FLUID-STRUCTURE INTERACTION SIMULATIONS OF A WIND TURBINE WITH CONSIDERATION OF ATMOSPHERIC BOUNDARY LAYER TURBULENCE

A Thesis in
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by
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Abstract

A reduced order, partitioned fluid-structure interaction (FSI) solver is developed to assess the importance of blade flexibility on the performance of a 5 MW wind turbine in the presence of atmospheric boundary layer (ABL) turbulence. The temporally and spatially varying wind conditions due to the ABL cause fluctuations in the aerodynamic forces and moments of the rotor blades. Analyzing the effect of blade flexibility on the estimated aerodynamic loading and power output is the primary objective of this project.

The FSI solver implemented in this work follows a partitioned approach that incorporates an existing OpenFOAM-based actuator line method (ALM) solver and a modal structural dynamics (SD) solver. This reduced-order solver incorporates a tightly coupled framework using a fixed-point iteration to ensure proper convergence of the flow field and structural displacements at each time step. Simulations using uniform flow are first performed to verify the solver’s functionality, and to evaluate the solver’s coupling characteristics for such things as the importance of blade bend-twist coupling, the effect of aerodynamic pitching moment, and the need for tight coupling.
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\( \tilde{u} \)  Filtered fluctuating velocity, p. 6
\( \rho \)  Flow density, p. 6
\( p^* \)  Resolved kinematic pressure, p. 6
\( \tau_{u}^{SFS} \)  Subfilter-scale stress tensor, p. 6
\( \vec{g} \)  Gravity acceleration vector, p. 6
\( \theta \)  Fluctuating temperature, p. 6
\( \theta_0 \)  Reference temperature, p. 6
\( \vec{f} \)  Planetary rotation rate, p. 6
\( U_g \)  Geostrophic wind velocity, p. 6
\( \eta \)  Gaussian spreading function, p. 7
\( \epsilon \)  Gaussian spreading width, p. 7
\( \phi \)  Modal matrix, p. 10
\( \xi \)  Displacement in modal space, p. 10
\( \omega \)  Natural frequency, p. 10
\( \zeta \)  Damping ratio, p. 10
\( \delta \)  Logarithmic decrement, p. 39
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Wind is an increasingly important alternative to conventional energy sources such as oil, coal, and gas. With coal and natural gas supplying the United States the majority of its energy consumed, the wind-turbine industry has faced an uphill battle in attempting to provide a more environmentally-friendly energy supply. In order for this to be a feasible endeavor, significant advances in technology are required to reduce the cost of electricity generation. One option, which has shown much interest in recent years, is to increase the turbine size. Turbines, capable of producing anywhere from 7 to 12 MW with rotor diameters approaching 200 m [20] (Figure 1.1) are being considered in place of 2 to 5 MW turbines. The lightweight construction and substantial length of current wind turbine blades causes the blades to be relatively flexible. As the size of wind turbine rotor diameters increases, so does their flexibility, which can lead to harsh vibration during their service life, ultimately causing dire consequences to a blade’s fatigue life [26].

Over the past several years, fluid-structure interaction (FSI) of wind turbine blades has been a steadily rising research topic. The wind energy community has placed a strong emphasis on understanding how the continuous interaction between the flow around a turbine and the turbine’s subsequent deformation as a result of this interaction affects performance and service life. Although it is a topic on the rise, it is still in its initial phase and requires much investigation.
1.1 Motivation

The work developed herein is part of a larger effort called the Penn State Cyber Wind Facility (CWF). It is a computational tool for modeling onshore/offshore wind turbine blades with selectable fidelity, meaning it can perform design-level simulations as well as highly resolved simulations in space and time. A schematic showing the different modules that make up the CWF is provided by Figure 1.2. Obtaining highly resolved four-dimensional (4D) cyber data for a fraction of the time and cost of a field experiment is one of the main benefits of the CWF. Additionally, the ability for the user to enable/disable the different modules is also one of the CWF’s more attractive qualities.

The aim of the research presented in this thesis is to model and study the interaction between wind turbine rotor blade deformation and the loads that cause this deformation in the presence of atmospheric boundary layer (ABL) turbulence. Turbulence present in the ABL causes the loading on a wind turbine to be chaotic and can amplify the peak loadings [24]. The loads experienced by the rotor blades are a result of the turbine’s interaction with the
oncoming wind. The blade flexibility impacts this interaction, causing a change in the blade loads, and hence the reaction loads within the turbine, which ultimately affects the life of the blades, gearbox [35] and bearings. Moreover, these load variations also have an impact on the turbine’s power output.

The majority of the FSI literature available for wind turbines deals with loosely coupled implementations, meaning that the flow and structure solvers do not communicate with each other during each time increment. Each solver works independently of the other and is advanced in time independently as well. In a type of framework where the fluid and the structure depend greatly on each other’s response, such as a wind turbine in the atmosphere, a tightly coupled solver is desired [7].

**Figure 1.2.** Schematic showing the different modules that make up the Penn State Cyber Wind Facility.
The partitioned FSI solver developed herein incorporates an actuator line method (ALM) based solver [17] to an author developed structural dynamics solver based on a modal summation approach. The ALM-FSI solver developed herein has the capability of implementing both loose and tight coupling schemes that are run-time selectable.

1.2 Previous and Related Work

Presently, the author is unaware of any efforts directed towards performing reduced-order or fully-resolved wind turbine simulations that incorporate both structural deformation and ABL turbulence, with the exception of the Simulator for Offshore Wind Farm Application (SOWFA) developed by Churchfield, et al. [8]. Among the capabilities present in the SOWFA package is the coupling of a computational fluid dynamics (CFD) solver for the ABL with NREL’s (National Renewable Energy Laboratory) aeroelastic computer-aided engineering (CAE) tool for horizontal wind turbines, FAST (Fatigue, Aerodynamics, Structures, and Turbulence) [19]. There are several limitations of FAST that may be important to the current research. FAST does not include a bend-twist coupling capability, which is one of central points of focus for this research. Bend-twist coupling is an effect present in wind turbine blades due to their construction and material layup, which is an important aspect of the present research. In simple terms, it is a phenomenon which causes the blade to respond in torsion as it deforms in bending. It is a widely featured topic in the literature, given that it has the potential to mitigate excessive loading on the turbine and improve fatigue performance [20]. Larwood and van Dam [22] built upon FAST by adding the required geometry and mode shapes for the bending and twisting motion of a swept blade. However, their modifications to FAST are not currently implemented in the publicly released version. In addition, Johnson [18] and Wang, et al. [36] have developed a new finite element (FE) beam model based on geometrically exact beam theory (GEBT) called BeamDyn. This new
beam model is capable of modeling the blade’s bend-twist coupling, in addition to blade nonlinear response. Although it is not yet available with the latest release of FAST, it is slated to replace the incumbent blade model in the near future.

The author is aware of only a limited number of publications related to FSI simulations of full-scale wind turbines, as in the recent work by Bazilevs, et al [3], as well as Hsu and Bazilevs [13]. The former exploits cyclic symmetry and solves only a third of the fluid domain to reduce computational cost, in addition to only simulating uniform and steady flow. The latter does not use cyclic symmetry, but also simulates only uniform and steady flow. Heinz developed an aeroelastic simulation tool coupling high-fidelity CFD with a multi-body-based structural solver to simulate the response of an entire wind turbine rotor [12]. This work did explore a variety of coupling strategies for loose and tight coupling schemes, but was limited to steady flow.
Chapter 2  
ALM-FSI Solver Implementation and Validation

2.1 Flow Solver

The flow solver implemented in this work is a variation of OpenFOAM’s \texttt{pimpleDyMFoam} solver, which is a transient solver based on the hybrid Pressure Implicit Split Operator - Semi-Implicit Method for Pressure Linked Equations (PISO-SIMPLE) algorithm [2], which also includes dynamic mesh motion capabilities. The ALM solver described here was primarily developed in support of the CWF project by Schmitz and Jha [16]. Its implementation is described herein for completeness. The existing \texttt{pimpleDyMFoam} source code is modified to include additional terms required to accurately model ABL turbulence (the Coriolis force due to planetary rotation and the buoyancy force) and the wind turbine via the ALM. The new solver is thus named \texttt{ablActuatorDyMFoam}. Turbulence in the flow is modeled using the Smagorinsky LES model in OpenFOAM. The governing equation for the conservation of momentum for the fluid using the Boussinesq approximation is:

\begin{equation}
\frac{\partial \vec{u}}{\partial t} + \nabla \cdot (\vec{u} \vec{u}) = \frac{-1}{\rho_0} \nabla p^* - \nabla \cdot \tau^{SFS}_u + \frac{\vec{g}}{\theta_0} (\theta - \theta_0) + \vec{f} \times (\vec{U}_g - \vec{u}) \tag{2.1}
\end{equation}

where:

- \( \frac{\partial \vec{u}}{\partial t} + \nabla \cdot (\vec{u} \vec{u}) \) represents the acceleration of a fluid particle

- \( \frac{-1}{\rho_0} \nabla p^* \) is the flow’s pressure gradient term
• $\nabla \cdot \tau_{u}^{SFS}$ represents the subfilter-scale momentum fluxes [31]

• $\frac{g}{c} (\theta - \theta_0)$ is the buoyant force

• $\vec{f} \times (\vec{U}_g - \vec{u})$ is the Coriolis force due to planetary rotation.

It is important to note that when using the ALM formulation to simulate the effect of the wind turbine on the flow, the surface of the blades is not explicitly modeled in the fluid mesh [32]. Rather, a body force term is added to the right-hand side of Eq. 2.1 to account for this effect. A summary of the ALM formulation used herein, along with a description of the aforementioned body force term, is provided next.

### 2.1.1 Actuator Line Method Formulation

The ALM represents each rotating blade as a line of discrete points referred to as actuator points, as depicted in Figure 2.1. The local flow velocity and angle of attack are used to interpolate coefficients of lift, drag, and pitching moment using two-dimensional (2D) lookup tables. The lift force, drag force, and pitching moment are estimated using these interpolated coefficients and are applied to the blade at each corresponding actuator point location as demonstrated in the last frame of Figure 2.1. The effect of the total aerodynamic force on the flow, however, is not applied discretely but rather projected into a volumetric force field:

$$\vec{F}_{\text{blades}}(x, y, z, t) = \sum_N \sum_m \vec{f}_{N,m}(x_{N,m}, y_{N,m}, z_{N,m}, t) \eta_{N,m}, \quad (2.2)$$

where $\vec{f}_{N,m}$ is the aerodynamic force vector at each individual actuator point, $N$ is the blade index (which ranges from 1 to the total number of blades for each turbine), $m$ is the actuator point index, $\vec{F}_{\text{blades}}$ is the volumetric force field, and

$$\eta_{N,m} = \frac{1}{\epsilon^3 \pi^2} \exp \left[ \left( \frac{|\vec{r}|}{\epsilon} \right)^2 \right] \quad (2.3)$$
describes how the force is projected as a function of the Gaussian spreading width $\epsilon$, and the distance from a cell center to an actuator point, $\vec{r}$. The volumetric force field $\vec{F}_{\text{blades}}$ is added to the right-hand side of Eq. 2.1, which provides a simple and computationally less expensive method of simulating the effect of the wind turbine on the flow field than using a blade-resolved fluid mesh.

There currently exist two approaches for the projection of the discrete aerodynamic forces: uniform and elliptic spreading. In the former, the spreading width $\epsilon$ is a function of the grid spacing only and is constant throughout the blade span. In the latter, $\epsilon$ is a function of an equivalent elliptic blade planform in addition to the grid spacing [15]. Jha, et al. conducted a study to investigate how each projection technique affects the wake characteristics and blade loads in the presence of an ABL [16]. This study determined that chord-based projection led to improved prediction of local blade loads when compared to measured data, and is therefore the approach used for this work.

### 2.2 Structural Solver

The structural solver developed for this research is based on a modal summation formulation, wherein a finite number of shapes (modes) are used to describe the motion of the structure

![Figure 2.1. Actuator Line Method discretization](image)
as it deforms. Because of this superposition of modes, this method is inherently linear. The aeroelastic module in FAST uses a similar approach, however, the number of mode shapes that can be used to define the blade’s response is limited to two flapwise modes and one edgewise mode. In contrast, the solver described herein allows an arbitrary number of modes. Furthermore, as first mentioned in section 1.2, the current version of FAST does not incorporate bend-twist coupling effects, whereas the present solver does.

The following subsections introduce the structure equation of motion and the process of transforming it from physical coordinate space to modal coordinate space, as well as introduce the implementation of the aerodynamic pitching moment and its relationship with the structural solver.

### 2.2.1 Modal Summation Formulation

The equation of motion in physical space for the structure can be arranged in the following form:

\[
[M] \{\ddot{x}\} + [C] \{\dot{x}\} + [K] \{x\} = \{F\},
\]

where \([M]\) is the mass matrix, \([C]\) is the damping matrix, \([K]\) is the stiffness matrix, and \(\{F\}\) is the force vector accounting for aerodynamic, gravitational, Coriolis, and centrifugal forces:

\[
F = f_{N,m} + F_{\text{gravity}} + F_{\text{Coriolis}} + F_{\text{centrifugal}}.
\]

The equation of motion can be transformed from physical to modal coordinates using

\[
\{x\} = [\phi]\{\xi\},
\]

where \(\{x\}\) is the displacement vector in physical space, and \([\phi]\) is the modal matrix. Using
this and premultiplying Eq. 2.4 by the transpose of $[\phi]$, the equation is modified to:


(2.7)

where $[\xi]$ is the displacement vector in modal coordinates. If the mode shapes (which are the columns of $[\phi]$) are mass-normalized, Eq. 2.7 becomes:

$$[I]\{\ddot{\xi}\} + \text{diag}[2\zeta \omega_n]\{\dot{\xi}\} + \text{diag}[\omega_n^2]\{\xi\} = \{f\},$$

(2.8)

where $\zeta$ and $\omega_n$ represent the damping ratio and natural frequency of each mode, respectively. This process decouples the equations, which are solved for $\xi$ using a Newmark time integration scheme [9]. Once the modal displacements are obtained, the physical displacements are recovered using Eq. 2.6.

### 2.2.2 Pitching Moment Implementation

Although the original ALM formulation does not incorporate an aerodynamic pitching moment calculation in its framework [32], the necessary modifications were made to include it in this research. However, it is important to note that its effect on the fluid-structure interaction is only partially implemented. This is due to the fact that only the effect of the pitching moment on the structure is considered, but not the effect on the flow. Incorporating the latter requires the body force projection algorithm to be modified to account for the pitching moment in a similar way to how the lift and drag forces are projected. Such a modification is outside the scope of this work, and thus only the effect of the pitching moment on the structure’s deformation is taken into account.

The sign convention used for the pitching moment is consistent to what is found in FAST’s aerodynamics module, Aerodyn [21]. Namely, a positive pitching moment is that which pitches the blade away from the oncoming flow, leading to an increase in AOA as seen in
2.2.3 Implementation of Gravitational, Coriolis, and Centrifugal Forces

To properly account for the effect of gravitational and inertial loads on the blades, the finite element principle of work-equivalent loads is implemented in the structural solver. This principle demonstrates that work done by nodal loads acting through nodal displacements is equal to the work done by a distributed load moving through a displacement field [9]:

\[
W = \{d\}^T \{f\} = \{d\}^T \int [N]^T \{F\} dV = \int \{u\}^T \{F\} dV,
\]

where \{d\} is the nodal displacement vector, \{f\} is the nodal loads vector, \([N]\) is the FE shape function matrix, and \{F\} is the vector of distributed loads on the structure. This approach allows for the gravitational and inertial loads, which are distributed loads, to be computed at each node. To compute these mass-dependent forces at the nodes, each node’s contribution to the total blade mass is determined via an optimal mass lumping approach, first introduced by Malkus and Plesha [25]. In this approach, the mass of each element is divided amongst its nodes as shown in Figure 2.3. The individual contributions from different elements to a particular node are then summed to find that node’s total mass. Thus the
gravitational, Coriolis, and centrifugal forces at each node are computed as:

\[{F_{\text{gravity}}}_i = m_i \{g\} \quad (2.10)\]

\[{F_{\text{Coriolis}}}_i = -2 * m_i \{\Omega\} \times \{v_i\} \quad (2.11)\]

\[{F_{\text{centrifugal}}}_i = -m_i \{\Omega\} \times (\{\Omega\} \times \{R_i\}) \quad (2.12)\]

where \(i\) is the node index, \(\{g\}\) is the gravity vector, \(\{\Omega\}\) is the blade’s rate of rotation, \(\{v\}\) is the \(i\)th node’s velocity relative to the spinning blade, and \(\{R\}\) is the \(i\)th node’s position vector.

2.2.4 Coupling Blade Deformation to the ALM

The modal matrix obtained using the commercial solver Abaqus (and FE model to be described in Chapter 3) contains information regarding the degrees of freedom of the entire structural model, which is required to obtain the displacements and velocities to compute the centrifugal and Coriolis forces (Eqs. 2.10, 2.11, 2.12). Because the actuator line is generally discretized by a much smaller number of points than the number of nodes in the FE model, a separate modal matrix is created to predict the deformation of the actuator line. This new
modal matrix contains mode shapes which have been interpolated from the model’s nodes to the appropriate spanwise locations of the actuator line. Note that the number of nodes in $[\phi]$ is arbitrary once a basis set is defined. Each airfoil section associated with an actuator point is defined by four nodes as demonstrated by Figure 2.4: one for the leading edge (LE), one for the trailing edge (TE), one for the aerodynamic center (AC), and one for the location of the pitch axis (PA).

![Figure 2.4. Example section showing the four nodes that define each airfoil section along the actuator line](image)

The number of actuator points is run-time selectable, but is limited to 41 in the present work. The motion at the LE, TE, and PA nodes is described by three translational DOF each, while the node at the AC has three translational and three rotational DOF; this results in a total of 15 DOF per airfoil section. Given that the airfoil lookup tables for the 5 MW turbine used for this work are the ones provided with FAST, it is necessary to maintain consistency as to where the lift, drag, and aerodynamic pitching moment coefficients act along the chord line. Figure 2.5 shows the location of the pitching axis and aerodynamic center along the span of the 5 MW blade. These values play an important role when interpolating the mode shapes extracted from the FE model to the appropriate locations for the actuator line, as will be described further in Section 3.1.2. For now, it suffices to state that the aerodynamic force and moment are applied at the AC location of each airfoil section, while the displacements applied to the actuator points are the ones computed at the PA location. This is due to the fact that the pitching axis location for the FE model coincides with the $z$-coordinate axis. As the blade bends due to the applied forces, it has a tendency to twist as well. This effect is
referred to as bend-twist coupling. This coupling is the main cause for variations in local angle of attack due to blade deformation, which in turn cause changes in the lift, drag, and pitching moment coefficients found via the lookup tables. Further details on how this effect is incorporated into the solver are provided in the following section.

2.3 Bend-Twist Coupling

The bend-twist coupling effect is a consequence of the principal material direction of laminas, as shown in Figure 2.6, being skewed with respect to the blade axis [5]. It is important to model this effect as accurately as possible, as it plays a crucial role in predicting the structure’s response and its interaction with the flow around it. The structural solver uses the flapwise component of displacement of the LE and TE, as well as the local chord, to compute the twist due to deformation:

\[
\Theta = \arctan\left(\frac{\text{flapwiseTEdisp} - \text{flapwiseLEdisp}}{\text{chord}}\right)
\]  

(2.13)
The twist computed in Eq 2.13, depicted in Figure 2.7, is added to the structural twist already present in the blade due to its design. The sign convention for the computed twist is such that a larger LE displacement results in a negative twist value, causing the blade to pitch away from the wind and increasing the AOA. Conversely, a larger TE displacement results in a positive value, decreasing the AOA due to the blade pitching into the wind. Both of these scenarios are demonstrated in Figure 2.8.

2.4 ALM-FSI Solver

Fluid-structure interaction solvers can generally be grouped into either monolithic or partitioned solvers. In a monolithic approach, the governing equations for both the flow and structural displacement are solved simultaneously with a single solver, usually in terms of the pressure and velocity fields [14]. In contrast, a partitioned approach employs two
distinct solvers to solve the governing equations for the flow’s pressure and velocity fields and structure’s displacement. As described in Section 1.1, the FSI solver developed herein couples the ALM solver and the structural solver using a partitioned approach. The solver is developed within a tightly coupled framework, but is still capable of performing loosely coupled simulations if desired.

2.4.1 Fluid-Structure Coupling

To ensure a tightly converged solution between the flow and structural solvers, multiple sub-iterations at each time step are required. The `ablActuatorDyM Foam` solver is thus
modified to incorporate an inner sub-iteration loop within the time marching scheme, as shown in Figure 2.9. In the first sub-iteration, the solver determines the aerodynamic loads on the blades based on the velocity field for the current time-step, and uses these loads to compute an initial displacement. The position of the actuator points is updated, and the effect of the deformed structure on the velocity field is taken into account for the next iteration. Once the loads are updated, the new displacement is calculated and a residual is computed based on the new and old values of displacement:

\[
Residual = \frac{\max(|\vec{x}^i_j - \vec{x}^{i-1}_j|)}{\text{blade span}}
\]  

(2.14)

where \(\vec{x}^i_j\) is the displacement at the current iteration, \(\vec{x}^{i-1}_j\) is the displacement at the previous iteration, \(j\) is the actuator point index, and \(i\) is the iteration index. If the residual is less than a prescribed tolerance criterion, the solver advances to the next time step. Otherwise, the velocity field and displacements are recomputed and convergence is checked until the solver reaches a converged solution.

![Figure 2.9. Outline of ALM-FSI solver](image.png)
2.4.2 Solver Flags

The new ALM-FSI solver created in this work includes a number of different flags to enable or disable features required for different simulations. These flags allow the solver to be stripped down from its full ALM-FSI functionality back to the original ALM version. The flags are identified in the following list, with a description of their purpose.

- **bladeInertia** enables the inertial terms in the structural equation of motion. If set to 'off,' the structural solver becomes a static solver.

- **simulationRestart** enables the solver's restart capability and imports the necessary files to load the latest turbine state available. If set to 'off,' the solver starts as usual from time = 0.

- **twist** when enabled the twist due to deformation is applied to the structure.

- **pitchingMoment** the aerodynamic pitching moments are applied to the structure when enabled.

- **chordSpreading** enables elliptic spreading of the body force term. If set to 'off,' uniform spreading is used.

- **numberOfModes** determines how many modes will be used to solve for the structure's response (must be \( \leq \) the number of modes in the basis set).

- **rigidBlades** turns off blade flexibility.

- **maxIter** determines the maximum number of iterations to be performed per time step for a tightly coupled simulation. If set to '1,' no subiterations occur and the solution is considered 'loosely' coupled.
Chapter 3  
Structure and Flow Model Development and Solver Validation

3.1  Structure Model

3.1.1  Finite Element Model

The structural solver for the present work is based on the modal summation approach, and thus requires eigenvalues and eigenvectors. A plethora of finite element solvers, including commercially available solvers, are capable of performing the necessary eigenanalysis, requiring only a valid finite element (FE) model. Therefore, a FE model is created using Abaqus CAE [30] based on an existing computer-aided design (CAD) model for the surface geometry developed by Vijayakumar [34] from the specifications in the NREL 5 MW report [19]. A schematic of an arbitrary wind turbine blade is provided in Figure 3.1, which shows how different types of airfoils are used to give the blade its desired geometry. Composite blades are made up of a variety of layups, which are combined to provide the blade with a wide range of thicknesses both along the span of the blade and the circumference of the airfoils. The internal members, i.e. the shear webs, are what allow the blade to maintain its aerodynamic shape, while also providing load-bearing capabilities [1]. For this reason, the CAD model from [34] is modified to include the shear webs based on the 5MW ANSYS model detailed in [28].
One of the most difficult aspects of modeling the NREL 5 MW blade is defining the original composite layup because it is not well defined in the literature. Several attempts have been made to reverse-engineer wind turbine blade layups, such as in [11], [10], and [28], although this type of approach is significantly expensive in terms of time required to develop. Given that the scope of this project is not to exactly reproduce the distributed structural properties detailed in [19], but rather to gain insight into how blade flexibility affects a wind turbine’s performance, the approximate layup definition developed by Resor [28] is used for the research described herein. The complexity of the composite layup of the 5 MW blade required each of the elements comprising the FE model to have its material definition, including elastic properties, ply thickness, and ply orientation. This produces a unique thickness distribution in the model, as can be seen in Figure 3.2.

The FE model developed for the 5 MW blade, which is clamped at the root and free at
Figure 3.2. Thickness distribution of the NREL 5 MW Finite Element model
the tip, is shown in Figure 3.3. This combination of boundary conditions causes the blade to behave as a cantilevered beam. The clamped condition at the root is an approximation made for the sake of simplicity, given that in actuality a turbine’s blades are fastened to the inside of the hub, which is a more complex condition to model. This root boundary condition appears to be a common approach, as detailed in [11].

The first six blade natural frequencies obtained with the Abaqus model compare reasonably well to those provided in [28], as demonstrated by Table 3.1. Because the most dominant forces experienced by the turbine are lift forces, flapwise bending is generally the most dominant form of deformation [1]. It is thus reasonable to estimate that the dominant modes in the blade’s structural response will be the flapwise bending modes. A contour depicting the first flapwise bending mode shape, colored by displacement magnitude, is shown in Figure 3.4.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Abaqus model (Hz)</th>
<th>Sandia report (Hz)</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.896</td>
<td>0.870</td>
<td>1\textsuperscript{st} flapwise bending</td>
</tr>
<tr>
<td>2</td>
<td>1.03</td>
<td>1.06</td>
<td>1\textsuperscript{st} edgewise bending</td>
</tr>
<tr>
<td>3</td>
<td>2.73</td>
<td>2.68</td>
<td>2\textsuperscript{nd} flapwise bending</td>
</tr>
<tr>
<td>4</td>
<td>3.94</td>
<td>3.91</td>
<td>2\textsuperscript{nd} edgewise bending</td>
</tr>
<tr>
<td>5</td>
<td>5.43</td>
<td>5.57</td>
<td>3\textsuperscript{rd} flapwise bending</td>
</tr>
<tr>
<td>6</td>
<td>6.34</td>
<td>6.45</td>
<td>1\textsuperscript{st} torsion</td>
</tr>
</tbody>
</table>

3.1.2 Mode Shape Interpolation

The eigenvectors extracted from the FE model form the columns of the modal matrix $\phi$ introduced in Eq. 2.6. As previously described in Section 2.2.4, each actuator line section requires 15 DOF to describe its motion: three translational DOF for each the TE, LE, and PA nodes, and six total (three translational and three rotational) DOF for the AC node. To obtain the appropriate modal matrix for the ALM simulations, the nodal DOF information contained in $\phi$ needs to be interpolated from the nodes in the FE model to the correct
spanwise (for all four nodes) and chordwise (for the PA and AC nodes only) locations that define the actuator line. The interpolation takes place in three phases:

- determine the fore and aft airfoil sections to be used for the spanwise interpolation to the corresponding actuator point
using the pitch axis and aerodynamic center chord offsets from Figure 2.5, determine
the nodes from the FE model to be used for the interpolation to the PA and AC 'ghost'
nodes for both the fore and aft airfoil sections

- linearly interpolate from the fore and aft airfoil sections to the required spanwise
  location for the actuator point in question

A visual depiction of this process is provided by Figure 3.5.

![Figure 3.5. Notional blade section demonstrating the mode shape interpolation from FE node locations to desired points](image)

### 3.1.3 Bend-Twist Coupling

As previously described, bend-twist coupling is the effect of the blade deforming in torsion as
it deforms in bending. One of the advantages of developing the 5 MW blade’s FE model is
that no further modeling is required to capture the bend-twist coupling effect. The FE solver
takes care of the necessary calculations for each element’s material stiffness matrix (where
the bending-twisting cross coupling terms arise) and of the assembly of the global stiffness
matrix. All of this information is inherent in the modal matrix $\phi$. 

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Figures 3.6 and 3.7 show the blade twist and TE displacement for the first flapwise mode computed using Abaqus, compared against similar results computed by ANSYS using the Sandia model [28]. Comparison of the model results in these figures shows they are in reasonable agreement.

![First flap mode graph](image)

Figure 3.6. Twist as a function of span location for the first flapwise bending mode

### 3.2 Flow Model

The fluid mesh used for the different validation studies of the full ALM-FSI solver is a 2.6 million cell, Cartesian grid generated with OpenFOAM’s blockMesh and refineMesh utilities. The mesh parameters and boundary conditions are outlined in Tables 3.2 and 3.3, respectively. The initial step that creates cells measuring 126 m in all three directions is performed with blockMesh. The grid is then refined systematically in six levels using refineMesh, with the refinement extending radially and axially from the rotor apex at the origin of the domain. Each refinement level splits the cell edges in half, resulting in the values specified in Table 3.4; details about the dimensions of each refinement level are also provided.
Figure 3.7. Trailing edge flapwise displacement as a function of span location for the first flapwise bending mode.

in this table. A visual representation of the domain and the different grid refinement levels is shown in Figures 3.8 and 3.9. The cell size in the innermost refinement region plays an important role in determining the appropriate time step size for the simulation; within a single time-step the blade tip must not traverse more than one grid cell [17]. The first five refinement levels extend far downstream of the turbine to accurately resolve the tip vortices being shed as the blades rotate.

<table>
<thead>
<tr>
<th>Table 3.2. Fluid mesh parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Domain size, x-direction</td>
</tr>
<tr>
<td>Domain size, y-direction</td>
</tr>
<tr>
<td>Domain size, z-direction</td>
</tr>
<tr>
<td>Number of cells</td>
</tr>
<tr>
<td>Smallest cell size, ( \Delta x )</td>
</tr>
</tbody>
</table>
Table 3.3. Fluid mesh boundary conditions

<table>
<thead>
<tr>
<th>Boundary</th>
<th>Velocity</th>
<th>Pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>top</td>
<td>slip</td>
<td>zeroGradient</td>
</tr>
<tr>
<td>bottom</td>
<td>zeroGradient</td>
<td>zeroGradient</td>
</tr>
<tr>
<td>inlet</td>
<td>fixedValue</td>
<td>zeroGradient</td>
</tr>
<tr>
<td>outlet</td>
<td>zeroGradient</td>
<td>fixedValue</td>
</tr>
<tr>
<td>left</td>
<td>slip</td>
<td>zeroGradient</td>
</tr>
<tr>
<td>right</td>
<td>slip</td>
<td>zeroGradient</td>
</tr>
</tbody>
</table>

Table 3.4. Details of grid refinement levels

<table>
<thead>
<tr>
<th>Refinement Level</th>
<th>Upstream</th>
<th>Downstream</th>
<th>Radially</th>
<th>Cell Size</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3D</td>
<td>10D</td>
<td>5.5R</td>
<td>R</td>
</tr>
<tr>
<td>2</td>
<td>2.5D</td>
<td>10D</td>
<td>4R</td>
<td>R/2</td>
</tr>
<tr>
<td>3</td>
<td>2D</td>
<td>10D</td>
<td>3R</td>
<td>R/4</td>
</tr>
<tr>
<td>4</td>
<td>1.5D</td>
<td>10D</td>
<td>2.5R</td>
<td>R/8</td>
</tr>
<tr>
<td>5</td>
<td>0.5D</td>
<td>10D</td>
<td>1.5R</td>
<td>R/16</td>
</tr>
<tr>
<td>6</td>
<td>0.5D</td>
<td>1D</td>
<td>0.75R</td>
<td>R/32</td>
</tr>
</tbody>
</table>

3.3 Solver Validation

The structural solver developed for this research is used to evaluate the deformation of wind turbines blades in the presence of ABL turbulence. It is validated by comparing its response to that of a commercial solver such as Abaqus. The ALM-FSI solver is tested by disabling all of the new features and effectively reducing it to a loosely coupled, rigid blade solver. While there is no formal validation of the ALM-FSI solver because of a lack of experimental data, each component of the solver is validated against other solver results. It should be noted that for these cases the `ablActuatorDyMFoam` solver is not used to solve the flow. Alternatively, a modified version of OpenFOAM’s `pisoFoam` solver that incorporates the ALM formulation is used. The ALM formulation in each of these solvers is identical, the only differences between them being the conservation of momentum equation (Eq. 2.1). Specifically, the Coriolis and buoyancy force terms are excluded due to the absence of ABL turbulence.
The simulation parameters are outlined in Table 3.5.

<table>
<thead>
<tr>
<th>Simulation parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor diameter, D</td>
</tr>
<tr>
<td>Time step</td>
</tr>
<tr>
<td>Inflow</td>
</tr>
</tbody>
</table>

### 3.3.1 Deformation: Structural Solver vs. Abaqus Static Case

The structural solver’s goal is to accurately model the deformation of the turbine’s blades, such that the effect of blade flexibility can be properly coupled to the flow solver. Flapwise bending and torsion about the blade axis are the dominant modes of deformation for a wind turbine blade, thus comparisons are focused on flapwise displacement and blade twist. The results from an Abaqus static load simulation, where representative values of aerodynamic
forces and moments from the ALM solver are applied throughout the blade span, are compared to results from a modal static case solved in Matlab using:

$$\{\xi\} = \text{inv}([\omega_i] \oslash \{\phi\})^T \{F\},$$

(3.1)

from which the physical displacements $\{x\}$ are retrieved using Eq 2.6. The twist due to deformation is then computed in both cases for all points along the blade span using Eq 2.13. Comparisons of TE displacement, LE displacement, and twist along the span between the modal static solution and the Abaqus simulation are provided in Figures 3.10, 3.11, 3.12, 3.14, and 3.13, respectively. These results show that the modal static solution and the full-model static solution differ by less than 5% in the LE and TE displacements and between 5-20% in the twist for the majority of the blade span. This discrepancy is attributed to the truncation...
Figure 3.10. Comparison of TE displacement between the Abaqus static load simulation and the modal static solution.

Figure 3.11. Comparison of LE displacement between the Abaqus static load simulation and the modal static solution.

error introduced by computing the structure’s response with only 20 modes in the modal static case. Therefore, it is reasonable to suggest that the difference in solutions will decrease as the number of modes increases. Fortunately, the structural solver is designed to allow for
Figure 3.12. Comparison of twist between the Abaqus static load simulation and the modal static solution

Figure 3.13. Percent displacement error between the Abaqus static load simulation and the modal static solution

An arbitrary number of modes to be used to represent the structure’s response. The close agreement demonstrated here suggests that 20 modes is sufficient for the aerodynamic loads anticipated for this work.
3.3.2 Deformation: Structural Solver vs. Abaqus Modal Dynamic Case

It is important to validate the solver’s ability to accurately predict the structure’s response in a time-dependent simulation, given that this is its main purpose within the ALM-FSI solver. Therefore, a modal dynamic analysis is performed in both the structural solver and Abaqus CAE, using the same representative aerodynamic forces and moments as in Section 3.3.1, applied instantaneously at the start of the simulation. The comparisons between the results of these analyses are again focused on flapwise displacement and blade twist, per the reasons previously described. Comparisons of TE displacement, LE displacement, and twist at the blade tip between the modal static solution and the Abaqus simulation are provided in Figures 3.15, 3.16, 3.17, 3.18, and 3.19, respectively.

These results show that the structural solver is in very good agreement with Abaqus’ modal dynamic solver. The error in the steady-state value of tip displacement and twist is less than 1%, which demonstrates that the structural solver is working correctly and is capable of predicting the structure’s response to time-varying loads representative of those
Figure 3.15. Comparison of tip TE displacement between the Abaqus modal dynamic simulation and the structural solver solution.

Figure 3.16. Comparison of tip LE displacement between the Abaqus modal dynamic simulation and the structural solver solution computed in the ALM solver.
3.3.3 Gravitational and Inertial Forces

As previously described in Section 3.3.4, the gravitational, Coriolis, and centrifugal forces are computed using the principle of work-equivalent loads. To ensure that the algorithm for
the implementation of these forces is correct, it is verified against a static load simulation in Abaqus. The algorithm for each force implementation is tested separately, starting with the gravitational force. Results for the first comparison are shown in Figures 3.20 and 3.21, which demonstrate good agreement between the proposed algorithm of Equation 2.10 and the Abaqus solver.

The next algorithm checked is the one for centrifugal force calculation. As detailed in Equation 2.12, the nodal positions are required to compute the centrifugal force with the algorithm developed for this work. Results for the second comparison are shown in Figures 3.22 and 3.23, which shows that there is also good agreement between the algorithm of Equation 2.12 and the Abaqus solver. Evaluation of the Coriolis force implementation is not as straight-forward as this force depends upon calculated velocities at each node and the force due to this effect is significantly less than the other forces involved. Validation of the Coriolis force implementation is therefore left to the overall solver validation against FAST in Section 3.3.5.

Figure 3.19. Percent tip displacement error between the Abaqus modal dynamic simulation and the structural solver solution.
Figure 3.20. Flapwise displacement for all FE model nodes under gravitational force

Figure 3.21. Edgewise displacement for all FE model nodes under gravitational force

3.3.4 Modified ALM Solver vs. Original ALM Solver

Disabling all of the new features added to the ALM-FSI solver effectively makes it identical to the original ALM solver intended for rigid blade, loosely coupled simulations. This implies that the results from these cases should be identical, and serves as a first check in determining
that the modifications to the ALM source code have not had any adverse effects.

This study determined that the modified ALM solver with all new features disabled produces the same solution as the original, unmodified ALM solver. A comparison of the power generated and mean aerodynamic from each case is provided in Figures 3.24 and 3.25,
respectively. These results demonstrate that the modifications made to the original solver have not negatively affected the core functionality of the ALM to model a rigid rotor in a loosely coupled scheme.

**Figure 3.24.** Time history of power generated, unmodified ALM solver vs. modified ALM solver with loose coupling and flexibility disabled

**Figure 3.25.** Mean aerodynamic force as a function of span, unmodified ALM solver vs. modified ALM solver with loose coupling and flexibility disabled
3.3.5 ALM-FSI Solver vs. FAST

In order to test the ALM-FSI solver’s functionality for system damping, a quiescent flow test was performed to compare with FAST. This entails setting the rotor to a parked condition, setting the initial flow velocity in the domain to zero, and applying an initial displacement to the blade. When the blade is released at $t = 0$, it will oscillate for a period of time before returning to its undeformed, natural state. A test such as this provides another form of validation by comparing the results from the ALM-FSI solver to a commonly used and accepted tool such as FAST.

As the results in Figures 3.26 and 3.27 show, the flapwise tip displacement in each case does in fact converge towards zero as expected, and shows very good agreement in decay rate, which is indicative of system damping. The damping ratio for each case is estimated using the relation between logarithmic decrement and damping ratio for small damping values:

$$\zeta \simeq \frac{\delta}{2\pi}, \quad (3.2)$$

where $\zeta$ is the damping ratio and $\delta$ is the logarithmic decrement, which is computed by:

$$\delta = \log(x_1/x_2), \quad (3.3)$$

where $x_1$ is the displacement at the first chosen peak, and $x_2$ is the displacement at the second chosen peak. Using these relations, the damping ratio for the ALM-FSI results is approximately 1.39% of critical, and 1.26% for the FAST results, whereas the structural damping is approximately 0.5% of critical for both cases. The discrepancy between the values is a result of local cycle-to-cycle variations in amplitude from which the peak values were determined. The overall agreement of decay rate shown in Figure 3.26 is very good.

A second test to compare the differences between rigid and flexible blade solutions
between FAST and the ALM-FSI solver is performed, this time with the parameters outlined in Table 3.5. Note that the damping level for this test is increased to 2.5% of critical damping, based on the damping values for composite blades reported in [4]. For the ALM-FSI cases
the solver is restricted to using a loosely coupled approach, as well as preventing the blades from twisting due to deformation. Limiting the ALM-FSI solver’s features this way allows for an appropriate comparison to FAST, given that FAST only supports a loosely coupled approach and restricts the blade deformation to pure bending. Figures 3.28 and 3.29 show the estimated power output from FAST for both rigid and flexible blades, while Figures 3.30 and 3.31 show results for the ALM-FSI solver. Note that the fluctuations observed in Figure 3.31 are caused by the discretization scheme used in the ALM formulation. Note also that absolute comparisons between FAST and the ALM-FSI solver are not performed here as it is the effect of blade flexibility that is of interest. Thus, comparisons between rigid and flexible blade cases are made.

![Figure 3.28. Time history of power generated via FAST, rigid vs. flexible blades](image)

Figures 3.28 through 3.31 show that the difference in mean power output between the rigid and flexible blade cases for both FAST and the ALM-FSI is approximately 0.32%. These results show that there is good agreement regarding the relative difference in power output between rigid and flexible blade solutions for both FAST and the ALM-FSI solver.
3.3.6 Loose coupling vs. Tight coupling with Rigid Blades

One of the most important features of the ALM-FSI solver is the capability to perform tightly coupled simulations to ensure proper convergence of the flow field and structural displacements. Especially considering that the body force computed by the ALM is a function of the actuator
Figure 3.31. Time history of power generated via the ALM-FSI solver, rigid vs. flexible blades - closeup view

line’s position, which changes throughout the ALM-FSI simulations as the structure deforms and the rotor rotates. The simulations performed for the following validation study use the same grid as the one in Section 3.3.4. Results from this validation study, as shown in Figures 3.32 and 3.33, demonstrate that even in the case of uniform inflow throughout the domain and without the effects of blade flexibility, the tightly coupled solutions approach a different steady-state value than the loosely coupled solution. The starkest difference occurs from one to two subiterations, while there’s a minimal difference between two and three subiterations. Note that the transient response shown in these figures is caused by the rotor instantly beginning to spin at the specified RPM at the start of the simulation.

3.3.7 Summary

While there is no data readily available for FSI validations in this effort, each solver component is validated separately. The structural solver developed for the ALM-FSI solver is validated through comparisons with Abaqus. The ALM-FSI solver is validated against the ALM solver
Figure 3.32. Time history of power generated, loose coupling vs. increasing levels of tight coupling using a rigid blade analysis.

Figure 3.33. Time history of power generated, loose coupling vs. increasing levels of tight coupling using a rigid blade analysis.
Chapter 4  
Uniform Inflow

The validated ALM-FSI solver is used to conduct a preliminary set of analyses of the NREL 5MW turbine with uniform and steady flow. The objective of this analysis is to verify the functionality of each solver component, as well as to evaluate the effect of aerodynamic pitching moments, the importance of blade bend-twist coupling, and the need for tight coupling. Recall that the latter two items are generally excluded from similar reduced-order modeling tools currently available, as described in Section 1.2. It also serves as a stepping stone towards the more complex analysis involving ABL turbulence inflow.

4.1 Overview

Although uniform and steady inflow conditions are not entirely representative of what a wind turbine experiences during operation, it is possible to obtain some useful insight into how blade flexibility will affect a turbine’s response by performing an analysis under such conditions. Specifically, this type of analysis provides an initial understanding into how much of an effect the twisting motion of the blades will have on the AOA along the span of the blade. This is of utmost importance due to the fact that variations in AOA lead to variations in the aerodynamic loads experienced by the turbine. Not only does this affect the estimated power output from the turbine, it also can have a significant impact on the overall life of the turbine itself.
The grid and parameters used for all uniform inflow simulations in this chapter are identical to those first introduced in Section 3.3.4 for the validation studies. In addition, the structural damping level for the blades is 2.5% of critical, which is on the higher end of values used for composite blades [4]. A total of five simulations were performed, the details of which are outlined in Table 4.1. The ALM-FSI solver settings for these simulations are chosen such that the effect of each solver feature can be isolated and quantified when making comparisons of their results. To facilitate the discussion of the results, the majority of the analysis is focused on comparing the mean values of the variables in question at different span locations. The means are computed over the last 250 seconds of the simulation.

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Flexibility</th>
<th>Deformation twist</th>
<th>Pitching moment</th>
<th>Coupling</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>Tight</td>
</tr>
<tr>
<td>2</td>
<td>×</td>
<td>×</td>
<td>×</td>
<td>Tight</td>
</tr>
<tr>
<td>3</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>Loose</td>
</tr>
<tr>
<td>4</td>
<td>✓</td>
<td>×</td>
<td>✓</td>
<td>Tight</td>
</tr>
<tr>
<td>5</td>
<td>✓</td>
<td>✓</td>
<td>×</td>
<td>Tight</td>
</tr>
</tbody>
</table>

### 4.2 Results

#### 4.2.1 Power Output

The first part of the analysis focuses on the estimated power output by the turbine. The mean generated power values for each of the five different cases are detailed in Table 4.2. It is evident that when enabling all of the solver’s features, such as blade flexibility and tight coupling, the generated power is considerably lower compared to the other cases. Comparing case 1 (all features enabled) and case 3 (ignoring twist deformation) reveals just how much of a role the blade’s twisting deformation plays on the overall turbine’s response, given that the generated power for the latter is only approximately 0.18% greater than that of the rigid
blade case (case 2), as seen in Figures 4.1 and 4.2. This suggests that bending deformation contributes much less to the changes in power output compared to twisting deformation for uniform and steady flow. Furthermore, it is also evident from Figure 4.1 that the aerodynamic pitching moment also plays a significant role in the structure’s response.

Table 4.2. Mean generated power output for uniform and steady flow simulations

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Mean Generated Power (MW)</th>
<th>% Difference From Case 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.73</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>5.08</td>
<td>7.3</td>
</tr>
<tr>
<td>3</td>
<td>5.09</td>
<td>7.5</td>
</tr>
<tr>
<td>4</td>
<td>5.04</td>
<td>6.5</td>
</tr>
<tr>
<td>5</td>
<td>5.36</td>
<td>13.2</td>
</tr>
</tbody>
</table>

Figure 4.1. Comparison of generated power output for the different uniform and steady flow simulations

As detailed in Section 2.2.2, a positive pitching moment is that which pitches the blade away from the oncoming flow. As Figure 4.3 shows, the pitching moment coefficient, $C_m$, at all span locations is negative for case 1. Therefore, the pitching moment causes the blades to pitch into the wind, which leads to a lower power output for case 1. As it will be demonstrated
in Section 4.2.2, this decrease in power is attributed to decrease in the aerodynamic force experienced by the blades, caused by a decrease in AOA. Figure 4.1 also illustrates the need for a tightly-coupled solver, given that the effect of blade deformation on the flow is not accurately captured in a loosely-coupled solution. As the blade is deforming it is affecting the flow around the turbine, and thus affecting the aerodynamic forces used to compute the body force. Recall that this body force is what effectively simulates the presence of the turbine in the domain. Therefore, if the flow velocity and displacement fields are only loosely-coupled, then the effect of the turbine on the flow is not accurately quantified.

4.2.2 Aerodynamic Load and AOA

The second part of the analysis focuses on the differences in AOA and, consequently, aerodynamic loads between the different uniform flow cases. The differences in mean generated power detailed in Table 4.2 are directly related to the differences in aerodynamic loads and AOA experienced by the blades. Specifically, the tangential component of force, which is
Figure 4.3. Comparison of mean Cm for the different uniform and steady flow simulations essentially the rotor torque-creating component, is of interest. This component of force at each actuator point is directly linked to the AOA at that point, meaning that if the force at a given actuator point decreases, it is reasonable to conclude that the AOA has also decreased. The mean tangential force as a function of span is shown in Figure 4.4, while the mean AOA as a function of span is shown in Figure 4.5. These figures confirm that the lower estimate for generated power in case 1, compared to the other cases, is due to a decrease in the aerodynamic load experienced by the blades. Special emphasis is placed again on the blade’s twisting deformation, given that the results from Figures 4.4 and 4.5 provide further evidence to the claim that bending deformation by itself does not have a significant impact on the aerodynamic loads. Furthermore, the effect of the aerodynamic pitching moment is evident in Figure 4.5, which shows that there is approximately a one degree difference between case 1 and case 5 in the last 15 m of the blade span. It is thus clear that the pitching moment contributes to the reduction in aerodynamic loads in case 1.
4.2.3 Deformation

The last part of the analysis focuses on both the translational and torsional deformation of the blades, and expands on the discussion from Sections 4.2.1 and 4.2.2 regarding this defor-
mation’s role in the overall turbine’s response. As the previous sections in this chapter have shown, the blade’s twisting deformation significantly affects the estimated power generated by the turbine. This is due to the fact that the twist due to deformation contributes to the variations in AOA, which consequently affect the aerodynamic loads on the blades. Moreover, previous results have shown that the pitching moment also has an effect on the aerodynamic loads. Namely, that alleviates the load on the blades, leading to a reduction in estimated power generated. This motion of pitching into the wind suggests that the blades experience a positive twist due to deformation, given the notation outlined in Section 2.3. Table 4.3 and Figure 4.6, which show the twist distribution as a function of span, confirm that this is indeed the case.

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Twist-Tip (deg)</th>
<th>Twist-75% span (deg)</th>
<th>Twist-50% span (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.77</td>
<td>0.5</td>
<td>0.21</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>3</td>
<td>0.76</td>
<td>0.48</td>
<td>0.20</td>
</tr>
<tr>
<td>4</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>5</td>
<td>-0.59</td>
<td>-0.52</td>
<td>-0.23</td>
</tr>
</tbody>
</table>

The effect of bend-twist coupling and tightness of coupling in the simulations demonstrated in this section for uniform and steady flow cases will be of paramount importance for the simulations with ABL turbulence, where the aerodynamic loads will fluctuate both spatially and temporally.

### 4.3 Summary

A preliminary analysis of the NREL 5 MW turbine with uniform and steady flow has been performed to verify the functionality of the ALM-FSI solver’s components. In addition, the effect of aerodynamic pitching moments, importance of blade bend-twist coupling, and
the need for a tightly coupled framework have been evaluated. The following chapter will introduce the analysis of the 5 MW turbine in the presence of ABL turbulence inflow, and using a similar set of comparisons as in this chapter the effects of the ABL on the performance of the turbine are studied.
Chapter 5  
ABL Turbulence Inflow

The main objective of this thesis is to assess the importance of blade flexibility on the performance of a wind turbine operating in the presence of ABL turbulence. More specifically, the goal is to analyze the effects of blade flexibility on the estimated aerodynamic loading and power output using the ALM-FSI solver demonstrated in previous chapters. This chapter discusses the application of the ALM-FSI solver to simulations for the NREL 5 MW wind turbine with ABL turbulence inflow. Compared to the previous chapter, only two simulations are performed for the ABL analysis. The two simulations performed are case 1 and case 2 from Table 4.1.

5.1 Mesh Generation and Boundary Conditions

The fluid mesh used in these analyses is not as straightforward as the one used for the uniform inflow analyses. Rather than systematically refining the mesh as before, the mesh was created from the bottom-up in the commercial software Pointwise by Lavely [23] in support of his ongoing PhD research, also part of the CWF project. It consists of an inner unstructured region containing 1,013,346 tetrahedral cells and an outer structured region containing 719,568 cells, for a total just shy of 1.75 million cells. A closeup view of the mesh is provided by Figure 5.1. The mesh extends approximately 480 m in the downstream direction, 494 m in the transverse direction, and 259 m in the vertical direction.
A precursor large eddy simulation (LES) of a day-time, moderately convective ABL is run using the LES code developed by Sullivan and Patton [33], which uses a pseudo-spectral algorithm. The domain for the simulation extends 5 km in both horizontal directions and 2 km in the vertical direction. Furthermore, the parameters of the simulation are chosen such that they are similar to the conditions found in the Great Plains or Midwest regions of the United States, and to obtain an approximate mean velocity of 14 m/s at the hub height for the 5 MW turbine [23]. The boundary conditions for the ABL mesh are outlined in Table 5.1.

Once the simulation reaches a quasi-stationary state, planes of data two rotor diameters upstream of the turbine location are extracted to be used as inflow for the ABL simulations performed with the ALM-FSI solver. The planes extracted contain velocity, temperature, and turbulent kinetic energy information, which is interpolated to regular half-second intervals to facilitate its implementation as boundary conditions in OpenFOAM.
Table 5.1. ABL fluid mesh boundary conditions

<table>
<thead>
<tr>
<th>Boundary</th>
<th>Velocity</th>
<th>Pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>top</td>
<td>zeroGradient</td>
<td>fixedMeanValue</td>
</tr>
<tr>
<td>bottom</td>
<td>Custom (surface stress)</td>
<td>zeroGradient</td>
</tr>
<tr>
<td>inlet</td>
<td>timeVaryingMapped</td>
<td>zeroGradient</td>
</tr>
<tr>
<td>outlet</td>
<td>zeroGradient</td>
<td>fixedMeanValue</td>
</tr>
<tr>
<td>sideInlet</td>
<td>zeroGradient</td>
<td>fixedMeanValue</td>
</tr>
<tr>
<td>sideOutlet</td>
<td>zeroGradient</td>
<td>fixedMeanValue</td>
</tr>
</tbody>
</table>

5.2 Results

5.2.1 Deformation

The first part of the analysis focuses on the translation and torsional deformation of the blades, given that the main aim of this work is to assess the importance of blade flexibility on the wind turbine’s loading in the presence of ABL turbulence. Furthermore, the response of the blades is expected to be significantly different than the response from the previous chapter where only uniform and steady flow is used. This is due to the fact that not only does the ABL turbulence inflow change in time, it also changes in space.

Recall that it was determined in Chapter 4 that the bend-twist coupling effect causes the blades to pitch into the wind as they deform downstream, effectively helping to alleviate the load on the blades in the process. A comparison of Figures 5.2 and 5.3 reveals that the response of the blades to the two different types of inflow is significantly different, as expected. There is one particular time period that stands out in the former figure, roughly around 100 to 120 seconds into the simulation. In this period, the blade appears to be deforming upwind rather than downwind, which is a behavior that is not experienced by the blades under uniform and steady flow. A closeup view of this time period is provided by Figure 5.4.

Given the results from Chapter 4 regarding the relationship between flapwise displacement and blade twist (that the blade will twist into the flow as it bends downstream), a similar sort
Figure 5.2. Flapwise tip displacement of blade 1 for flexible blade case with ABL turbulence inflow of behavior is predicted for the ABL simulations. However, it appears that this is not entirely the case. As Figures 5.5 and 5.6 demonstrate, the overall trend in the aforementioned period of interest (100 to 120 seconds) is for the blade to twist into the wind when it is bending upstream.

Figure 5.3. Flapwise tip displacement of blade 1 for flexible blade case with uniform and steady flow
Figure 5.4. Close up view of Figure 5.2

Figure 5.5. Comparison of flapwise bending and twist at the tip of blade 1 for flexible blade case with ABL turbulence inflow - 100 to 110 seconds

Although this is the overall trend in this 20 second period, notice that there are noticeable oscillations at various points in time. For example, from approximately 112 to 113 seconds there appear to be at least five of these oscillations, where the blade momentarily pitches into the wind as it deforms upwind before pitching out of the wind again. It appears that there
are two different opposing mechanisms in play here as the blade deforms upwind: one forcing it to pitch out of the wind, and another forcing it to pitch into the wind. As previously shown in Section 3.1.3, Figure 3.6 shows that for the first flapwise bending mode (global mode 1), downstream flapwise bending results in the blade pitching into the wind. However, Figure 5.7 shows a significantly different response for the first edgewise mode (global mode 2). For this particular mode, the parts of the blade where flexibility is more noticeable experience a twisting motion which, for the most part, pitches the blade out of the wind. It is possible that mode 2 is contributing to the blade’s response to a large enough extent that it is causing the back and forth into the wind/out of the wind twisting motion. To verify whether or not this is the case, the modal participation factors over time for the uniform and steady flow case are compared to those from the ABL case. It is clear from Figure 5.8 that the first flapwise mode dominates the response over the first edgewise mode. Note that the remaining modes from the basis set are not featured in Figure 5.8 because their amplitudes are orders of magnitude smaller compared to the first mode. However, Figures 5.9, 5.10, and 5.11 show
that the first edgewise mode’s contribution to the response is of the same order as the first 
flapwise mode in the ABL case.

![Figure 5.7. Twist as a function of span location for the first edgewise bending mode](image)

![Figure 5.8. Modal participation factors for flexible blade case with uniform and steady flow](image)

Given this finding, in addition to the insight provided by Figure 5.7, it is reasonable to 
conclude that the interaction between the first flapwise mode and the first edgewise mode is
Figure 5.9. Modal participation factors for flexible blade case with ABL turbulence inflow

Figure 5.10. Closeup view of Figure 5.9 - 100 to 110 seconds

responsible for the response shown in Figures 5.5 and 5.6.
5.2.2 Aerodynamic Load and Power Output

Continuing to focus on the 20 second period highlighted in Section 5.2.1, the next part of the analysis involves determining the impact of the blade’s bending and twisting on the aerodynamic loads. It has been shown in Figure 5.4 that around 100 seconds the tip of blade 1 bends back toward the rotor plane, up until approximately three seconds later when it begins to bend upwind. During those three seconds, the tip of the blade unloads itself as demonstrated by Figure 5.12. This reduction in load is evidently not caused by the blade bending or twisting, given that a similar behavior is observed in the rigid blade case. Therefore, it is reasonable to assume that it is caused by a considerable change in the flow characteristics during that period of time. This is confirmed by Figures 5.13 and 5.14, where the contours of instantaneous velocity in the downstream direction at the rotor plane show that in both the rigid and flexible blade cases, the blade in question (blade 1, in blue) passes through a low-speed region at 103 seconds. Recall that the blade surface is not explicitly modeled in ALM simulations; the lines representing the blades in these
images are superimposed on the velocity contours for the purpose of providing the reader with an approximate location of the rotor’s position. A similar unloading event takes place around 107-108 seconds, as the blade passes through another low-speed region, as observed in Figures 5.15, 5.16, and 5.17. These images help reinforce the idea that the velocity field is changing not only temporally, but also spatially. Moreover, these fluctuations in space and time have a strong impact on the performance and longevity of the turbine. It is reasonable to say that if these sort of events take place over the span of a few seconds and can significantly affect the loading on the rotor blades, they can cause large load fluctuations and could impact fatigue performance over long periods of time.

![Tangential force at the tip](image)

**Figure 5.12.** Comparison of tangential force at the tip of blade 1 with ABL turbulence inflow - 100 to 110 seconds

Although these unloading events occur in both the rigid and flexible blade cases, Figure 5.12 shows that in the flexible case, blade 1 experiences lower magnitude loads. As can be inferred from Figures 5.13 through 5.17, the other two blades also experience unloading events, albeit at different instances in time. Moreover, these two blades also experience lower magnitude loads in the flexible case, similar to blade 1. This can be seen in Figures 5.18 and 5.19. This
Figure 5.13. Slice at the rotor plane showing contours of instantaneous velocity in the downstream direction for the flexible blade case with ABL turbulence inflow at $t = 103$ s

Figure 5.14. Slice at the rotor plane showing contours of instantaneous velocity in the downstream direction for the rigid blade case with ABL turbulence inflow at $t = 103$ s
Figure 5.15. Slice at the rotor plane showing contours of instantaneous velocity in the downstream direction for the flexible blade case with ABL turbulence inflow at $t = 107$ s

Figure 5.16. Slice at the rotor plane showing contours of instantaneous velocity in the downstream direction for the flexible blade case with ABL turbulence inflow at $t = 107.5$ s
ultimately leads to the estimate of mean power generated in the flexible blade case to be 11.3% lower than in the rigid blade case, as evidenced by Figures 5.20 and 5.21.

**Figure 5.17.** Slice at the rotor plane showing contours of instantaneous velocity in the downstream direction for the flexible blade case with ABL turbulence inflow at $t = 108$ s

**Figure 5.18.** Comparison of tangential force at the tip of blade 2 with ABL turbulence inflow - 100 to 110 seconds
Figure 5.19. Comparison of tangential force at the tip of blade 3 with ABL turbulence inflow - 100 to 110 seconds

Figure 5.20. Comparison of estimated power output between rigid blade and flexible blade cases with ABL turbulence inflow
5.2.3 Out of Plane Bending Moment

Non-torque loads on a wind turbine, primarily caused by out-of-plane bending moments, cause misalignment of gears and bearings, ultimately leading to these components failing sooner than expected in the turbine’s gearbox [6]. Therefore, the last part of this analysis focuses on these moments, and how they evolve over time for both rigid blades and flexible blades.

Time histories of out-of-plane bending moment for each of the rotor’s three blades are shown in Figures 5.22 through 5.24, from which it is evident that the magnitude of these moments is lower in the flexible blade case compared to the rigid blade case. Moreover, it is also clear that the rotor is experiencing these moments with magnitudes on the order of the torque loads. Although the magnitude of the out-of-plane bending moment for the blades may be lower when the blades can deform, a closer look at the 100 to 120 second time period, which was the focus of the 2 previous parts of the analysis, could potentially reveal some important details. Figure 5.25 shows a closeup view of the out-of-plane bending moment
comparison during this 20 second interval.

Figure 5.22. Comparison of out-of-plane bending moment between rigid blade and flexible blade cases with ABL turbulence inflow - blade 1

Figure 5.23. Comparison of out-of-plane bending moment between rigid blade and flexible blade cases with ABL turbulence inflow - blade 2

Note that there are some instances in time where it appears that in the flexible blade case, the changes in the magnitude of the moment take place over a shorter period of time.
**Figure 5.24.** Comparison of out-of-plane bending moment between rigid blade and flexible blade cases with ABL turbulence inflow - blade 3

**Figure 5.25.** Comparison of out-of-plane bending moment between rigid blade and flexible blade cases with ABL turbulence inflow - blade 1, closeup view

compared to the rigid blade case. The changes in magnitude might be different, but the time it takes for these changes to occur is also different. This sharper gradient in moment magnitude could potentially play a significant role in future studies focused specifically on
gear and bearing failure. However, such an analysis is out of the scope of this work.

5.3 Summary

An analysis of the NREL 5 MW turbine with ABL turbulence inflow has been performed to understand the role of blade flexibility on the turbine’s loads and generated power. The initial insight from Chapter 4 regarding the relationship between blade bending and twisting has been further explored, and has been found to be heavily influenced by the spatially and temporally varying velocity field due to ABL turbulence. These variations appear to cause opposing twisting motions from the first flapwise bending mode and the first edgewise bending mode. The loads on the blades and the turbine’s estimated generated power are both lower in the flexible blade case compared to the rigid blade case, which is consistent with what was found in Chapter 4.
Chapter 6
Summary, Conclusions, and Future Work

6.1 Summary

The objective of this research was to analyze the effects of blade flexibility on the loadings experienced by the rotor blades, and the consequences of this on the estimated power generated by the turbine. A partitioned, reduced order, ALM-FSI solver capable of performing tightly coupled simulations was developed using an OpenFOAM-based ALM solver and an author-developed, modal-based structural dynamics solver. A strong emphasis was placed on the solver’s ability to incorporate the blade’s twisting motion to ensure its bend-twist coupling features were captured properly. The tight coupling and incorporation of blade twisting are solver features that are currently not widely available in wind turbine analysis software. In lieu of experimental data for validation, the different components of the solver were systematically checked via a series of comparisons to existing software.

Prior to beginning the analysis with ABL turbulence inflow, a uniform and steady flow analysis was performed to gain initial insight into the rotor’s structural response and its effect on the turbine’s aerodynamics. It was determined that there is a significant difference in the turbine’s operation when blade flexibility is enabled. Mainly, the aerodynamic loads and estimated power output have lower values in the flexible blade cases compared to the rigid blade cases. This analysis also revealed that the majority of the difference between rigid blade and flexible blade cases is attributed to the blade’s bend-twist coupling. A non-
discernible difference was observed when comparing rigid blade results to flexible blade results without blade twisting. The importance of including the aerodynamic pitching moment in the formulation of the structural solver was also demonstrated. The response of the rotor is substantially different when this load is omitted, which leads to an incorrect solution.

Upon completion of the preliminary analysis, the main analysis focusing on ABL turbulence inflow was conducted. Results showed that the structural response of the rotor is significantly more complex than in the uniform and steady flow analysis. Such a response was anticipated due to the nature of the unsteady loadings caused by temporal and spatial fluctuations in the ABL, and confirmed by studying the blade’s translational and torsional deformation. Special focus was placed on a 20 second period of the simulation, where the rotor deforms upwind, a phenomenon not previously encountered in the uniform and steady flow analysis. The ABL analysis revealed that blade flexibility plays a substantial role in the overall operation of the turbine, given its effects on the turbine’s loads and power output.

6.2 Conclusions

The research described in this thesis has demonstrated the following:

1. wind turbine blade flexibility plays an important role in the aerodynamic loads and power output

2. a tightly coupled, ALM-FSI solver is required to accurately capture the effects of the rotor on the flow, and vice versa. This is even the case for rigid blades, where a substantial difference was revealed when comparing loosely coupled and tightly coupled solutions for the ALM solver

3. the majority of the difference in response between the flexible and rigid blade cases results from twisting deformation of the blades. Without the ability to twist, the blade’s
response is significantly different, and thus negatively affects the solution.

6.3 Future Work

Although the different solver components were verified systemically by comparing them to existing software, proper validation requires some form of comparison to experimental data. In addition, the FE model could be improved by obtaining further information on the blade’s composite layup and damping. This would allow the model to be a more exact representation of the physical wind turbine blade under consideration. The simulations involving ABL turbulence inflow should be allowed to run for an extended period of time to allow both the structure and the fluid to reach their respective converged solutions. This could then be used to compare the cyber data to actual experimental data obtained from a wind turbine in the field.

The effects of geometrical nonlinearities in the structural response should be studied, and comparisons should be made between that nonlinear response and the linear response used in this work. Nonlinear effects could potentially be an important factor to consider for the development of new blade models and/or turbines. Furthermore, blade-resolved FSI simulations should be performed using the FE model developed for this work to compare to the ALM-FSI solver and further refine this reduced-order modeling tool.
Bibliography


