SAND2009-2192C

Structural Dynamics Analysis and Model Validation of Wind Turbine Structures

D. Todd Griffith*
Sandia National Laboratories†, Albuquerque, New Mexico 87185

The focus of this paper is the development of validated structural models of wind turbine structures and their substructures. A typical modern wind turbine is a large structure composed of a single tower, a nacelle located atop the tower which houses the drive train mechanical components, and three rotor blades. In very broad terms, a wind turbine design team must consider the dynamic response of the full system in the design process along with detailed design for each individual substructure. Blades are a critical substructure of a wind turbine as they carry large loads in capturing the energy from the wind, and must be designed to maximize performance and reliability while minimizing their cost. The work summarized in this paper has two major parts. First, we discuss the structural dynamics analyses that are performed to design a modern wind turbine structure. Secondly, a methodology for validating blade structural models is presented. A number of blades incorporating innovations for large blade technologies have been designed and tested at Sandia National Laboratories in recent years. Results from recent tests and validation of models for these blades will be presented.

I. Introduction

THE focus of this paper is the development of validated models of wind turbine structures and their substructures. Validated models are useful for improved decision making for new designs of wind turbines. There are a number of unique challenges in testing and modeling wind turbine structures including novel blade composite material architectures, complex aerodynamic loading, and rotational effects of the rotor. Model validation is a comprehensive undertaking which requires carefully designing and executing experiments, proposing appropriate physics-based models, and applying correlation techniques to improve these models based on the test data. The result of the validation process is an understanding of the credibility of the model and the modeling process for making useful analytical predictions.

There are two common types of wind turbine structures, which include horizontal axis wind turbines (HAWTs) and vertical axis wind turbines (VAWTs) (See Figure 1). The vast majority of modern wind turbines is of the HAWT type, which consists of a tower, nacelle, and blades. A typical machine has three blades, which form the rotor. The housing on the top of the tower is the nacelle which contains the mechanical drive train components such as the gearbox and generator. HAWTs are designed with a number of important control systems, and we now discuss them as they relate to turbine operation. Variable speed controllers are designed to maximize energy capture from the cut-in wind speed up to the wind speed at which rated power is achieved. Above this rated power wind speed, blade pitch control is used to maintain power at the rated design level. At wind speeds above rated, the blades are pitched out of the wind in an effort to maintain the constant rated power output. Typically, this has been accomplished with collective pitch control, with all three blades pitching in a coordinated fashion; however, more recent designs have considered independent blade pitch control. Additionally, yaw control of the rotor is used to orient the rotor into the direction of the wind for increased energy capture. Furthermore, each of these controllers can be used as a safety feature to avoid large, potentially damaging, loads on the blades. A variety of wind loading cases and controller scenarios are considered to certify a wind turbine design. Good general references on turbine operation and design include References 1 and 2.

---

* Technical Staff, Analytical Structural Dynamics Dept., MS 0557, Senior Member AIAA.
† Sandia is a multiprogram laboratory operated by Sandia Corporation, a Lockheed Martin Company for the United States Department of Energy's National Nuclear Security Administration under contract DE-AC04-94AL85000.

Figure 1. (a) Horizontal Axis Wind Turbines (left), (b) Vertical Axis Wind Turbine (right)

The trend in the industry has been and continues to be growth in machine size. The evolution of machine size over the past 24 years is illustrated in Figure 2. A modern challenge lies in designing longer blades to increase the rated power output. The power rating of a machine is proportional to the swept area of the rotor, or roughly the blade length squared. Thus even small increases in blade length can greatly increase the amount of energy capture. The desire to increase power output with larger machines challenges structural designers to design blades with high structural efficiency to limit the weight growth associated with larger blades. Currently, machines with rated power of 2 MW and greater are common and machines with ratings of 5 MW are available.

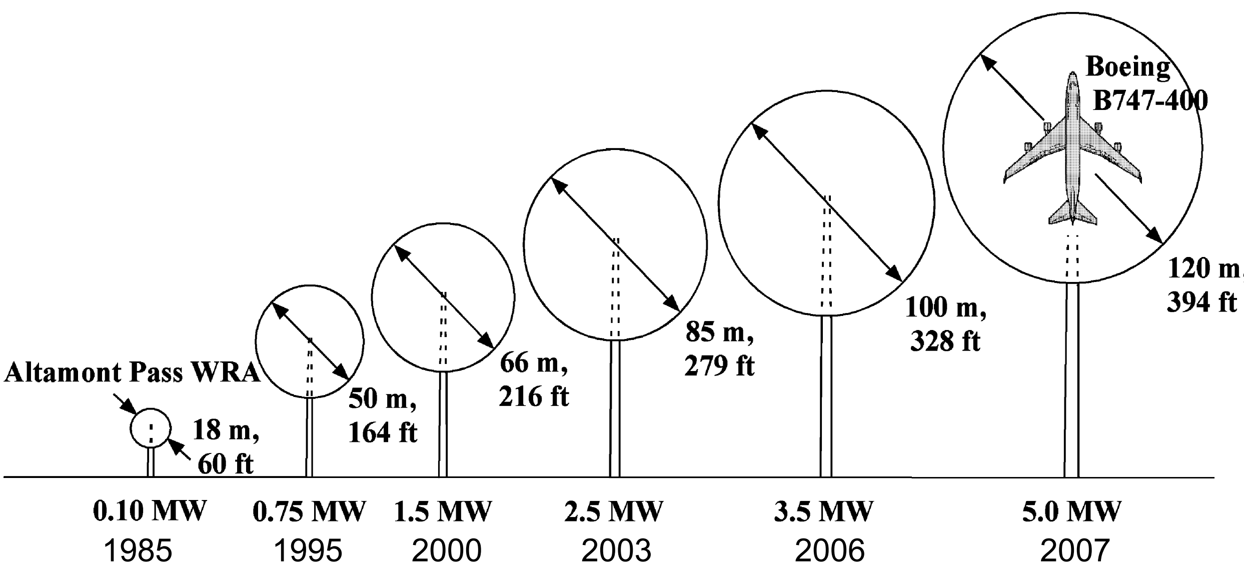

Figure 2. Wind Turbine Size Growth Trends

The challenge in designing modern machines is considerable. The cost of building prototype blades is quite large, and there is a strong need for the use of predictive analysis for early design studies to replace some aspects of the traditional build, test, and redesign process. The process of model validation supports improved predictive modeling capabilities, and the hope is that validated predictive modeling tools can be used to reduce development time and prototyping costs.

For blades, one is faced with a challenging structural design to achieve the very high structural efficiency of modern wind turbine blades. A typical blade is composed of a hollow airfoil, reinforced with “box-beam” or “I-beam” type construction. These reinforcements are comprised of spar caps along the interior of the blade top and bottom surface, which support the loads of the internal shear web(s). The shear webs are oriented nominally normal to the spar caps and carry the flap-wise loads. The leading and trailing edge sections are designed with balsa or foam to form the desired airfoil shape and thicken these unsupported panels to prevent buckling. Several blade cross-sections are shown in Figure 3, which have a single shear web and a short spar cap. The composite construction of wind turbine blades is unique. Their construction and material selection has many similarities with boat construction rather than modern aerospace composites. Typical materials include balsa and foams as core sandwich materials and unidirectional fiberglass (or carbon fiber in some of the more recent designs) as major load carrying elements in the spar caps and leading/trailing edge reinforcements. Many of the manufacturers use vinyl ester or polyester resins. A major driver for wind turbine structures is cost, thus selective use of carbon fiber. Of course, blade weight must be minimized to not only save material cost, but principally to avoid large gravitational loadings. The challenge of designing and modeling modern blades is accentuated by tight margins on performance and cost which necessitate the need for innovative structural mechanics concepts.

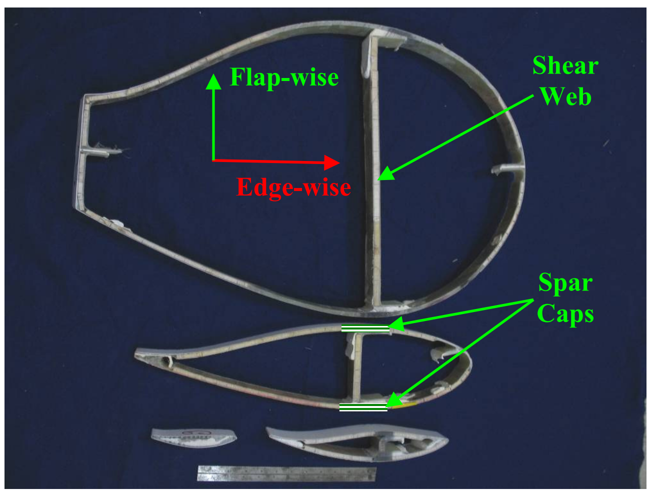

Figure 3. Examples of Wind Turbine Blade Cross-sections

A wind turbine experiences large dynamic loads during operation as the aerodynamic and gravitational loads are changing with time. Thus, it is important that a model of a wind turbine blade accurately predict structural response due to dynamic loads. Structural dynamics analysis of wind turbine structures is very important, and structural dynamics models are needed for turbine substructures as well as the full system. Modal testing, an experimental method for measuring the natural vibration characteristics of a structure, is used to evaluate the structural dynamics model. References 3 and 4 provide good general references on experimental and analytical modal analysis.

The organization of the paper is as follows. First, a description of a number of the structural dynamics analyses performed in designing wind turbine structures is discussed. This description is typical of blade design, although it is likely not comprehensive as different organizations may have different processes. Next, the issue of validation of these models is addressed by presenting a methodology for blade and turbine model validation. Then, results for a blade structural model validation effort will be detailed. The blades tested in this program will be described and relevant tests will be summarized. A discussion of important test considerations, in particular the root boundary condition, and evaluation of blade structural models through calibration studies and validation tests is provided.

II. Structural Dynamics Analysis of Wind Turbine Structures

It is important, in the design of wind turbine structures, that adequate models are used in design to understand operation and lifetime performance. These issues are accentuated as turbines grow larger and more costly; particularly for offshore machines where high reliability will be required. In the design process, resonant response is considered at all levels in the modeling process from blade substructure models to full system models. As a result of the architecture for wind turbine blade construction, an analyst is interested in several types of resonant responses when developing blade structural models. These include bending modes, torsional modes, and localized resonances. Traditionally, only the low-order bending modes have been of interest. However, some modern blade designs take advantage of torsional dynamics as a passive load shedding mechanism, thus these coupled modes are also drawing interest with analysts and designers.

The primary loads on the system come through the blades. The incoming wind combined with the motion of the rotor produces aerodynamic loads, lift and drag components, on the rotor blades. Additionally, there are gravitational loads on the blades, although they have not been a primary design driver in the past. Characterization of the aerodynamic loading is also an important area of research because it must be adequately modeled to perform transient dynamics calculations. Transient response simulation of a wind turbine is quite important for understanding turbine response for the wide range of wind input conditions and system operating scenarios needed to be evaluated to successfully certify a design.

At the full structure level, one combines the three major substructures, the tower, nacelle, and blades. Modal analysis can be performed for the parked rotor to evaluate the structural model without consideration of rotational effects. This simplifies the analysis and can serve as a baseline evaluation of the structural model. Since the natural frequencies of the coupled tower/blade modes vary with rotational speed of the rotor, modal analysis must be performed over the range of desired operating speeds of the machine. These calculations are useful in designing to avoid resonance conditions, which can occur when the operating speed or an integer multiple of the operating speed of the machine is near a resonant frequency. Thus adequate modeling of the rotational effects of the rotor is quite important in the design of wind turbines. Resonance issues can be evaluated by creating Campbell diagrams, which graphically plot the resonant mode frequencies as a function of rotor speed[1]. Integer multiples of the operating speed are plotted as straight lines, and the intersection of these modal frequencies and operating speeds indicate the operating speeds at which resonance is possible.

We now briefly review the kinematics of the rotor blades in an effort to describe the important rotational effects. We neglect the motion of the tower in this illustration. The velocity level kinematics expression for a point on a rotor blade is given by:

$$\begin{aligned} \underline{v}^N &= \frac{{}^N d}{dt}\left(\underline{r}+\underline{u}\right) \\ &= \underline{\dot{u}}^B + {}^N\underline{\omega}^B \times \underline{r}^B + {}^N\underline{\omega}^B \times \underline{u}^B \end{aligned} \tag{1}$$

where $\underline{v}^N$ is the inertial velocity of a point on the rotor. Note, that the B frame is fixed in the rotor. The position vector locating this point in the B frame for the undeformed rotor is denoted by $\underline{r}$, and $\underline{u}$ is the deflection from this point in the B frame. The constant angular velocity of the rotor is ${}^N\underline{\omega}^B$. Rotor speed can vary in operation; however, for modal analysis it is chosen to be a constant.

The acceleration level kinematics expression for a point on a rotor blade is given by:

$$\begin{aligned} \underline{a}^N &= \frac{{}^N d}{dt}\left(\underline{v}^N\right) \\ &= \underline{\ddot{u}}^B + 2\left({}^N\underline{\omega}^B \times \underline{\dot{u}}^B\right) + {}^N\underline{\omega}^B \times \left({}^N\underline{\omega}^B \times \underline{u}^B\right) + {}^N\underline{\omega}^B \times \left({}^N\underline{\omega}^B \times \underline{r}^B\right) \end{aligned} \tag{2}$$

where $\underline{a}^N$ is the inertial acceleration of a point on the rotor. The acceleration is the sum of four contributions. The first term on the right hand side of (2) represents the acceleration of a point on the structure relative to the undeformed rotor in the rotating B frame. The second term is the Coriolis term, which results in a skew-symmetric damping matrix in the system equations of motion. The third term represents centrifugal softening, and results in a stiffness matrix term in the equations of motion. The fourth and final term represents constant centrifugal forces, which preload the structure at the constant spin rate. Choosing to perform the modal analysis in the rotating frame simplifies the analysis by avoiding consideration of large rigid body motions. These kinematic expressions are the basis for development of finite element or assumed modes based models for performing modal analysis. The resulting model based on (2) is a conservative gyroscopic system, which can be analyzed by existing methods[5]. First, one spins up the structure to the desired constant operating speed and computes the tension stiffening effect of the constant centrifugal loads. Then, modal analysis is performed about this pre-stressed state in the rotating frame.

Recently, some new techniques have been considered for performing modal analysis on wind turbines. Floquet modal analysis[6] has been investigated for determining the modal parameters in order to account for the periodic coefficients in the equations of motion. And, the multi-blade coordinate transformation[7] has also been considered to map the motion of the rotor blades into the fixed frame of the tower to perform dynamics analysis including modal and stability analysis.

In addition to structural modeling and analysis, aeroelastic phenomena are quite important for wind turbine structures. Flutter and stall can cause instability and must be adequately addressed in the design process. This requires modeling of aerodynamic loads coupled with structural models incorporating the rotational effects discussed in this section. Again, one solves a complex eigenvalue problem (as is done in modal analysis described above), and the practice for determining if a mode will go unstable in operation has been to determine the operating speed at which damping in the mode goes to zero.

III. Model Validation of Wind Turbine Structures

We now move to the second major topic of this paper, a model validation methodology for wind turbine structural models. Model validation is a comprehensive undertaking to determine the usefulness of a model for its intended purpose. One starts by proposing an appropriate physics-based model. Then, tests are designed to acquire data for calibration of model parameters or evaluation of the validity of the model for its intended purpose. Typically, one set of calibration experiments provide data to calibrate a model, and an independent set of "validation experiments" are performed to assess the adequacy of the calibrated model. Each of the components of model validation is briefly discussed in the following sections, with emphasis on validation of blade structural models. Our motivation is the development of adequate blade models for inclusion in the full assembly wind turbine structural model.

A. Model Development

There are a number of important considerations for developing a useful blade structural model. These include the form of the model, its correspondence with the test article and the conditions of the test, and the parameters that comprise the model. The primary design drivers for a wind turbine include system resonance, root strains, blade tip-tower clearance, fatigue, and blade panel buckling. The ability to accurately predict these often dictates the type of model that is selected.

With regard to model fidelity, one can choose low fidelity beam type models or high fidelity solid models to evaluate the design drivers. The choice depends, of course, on the type of analysis that is desired. First, we describe the use of low fidelity beam models which are used to perform modal analysis on blade substructures in addition to modal analysis of the full turbine assembly. Modal analysis of the full assembly provides a means to evaluate the mass and stiffness properties of the model, as well as the rotational effects on the modal parameters. Low fidelity turbine models are also used for transient dynamics calculations to evaluate tower clearance and blade root bending moments. Codes for transient dynamics calculations include the MSC.Adams software[8] and FAST (Fatigue, Aerodynamics, Structures, and Turbulence)[9]. FAST is a tool specially designed for transient simulation of wind turbine performance. FAST models the elastic elements of the structure (tower and blades) by truncating the structural dynamic response to a limited, fixed set of modal degrees of freedom for the low-order bending modes of the tower and blades. The MSC.Adams capability on the other hand offers a higher fidelity blade modeling capability; however, the turnaround time for this analysis is significantly longer. For transient dynamics calculations using either code, the transient wind input is provided by the AeroDyn code[10].

The blades require higher fidelity analysis predictions in some cases, which simple beam type models cannot provide. Highly detailed solid models offer analysis of buckling of unsupported blade panels, prediction of detailed stresses at joints or material boundaries, and evaluation of localized resonance. Development of a high fidelity model can be expedited by using specialized modeling codes. To aid model development, a modeling tool has been developed at Sandia Labs which automates a great deal of the model development process specifically for wind turbine blades. A code called NuMAD (Numerical Manufacturing and Design Tool)[11] has been developed over the years to aid in creating an accurate geometry for a wind turbine blade finite element model (FEM) by incorporating unique blade span-wise features such as varying airfoil cross-section, chord, and structural twist; placement of internal shear webs; and meshing into a FEM with suitable element types and element properties. Accompanying this effort has been the development of an extensive material property database for wind turbine composite materials[12]. The combination of NuMAD and the material property database forms the basis for a predictive modeling capability. Another code that is useful for evaluating detailed composite layup designs is PreComp[13], which was developed at NREL (National Renewable Energy Laboratory). PreComp produces the equivalent beam properties for a detailed wind turbine composite layup.

B. Experiment Design

A program has been underway at Sandia Labs for several years to evaluate innovative structural mechanics concepts for wind turbine blades. These 9 meter research-sized blades have been evaluated with static, fatigue, and free boundary condition modal tests. See Figure 4 for a sketch of the planform for each of the blades developed in this effort. The CX-100 design incorporates carbon fiber in the spar cap, as indicated in the sketch. The TX-100 blade incorporates off-axis fiber in the skins to produce twist-bend coupling. The CX-100 and TX-100 blades have identical external geometries. The BSDS (Blade System Design Study) blade was designed with a new planform, new selection of airfoils, and a larger root diameter. Figure 5 shows the modal test setups for free boundary condition modal tests conducted to evaluate each of the three research blades. Each test was designed with particular focus on the unique innovation used in each design. Validation of one blade, the BSDS design will be discussed in more detail later in this paper. The BSDS blade is a research-sized blade and is nominally 8.325 meters (27.3 feet) and 127 kilograms (290 lbs). A key feature of this blade design is the flatback airfoil, an airfoil with a thick blunt trailing edge. The flatback airfoil was designed to improve the structural efficiency of the inboard portion, while minimizing loss of aerodynamic performance.

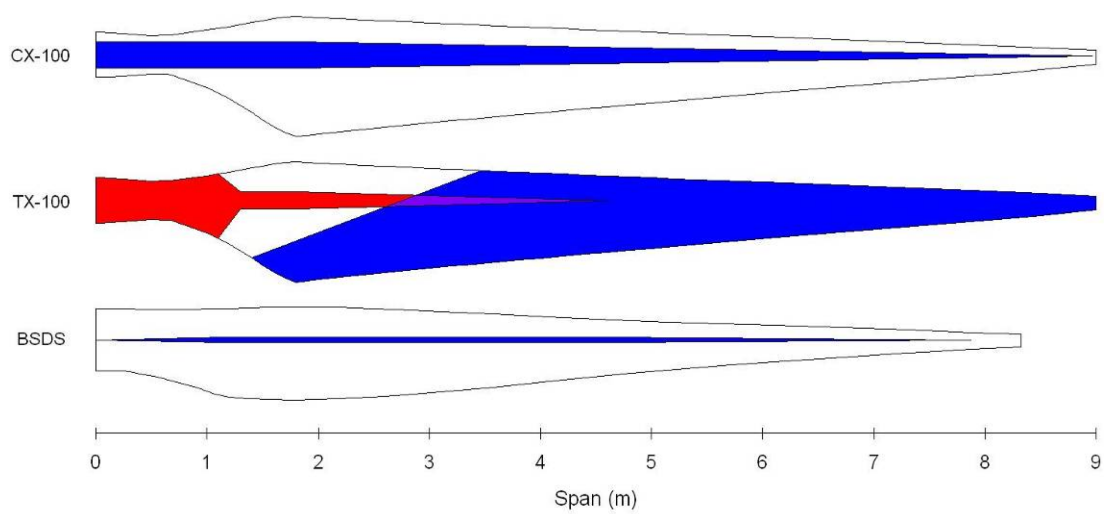

Figure 4. Research-sized blades as part of Blade Innovation Study

Figure 5. Photos of Modal Tests Performed on the Blades in Figure 4.
(a) CX-100 (left), (b) TX-100 (center), (c) BSDS (right)

Static tests were performed for each of the three blades, and the setup for the BSDS blade is shown in Figure 6. The whiffle-tree apparatus is visible above the blade which provides the upward vertical load at three locations, while the blade is constrained at the root. From this test, strain and deflection data was obtained as a function of the measured load input, which provides a means to calibrate the stiffness properties of the blade model. Note, however, that this test provides no information regarding the properties of the blade section outboard of the outer loading position because this portion of the blade is not strained in this loading arrangement. This is important when considering calibration of structural models based on static tests.

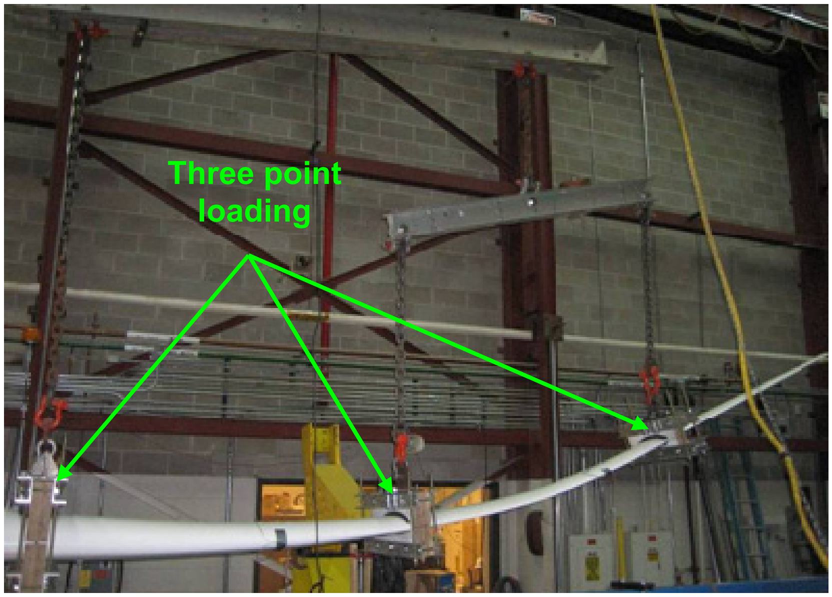

Figure 6. Static Test of the BSDS Wind Turbine Blade

For full system model validation, field testing of the turbine would be conducted. The NExT (Natural Excitation Technique) was an early operational modal analysis (OMA) technique developed to measure the modes of an operating wind turbine[14]. Traditional input-output modal analysis techniques are difficult to use for modal testing of large structures like wind turbines because it is difficult to apply sufficiently large and measurable input excitation. OMA provides a means to measure the modes during operation, and can be performed without requiring measurement of the input aerodynamic forces. Both of these are significant advantages for measuring the modal properties of an operating wind turbine, and have led to recent interest and developments in OMA methods.

C. Test-Analysis Reconciliation

At the heart of validation is a comparison of analysis predictions and test observations. For structural dynamics, we are most often interested in comparing natural frequencies of important response modes. Direct search or gradient-based optimization algorithms are used to determine a set of model parameters which reconcile the differences between predictions and observations. When performing tests, one is concerned with the accuracy of the measurements as bias errors in the test setup must be quantified for proper test-analysis correlation. Another important consideration when doing test-analysis correlation is the boundary condition of the test because the boundary condition, which must be included in the model, can have a significant effect on the modal parameters. Several studies related to modal test techniques for wind turbine blades have been published previously on various topics including experimental quantification of uncertainty, and measurement and analysis of bias errors due to instrumentation and support conditions[15]. And, consideration of the boundary condition has been considered in more recent work[16] and is summarized in the results section of this paper. Work was completed to calibrate a structural model of the BSDS blade through calibration with test data[17]. Typically, only modal test data is used for validation of structural models. However, a novel aspect of our calibration approach was to use both static test load-deflection data and modal data together to calibrate the model with the hope that the calibrated model is better as a result.

The final step in model validation is design of independent “validation tests” used to determine the quality of the calibrated model. If the calibrated model can predict the behavior of the validation experiments meeting a pre-determined adequacy criterion, then the model is considered valid. In the next section, a description of validation tests conducted with a new boundary condition, provided by a seismic mass on airbags, for a modal test of the BSDS blade is provided. Results of these tests and the accompanying model validation effort are the subject of the following section.

IV. Results of Wind Turbine Blade Validation Experiments

Thus far, we have discussed the types of structural dynamics analysis one must perform in designing a wind turbine. And, we have discussed a model validation methodology that has been employed in the development of structural models in our work. Now, we focus on one application, the development of a validated structural model for a wind turbine blade. In this section, we provide details on calibration of the structural model; and key aspects of the test design and results from the recently performed validation tests.

A beam finite element model (FEM) was chosen to investigate calibration of a blade structural model. The goal was to determine the blade span-wise mass and bending stiffness properties. A calibrated model can then be utilized to make predictions under different conditions; for example, for a different root boundary condition. The results of different calibration approaches in this calibration study are given in Table 1, which were described in more detail in Reference 17. We consider three approaches for calibration: 1) using only load-deflection data from static tests, 2) using only natural frequencies from free boundary condition modal tests, and 3) using both load-deflection and natural frequencies. When only static test data was used, the post calibration static deflection residuals were, of course, small; however, prediction of the first flap-wise mode for free boundary conditions was under predicted by 12.3%. Likewise, when only natural frequency data from free boundary condition tests was used for calibration, the natural frequency was in close agreement, but the static deflection prediction errors were large. This suggested that static test data and modal data should be used together to create a better model. The result of the third calibration approach indicates that the residual errors can be reduced for natural frequency and static deflection predictions, and demonstrate the importance of both static and modal data for calibration of a structural model. Use of static test data provides key calibration data for root end property estimates, while modal calibration data provides data for the outboard section of the blade. Also, the boundary condition has a significant effect on the modal frequencies as the measured first flap-wise frequency for a free boundary condition is 47% higher than the predicted first flap-wise frequency for a cantilever root boundary condition, based on calibration by the third approach. Thus, the boundary condition needs to be well-characterized in a test for inclusion in a model.

Table 1. Predictions of Calibrated Models and Prediction Residuals

Calibration Type	Calibration Error for First Flapwise Mode with Free Boundary Conditions (%)	Norm of Static Deflection Error (meters)
1) Statics Updating	-12.3%	0.01
2) Modal Updating	1.1%	0.31
3) Modal and Statics Updating	-1.1%	0.03

Validation tests were designed with the objective of providing a boundary condition which exercised the root as in service while minimizing boundary condition uncertainty for inclusion in the structural analysis. One could consider a cantilever root boundary condition for the validation test; however, it is difficult to realize a truly rigid, cantilever boundary condition in practice. Therefore, we chose an alternative boundary condition which exercised the root as in service and which was also simple to model. The test design includes the selection of the boundary condition, test fixturing, the lift procedure, pre-test model calibration, pre-test model prediction, test instrumentation layout, and test execution, which is discussed in more detail in Reference 16.

Again, it was desired that the boundary condition for the validation experiment be very well-characterized for inclusion in the structural model. This was accomplished by choosing a root boundary condition which could be completely measured and modeled accurately – a seismic mass supported by airbags. The seismic mass is composed of steel and has a mass of 9858 kg (21,740 lbs) with dimensions of 66 inches by 72 inches (1.67 by 1.83 meters) and 24 inches (0.61 meters) thick at the thickest point. The four airbags were placed near the corners of the mass; the center-to-center distances between the airbags were 51.5 inches (1.31 meters) and 59.75 inches (1.51 meters). When pressurized, the seismic mass is lifted from the floor, providing a soft boundary condition. The natural frequencies of the six rigid body modes associated with the seismic mass on airbags are completely dependent on the mass properties of the seismic mass and the stiffness properties of the airbags. The schematic in Figure 7 demonstrates the approach of the test, showing the blade mounted on the seismic mass/airbag system.

The primary objective of the work was to evaluate the seismic mass on airbags boundary condition for validation of a blade structural model. Mass properties measurements were used to develop an accurate mass representation for the blade FEM. Then, static tests and free boundary condition modal tests were conducted, and the data from these tests was then used to calibrate the stiffness properties of the blade FEM as previously described. This calibration resulted in a pre-test calibrated blade FEM to be evaluated in this validation effort. Then, a model of the seismic mass boundary condition, calibrated with modal data from independent tests of the mass on airbags, was combined with the pre-test calibrated blade FEM to make predictions for the combined assembly for the new test. This is a validation experiment – if the pre-test calibrated blade model is valid, then our predictions with the new boundary condition should be in agreement with the measurements according to our predetermined adequacy criterion of less than 5% error in the first four flap-wise bending mode natural frequencies. The results are now presented.

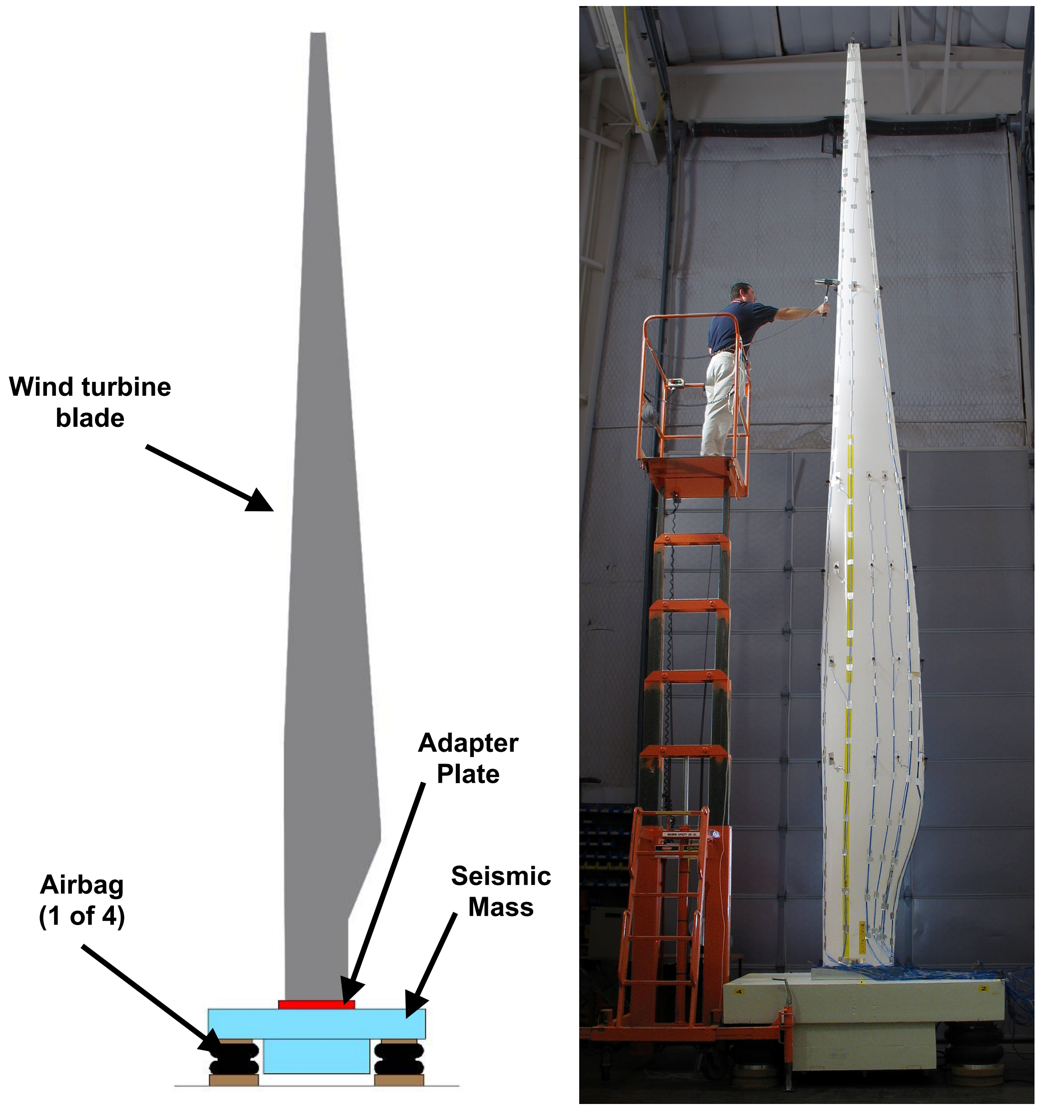

Figure 7. Validation Test for BSDS Blade (a) Sketch (left), (b) Photo of Test Setup (right)

A large number of modes were measured for comparison with the predictions, which included all six rigid body modes of the system and five flap-wise bending modes of the blade. The rigid body mode natural frequencies are examined first. Table 2 lists the predictions of rigid body modes for the system using the pre-test calibrated model for the blade with the seismic mass boundary condition; compared with the measurements from the validation test. This serves as a check of the boundary condition model and the mass properties of the blade model. The first three modes changed less than 1.5% by the presence of the blade on the seismic mass, that is, these modes are at nearly the same frequency with or without the blade on the mass. These modes depend on the total mass of the blade, which is small compared to that of the seismic mass. Conversely, the two pitch modes are significantly affected by the presence of the blade on the mass because these modes depend on the mass moments of the system. The blade has a significant contribution to the system mass moment of inertia through its mass and CG offset from the seismic mass, which reduce the pitch rigid body mode frequencies by 20% with the addition of the blade to the mass. The twisting mode is in error by 3.6%, which indicates a possible inaccuracy in the mass moment of inertia of the blade model about the vertical axis. However, this mode has an insignificant effect on the flap-wise bending modes. Finally, the pitch mode in the flap-wise direction is in error by only 1.1% which provides certainty in this boundary condition model as this mode most strongly affects the flap-wise bending modes.

Table 2. System Rigid Body Modes Natural Frequency: Prediction versus Measured

Mode	Prediction	Measured	Percent Difference
Lateral motion flap-wise	1.08	1.08	0.0%
Lateral motion edge-wise	1.07	1.09	1.5%
Axial mode in vertical direction	1.75	1.77	1.1%
Twisting about vertical	1.80	1.86	3.6%
Pitch in flap-wise direction	1.79	1.81	1.1%
Pitch in edge-wise direction	2.26	2.22	-1.9%

Given that the rigid body modes of the full system were accurately predicted, we continue to compare the natural frequencies of the flap-wise bending modes, which is an evaluation of the blade model mass and stiffness properties. Table 3 lists the predictions of the flap-wise bending modes of the pre-test calibrated blade model with the seismic mass boundary condition and compares them with the measurements from the validation test measurements.

Table 3. Pre-test Flap-wise Bending Modes Natural Frequency: Prediction versus Measured

Mode	Prediction	Measured	Percent Difference
1st Flap-wise	3.95	4.20	5.9%
2nd Flap-wise	9.56	9.57	0.0%
3rd Flap-wise	18.25	18.29	0.2%
4th Flap-wise	29.38	29.77	1.3%
5th Flap-wise	44.84	43.66	-2.7%

The predictions were quite good for the flap-wise bending modes as the largest error was found in the 1st flap-wise mode with an error of 5.9%. However, this error is slightly larger than our predetermined adequacy criterion of 5% error in the first four flap-wise bending mode frequencies. Possible contributors to this discrepancy include the size of the elements in the root section of the FEM (which were initially the largest size elements in the model) and the correctness of the assumption of the adapter plate rigidity. FRFs of the co-located root instrumentation (corresponding sensors were placed on the blade root and the adapter plate) showed that there was no significant difference in these sensors, therefore, it was decided that improvements in the model could be made by considering re-calibration of the blade model. To test this idea, the two elements at the root end of the model were split in two, which created four elements. This provides better resolution for the root end elements, and a better ability to capture the large gradients in the root end mass and stiffness properties.

A slightly different approach was taken for the re-calibration process. In the initial calibration, a hybrid calibration was used as the static test and modal test data were used simultaneously to calibrate the model. In the re-calibration, the new 22 element FEM was first calibrated with the static test load-deflection data. The stiffness parameters obtained from this calibration were held constant for the portion of the blade from the root to the third loading point of the static test (approximately at the 6 meter (236 inches) span), while the parameters for the remaining outboard elements were calibrated using the free boundary condition natural frequencies. The re-calibrated 22 element model was then used to make predictions for the blade with the seismic mass on airbags boundary condition. These predictions are listed in Table 4.

Table 4. Re-calibrated Flap-wise Bending Modes Natural Frequency: Prediction versus Measured

Mode	Prediction	Measured	Percent Difference
1st Flap-wise	4.05	4.20	3.5%
2nd Flap-wise	9.73	9.57	-1.7%
3rd Flap-wise	17.88	18.29	2.2%
4th Flap-wise	29.11	29.77	2.2%
5th Flap-wise	43.27	43.66	0.9%

Improvements were found for the 1st flap-wise mode, but at some expense to the accuracy of the other modes. One does not expect to predict all natural frequencies precisely; however, from the standpoint of reduction of the maximum prediction error the re-calibrated model is an improvement. Also, all errors are below 3.5% which indicates that this model is valid for our chosen adequacy criterion. Additionally, the fifth flap-wise bending mode also meets our adequacy criterion. It is worth noting that for a free boundary condition the 1st flap-wise natural frequency was measured at 5.25 Hz and for a fixed boundary condition the 1st flap-wise natural frequency was predicted to be 3.57 Hz. For the free boundary condition, the natural frequency is 47% higher. This demonstrates that the boundary condition has a significant effect on the natural frequencies.

In our study, we have validated a model of the blade span-wise properties using a beam FEM. The validated bending stiffness (EI) and mass distributions can now be used in a number of structural dynamics analyses to assess the blade design and its performance in operation. For example, we can run full system dynamics analysis using MSC.Adams or the FAST code to compute modes of the full assembly turbine or transient dynamics analysis for chosen wind conditions. Further, this validated EI and mass distributions could be compared with equivalent beam properties derived from high-fidelity solid models as an evaluation the detailed composite layup design[11,13].

V. Concluding Remarks

In this paper we presented an overview of structural dynamics analysis for wind turbine structures and their substructures. A focus of this work is validation of these models using static and modal testing. The unique features of wind turbine blade architectures were overviewed to describe the types of structural models and structural analysis needed to certify the designs. Structural dynamics of wind turbine structures is very important because resonant conditions can lead to turbine damage or even failure. Also, an analyst must consider the rotational effects of the rotor to properly perform modal analysis for an operating wind turbine.

A model validation methodology was presented and utilized to validate a blade structural model. A hybrid calibration approach incorporating static test and modal test data was used to calibrate the structural model, and a novel boundary condition was used for validation experiments. The boundary condition is one of the important considerations when performing structural dynamics analysis; any test designed to validate a structural dynamics model should provide information that well characterizes the boundary condition for inclusion in the model. A seismic mass on airbags provided the boundary condition for validation modal tests, which can be accurately characterized from test observations and simply modeled. This capability could be considered a new test technique for modal testing of wind turbine blades because it shows promise as an alternative to boundary conditions traditionally selected for wind turbine blade modal testing, which are free and fixed boundary conditions. Free boundary condition tests do not exercise the root end of the blade as is present in service although the effect of the soft supports on the modal parameters is typically small. Cantilever boundary conditions cannot be realized in practice because of compliance and fixture resonance interference issues, although it provides service type strain at the root. The seismic mass boundary condition offers the advantage of the fixed boundary conditions by straining the root as in service, and it also reduces uncertainty in the boundary condition model.

VI. References

[1]Burton, T., Sharpe, D., Jenkins, N., and Bossanyi, E., *Wind Energy Handbook*, 2nd Edition, Wiley, Chichester, UK, 2008.

[2]Manwell, J.F., McGowan, J.G., and Rogers, A.L., *Wind Energy Explained: Theory, Design, and Applicaton*, 2nd Edition, Wiley, Chichester, UK, 2006.

[3]Ewins, D.J., *Modal Testing: Theory, Practice and Application*, 2nd ed. Research Studies Press, Ltd, 2000.

[4]Craig, R.R., and Kurdila, A.J., *Fundamentals of Structural Dynamics,* 2nd Edition, Wiley, Hoboken, New Jersey, 2006.

[5]Meirovitch, L., *Principles and Techniques of Vibrations*, 2nd Edition, Prentice Hall, Upper Saddle River, NJ, 2000.

[6]Stol, K., Bir, G., & Balas, M., "Floquet Modal Analysis of a Teetered-Rotor Wind Turbine," ASME Journal of Solar Energy Engineering, Vol. 124, No. 4, 2002, pp. 364-371.
[7]Bir, G., "Multi-Blade Coordinate Transformation and Its Applications to Wind Turbine Analysis," Proceedings of the AIAA/ASME Wind Energy Symposium*,* Reno, Nevada, 7-10 January 2008.
[8]MSC.ADAMS, http://www.mscsoftware.com/products/adams.cfm.
[9]*NWTC Design Codes (FAST by Jason Jonkman)*. http://wind.nrel.gov/designcodes/simulators/fast/. Last modified 12-August-2005; accessed 12-August-2005.
[10]*NWTC Design Codes (AeroDyn by Dr. David J. Laino)*. http://wind.nrel.gov/designcodes/simulators/aerodyn/. Last modified 05-July-2005; accessed 05-July-2005.
[11]Laird, D. and Ashwill, T., "Introduction to NuMAD: A Numerical Manufacturing and Design Tool," Proceedings of the ASME/AIAA Wind Energy Symposium, Reno, NV, 1998, pp. 354-360.
[12]Mandell, J.F. and Samborsky, D.D., "DOE/MSU Composite Material Fatigue Database", Sandia National Laboratories Technical Report, SAND97-3002, March 2008.
[13]*NWTC Design Codes (PreComp by Gunjit Bir)*. http://wind.nrel.gov/designcodes/preprocessors/precomp/. Last modified 26-March-2007; accessed 26-March-2007.
[14]James, G.H., Carne, T.G., and Lauffer, J.P., "The Natural Excitation Technique (NExT) for Modal Parameter Extraction from Operation Structures," Modal Analysis: the International Journal of Analytical and Experimental Modal Analysis, Vol. 10, No. 4, Oct 1995, pp. 260-277.
[15]D.T. Griffith, T.G. Carne, and J.A. Paquette, "Modal Testing for Validation of Blade Models," Journal of Wind Engineering, Vol. 32, No. 2, 2008, pp. 91-102.
[16]D. T. Griffith, Hunter, P., Kelton, D., Carne, T., and Paquette, J., "Boundary Condition Considerations for Validation of Blade Structural Models," To be presented at the SEM Annual Conference and Exhibition, June 1-3, 2009, Albuquerque, NM, Paper No. 521.
[17]D.T. Griffith, J.A. Paquette, "Development of Validated Blade Structural Models," Proceedings of the AIAA/ASME Wind Energy Symposium, January 2008, Reno, NV, AIAA-2008-1297.