the simulated normal force coefficient curves was similar, predictions of normal force coefficient diverged in the downstream region around 180° . The DMS model accounted for reductions in flow velocity in the downstream region, but it did not account for the increased levels of turbulent flow over the downstream blades, which reduces lift and hence normal force coefficients. However, these turbulent flow effects were simulated by the CFD model, resulting in discrepancies between the two models in the wake-influenced downstream regions. The jump in force coefficient around 22.5° to 45° was caused by jumps in the lift and drag tables used in the DMS model, as well as the by the rapid reduction in the additional lift determined by the dynamic stall model.

Figure 6 also shows the normal force simulations at $\lambda=3.5$ as determined using the DMS-Beam and CFD-Beam models. The predicted azimuth location of maximum force coefficients agreed well, however reduced correspondence was found when comparing maximum force coefficient values predictions, which were within 21% of each other. This reduction in force coefficient similarities between the numerical models when compared to the $\lambda=1.5$ and 2.75 results may be due to the over prediction of the increasing influence of strut drag on the turbine as λ increases by the DMS model. Similar to the simulations of normal force coefficient at $\lambda=1.5$ and 2.75, differences in the downstream region between the CFD and DMS model were apparent.

4.2 Straight-Bladed Turbine Loading and Deflection Simulations

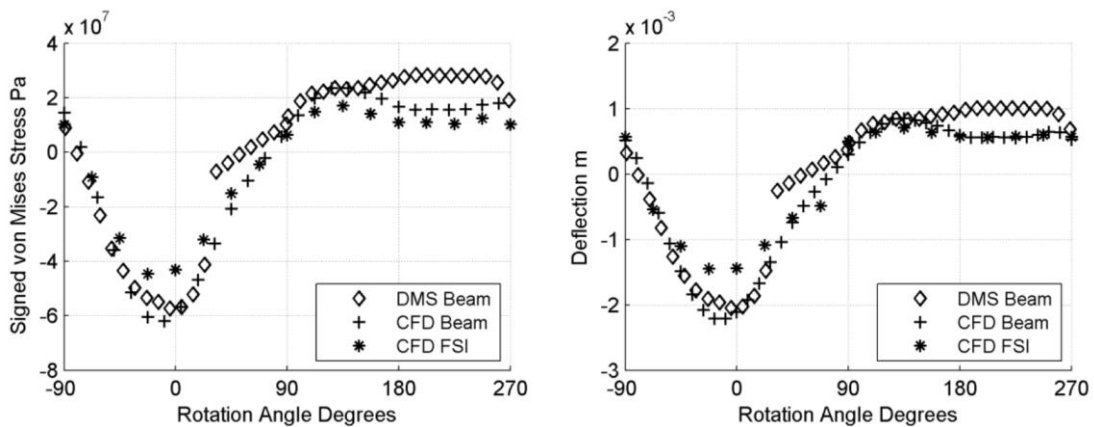
Figure 7 compares von Mises blade stress and deflection levels at $\lambda=1.5$ for the DMS-Beam, CFD-Beam, and CFD-FEA models. The CFD-FEA blade results ignored the stress concentrations at the blade-strut joints, allowing comparison between the simulation models. The highest blade stress and deflection levels were found around -22.5° coinciding with the peaks in the normal force coefficients shown in Figure 6. Similarities across all λ were found between the three simulation models, with the location of maximum stress and deflection found mostly at the middle of the blade span. The maximum stress and total deflection results determined using the DMS-Beam and CFD-Beam models were within 8.4% of each other, as they were calculated using similar values of normal force coefficient as shown in Figure 6. At high

310 absolute values of force coefficients the DMS–Beam and CFD–Beam results diverged from the CFD–FEA
 311 simulations due to differences in the structural support conditions at the blade ends. In the CFD model the
 312 deflection of the struts reduced the blade stress levels, whereas the beam theory models assumed that the
 313 blade was simply supported, resulting in increased stress levels. The stress on the blades was cyclic;
 314 however it is not fully reversed, with reduced levels found in the downstream region around 180°.
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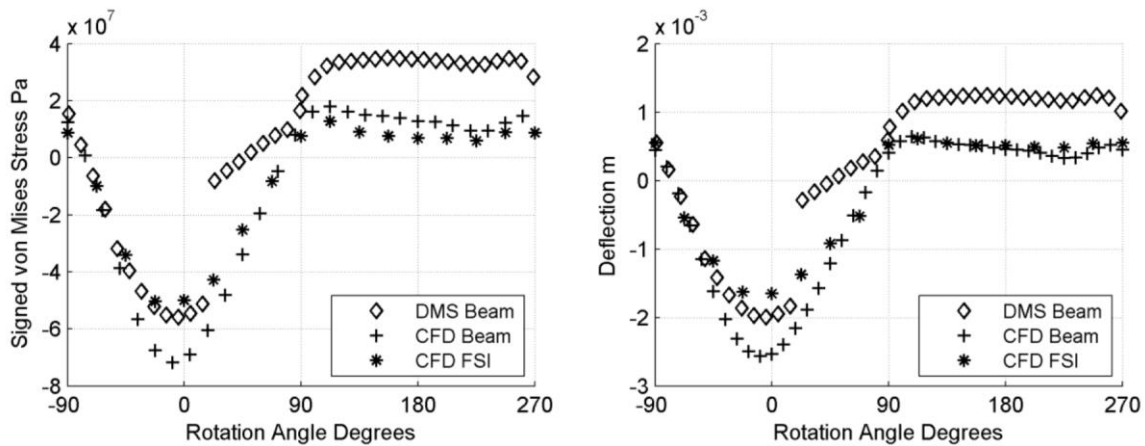
316
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 318 Figure 7: Signed maximum von Mises blade stress and total deflection comparisons for the DMS-Beam,
 319 CFD-Beam, and CFD-FEA straight-bladed turbine models at $\lambda=1.5$. Positive deflection is outwards away from
 320 the shaft
 321

322 Comparisons of blade von Mises blade stress and deflection at $\lambda=2.75$ are shown in Figure 8. The
 323 three simulation model curves prescribe similar stress and deflection curves, with maximum values located
 324 at the middle of the blade span. The highest stress and blade deflection was found at approximately 0°, with
 325 peak stress loads increased on average by 45% when compared to the $\lambda=1.5$ case. This increase in stress
 326 was caused by increases in blade lift due to the blade angle of attack variations reducing to more favorable
 327 levels below stall as λ increased. Similar to that found at $\lambda=1.5$, the DMS-Beam and CFD-Beam models
 328 differed in maximum stress level prediction from the CFD-FEA model, as a result of the blade end support
 329 conditions. The von Mises stresses were not fully reversed, due to reductions in flow velocity and increased
 330 flow turbulence generated by the wake of the upstream blades. The DMS model predicted higher stress and
 331 deflection levels in the downstream regions, as it was unable to simulate the influence of this upstream
 332 blade vortex shedding on the downstream blades.
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 335
 336 Figure 8: Signed von Mises blade stress and total deflection comparisons for the DMS-Beam, CFD-Beam,
 337 and CFD-FEA straight-bladed turbine models at $\lambda=2.75$. Positive deflection is outwards away from the shaft

338 Figure 9 shows the simulated von Mises blade stress and total deflection at $\lambda=3.5$, with the maximum
 339 values located at the middle of the blade span. The maximum stresses were found at approximately 0° , as a
 340 result of peaks in normal force coefficient in the upstream region as shown in Figure 6. Peak stress values
 341 were found to increase on average by 10.6% when compared to the $\lambda=2.75$ case. This increase was less than
 342 that found between $\lambda=1.5$ and 2.75, as the increase in λ resulted in increased centrifugal forces on the
 343 blades which oppose the hydrodynamic forces in the upstream direction. Similar to results in Figures 6 and
 344 8, the maximum stress levels simulated by the CFD-FEA model were reduced when compared to the DMS
 345 and CFD-Beam Theory models.
 346



347
 348
 349 Figure 9: Signed von Mises blade stress and deflection comparisons for the DMS-Beam, CFD-Beam, and
 350 CFD-FEA straight-bladed turbine models at $\lambda=3.5$. Positive deflection is outwards away from the shaft
 351

352 For all simulation models, the highest magnitude of the blade deflection versus blade span was 0.4%.
 353 The small blade deflections found would have minimal impact on the lift and drag generated over the blade,
 354 allowing one-way FSI models to be used. However, if the turbine was constructed from a more flexible
 355 material with a lower modulus of elasticity, these deflection levels would be much higher as a percentage of
 356 the blade span, possibly requiring a two-way FSI approach.
 357

358 Figure 10 illustrates strut and blade deflection over one rotation using the CFD-FEA model. The
 359 blades can be seen to deflect inwards between the rotational angles of -90° to 45° , after which they
 360 deflected outwards for the rest of the rotational cycle. This cyclic pattern repeats over each revolution,
 361 generating tension and compression cycles on the blades. The struts can also be seen to deflect with the
 362 blades, particular at the blade-strut joints.
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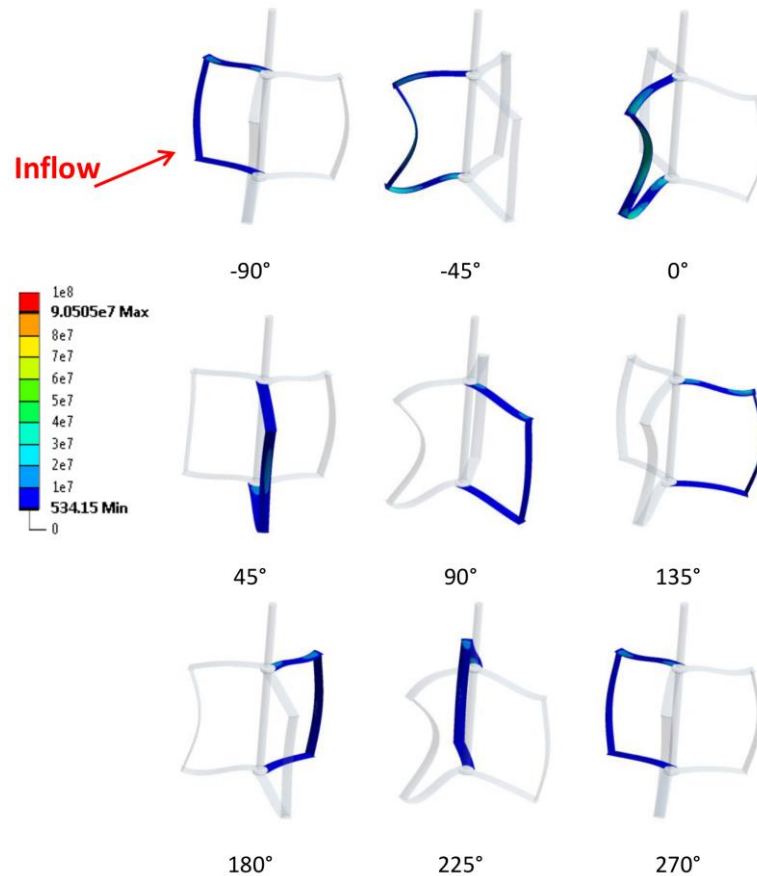


Figure 10: Turbine von Mises stress magnitudes for one turbine rotation at $\lambda=2.75$. Deflection scale increased by 150 to highlight structural deformation

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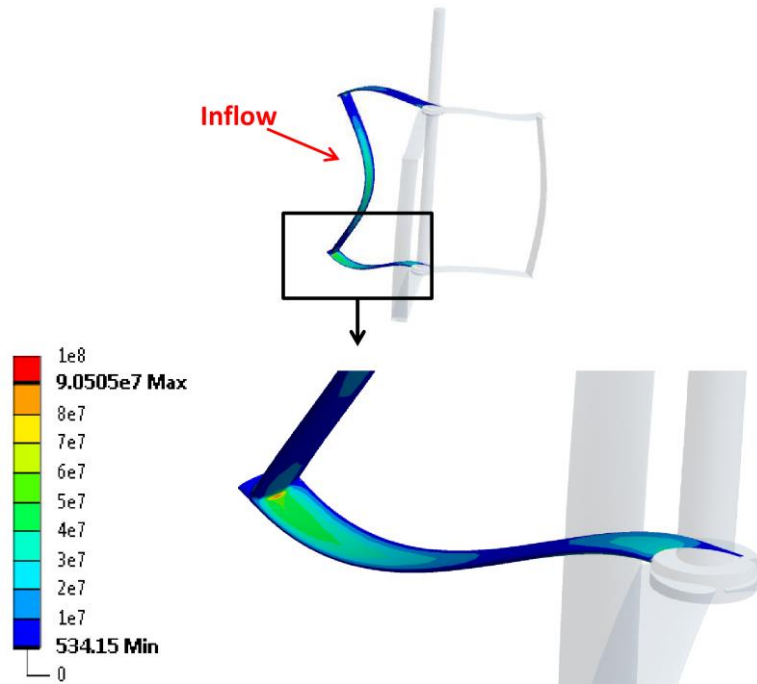
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370 The centrifugal forces generated by the turbines rotation opposed the hydrodynamic forces in the
371 upstream region from approximately -90° to 90° , reducing blade stress and deflection levels, whereas in the
372 downstream region from 90° to 270° the hydrodynamic and centrifugal forces combined. However, the
373 hydrodynamic normal blade forces in the downstream region were significantly reduced when compared to
374 upstream normal force values as shown in Figure 6, due to the reduction in flow velocity in the downstream
375 region and the turbulent flow effects of the preceding blades wake. Thus, the combined downstream total
376 hydrodynamic and centrifugal forces and hence blade stress and deflections were reduced when compared
377 to upstream values. For the turbines studied here the hydrodynamic force was dominant, with upstream
378 force magnitudes and hence blade stress and deflection levels higher than downstream values for all λ
379 simulated.

380

381 The CFD-FEA model was then used to predict the maximum stress magnitudes within the blades and
382 the struts. The maximum stress was found to occur at the bottom blade-strut joint for all λ , as a result of
383 the combination of hydrodynamic and gravitational loading, with levels significantly higher than blade stress
384 levels shown in Figures 7, 8 and 9. An example at $\lambda=2.75$ is shown in Figure 11, with results in Figure 12
385 showing the maximum stress relationships with azimuth angle for each λ simulated. The maximum stress
386 occurred at approximately 0° at the bottom blade-strut joint, as the maximum normal force occurs at this
387 azimuth angle as shown in Figure 6. These normal force peaks generated large bending moments, and
388 hence large stress concentration at the blade-strut joints, with peak magnitudes of approximately 101 MPa
389 noted. The use of beam theory models will not resolve this depending on the location of the strut on the
390 blades.

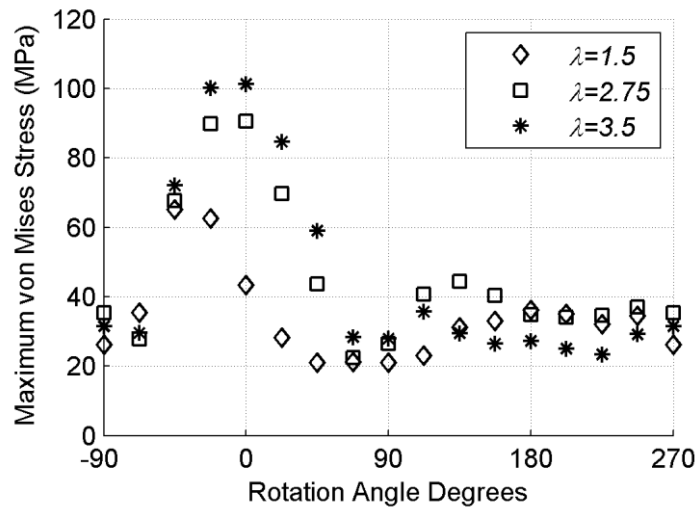
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394 Figure 11: Stress concentration at bottom blade-strut fillet showing the location of maximum von Mises
395 Stress of 90.51 MPa at the azimuth angle of 0° at $\lambda=2.75$
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400 Figure 12: Maximum von Mises Stress at the bottom blade-strut fillet over one revolution determined using
401 the straight-bladed CFD-FEA turbine model at $\lambda=1.5$, 2.75, and 3.5
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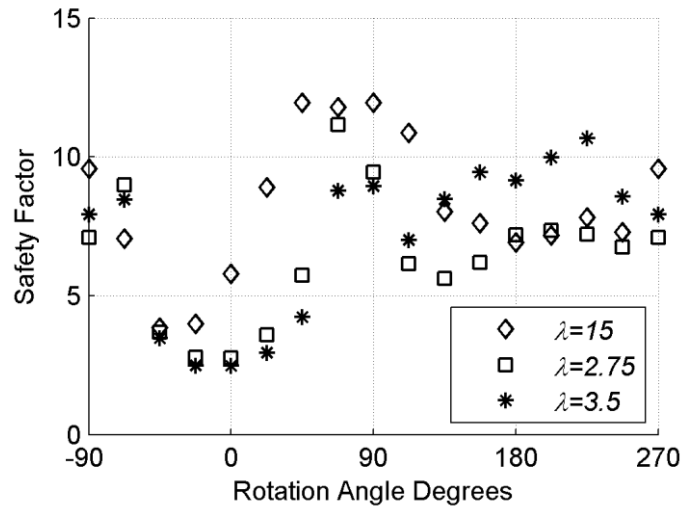
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Comparison of yield safety factors are shown in Figure 13, where the yield safety factor was defined as the ratio of the material yield stress shown in Table 3 to the maximum stress. For each λ , the maximum stress levels were below the material yield strength, with minimum safety factors of 3.84, 2.76, and 2.49 found for $\lambda=1.5$, 2.75, and 3.5. However, the analysis of yield safety factors does not take into consideration any fatigue issues as a result of the cyclical loading. If the tidal velocity distribution is known, the models

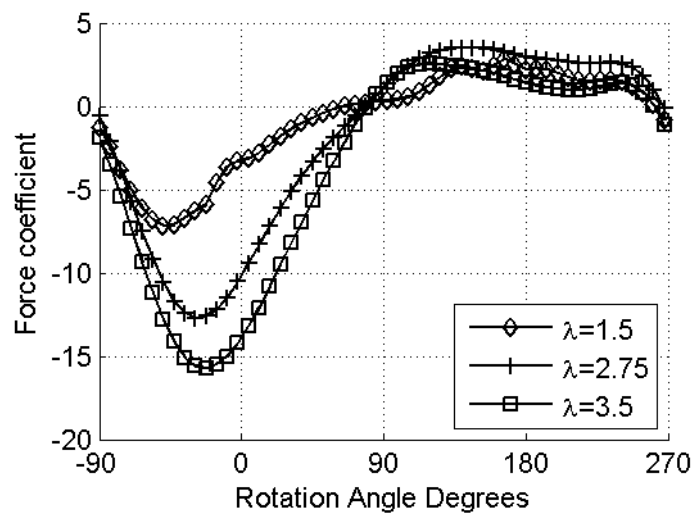
408 developed here can be used to determine the fatigue life of turbine using rainflow counting methods
 409 combined with fatigue models such as Miners rule [26].
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 413 Figure 13: Yield safety factor for the straight-bladed CFD-FEA simulation results for one revolution at $\lambda=1.5$,
 414 2.75, and 3.5

415 **4.3 Helical Turbine Normal Force Coefficients**

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 417 Using the CFD model, the normal blade coefficients were determined for the helical turbine at $\lambda=1.5$,
 418 2.75, and 3.5 as shown in Figure 14. Similar to the coefficient curves determined for the straight-bladed
 419 turbine shown in Figure 6, maximum force was found at approximately -45° to -22.5° . The normal force
 420 coefficients for the helical turbine shown in Figure 14 were reduced when compared to the values found for
 421 the straight-bladed turbine shown in Figure 6, as the distribution of the helical blade around the rotational
 422 axis does not generate lift force peaks simultaneously along its full length as it rotates in the upstream
 423 section at azimuth angles from -90 to 0° .
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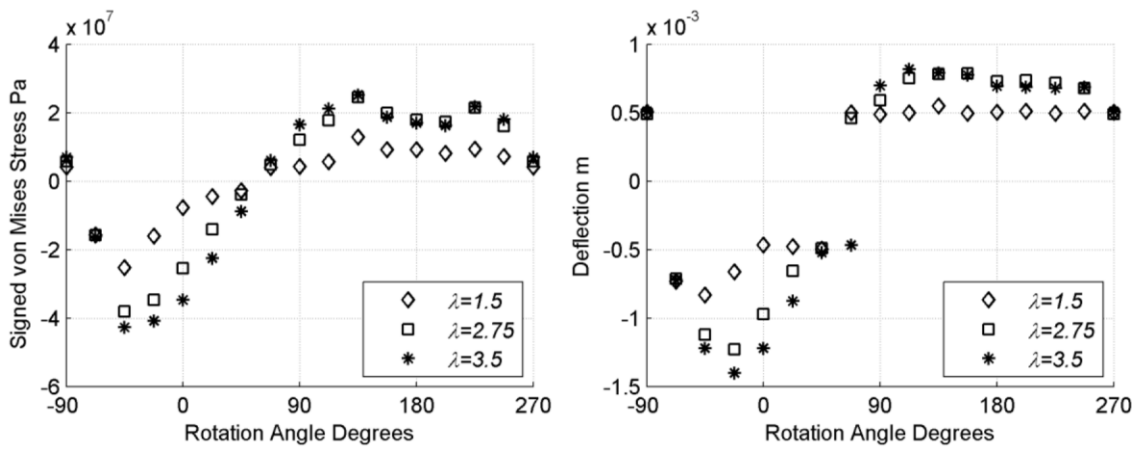


425
 426 Figure 14: Normal force coefficient simulations for one revolution for the helical CFD model at $\lambda=1.5$, 2.75,
 427 and 3.5

428 **4.4 Helical Turbine Loading and Deflection**

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Figure 15 shows the helical turbine von Mises blade stress magnitudes and deflection using the CFD-FEA analysis model. These results focused on the blades and ignored the stress concentrations at the blade-strut joints to allow for comparison with the blade force simulations shown in Figures 7, 9, and 10. Peaks in stress and total deflection occurred for all λ at approximately -45° to -22.5° , with the blades deflected inwards by up to 0.0014 m. In the downstream region the blade deflected outwards, however the stress magnitudes were not fully reversed, similar to that found for the straight-bladed turbine. The helical blade stress and deflection levels were reduced when compared to the straight-bladed turbine results shown in Figures 7, 8 and 9 as the normal force coefficient levels were lower, shown when comparing CFD force predictions in Figures 6 and 15.



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Figure 15: Helical turbine signed von Mises blade stress and deflection comparisons found using CFD-FEA models at $\lambda=1.5, 2.75,$ and 3.5 . Positive deflection is outwards away from the shaft

Figure 16 compares the blade and strut maximum von Mises stress magnitudes at $\lambda=1.5, 2.75,$ and 3.5 . Similar to the straight-bladed turbine results shown in Figure 11, stress peaks occurred at the bottom blade–strut joint due to the combination of hydrodynamic and gravitational forces. Peaks in maximum stress levels were found to occur at azimuth angle of -45° to -22.5° , due to the peaks in normal force generated by the blade in the upstream regions.

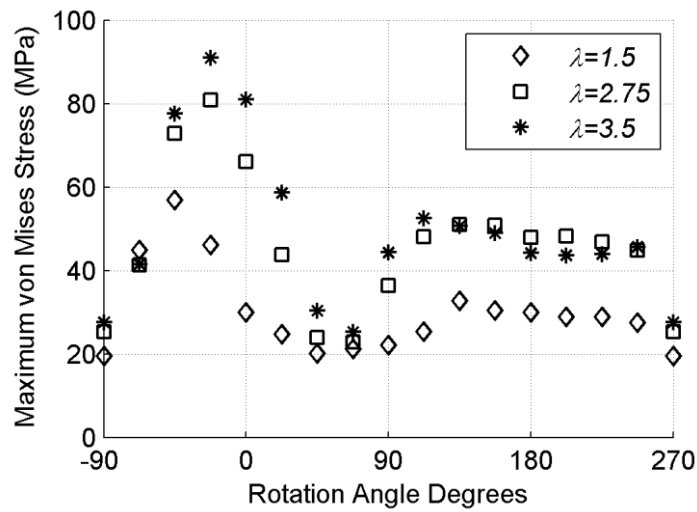


Figure 16: Maximum helical-bladed turbine von Mises stress levels comparing $\lambda=1.5$, 2.75, and 3.5

4.5 Straight and Helical Bladed Turbine Loading Comparisons

Comparisons of maximum von Mises stress levels for the straight and helical turbines are shown in Figure 17. For all λ , the straight-bladed turbine maximum stress levels were approximately 12.9% higher than for the helical turbine values. The straight-bladed turbine stress peaks were higher as the blade generates peaks in lift along its full length simultaneously, whereas the helical turbine blade lift peaks occur along the blade span at differing rotational angles due to the blades distribution around the rotational axis. The decrease in blade bending moment levels found for the helical turbine reduces blade stress when compared to the straight-bladed turbine. In addition, the moment of inertia of the helical blades is better suited to resist bending when compared to the straight blades, again due to their distribution around the rotational axis. Similarly, the blade stress and deflection levels of the helical-bladed turbines were lower than that of the straight-bladed turbines for all λ .

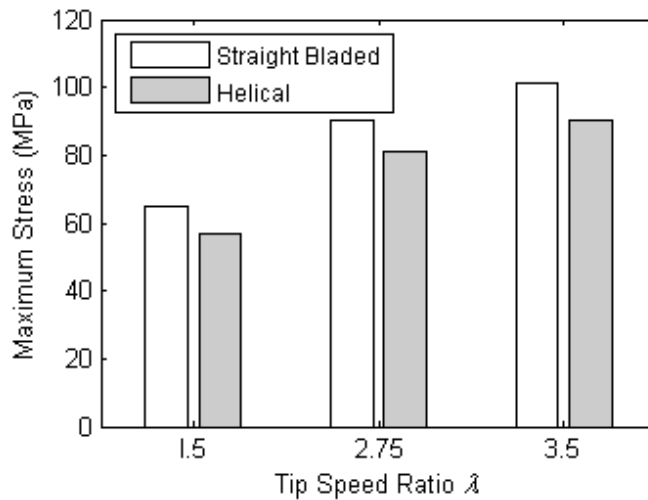


Figure 17: Comparisons of the maximum von Mises Stress magnitudes determined using the CFD–FEA models for the straight and helical turbine models at $\lambda=1.5$, 2.75, and 3.5

472 For ocean and tidal power installations, the authors suggest that straight-bladed turbines are more
473 suitable than helical-bladed turbines as they generate 8% more power for the same frontal area [2], without
474 any significant increase in stress levels as shown in Figure 18. These factors will increase installed power
475 generation capacity while not reducing turbine longevity. Additionally, straight-bladed turbines are much
476 simpler to manufacture than the curved blades of helical turbines, reducing blade manufacturing costs.
477

478 Although no EFD data was available to validate force coefficient simulations for the DMS and CFD
479 models, close agreement between the two models provides some verification and gives confidence in the
480 predicted results. Although the two numerical methods use different techniques, one based on EFD lift and
481 drag data tables and the other on solutions to the Navier-Stokes equations, the normal force coefficient
482 predictions found were on average within 12% of each other for all rotational rates. Combined with the
483 previous validation of the DMS [13] and CFD [2,12-15] models, this high level of agreement gives confidence
484 in the hydrodynamic simulation results presented in this paper. Additionally, although no validation data
485 was available for the structural simulations, the level of agreement between the predicted blade stress and
486 deflection results through the use of two separate structural analysis methods gives confidence in the
487 results presented.
488

489 4.6 Computational Requirements

490 Significant differences in total simulation time and files sizes were required between simulation
491 models as shown in Table 4. All numerical solutions were performed on an Intel i7 860 2.8 GHz based
492 cluster with 2GB ram per core. The significant variations in simulation time suggest that the turbine design
493 process should be performed in two stages. For initial geometrical design studies DMS-Beam models allow
494 the quick estimation of normal forces, blade stress, and deflection levels; enabling the optimization of both
495 power output and blade loading. However, the determination of maximum stress magnitudes as found at
496 the blade-strut joints required the use of CFD-FEA models, as beam theory-based models were unable to
497 resolve the blade-strut stresses.
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500 Table 4: Computational Requirements for One Revolution of the Straight-Bladed Turbine at $\lambda=2.75$

Model	Hydrodynamic	Cores	Structural	Cores	File Size
DMS-Beam	1 minute	1	1 minute	1	1 Mb
CFD-Beam	2400 minutes	24	1 minute	1	80 Gb
CFD-FEA	2440 minutes	24	500 minutes	2	160 Gb

501 Simulations using coupled two-way FSI models were attempted, however they were not completed as
502 it was estimated that the simulations would take around 140 days to complete one revolution, due to the
503 combination of large CFD mesh element counts and reductions in numerical speed due to the coupling of
504 the CFD and FEA models. This compared poorly with the one-way FSI simulations reported here, with total
505 run times of less than 2 days.
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508 5. Conclusions

509 Numerical evaluations of the hydrodynamic and structural loading of straight and helical-bladed
510 turbines were performed using DMS, CFD, beam theory, and FEA methods. These simulations were
511 performed at multiple rotational rates to characterise blade and strut loading. This study revealed three key
512 findings:
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- 515 • straight-bladed turbines exhibit higher maximum stress and deflection levels than helical
516 turbines;
517 • maximum stress levels were found at the bottom blade-strut joints for both straight and
518 helical-bladed turbines; and
519 • maximum stress levels for straight and helical turbines were well below yield strength at an
520 inflow velocity of 1.5 ms^{-1} .

521
522 Combined, the key outcomes listed above lead to an important finding; that straight-bladed turbines
523 are better suited for ocean power than helical turbines, as they generate higher power output without any
524 significant increases in blade loading.

525
526 The simulation models developed in this paper open up considerable possibilities to improve vertical
527 axis turbine designs from both hydrodynamic and structural perspectives. Based on this work the following
528 is recommended:

- 529
530 • investigate blade-strut joint designs using FEA to reduce maximum stress concentration levels; and
531 • conduct EFD using strain gauges to evaluate turbine loading characteristics and provide validation
532 data for the models developed in this work.
533

534 Nomenclature

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c	Blade chord (m)	V_1	Upstream Velocity (ms^{-1})
E	Youngs modulus (Pa)	V_2	Downstream Velocity (ms^{-1})
F_c	Centrifugal force (N)	w	Distributed load (kg/m)
K_I	Moment of inertia proportionality coefficient	y	Maximum blade thickness /2 (m)
l_e	Effective Blade Length (m)	ϵ	Blade camber (%)
I	Area moment of inertia (m^4)	λ	Tip speed ratio
m	blade mass (kg)	σ	Blade stress (Pa)
M	Blade moment (Nm)	ρ	Density (kgm^{-3})
r	Radius (m)	τ	Blade thickness (%)
S	Turbine Frontal Area (m^2)	Φ	Blade overlap angle (degrees)
V	Inflow Velocity (ms^{-1})	ω	Rotational Rate (rads^{-1})

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