The Development of a Phase Locked Excitation Testing Method for Full-Scale Wind Turbine Blades

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The Development of a Phase Locked Excitation Testing Method for Full-Scale Wind Turbine Blades

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Master of Science in Mechanical Engineering

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The Development of a Phase Locked Excitation Fatigue Testing Method for Full-Scale Wind Turbine Blades

By

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This thesis was prepared under the direction of the candidate's thesis committee chairman, Darris White, Department of Mechanical Engineering, and has been approved by the members of his thesis committee. It was submitted to the Mechanical Engineering Department and was accepted in partial fulfillment of the requirements for the degree of Masters of Mechanical Engineering.

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Full scale blade testing provides blade manufacturers with quantitative data regarding their blade design, manufacturing processes and durability. Structural fatigue tests are designed to assess the structural health of the turbine blade as well as simulate a lifespan worth of damage in a laboratory environment. These tests also provide a further understanding of the dynamics involved in modern turbine blades. Blade tests are typically conducted in one of the following manners: a dual-axis forced displacement, single axis or dual axis resonant configurations. Historically, fatigue testing has been performed by utilizing forced displacement systems. These systems do not allow for the load phase angle to be controlled, which leads to inaccurate loading distributions, when compared to actual field gathered data. The PhLEX (Phase Locked Excitation System) testing method outlined herein utilizes resonant excitation system in order to reduce energy requirements, decrease test duration and improve overall loading distributed.
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CHAPTER 1: INTRODUCTION AND BACKGROUND

Introduction

The purpose of this research was to develop a Finite Element Model and test method to further enhance the current dual-axis fatigue test of full-scale wind turbine blades currently in use at the National Renewable Energy Laboratories (NREL) in Boulder, Co. The analysis described herein outlines the design of a fatigue test for wind turbines using a newly developed Phase-Locked Excitation testing method. The test takes advantage of resonance testing in order to decrease the overall energy requirements of full-scale wind turbine blades while simultaneously decreasing test duration. A background and history of wind energy, wind turbines and wind turbine blade testing is provided in the opening chapter to provide the reader with a basic understanding of the topic at hand. The following section will cover the following: wind turbine history, wind turbine configurations and testing methods.

History of wind turbines

Wind is a form of solar energy that is generated by the uneven heating of the atmosphere by the sun, the rotation of the earth and the irregularities of the earth’s surface [1]. For centuries, humans have taken advantage of nature to accomplish their biddings. The earliest known use of wind energy was in sailboats that harnessed the energy from the wind to aid the ancient Egyptians traveling along the Nile River [2]. The Persians, who are credited with building the first windmills, later adopted this concept. See Figure 1-1: Persian Windmill. These machines were mainly used to perform tasks such as grinding wheat and pumping water. The expansion of Islam saw this technology spread throughout its empire and along its trading routes as far as China. These devices did not make their debut in Europe until the 11th century [3]. Wind turbines later became an important tool for daily life in Holland. Tasks such as grinding wheat, pumping water and cutting wood were accomplished by means of these machines.
Windmills were such an integral part of life in Holland, that living quarters were built into their structures. This way nearby villagers always had someone to grind wheat [4]. Today, wind turbines are still associated with Holland, even though heavy industrial machinery has long since taken its place in performing such tasks. Likewise, wind-driven water pumps, cereal grinders, and sawmills were a contributing factor in the development of the American West [3].

The main difference between windmills and modern wind turbines is that wind machines drive their loads directly, while the latter generates electricity that can be utilized in a wide variety of ways [3]. Charles Bush is widely considered as the builder of the first wind turbine that was designed specifically for generating electricity [3]. It was built in 1888 for the sole purpose of supplying the electrical needs of his mansion in Cleveland, Ohio. This 40-ton piece of machinery was of the multivane type, which supported 144 blades. Owing to the intermittent nature of the wind, the electricity generated by the wind turbine was stored in some 400 storage cells. Due to financial restrictions placed on this effort, from maintenance and initial investment costs, the wind turbine was put out of operation.
Factors such as these drove the cost of electricity to be much higher than that produced by steam
plants, rendering the wind machine financially unfeasible [3].

As the industrial revolution kicked into full swing, the construction costs associated with Large-
scale projects became more affordable. In 1939, work began on the famous Smith-Putnam machine, in
Vermont. This machine was rated at 1.3MW for wind speeds of 15 m/s. It was a propeller type-machine
with a rotor diameter of 53-m. The wind turbine began providing the power grid with direct electricity in
1941. Not too long thereafter, the Smith-Putnam machine’s service life was cut short due to blade
failure, in March of 1945. This mechanical failure had been predicted; however, the events of World War
II offered no opportunity to redesign the propeller hub [3]. The low cost of oil that followed World War II
discouraged much of the research effort toward alternative energy. Of this period, one person’s efforts
worth mentioning are Johannes Juul, who developed the Gedser turbine during the 1950’s [6]. This
design set the stage for the modern day wind turbine, which had similar features such as the 3-blade
design, aerodynamic stall regulation of power, an electromechanical yaw drive and asynchronous power
generation.

As environmental concerns surrounding fossil fuels became more evident in the 1960’s interest
in wind energy began to take hold once again. It wasn’t until the oil crisis of the 1970’s that significant
research began in the U.S [3]. This research resulted in the overall improvement of the following:
aerodynamics design, advanced materials, electrical components, control strategies and component
testing [6]. These advancements drove the cost of generating electricity from wind to rates that allowed
it to be competitive with standard methods of electricity production. By 1996, the cost of 1KWh was
approximately 5 cents, compared to 25 cents in the early 1980’s [6].Figure 2-1: Calculated cost for
different capacity turbines below, illustrates the calculated cost for different capacity turbines. (All costs
are converted into constant 2006 prices.)
Figure 2-1: Calculated cost for different capacity turbines [7]

These numbers reflect the production estimate for coastal and inland sites. The starting point on the graph reflects machines that were typically installed in the mid 1980’s and ends with machines that were installed around 2003 and onwards [7]. The associated cost of producing 1KWh is dependent on the cost of the investment and its return. Typically, the investment cost of producing 1KWh by means of wind turbines is comparable to that of a fossil fuel plants. However, wind-power plants have a capacity factor much lower than fossil fuel/hydrocarbon plants, due to the following factors [3]:

1. Variable nature of the wind
2. The cost of leasing the land on which the wind farm sites are situated
3. Maintenance and operational costs
4. Decommissioning costs, which are the costs associated with non-operational wind farm sites

For the sake of being competitive with fossil fuel energy production, wind turbine energy efforts again slowed down until the Bush administration. This new leadership provided an opportunity for the federal wind program to resume and further develop research efforts in this relatively new field. By 2009, some 35GW of the power generated in the U.S was from its wind turbine fleet, of which 10GW were installed in various states such as Iowa, Texas and Indiana that same year. The driving force behind
This effort was the Wind Program, funded by the Department of Energy [8], whose aim was to have 20% of the US energy demands supplied by wind turbines [9] [8]. Outlined in its Wind Energy Report, the DOE foresees wind turbine energy supply reaching 300GW by 2030 [10], as indicated in Figure 3-1: Wind Energy Capacity trend below. The figure also shows how the US is currently ahead of the growth curve thanks to the effort put forth towards renewable energy in 2009.

Figure 3-1: Wind Energy Capacity trend [8]

Figure 4-1: Geography of Wind Energy below illustrates the installed capacity by region for the year 2009. As can be seen in the figure, the geographic application of renewable energy is continually changing, which suggests a new era of its geographic diversity [11]. In the 1990s, wind energy only existed in a handful of countries.
Today, wind energy operations can be found in over 82 countries. Current wind turbine technology has made wind a viable alternative energy source in the current energy market. The developments brought forth by ongoing research efforts, such as this project, will continue to keep wind energy growing in use and popularity [11].

Wind Turbine Configurations

Wind turbines are classified by the axis of rotation about which they spin. They fall into one of two categories: Vertical Axis Wind Turbine (VAWT) that spin about the Vertical axis and Horizontal Axis Wind Turbines (HAWT) that rotate about the horizontal axis [12]. These two configurations are further subcategorized into Vertical Drag-Type/Lift-Type and Horizontal Upwind/Downwind devices.

Vertical Axis Wind Turbines (VAWT)

The drag-type vertical axis turbine is an apparatus with radially mounted disks that has a higher drag coefficient in one direction than the other. Vertical axis turbines are not as common as their horizontal counterpart for several reasons, which will be discussed in further detail; however, their use is advantageous for certain applications, such as wind speed measuring devices [12]. Its limitations are
outlined by the turbine tip speed, which cannot exceed wind speeds [6]. See Figure 5-1: Drag-Type Vertical-Axis Wind Turbine below.

![Figure 5-1: Drag-Type Vertical-Axis Wind Turbine][13]

Lift-type vertical axis turbines are more commonly used for power generation than the drag-type vertical axis. These wind turbines generate power by means of two radially situated airfoils that generate lift, which drive the generator, as shown in Figure 6-1: Darrieus Turbine below. In this configuration, the turning moment that spins the wind turbine is generated by the tangential lift component. This characteristic is adventitious as it allows the machine to operate at a tip speed that is greater than wind speed [6] [14]. The Darrieus turbine is distinctive, as it does need to be directed into the wind to be effective.

![Figure 6-1: Darrieus Turbine][15]
This configuration is also advantageous from maintenance perspective because of the location of the generator. As indicated in Figure 6-1: Darrieus Turbine above, the generator is located at ground level, which makes access to key mechanical and electrical components easy as well as lowering construction costs [4]. The Darrieus turbine is distinctive, as it does need to be directed into the wind to be effective. Its limitations however, are mainly due to the low starting torque, which requires an initial boost at start up. Also, because it is so low to the ground, there is less wind speed to harness energy. This drives their overall production of energy down, when compared to Horizontal Axis Turbines [6] [16].

**Horizontal Axis Wind Turbines (HAWT):**

Although similar in function, HAWT are capable of generating more electricity than VAWT and are typically the choice for large-scale utility projects [12]. These wind machines are generally designed with an odd number of blades due to stability reasons. In the case of a 3-bladed wind turbine, when performing dynamic calculations on the structure, the rotor can be treated as a disk. Rotors with even number of blades tend to be unstable, especially with stiff structures. This is mainly due the asynchronous maximum and minimum loading applied at the 12 and 6 o-clock positions. The 3-bladed configuration, typically known as the classical Danish concept, is most common in modern wind turbines [17] as seen in Figure 7-1: Horizontal Axis Wind Turbines below.

![Figure 7-1: Horizontal Axis Wind Turbines](image-url)
As is the case with VAWTs, HAWT are also grouped into two categories: Upwind and Downwind devices. Downwind HAWT are designed to freely rotate into the best place that is conducive to the greatest power production, as seen in Figure 7-1: Horizontal Axis Wind Turbines above. Feathering the turbine blades allows optimized operating efficiency by controlling blade speed [6]. This configuration is easier and cheaper to manufacture than the upwind type for the following reasons: downwind turbines do not require the extra mechanical component or controllers needed to yaw the hub in the direction of the wind; they do not require the additional stiffness needed to prevent blade strikes.

Similar in appearance to the downwind turbine, the upwind turbine varies in mechanical hardware and conceptual design features. As opposed to freely yawing into the wind direction, upwind turbines are controlled by means of a drive mechanism that positions them into a direction, relative to the wind, that yields the greatest power see Figure 8-1: Upwind Horizontal-Axis Wind Turbine below [19].

![Figure 8-1: Upwind Horizontal-Axis Wind Turbine](image)

The advantage to this design over the downwind design lies in turbulence-free air that strikes the blade head on. Upwind turbines do not experience the problematic wake turbulence generated by
the tower, which is inherent to the downwind design. This causes a reduction in the amount of power that can be captured by the turbine.

CHAPTER 2: WIND TURBINE FATIGUE TESTING

According to the International Electrotechnical Commission (IEC), “the fundamental purpose of a wind turbine blade test is to determine to a reasonable level of certainty that the blade type, when manufactured according to a certain set of specifications, has the prescribed reliability with reference to specific limit states, or, more precisely, to verify that the specified limit states are not reached and the blade therefore possess the strength and service life provided for in the design” [21] [22]. The following sections discuss the history and type of wind turbine structure testing.

History of Fatigue Testing

The turbine blade is designed to transfer aerodynamic (lift) and inertial loads (gravity) to the rotating hub, which is then converted into electricity [23]. The nature of the loads generated by the wind put the wind turbine blades under considerable stress. Herein lies the importance of the turbine blade to the functionality of the machine; without it, no power can be generated. This also underlines the importance of performing fatigue tests on wind turbines blades. Since wind turbine blades are subjected to a variety of vibration and other loading conditions throughout the span of their twenty-some year life, it is critical to simulate these conditions before the production phase in order to guaranty this life span [6] [24] [25] [26].

Currently, there are a handful of facilities around the world that are capable of performing full-scale fatigue tests. These facilities are located in Greece, the Netherlands, the United Kingdom, Denmark and the U.S.A [6]. The National Renewable Energy Laboratories (NREL) in Golden, Colorado, is home to one of the full-scale wind turbine fatigue testing facilities in the U.S. This National Wind Technology Center (NWTC) is capable of performing static, fatigue and modal testing on blades up to 30-m in length
More recently, NREL opened a testing facility in Massachusetts and will be opening a similar facility in Texas, which will be capable of accommodating blades up to 100‐m in length [10]. The driving force behind these costly efforts dedicated to building such testing facilities lies in the critical information they relate to the turbine blade manufacturers regarding their designs. The required output demand of wind turbines has increased in recent years, which has led to an increase in the size of the turbine blades. In order to effectively meet these demands while maintaining low weight, new design concepts and materials have been introduced to wind turbine blades.

Modern wind turbine blades have varying airfoil cross‐sections that culminate in a circular cross section at the root to allow for connection to the hub. The length of these blades can vary from 9 meters to 100 meters. Typically, the maximum root occurs at 20% span of the blade [23] [27]. Wind turbine blades are typically made of composite material, fiberglass with epoxy or vinyl ester matrices, while shear webs help to distribute loads while also adding stiffness, see Figure 9‐2: Typical Wind Turbine Blade Construction below.

![Figure 9-2: Typical Wind Turbine Blade Construction][28]

Since wind turbine blades have become longer, materials such as carbon fiber have been added in order to keep the blades light while maintaining structural integrity [27] [23]. This allows them to be designed with the maximum power output in mind; maximizing the blade length, while minimizing
weight induced fatigue loads [29]. This also reduces the loads induced on the tower and foundations [28]. Composite structures are generally very resistant to fatigue. However, it is very difficult to predict when a composite structure will fail [28]. Fatigue tests are a concrete method of testing the durability of turbine blade designs.

**Dual-Axis Force Displacement Test**

In order to increase the power captured by wind turbines, the swept area of the blades must also increase. This requires longer and more expensive blades. A number of tests are used to validate the design of wind turbine blades. Currently, there are three methods for testing of wind turbine blades: single-axis resonance method, dual-axis forced displacement and dual-axis resonance method [6]. The most common of these tests has been the dual-axis forced displacement test, as seen in Figure 10-2: Dual-Axis Force Displacement Test below.

![Dual-Axis Force Displacement Test](image)

*Figure 10-2: Dual-Axis Force Displacement Test [23]*

This method uses hydraulic actuators to exercise the blade in both the flap and lead lag directions [30] [31] [32]. The strain profiles created by these test more accurately depict the conditions seen in service than single axis tests. One drawback to the dual-axis forced displacement test, however,
is the test duration. This is due to the natural frequency at which this type of test is conducted, typically much lower than the first fundamental natural frequency of the blade [6]. As wind turbine blades increase in length and weight, the resulting lead-lag bending moments also increase due to gravity. As a result, the flap and edge moments are on the same order of magnitude.

**Single-Axis Resonance Test**

The single axis resonance test was developed for small blades in which the flap moment is much greater than the edge moment. In this test method, blade excitation is achieved by rotating an eccentric mass applied in the desired blade direction, as seen in the Figure 11-2: Single-Axis Resonant Test below.

![Figure 11-2: Single-Axis Resonant Test](image)

The single-axis resonance test is limited to testing a blade in any given direction at a time. This leads to less accurate results than the aforementioned testing method. However, it is possible to fine-tune the bending moments by adding masses along the blade to closely match actual bending moments. Furthermore, the excitation force can be drastically reduced due to the magnified displacements that occur at resonance. This in turn reduces the energy requirements to conduct such tests. Of the many advantages to resonance testing, lower testing costs and faster results have made this test a viable option [10] [6].
Hybrid Testing

As previously indicated, force displacement tests exercise the flap and edge directions independently, which can cause some inaccuracies in results [6]. To fine-tune these simulation results, the Universal Resonant Excitation system (UREX), which uses the dual-axis resonant method, is a commonly adopted test nowadays [34], see Figure 12-2: Dual-Axis Hybrid Test below.

![Dual-Axis Hybrid Test](image)

**Figure 12-2: Dual-Axis Hybrid Test [10]**

The dual-axis resonance method excites the flap resonance frequency while forcing the edge displacement. This method provides a more accurate stress distribution than either of the previously discussed methods [35]. For similar reasons, this test requires less energy, less expensive equipment and less testing time than the dual-axis test method. One limitation to these test system is that the phase angle is unable to be controlled, which results in loading conditions that are not seen in the field [23] [9].
CHAPTER 3: BLADE TESTING CONSIDERATIONS

This section outlines the important properties unique to wind turbine blades and the terminology associated with wind turbine blade testing. Blade material and construction along with blade properties will be discussed, concepts such as phase angle will be defined and the three primary directions used in the analysis of wind turbine blades will also be defined [6].

Introduction

In this section, blade properties are presented as a set of normalized data points derived from manufacturer data that is unique to one specific blade. Modern large-scale wind turbine blades are typically constructed from fiber-reinforced glass-epoxy composites. The blades transmit aerodynamic and gravitational loads to the hub assembly by the root. Owing to its complex geometry, layered composite structure and excessive loading, the root is most prone to failures. Through many years of blade testing, large sets of data pertaining to turbine blades have been collected at NREL facilities. As blades scale up, the mass per unit length increases, and stiffness in both the edge and flap directions increase significantly in the root of the blade. Chord lengths also increase significantly as blades are scaled up in size [36] [26]. As more energy is required from wind turbines, the rotor diameter increases, meaning the blade length and weight increases. This increase in weight of the blade increases the stresses not only on the blade itself but also on the hub and driveshaft and all of the other components of the turbine [9].

As discussed previously in the testing methods section, typically the flap and lead-lag directions along with their corresponding stiffnesses were considered separately, as both directions were tested under different conditions. The goal of the PhLEX test is to test the blade in both directions simultaneously at one natural resonant frequency with the ultimate goal of a 72 degree phase angle [37] [26]. To perform these tests, solid data is acquired as a basis of test result comparison. This data has
been and continues to be gathered from wind turbines currently in field use. Testing facilities, such as the one at the WTC in Colorado, provides the controlled conditions under which these fatigue tests may be carried. According to White “By precisely applying fatigue loads to the wind turbine blades, it is possible to compare the results of the actual blade to the finite element model, find manufacturing defects, and accelerate the fatigue test to take months instead of decades.” [6]

Normalized Blade Properties

This section provides a basic understanding of a typical wind turbine blade. Since manufacturer specific blade properties are proprietary, normalized blade data will be used to convey the typical structure of a blade. Figure 13-3: Wind Turbine Testing Directions defines the directions of the blade properties and shows a typical cross sectional view of a wind turbine blade. The data illustrated in this section is for LANL 9-meter CX-100 wind turbine blade. Blade construction materials include fiberglass epoxy with carbon fiber spar caps. The root transitions into an airfoil profile from 25% span location to the tip.

Flap & Edge Stiffness

The flap stiffness is defined as the resistance to displacement for a given force in the flap direction. As previously mentioned, due to the blade geometry, the stiffness in the edge direction is typically higher than in the flap. Normally, the stiffest portion of the blade occurs at the root, as shown in Figure 13-3: Wind Turbine Testing Directions. The attachment fasteners are embedded into the blade material at this section. This in turn adds stiffness in relation to the rest of the blade. The stiffness of the

Figure 13-3: Wind Turbine Testing Directions [9] [36]
mounting area of the blade is shown in Figure 14-3: Flap Stiffness, which is large in the flap direction, starting at the root, and drops significantly as the geometry changes [37] [26].

The edge stiffness refers to the resistance to bending in the lead-lag direction. The edge stiffness is similar in characteristics to the flap stiffness distribution but varies typically by having higher values and a shallower decay. The increase in stiffness at the 15% station in Figure 15-3: Edge Stiffness bellow is not a common occurrence in wind turbine blades [26] [36] [23].
Figure 15-3: Edge Stiffness

Mass Per Unit Length

Figure 16-3: Mass per Unit Length below shows a significant mass associated with the blade towards the root. This is primarily due to the fact that wind turbine blades are structurally reinforced at the root for blade attachment purposes, which is accomplished through bolts laid directly into the root. The figure also shows this particular blade increases in mass during the transition from the root circle geometry to the maximum chord, which occurs generally around the 25% span. The last 75% span can be best described as a linear decreasing profile from the maximum chord to the tip. Almost all wind turbines share this mass distribution [27].
Chord Length

Figure 17-3: Chord Length above shows the chord length of the blade. The maximum chord length occurs at twenty percent blade station, which coincides with a linear decrease in mass per unit
length to the tip of the blade. This indicates a consistent composition from twenty percent to the blade tip. From the twenty percent station, the mass per length is only changing due to the changing geometry [37].

**Blade Twist**

The angle of twist depicted in Figure 18-3: Angle of Twist, shows the relation between each element with respect to the global coordinate axes. These angles are used when assembling the global stiffness matrix. This particular blade has maximum twist in the blade close to the hub, which gradually decreases to approximately zero degrees of twist towards the blade tip [27].

![Figure 18-3: Angle of Twist](image)

**Axial Stiffness**

Axial deformation in blades should be insignificant when compared to the flap and edge deformation that will be present during the blade test. Axial blade stiffness resists elongation along the length of the blade. Figure 19-3: Axial Stiffness below shows an approximation of the axial stiffness of this particular blade as this data was unknown. Such estimations were provided by empirical formulas.
that have been developed by the long history of blade testing at NREL. This estimation is based upon the materials used to manufacture the blade [26].

![Blade Properties - Axial Stiffness](image)

**Figure 19-3: Axial Stiffness [26]**

**Torsional Stiffness**

The torsional stiffness of the blade describes the ability of the blade to resist bending moments along the length of the blade, as seen in Figure 20-3: Torsional Stiffness above. Torsional stiffness of the blade drops rapidly, which allows for a coupling of the edge and flap deformations further down the blade. This data is also approximated since manufacturers do not typically disclose the torsional stiffness data of their blades [37].
Figure 20-3: Torsional Stiffness

Figure 21-3: Stiffness Ratio
Stiffness Ratio

Figure 21-3: Stiffness Ratio shows the relationship between the edge and flap stiffness along the length of the blade. At the root, the flap and edge stiffness are approximately equal. The ratio between the two quickly drops as the geometry changes along the blade length. At approximately twenty percent of the blade length the ratio between the two stiffnesses stabilizes [27].

Wind Turbine Blade Loads

The three primary directions used in the analysis of wind turbine blades are: span-wise, flap-wise (flap) and edge-wise (edge) directions. Figure 22-3: Cross Sectional View with Testing Directions below shows the primary directions on a cross sectional view of a blade. Wind turbine blades are subject to a wide variety of loads, such as bending, torsion, compression and non-deterministic loads caused by the variable nature of the wind (SuperGen).

![Figure 22-3: Cross Sectional View with Testing Directions [6]](image)

For the purpose of this research, the loads are categorized as either aerodynamic loads (lift, drag and shear) or inertial loads (gravity and blade dynamics) [6], as illustrated in Figure 23-3: Blade Loads below. Aerodynamic and inertial loads typically occur in orthogonal (perpendicular) bending directions. The flap forces are applied out of the hub plane of rotation. These typically result in bending moments generated by wind loads. These loads are categorized as either stochastic or deterministic loads. The lead-lag forces occur in the rotor plane of rotation. For smaller blades, these loads are not considered a major source of fatigue [38] [39].
In the case of smaller wind turbines, most of the applied loads were in the flap direction, which were caused by wind loads. Larger blades experience almost equal loads in both the flap and edge direction. This is mainly due to loads caused by gravity in the edge direction. There is also an induced bending moment caused by generator torque which must also be considered. These loads factor into the resulting stiffness inherent to the turbine blade; however, the airfoil shape is largely considered the determining constraint. As a result, wind turbine blades tend to be much stiffer in the lead-lag direction than in the flap direction [38] [41].

**Figure 23-3: Blade Loads [40]**
CHAPTER 4: PhLEX MODELS

The Phase angle of a blade is defined as the degrees of rotation a blade will experiences between maximum loading in both the lead-lag and flap directions. This angle was previously determined from data gathered in the field and was found to be approximately 72 degrees. The Phase Locked Excitation test method was designed with the main goal of allowing this angle to be controlled by means of a stiffening actuator. The following section discusses the design Phases and results of this system.

Phase I Introduction

In Phase I of the design, a proposed stiffening element was added in the flap direction in order to modify the natural frequency of the blade in the flap direction, and make it approximately equal to the natural frequency in the edge direction. The modeling was performed utilizing MATLAB scripts that were based on previous blade testing models. This test was designed with the ability to lock the phase angle between the edge and flap directions of the blade for more realistic blade loading conditions. This would ultimately reduce the blade test duration by allowing both the edge and flap tests to be completed simultaneously while also decreasing testing energy requirements [35] [26]. Figure 24-4: Phase I of PhLEX Model describes the system that was modeled. A simple linear spring was placed between the ground and the blade as a method to add stiffness in the flap direction of the blade. It is assumed that the mass of the spring will be supported by the ground, and will not affect the blade. [23]

Figure 24-4: Phase I of PhLEX Model [23]
The analysis in this phase was performed using normalized blade properties of a 9-meter blade. Results and findings will be discussed in the following sections. The results from this system were promising when compared to previous test methods, however, they did not hold when the model was scaled to accommodate for larger blades. This model was unsuccessful with larger blades due to the greater difference in Eigen values in the flap and edge directions. A scaling study showed that the amount of stiffness required to match the flap and edge natural frequencies for a 44.5-meter blade was very large. The difference was large enough that the stiffness being added by the actuator acted as a restraint, essentially cutting off the length of the blade before the actuator. This would effectively hold the blade stationary at the location of the stiffening element. These issues were addressed in the second phase of the PhLEX design process, which will be discussed in the following section. [23]

**Phase II Introduction**

The second phase of the design took a more direct modeling approach toward achieving the same goals outlined in phase I, while addressing the encountered scaling issues. Instead of modeling the system as a quasi-static model, a linearized dynamic Finite Element Model (FEM) was developed using MATLAB [23]. Figure 25-4: Phase II of PhLEX Model below illustrates the concept of the second phase of the PhLEX test analysis. Resonance excitation was applied via the UREX actuator in the lead-lag direction at the blade’s first fundamental lead-lag natural frequency. The flap direction was excited by means of the PhLEX actuator. The actuator applied a force that was also a function of the lead-lag fundamental natural frequency. This method avoided matching the flap and lead-lag natural frequency through augmenting the flap stiffness, which proved problematic in the first phase of the design.
The results from this analysis are promising. Aside from a control system limitation that was found during the analysis, this model was successful in predicting the transient response of the blade. The current and final design phase has taken this model and improved the control system to achieve more accurate and promising results.

MODELING CONCEPTS

This section highlights the theoretical approached used in the modeling and analysis process of both Phase I and Phase II of the design process. Concepts such as Finite Element Models and State Space representation will be discussed.

Finite Element Analysis

R. Courant first developed finite Element Analysis (FEA) in 1943. He utilized the Ritz method of numerical analysis and minimization of variation calculus to obtain approximate solutions to vibrating systems [42]. FEA programs use a system of points called nodes, which make a grid that is referred to as a mesh. The mesh is programmed to contain the material and structural properties of the model, which will define how the structure reacts to various loads that are applied [42] [23].

Bernoulli-Euler & Timoshenko Analysis

Since the Timoshenko beam theory is of a higher order than the Bernoulli-Euler beam theory, it is known to be superior in predicting the transient response of a beam [23] [43]. It is shown that the
Timoshenko model superiority is more pronounced for beams that have a low aspect ratio. Beam elements in Timoshenko beam theory have transverse shear strain constants through the cross section. The cross section remains undistorted after deformation of the beam. Due to this limitation of first order shear deformation, Timoshenko beam theory can only be used on thick beams. Unlike in the Timoshenko beam theory, the shear deformations in the Bernoulli-Euler beam theory are neglected. Thusly, the Bernoulli-Euler theory was used in the analysis of both development phases.

**State Space Representation**

A state space model was developed using MATLAB to more accurately model the dynamic response of the system in phase two of the design process. State space models are constructed by physically modeling a system with a set of inputs and outputs [23]. The system consists of state variables, which are defined by its equations of motion, Equation 1: state differential equations. The state differential equations relate the rate of change of the state of the system to the input signals [44]. These state variables fully describe the system and its response to a given input. The state differential equations are as follows:

\[
\dot{x} = Ax + Bu \\
y = Cx + Du
\]

**Equation 1: state differential equations**

The column consisting of the state variables is called the state vector, and is denoted as \( x \). Vector \( u \) is the input vector. The output signals are expressed in the output vector \( y \). The dynamic response of the blade can be monitored thusly to view the response of the blade for any given input [44] [23]. Figure 26-4: Phase II flow Diagram below illustrates a flow diagram of the PhLEX test method.
Finite Element Model

A finite element model utilizing Bernoulli beam theory was developed in order to find Eigen values of the system. This model was developed as a lumped mass model where, each node had a unique mass and stiffness. The connecting segments between each node were assumed to be massless [45] [26] [23]. The resulting stiffness matrix for a 2-dimentional element under pure bending is shown in Equation 2: 2-dimentional element under pure bending below [46]. This is the local stiffness matrix of an element comprising of the shear force and moment of each node of the element. The FEA model segmented the blade into 100 and 10 elements for Phase I and Phase II analysis respectively, with each element consisting of six degrees of freedom. The degrees of freedom are: Axial displacement, edge displacement, flap displacement, axial rotation, edge rotation and flap rotation. Each node was modeled with a mass and stiffness specified by an input data file, which was provided by the manufacturer. The eigenvalues and corresponding eigenvectors were found using standard eigenanalysis. This analysis returned important information regarding blade response characteristics such as flap and edge natural frequencies and their corresponding mode shapes.

\[
\begin{bmatrix}
12 & 6L & -12 & 6L \\
6L & 4L^2 & -6L & 2L^2 \\
-12 & -6L & 12 & -6L \\
6L & 2L^2 & -6L & 4L^2 \\
\end{bmatrix}
\begin{bmatrix}
v_1 \\
v_2 \\
\theta_1 \\
\theta_2 \\
\end{bmatrix}
\]

Equation 2: 2-dimentional element under pure bending [46]

Phase I Modeling

This section outlines the methods used to calculate important information pertaining to the system such as the eigenvalues and eigenvectors, natural frequencies and displacements.
Stiffness

The stiffening element was incorporated into the FEM by first determining the support reactions of the actuator, and applying the boundary conditions to the entire system. “By applying boundary conditions of zero rotation and displacement at both the root of the blade, and the point the spring is attached to the ground, the stiffness matrix was modified by only adding stiffness at a chosen node” [47] [26]. Equation 3: Spring Stiffness Matrix below shows the stiffness matrix of the spring. This matrix was added into an optimized location in the local elemental stiffness matrix, which was finally assembled into the global stiffness matrix [26] [47].

\[
\begin{bmatrix}
K & -K \\
-K & K
\end{bmatrix}
\]

**Equation 3: Spring Stiffness Matrix [46]**

The assembled global stiffness matrix is shown in Equation 4: Assembled Global Stiffness Matrix with Spring, below. This stiffness matrix was created based upon a three-element model, which produced four nodes along the blade model. A fifth node was introduced into the model to represent the ground reaction of the spring [26].

\[
\frac{EI}{L^3} \begin{bmatrix}
12 & 6L & -12 & 6L & 0 & 0 & 0 & 0 & 0 \\
6L & 4L^2 & -6L & 2L^2 & 0 & 0 & 0 & 0 & 0 \\
-12 & -6L & 24 & 0 & -12 & 6L & 0 & 0 & 0 \\
6L & 2L^2 & 0 & 8L^2 & -6L & 2L^2 & 0 & 0 & 0 \\
0 & 0 & -12 & -6L & 24 + K & 0 & -12 & 6L & -K \\
0 & 0 & 6L & 2L^2 & 0 & 8L^2 & -6L & 2L^2 & 0 \\
0 & 0 & 0 & 0 & -12 & -6L & 12 & -6L & 0 \\
0 & 0 & 0 & 0 & 6L & 2L^2 & -6L & 4L^2 & 0 \\
0 & 0 & 0 & 0 & -K & 0 & 0 & 0 & K
\end{bmatrix}
\]

**Equation 4: Assembled Global Stiffness Matrix with Spring [46]**

By applying the boundary conditions of a cantilevered connection at both the node of the mounting surface and the spring that was attached to the ground will cause their corresponding rows and columns
to be equal to zero [26]. Equation 5: Final Assembled Global Stiffness Matrix below displays the reduced global stiffness matrix [47].

$$
\begin{bmatrix}
24 & 0 & -12 & 6L & 0 & 0 \\
0 & 8L^2 & -6L & 2L^2 & 0 & 0 \\
-12 & -6L & 24 + K & 0 & -12 & 6L \\
6L & 2L^2 & 0 & 8L^2 & -6L & 2L^2 \\
0 & 0 & -12 & -6L & 12 & -6L \\
0 & 0 & 6L & 2L^2 & -6L & 4L^2 \\
\end{bmatrix}
$$

Equation 5: Final Assembled Global Stiffness Matrix [46]

Natural Frequency

An Eigen analysis was performed using the global mass and stiffness matrices to determine the natural frequencies and mode shapes. The EIG command in MATLAB returned a diagonal matrix of eigenvalues and their corresponding eigenvectors. Recall, the natural frequency is related to the eigenvalue through the following equation:

$$\omega_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}}$$

Equation 6: Natural Frequency [46]

Phase I Results

The primary objective in this phase was to augment the flap natural frequency to match that of the edge. In this phase, this was accomplished by adding stiffness, by means of a theoretical spring or stiffening element (an actuator), in the flap direction. An optimization routine was performed to determine the location of the stiffening element that resulted in the lowest difference in flap and lead-lag natural frequencies. The difference in flap and lead-lag natural frequencies as stiffness was added at varying locations along the length of the blade is tabulated in the Table 1: Flap and Edge Difference in Natural Frequencies below.
Table 1: Flap and Edge Difference in Natural Frequencies [26]

<table>
<thead>
<tr>
<th>Blade Station</th>
<th>Stiffness Added</th>
<th>Blade Angle</th>
<th>Natural Frequency Difference (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>55.00%</td>
<td>5802463</td>
<td>1.271622</td>
<td>0.00038</td>
</tr>
<tr>
<td>60.00%</td>
<td>196147.9</td>
<td>7.132459</td>
<td>0.00134</td>
</tr>
<tr>
<td>65.00%</td>
<td>110673.5</td>
<td>8.573477</td>
<td>0.00059</td>
</tr>
<tr>
<td>70.00%</td>
<td>76081.06</td>
<td>8.641049</td>
<td>0.00053</td>
</tr>
<tr>
<td>75.00%</td>
<td>55191.43</td>
<td>8.210615</td>
<td>0.00107</td>
</tr>
<tr>
<td>80.00%</td>
<td>42302.92</td>
<td>6.868813</td>
<td>0.00090</td>
</tr>
<tr>
<td>85.00%</td>
<td>31418.89</td>
<td>5.997127</td>
<td>0.00067</td>
</tr>
<tr>
<td>90.00%</td>
<td>23274.77</td>
<td>4.987597</td>
<td>0.00097</td>
</tr>
<tr>
<td>95.00%</td>
<td>17048.8</td>
<td>3.812021</td>
<td>0.00086</td>
</tr>
</tbody>
</table>

Figure 27-4: Natural frequency at various blade stations below, shows the effects of adding stiffness to the blade. The stiffness caused a decrease in the edge natural frequency.

As seen in the figure above, the natural frequency linearly decreased as the stiffening element was moved along the length of the blade. It was found that by lowering the natural frequency resulted in an increase in the overall controllability of the system as the system underwent slower accelerations [26]. The acceleration in the edge direction is typically lower than that in the flap. This decrease in
natural frequency, however, resulted in an increase in test duration time as the number of cycles is reduced for a given unit of time. Figure 28-4: Stiffness Ratio below illustrates the flap to edge stiffness ratio. It can be seen that at approximately 65% blade station, the flap and edge stiffness begin to converge. This was the basis of selecting this location for the stiffening element.

![Stiffness Ratio Graph](image)

**Figure 28-4: Stiffness Ratio [26]**

As can be seen in the figure above, this particular blade exhibits high stiffness at approximately 55% blade station. In order to induce any displacement at this location, a very large load would have had to be applied. The stiffness of the flap and edge modes begins to converge quickly after 55% blade station. The ratio of the flap and edge stiffness decreases and approaches a ratio of 1, at approximately the 65% blade station.

The displacement of the blade was based on its mode shape. The target loads were identified then the mode shapes in the flap and edge directions were scaled to meet these target loads. Figure 29-4: Mode Shapes for First Natural Frequency show the corresponding mode shapes for the first natural frequency [26].
Phase I Controller

In phase 1, an adaptive PID controller was used to implicitly control four actuators by controlling the phase angle between the edge and flap motions [9] [23]. The Phase Angle is defined as the degrees of rotation a blade will experience between maximum loading in the flap and edge directions [6]. Field-testing showed that this angle occurs at 72 degrees. This controller had the ability to handle disturbances such as blade softening or heating of the blade during testing. Environmental disturbances, such as heating caused by the sun, cause changes in the material properties of the blade, which lead to variation in the system. The phase angle is detected by means of a peak detection algorithm that utilized the flap and edge displacements. The following Figure 30-4: Phase Angle vs. Time illustrates the response of the 9-meter [23].
The control strategy was used to force the phase angle between flap and edge maximum loading to 72 degrees. This was accomplished by subtracting the current phase angle from the desired phase angle of 72 degrees to get an error signal, which generated a command signal to the actuator [9][23].

**Scaling Issues**

The scaling analysis was performed to see if the current model would produce the same result for larger blades. To confirm this, manufacturer provided data for a 45-meter blade was used to run the analysis outlined in Phase I section. After the optimization routine was performed, the required additional stiffness for the PhLEX test was calculated. This stiffness was then added to the model. The simulation results showed that the required stiffness by PhLEX actuator was large enough to virtually eliminate any displacement at the optimized station.

This led to the conclusion that the model was unsuccessful with determining the response of larger blades, due to the greater difference in Eigen values in the flap and edge directions. It was found that the difference was large enough that the stiffness being added by the actuator actually acted as a restraint, essentially cutting off the length of the blade before the actuator [23]. In order to solve this issue, it was decided that the system would be modeled as a dynamic system as opposed to the system
described thus far. Ultimately, the goal was to use the results of the dynamic model to simulate and determine the nonlinear responses of turbine blades of all sizes.

**Phase II Modeling**

As previously stated, Phase II of the PhLEX test development process began with performing a scaling analysis on the current model. The simulation was run using manufacturer specific data for a 45-meter blade. This analysis targeted the model’s response to larger blades. In Phase II of the PhLEX test development, the blade was modeled as a dynamic lumped mass-spring-damper system. This allowed for the system to be modeled taking parameters such as blade damping and the coupling of flap and edge responses of larger blades into account. This analysis modeled the six degrees of freedom discussed earlier in the modeling section [23].

The initial number of elements was reduced to 10 elements, compared to the 100 elements used in the previous Phase I model. This was done in order to lessen the computational complexity and processor speeds required to perform the calculations for the proof of concept phase. Future works will consider the inaccuracies caused by the current number of elements used to model the system. First the equations of motion were derived by treating the system as a mass-spring-damper system, see Equation 7: Equation of motion of blade below. The left-hand side of the equation represented the actual blade dynamics, while the right-hand side represented the input force [23]. This forcing function was derived based on the flap and edge target bending moments.

\[
m \ddot{x} + c \dot{x} + kx = F(flap, edge) \omega t
\]

*Equation 7: Equation of motion of blade [44] [23]*

Figure 31-4: Blade elemental FBD below illustrates the Freed Body Diagram (FBD) the following equations of motion were derived from.
The differential equations describing the motion of the system were then solved:

\[ m_1 x(t)_{1} = u - k_1 (x(t)_{1} - x(t)_{2}) - c_1 (\dot{x}(t)_{1} - \dot{x}(t)_{2}) \]

\[ m_1 x(t)_{1} + c_1 x(t)_{1} + k_1 x(t)_{1} = u + k_1 x(t)_{2} + c_1 x(t)_{2} \]

Equation 8: Equation of motion of blade [44] [23]

The accelerations of both masses are denoted as \( x(t)_{1,2} \) respectively.

\[ m_2 x(t)_{2} = k_1 (x(t)_{1} - x(t)_{2}) + c_1 (x(t)_{1} - \dot{x}(t)_{2}) - k_2 x(t)_{2} - c_2 x(t)_{2} \]

\[ m_2 x(t)_{2} + (k_1 + k_2) x(t)_{2} + (c_1 + c_2) \dot{x}(t)_{2} = k_1 x(t)_{1} + k_2 x(t)_{2} \]

Equation 9: Equation of motion of blade [44] [23]

After the equations were derived, the model was put in State Space format. The state variables, flap/edge displacements and velocities are as follows:

\[ x_1 = x(t)_{1} \]
\[ x_2 = x(t)_{2} \]
\[ x_3 = \dot{x}_1 = x(t)_{1} \]
\[ x_4 = \dot{x}_2 = x(t)_{2} \]

Equation 10: State Variables [44] [23]
The resulting system in state space format becomes:

\[
\dot{x} = Ax + Bu
\]
\[
x = \begin{pmatrix}
x_1 \\
x_2 \\
x_3 \\
x_4
\end{pmatrix} = \begin{pmatrix}
x(t)_1 \\
x(t)_2 \\
x(t)_3 \\
x(t)_4
\end{pmatrix}, \quad B = \begin{bmatrix} 0 \\ 0 \\ \frac{1}{m_i} \\ 0 \end{bmatrix}
\]
\[
A = \begin{bmatrix}
0 & 0 & 1 & 0 \\
0 & 0 & 0 & 1 \\
-k_1/m_1 & k_1/m_1 & b/m_1 & b/m_1 \\
k_1/m_1 & k_1 + k_2/m_2 & b/m_1 & -k_1 + k_2/m_2
\end{bmatrix}
\]

**Equation 11: State Space System [44] [23]**

Substituting Equation 9: Equation of motion of blade into Equation 1: state differential equations resulted in a 120x120 A matrix. See equation Equation 11: State Space System. Using this format allowed for the system to be analyzed as a Multi Input Multi Output (MIMO) model [23]. The model inputs were the flap and edge forcing functions, which resulted in displacements and velocities as outputs. A step response along with standard Eigen analysis was initially used to find the eigenvectors and eigenvalues.

**Natural Frequencies**

Since the objective of the test was primarily to excite the blade in the edge direction at its natural frequency while forcing the flap at that same natural frequency, only the first few eigenvalues were of interest. Table 2: Flap and Edge Difference in Natural Frequencies below tabulates the first and second flap and edge natural frequencies. The flap and edge eigenvalues corresponding to the first and second modes were selected and used in Equation 6: Natural Frequency to find their corresponding natural frequencies [23]. The flap natural frequency was of no interest for this part of the analysis as the
blade was being excited at the edge natural frequency; however, it was later utilized in the filtering method, which will be discussed in the controller section.

<table>
<thead>
<tr>
<th></th>
<th>Flap Natural Frequency (rad/sec)</th>
<th>Edge Natural Frequency (rad/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1&lt;sup&gt;st&lt;/sup&gt;</td>
<td>5.83</td>
<td>10.47</td>
</tr>
<tr>
<td>2&lt;sup&gt;nd&lt;/sup&gt;</td>
<td>16.37</td>
<td>34.76</td>
</tr>
</tbody>
</table>

Table 2: Flap and Edge Difference in Natural Frequencies

**Target Loads & Excitation Forces**

Next, the excitation forces were calculated in order to match the bending moment specified by the manufacturer. This was accomplished by running the analysis with a 1-Newton of force to the input forcing function then calculating the resulting output bending moment at the root of the blade. The required flap and edge forces were then calculated by finding the additional flap and edge forces to match the manufacture specified bending moments. After the forcing functions were derived, they were applied to the model and the resulting bending moments at the roots were found to be similar [23].

<table>
<thead>
<tr>
<th></th>
<th>Input Force (KN)</th>
<th>Resulting Root Bending Moment (KN.m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flap</td>
<td>0.6</td>
<td>2500</td>
</tr>
<tr>
<td>Edge</td>
<td>400</td>
<td>2900</td>
</tr>
</tbody>
</table>

Table 3: Root Bending Moments and Input Forces

After the simulation was run, it was found that the response of the blade in the edge direction was affected by the flap input forcing function. This resulted in very large edge displacements caused by the flap input forcing function. The exaggerated edge response was primarily due to the close coupling between the flap and edge response. In an attempt to reduce the effects of the flap and edge coupling, the actuator was tilted. An optimization routine was used to find the appropriate angle to position the actuator at in order to minimize the response of the blade in the lead-lag direction from the flap input [23]. As a result, the actuator was modeled at an angle of -2 degrees, which significantly reduced the influence of the flap input on the lead-lag response.
Phase II Results

As previously stated, the idea behind this phase of the analysis was to excite the blade in the edge direction while forcing it in the flap direction at the edge fundamental natural frequency. The blade response was as predicted; however, it was very noisy. This was because, although the forcing input in the flap direction was a function of the edge natural frequency, the fundamental flap natural frequency still impacted the system. In order to smooth out this response, the effects of the flap natural frequencies were minimized by means of a filtering algorithm [23].

A MATLAB function was implemented to create a second state space system based on a derived transfer function that would cancel any disturbances caused by the flap natural frequency. The simulation was carried out using the “LSIM” command, which plots the time response of a linearly time-invariant (LTI) model to the input signals described by \( u \) and \( t \). The time vector \( t \) consists of regularly spaced time samples and \( U \) is a matrix with as many columns as there are inputs, whose \( i_{th} \) row specifies the input value at time \( t \) [48]. This command also allows for more than one state space model to be concatenated. The results from the simulation are illustrated in the following figures.

Figure 32-4: Bending Moment Target Loads [27]
Figure 33-4: Uncontrolled Flap & Edge Response shows the unfiltered Normalized Flap and Edge Displacements. As shown in the figure, the noisy response of the blade is particularly in the flap direction. As discussed earlier, this is due to remnants of the flap natural frequency exciting the blade in addition to the flap forcing function. Figure 34-4: Normalized Unfiltered Flap vs. Edge Displacement shows the Normalized Flap vs. Edge Displacements.

Figure 33-4: Uncontrolled Flap & Edge Response [23]

Figure 34-4: Normalized Unfiltered Flap vs. Edge Displacement [23]
Phase II Control System

The objective of this phase was to excite the blade in the edge direction while forcing it in the flap direction at the edge natural frequency [23]. The response was very noisy however. This was because the blade’s response in the flap direction was still affected by its fundamental natural frequency. In order to smooth out the response, a filter was designed using pole placement. The desired output of the system is called the reference signal. The purpose of the controller was to manipulate the inputs of the system to achieve a desired output, or reference. In the PhLEX model, the desired output was a smooth blade displacement, which was a result of the input forcing function. In order to achieve this, corrective measures were made to the input forcing function, by means of the PhLEX controller.

After the flap and edge natural frequencies were identified, a continuous-time transfer function containing the edge natural frequency in the numerator and the flap frequency in the denominator was applied to the system [23]. This method would effectively cancel the flap natural frequency of the input signal and replace it with the edge natural frequency. The result was a smooth response in both the flap and edge displacements. Figure 35-4: Normalized Filtered Flap vs. Edge Displacement below shows the filtered Normalized Flap vs. Edge Displacements.

Figure 35-4: Normalized Filtered Flap vs. Edge Displacement [23]
Figure 36-4: Controlled Flap & Edge Response below illustrates the response of the system after the flap natural frequency was filtered. This method was effective in smoothing the response of the blade; however, the force requirement to meet the flap and edge target loads increased significantly [23].

![Controlled Flap & Edge Response](image.png)

**Figure 36-4: Controlled Flap & Edge Response [23]**

In addition, the power analysis performed showed that the energy required to perform the task outlined above would be very power costly. Also, the hardware control capabilities at the NREL testing site does not permit for the pole placement filtering method. Figure 37-4: Normalized Total Flap & Edge Displacement below illustrates the response of the blade at every element to the flap and edge forcing inputs. The normalized deflections below depict the response of the blade to the input forces with the implementation of the pole-placement filtering method [23].
ENERGY ANALYSIS

Phase I

In order to weigh the costs associated with such testing methods, a power analysis was performed in order to assess the energy requirements. The power requirements of the system in phase I was found by taking the flow rate and multiplying it by the hydraulic pressure [23]. The flow rate was found using the following equation

\[
\text{FlowRate} = \text{AreaPiston} \times \text{StrokeActuator} \times \text{ExitationFrequency} \quad \text{[26]}
\]

The hydraulic pump was rated at 3000 psi. The sum of the flow rate at each cycle was multiplied by the pump pressure. Results show that the PhLEX system will take approximately 1.62 times more power to conduct the test than previous test methods. However, the time to reach the number of cycles will be decreased by a factor of 2.5. This is because the test is conducted at a higher natural frequency. The
power required to test the 9-meter test specimen using the PhLEX method was found to be approximately, 60 Horse-Power [26]. Figure 38-4: Phase I Power Analysis below illustrates the power requirements from phase I for the PhLEX test.

![PhLEX Power Analysis](image)

**Figure 38-4: Phase I Power Analysis [26]**
Phase II

In phase II of the analysis, the power was calculated from data internal to the model. The forcing function was essential to the power calculations, since they were derived from the target loads. The total power can thusly be calculated by multiplying the Force by the velocity. The resulting velocities the blade underwent during the simulation were used to obtain the power requirements of the system. The resulting power needed for a full-scale 45-meter blade test was found to be approximately 120 Horse-Power [23]. Figure 39-4: Phase II Power Analysis shows that the power requirements greatly increased to more than twice the amount after the rudimental filtering method was applied.

![Figure 39-4: Phase II Power Analysis](image-url)
CHAPTER 5: Conclusions and Future Work

Conclusions

The analysis herein outlines the design of a fatigue test for wind turbines using a newly developed Phase-Locked Excitation testing method. The test takes advantage of resonance testing in order to decrease the overall energy requirements of full-scale wind turbine blade while simultaneously decreasing test duration. Theoretically, the results from both phases of the PhLEX design process indicated that both systems simulated a dual-axis resonant fatigue test with a good deal of certainty. Towards the end of phase one, a scaling analysis was performed on the PhLEX model, which showed that the simulation method was sufficient for smaller blades. The dynamics of larger blades affected the results of this model. In phase two of the design, the model was changed to accommodate for the testing of larger blades. The results from this model are promising. The Power analysis performed on the system showed a weakness in the control system methodology, which lead to large power requirements.

Future Work

Future works will consider the inaccuracies caused by the current number of elements used to model the system. This can be solved by increasing the number of elements for more accurate results. As discussed in the power analysis section of this paper, the current model from phase II gives realistic results for full-scale testing of a 45-meter blade. However, it was found that the power required to perform the test with the current control algorithm will be very energy costly. Future work also exists in upgrading the control system to reduce the overall power requirement. In the coming months, the research team will be performing a proof of concept of the testing methods outlined in Phase II of this paper at the National Renewable Energy Laboratories in Boulder, Colorado. This will give the research team a better idea of the control system options based on the existing hardware at the testing facility. Until then, improvements will be made to the overall accuracy of the model.
Appendix A  Source Code

clc;
clear all;
close all;
format long
useBasicEA = 0;
plotModes = 0;
umele = 10;
useTimo = 0;
cutlength = 0;
excMass = 0; %lb
excMass = excMass * 1; % Convert from lb to kg
position2 = 0;
ActuatorAngle = -2.441033038692329;
stiffness2 = 0; % Timoshenko beam
outboardMassChange = 0;
rotateBladeAngle = 0;
sload(1) = (4.306)*1000+excMass*9.8; %kN converted to N
sloc(1) = 8.9;
sload(2) = 0; %1.401*1000+outboardMassChange*9.8; %kN converted to N
sloc(2) = position2;
global ASP XDot Css Eig_Value;

%Target Loads
rangeFlapLoad = xlsread('Mhi_opt(1).xls','C2:C101');
rangeEdgeLoad = xlsread('Mhi_opt(1).xls','D2:D101');
meanFlapLoad = xlsread('Mhi_opt(1).xls','B2:B101');
segments = xlsread('Mhi_opt(1).xls','A2:A101');

[K, M, mpl, R, eL, EA] = fn_rotateBlade(rotateBladeAngle, sload, sloc, numele, cutlength, useBasicEA, useTimo);

rtsne = sloc(1)/eL+1;
R=R';
R(:,2) = abs(R-position2);
R(:,3)=1:length(R);
R = sortrows(R, 2);
if R(1,2) >= 1e-12
    surroundingNodes = R(1:2,:);
    surroundingNodes = sortrows(surroundingNodes, 3);
    NodeA = surroundingNodes(1,3)*6-3;
    NodeB = surroundingNodes(2,3)*6-3;
    K(NodeA, NodeA) = K(NodeA, NodeA) + (1-
    surroundingNodes(1,2)/eL)*stiffness2;
    K(NodeB, NodeB) = K(NodeB, NodeB) + (1-
    surroundingNodes(2,2)/eL)*stiffness2;
    nodalNum = (NodeB - 3) / 6;
\[
\text{else} \\
\quad \text{NodeA} = R(1,3)*6-3; \\
\quad K(\text{NodeA, NodeA}) = K(\text{NodeA, NodeA}) + \text{stiffness2}; \\
\quad \text{nodalNum} = R(1,3);
\]

\% Set Up of State Space Matrices
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%% %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%% %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%% % A matrix

\text{ASP} = \text{zeros}(120,120); \quad \% \text{Upper Left Corner of Zeros}
\text{ASP}(1:60, 61:120) = \text{eye}(60,60); \quad \% \text{Upper Right Corner Identity Matrix}
\text{ASP}(61:120,1:60) = -\text{M}(7:end,7:end)\text{\textbackslash}K(7:end,7:end); \quad \% \text{Lower Left Corner Stiffness Matrix}
\text{ASP}(61:120,61:120) = -(7.608356987615543e-004)\times(\text{M}(7:end,7:end)\text{\textbackslash}K(7:end,7:end)); \quad \% \text{Lower Right Corner Damping Matrix}
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%% %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%% %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

[\text{Vector},\text{Value}] = \text{eig(ASP, 'nobalance')};
[\text{Eig\_Value},b] = \text{sort(diag(Value))};
[\text{Eig\_Vector},\text{index2}]= \text{sort(Vector)};
\% Calculate the natural frequency of the the second flap
\text{New\_Eig\_Value} = \text{conv(abs(Eig\_Value(3)),abs(Eig\_Value(4)))};
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%% %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%% %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

\% B matrix
\text{Bss}=\text{zeros}(120,2);
\text{Bss}(93,1)=.3\times\cos(\text{ActuatorAngle}); \quad .3
\text{Bss}(92,1)=.3\times\sin(\text{ActuatorAngle}); \quad .3
\text{Bss}(87,1)=.7\times\cos(\text{ActuatorAngle}); \quad .7
\text{Bss}(86,1)=.7\times\sin(\text{ActuatorAngle}); \quad .7
\text{Bss}(92,2)=1;
\% Add mass to B matrix
\text{MInv=}\text{inv(M)};
\text{XDot} =\text{zeros}(120,2);
\text{XDot}(61:120,1)=\text{MInv}(7:66,7:66)\times\text{Bss}(61:end,1);
\text{XDot}(61:120,2)=\text{MInv}(7:66,7:66)\times\text{Bss}(61:end,2);
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%% %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%% %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%C matrix
Css = eye(120);

%D matrix
Dss = zeros(1,1);

%State Space Representations
Sys1_mimo = ss(ASP,XDot,Css,Dss);

[Angle, FVAL, EXITFLAG] = fminbnd(@(Angle) Sen_Optim(Angle,ASP,M),-100,100);
Sys_Controller = tf(conv([1 -Eig_Value(1)],[1 -Eig_Value(2)]), conv([1 abs(Eig_Value(3))],[1 abs(imag(Eig_Value(4))))]);
Sys_Controller_Nu = [Sys_Controller 0; 0 1];
Sys_Controller_SS = ss(Sys_Controller_Nu);

%Forcing Functions
t = 0:.001:200;
Fx1 = 400000*sin(abs(Eig_Value(3)*t));
Fx2 = 600*cos(abs(Eig_Value(3)*t));
U = [Fx1; Fx2];

%System Simulation
Controller_sim = lsim(Sys_Controller_SS,U,t);
%System = lsim(Sys1_mimo*Sys_Controller_SS,U,t);
System = lsim(Sys1_mimo,Controller_sim,t);
%System = lsim(Sys1_mimo,U,t);

%Moment Calculations
SysDeflections = System(length(System)-150000:end,1:60);
ForceSys = K(7:end,7:end)*SysDeflections';
ForceSys = ForceSys;
Lengths=[4.45;4.45*2;4.45*3;4.45*4;4.45*5;4.45*6;4.45*7;4.45*8;4.45*9;4.45*10];
zeroVector=zeros(60,2);
zeroVector(2:6:end,2)=Lengths;
zeroVector(3:6:end,1)=Lengths;
Moments=zeroVector'*ForceSys';

power=one(length(vvect1)*FlapEnergy);
vvect2=120000*Sys_Controller_SS(150000:end,1);
vvect3=vvect2.^2;
ControllerEnergy = sqrt(sum(vvect3)/length(System(150000:end,1)))

A1=[1:10];
A2=[0:9];
A3=[0,0:8];
A4=[0,0,0,0:7];
A5=[0,0,0,0,0:6];
A6=[0,0,0,0,0,0:5];
A7=[0,0,0,0,0,0,0:4];
A8=[0,0,0,0,0,0,0,0:3];
A9=[0,0,0,0,0,0,0,0,0:2];
A10=[0,0,0,0,0,0,0,0,0,0:1];
length=((length-1)>0).*(length-1);
moment=(ForceSys(:,3:6:end))*length;
for i= 1:10000:150000
    figure(1)
    plot(moment(i,:))
    axis([1 10 -5e5 5e5])
    title(num2str(i))
end

for i = 19000:200001
    figure(1)
    subplot(2,1,1)
    plot(System(i,3:6:60))
    axis([0 10 -2 2])
    grid on
    subplot(2,1,2)
    plot(System(i,2:6:60))
    axis([0 10 -2 2])
    grid on
end
Appendix B  Works Cited

Bibliography


[21] M. MALHOTRA, "ADVANCED BLADE TESTING METHODS FOR WIND TURBINES," University of Massachusetts Amherst Department of Mechanical and Industrial Engineering, Amherst, September 2010.


Abstract

Collaborative efforts between Embry-Riddle Aeronautical University (ERAU) and the National Renewable Energy Laboratories (NREL) have resulted in an innovative dual-axis phase-locked resonant excitation (PhLEX) test method for fatigue testing of wind turbine blades. The Dual-axis phase-locked test method has shown to provide more realistic load application as compared to wind loading experienced through field operation conditions. The current concepts involved exciting the blade at its fundamental edgewise natural frequency while applying a force in the flap direction at that same frequency. This advanced test method incorporates existing commercially available test hardware, known as the Universal Resonant Excitation (UREX), combined with an additional hydraulically actuated member to dynamically force the blade using adaptive algorithms and advanced control strategies in order to provide cycle-to-cycle phase control and decreased testing time. In short, this paper will outline the development of a finite element model for predicting performance and evaluation of the results.

INTRODUCTION

The purpose of this paper was to develop a dynamic model of a dual-axis Phase-Locked Excitation (PhLEX) fatigue test for wind turbine blades. It was crucial to first gain an understanding of the loads that these turbine blades are subjected to, along with characteristics such as blade materials and current blade testing methodologies. The loads described herein correspond to those experienced by modern multi mega watt horizontal wind turbine blade in service. Results from such tests, give wind turbine blade manufacturers important information regarding the structural integrity of their designs. Also, they offer solutions on how to further increase the strength of these blades. This paper will discuss the development of the dynamic Finite
Element Analysis (FEA) model and results from modeling a 9-meter and 44.7-meter blade.

**History of Wind Turbine Blade Fatigue Testing**

The importance of structural testing of wind turbine blades lies in the quantitative information it relays to the manufacturers regarding the structural integrity of their blades, manufacturing quality, and design durability [1]. These full-scale blade fatigue tests are conducted at the National Renewable Energy Laboratory’s (NREL) National Wind Technology Center (NWTC). The NWTC is ever developing more efficient methods of fatigue testing that employ resonant excitation systems with goals such as: increasing testing speed, reducing hydraulic requirements, and improving load distribution, in mind. Structural testing of wind turbine blades began at the National Renewable Energy Laboratory in Boulder, CO, in 1990[White, 24]. Within a couple of years the NREL testing facility was capable of testing blades up to 28 meters in length. The motivation behind these fatigue tests was verifying that a given turbine blade would function reliably while meeting its expected life cycle [19-22]. Typically, a single equivalent load case is applied during testing. This load case is calculated either experimentally or from design load conditions specified by the manufacturer and usually includes magnification factors [7]. There are only a handful of facilities capable of performing full-scale structural blade testing, such as the one outlined herein. These facilities employ either one of two testing methods: the single-axis resonance excitation test or the dual axis forced-displacement system [24-26]. In the case of a dual-axis resonant test, the blade was excited through multiple actuators at two distinct frequencies corresponding to the flapwise and lead-lag frequencies. A primary objective of a dual-axis test is to test the blade to equivalent damage moments in multiple axes [7]. This type of testing concept was modified in order to develop the PhLEX test system.

**BLADE PROPERTIES**

**Construction and Material**

Modern wind turbine blades have a varying airfoil cross-section that culminates in a circular cross section at the root to allow for connection to the hub. The length of these blades can vary from 9 meters to anywhere up to 100 meters. Typically, the maximum root occurs at 20% span of the blade [1]. The turbine blade is designed to transfer aerodynamic (lift) and inertial loads (gravity) to the rotating hub, which is then converted into electricity. Wind turbine blades are typically made of composite material, fiberglass with epoxy or vinyl ester matrices, while shear webs help to distribute loads while also adding stiffness. Since wind turbine blades have become more prevalent in length, materials such as carbon fiber have been added in order to keep the blades light while maintaining structural integrity [1].

**Figure 1** Coordinate geometry of wind turbine blade

The three primary directions used in the analysis of wind turbine blades are: span-wise, flap-wise (flap) and edge-wise (edge) directions. Figure 1. shows the directions on the geometry of a blade. Due to the orientation of the blade while operational and its shape, the turbine blade is normally much stiffer in the edge direction than in the flap direction. Also, as wind turbines have grown in size, so have their respective loads. In the case of smaller wind turbines, most of the applied loads were in the flap direction, which were caused by wind loads. Larger blades however, experience almost equal loads in both the flap and edge direction. This is mainly due to loads caused by gravity, in the edge direction. This will be further discussed in the following sections.

**Normalized Properties**

Due to the proprietary nature of wind turbine blade properties, data such as blade stiffness and mass cannot be published. However, information such as flap/edge stiffness and mass per length can be conveyed in a normalized distribution manner for most blades [1]. The following figures illustrate the normalized Mass Per unit Length (MPL) and stiffness in both the flap and edge direction as a function of the normalized blade station.
From the Figure 2, it can be seen that there is significant mass associated with the blade, towards the root. This is primarily due to the fact that wind turbine blades are structurally reinforced at the root for blade attachment purposes, which is accomplished through bolts laid directly into the root. Not as pronounced in this blade is the increase in mass during the transition from the root circle geometry to the max chord which occurs generally around the 25% span. The last 75% span can be best described as a linear decreasing profile from the max chord to the tip. Almost all wind turbines share this mass distribution [1].

The flap stiffness is defined as the resistance to displacement for a given force in the flap direction. As previously mentioned, due to the blade geometry, the stiffness in the edge direction is typically higher than in the flap. Typically, the stiffest portion of the blade occurs at the root, as shown in Figure 3. The attachment fasteners are embedded into the blade material at this section. This in turn adds stiffness in relation to the rest of the blade. The effects of the mounting area of the blade are shown in the figure 3, which in the flap direction the stiffness is quite high at the root, and drops significantly as the geometry changes [1,8].

The edge stiffness refers to the resistance to bending in the edge direction. The edge stiffness is similar in characteristics to the flap stiffness distribution, but varies typically by having higher values and a shallower decay. The increase in stiffness at the 15% station in figure 4 above is not common occurrence in wind turbine blades [1].

**BLADE LOADS**

As previously stated, there are two essential categories of loads that a wind turbine is subjected to: aerodynamic loads and inertial loads. These loads occur orthogonally to one another. Taking a closer look, we find that the aerodynamic forces are a combination of stochastic and deterministic loads which result in lift, drag and shear forces which generally act in the flap direction. Steady, non-uniform aerodynamic loads cause cyclic loading of the turbine blade due to an increase in wind speed at higher elevations, turbulence caused by the tower, and cross-flow on the rotor [18].

The inertial loads are mainly due to gravity and act in the lead-lad direction [7]. The cyclic nature of inertial loads is due to the gravity force reversing as the
blades rotate around the hub [18]. For smaller wind turbine blades, the bending moment in the flap direction is the predominant fatigue factor [9,10]. However, since lighter and stiffer materials have been developed and incorporated into the design of wind turbines, the blades have grown considerably. Thusly, the inertial loads due to gravity have continually been growing and are as equally important as the flap loads. This results in much higher loading at the rotor in the edge direction than what was previously experienced in smaller blades. Since the aerodynamic and inertial loads are applied in a cyclic manner, fatigue is the most common mode of failure [7]. This highlights the importance of performing fatigue tests to better ensure the reliability of turbine blades before they enter the production phase.

**TESTING METHODOLOGIES**

To increase the power captured by wind turbines, the swept area of the blades must increase. This requires longer and more expensive blades. A number of tests are used to validate the design of wind turbine blades. Fatigue testing of the complete blade allows for demonstration of the blade’s lifespan. Currently, there are three methods for testing of wind turbine blades: single-axis resonance method, dual-axis forced displacement, and dual-axis resonance method [7].

The single axis resonance test was developed for small blades in which the flap moment is much greater than the edge moment. As wind turbine blades increase in length and weight, the edge moment has increased more than the flap moment due to gravity. As a result, the flap and edge moments are on the same order of magnitude. To more accurately test the turbine blades, the dual-axis forced displacement method was developed to more closely represent the moments experienced in the field.

The dual-axis forced displacement tests the flap and edge directions independently, which can cause inaccurate results [7]. The most widely used test system today is the Universal Resonant Excitation system (UREX), which utilizes the dual-axis resonant method [11]. The dual-axis resonance method excites the flap resonance frequency while forcing the edge displacement. This method provides a more accurate stress distribution than either of the previously discussed methods [12]. By using both resonance and forced displacement to load the blade, the phase angle is unable to be controlled. This results in a load on the blade that is not seen in the field [13].

**PHLEX MODEL CONSIDERATIONS**

The Phase-locked excitation fatigue test was developed in two stages. The first conceptual model design allowed for testing of a turbine blade at a predetermined phase angle in order to load the blade during testing in the same manner it is loaded in the field [7].

**Phase 1**

In order to lock the phase angle of the resonant test system, a solution of adding a stiffener in the flap direction was proposed in order to modify the natural frequency of the blade in flap direction, and make it approximately equal to the natural frequency in the edge direction. The modeling was performed utilizing a MATLAB script that was developed based upon previous quasi-static blade testing models. This test was designed with the following in mind: the ability to lock the phase angle between the edge and flap directions of the blade for more realistic blade loading conditions, and decreasing the blade test duration by allowing both the edge and flap tests to be completed simultaneously [12,8].
The analysis in this phase was performed using normalized blade properties of a 9-meter blade. Figure 7 illustrates a sketch of the PhLEX test set up. Results and findings will be discussed in the following sections. This method proved effective for smaller blades, however, the results did not hold when the model was scaled to accommodate for larger blades. The model was unsuccessful with larger blades due to the greater difference in Eigen values in the flap and edge directions. The difference was large enough that the stiffness being added by the actuator actually acted as a restraint, essentially cutting off the length of the blade before the actuator.

**Phase II**

Phase two of the design processes took a different approach toward achieving the same solution as in phase I, with the scaling problem in mind. The major difference being in the way the Finite Element Analysis (FEA) was performed. Instead of modeling the system as a quasi-static model, a linearized dynamic Finite Element Model (FEM) was also developed using MATLAB.

This model took advantage of exciting a 45-meter blade at its edge natural frequency, in the edge direction. Simultaneously, a forcing function that was a function of that same natural frequency, was applied in the flap direction. Figure 8 illustrates a sketch of the PhLEX test set up used in phase II of the testing analysis.

**MODELING CONCEPTS**

**Finite Element Analysis**

Finite Element Analysis (FEA) was first developed in 1943 by R. Courant, who utilized the Ritz method of numerical analysis and minimization of variation calculus to obtain approximate solutions to vibrating systems [14]. FEA programs use a system of points called nodes which make a grid that is referred to a mesh. The mesh is programmed to contain the material and structural properties of the model which will define how the structure reacts to various loads that are applied [14]. A finite element model was developed to simulate a dynamic test in or to produce loads and deflections that are based on the wind turbine blades properties. The target actuator loads were previously calculated so that bending moments in the flap and edge directions would match the specified moments. The purpose of performing dynamic tests simulations are to simulate the dynamic loading applied to the blade that would occur during a resonant fatigue test.

**Bernoulli-Euler vs. Timoshenko**

Since the Timoshenko beam theory is of a higher order than the Bernoulli-Euler beam theory, it is known to be superior in predicting the transient response of a beam [15]. It is shown that the Timoshenko model superiority is more pronounced for beams that have a low aspect ratio. Beam elements in Timoshenko beam theory, have transverse shear strain constant through the cross section; the cross section remains undistorted after deformation of the beam. Due to this limitation of first order shear deformation, Timoshenko beam theory can only be used on thick beams. Unlike the Timoshenko beam theory the Bernoulli-Euler beam theory shear deformations are neglected and the plane sections remain plane and normal to the longitudinal axis. A finite element model utilizing Bernoulli beam theory was developed in order to find Eigen values of the system.

**State Space Representation**

In order to develop a dynamic representation of the system in phase two of the design process, a state space model was developed in MATLAB. The concept of a state space model is physically modeling a system with a set of inputs and outputs. The system consists of state variables, which are defined by its equations of motion. The state differential equations relate the rate of change of the state of the system to the input signals. These state variables fully describe the system and its response to a given input. The state differential equations are as follows:

\[
\dot{x} = Ax + Bu
\]

\[
y = Cx + Du
\]

The column consisting of the state variables is called the state vector, and is denoted as \( x \). Vector \( u \) is the input
vector. The output signals are expressed in the output vector \( \mathbf{y} \). The dynamic response of the blade can be monitored thusly to view the response of the blade to any given input [17]. In Figure 9 below, is a flow diagram of the PhLEX test method.

Figure 9 PHLEX Flow Diagram

MODAL ANALYSIS
The purpose of the modal analysis is to verify that the finite element models correctly portray the blade properties and produced realistic outputs with no external forces applied. This corresponds to static loading of the blade (without external forces being applied [1]). The curve shape for the loads in the flap and edge directions are very similar, however, the magnitude of the load in the flap direction is roughly twice that of the in the edge direction [1].

Introduction
The code consists of a main driver MATLAB file that performs the PhLEX test simulations. The driver file calls a number of functions that perform a variety of tasks, from reading input data files to performing optimization routines. To briefly explain the input Blade Data file, there are five tabs, which contain normalized blade data, flap and edge load data. The blade data specifies the normalized properties given by the manufacturer which include station, mass-per-unit-length, chord, twist angle, flap stiffness, edge stiffness, torsional stiffness, and axial stiffness. Due to the extensive research and case studies, performed at NREL, to develop curve fits for blade properties this code has the ability to interpolate any missing blade properties [16].

Phase I
In phase one of the model development, the 9-meter blade was modeled as a lumped mass system. The code broke the blade into 100 elements with each element consisting of six degrees of freedom. The degrees of freedom are: Axial displacement, edge displacement, flap displacement, axial rotation, edge rotation and flap rotation. Each node was modeled with a mass and stiffness based on the normalized manufacturer’s data from the input data file. The connecting material between nodes was assumed to be massless. The eigenvalues and corresponding eigenvectors were found using standard eigenanalysis. This analysis returned important information regarding blade response characteristics such as flap and edge natural frequencies and their corresponding mode shapes.

Figure 10 Modal Analysis

Figure 11 Natural Frequency vs. Stiffness

The primary objective was to augment the flap natural frequency to match that of the edge. In this phase, this was accomplished by adding stiffness, by means of a theoretical spring or stiffening element (an actuator), in the flap direction. Figure 11 shows that adding stiffness to the blade causes a decrease in the edge natural frequency. It was also found that by lowering the natural
frequency effectively increased the overall controllability of the system [8]. Figure 12 below illustrates the flap to edge stiffness ratio. It can be seen that at approximately 67% blade station, the flap and edge stiffness begin to converge. This was the basis of selecting this location for the stiffening element.

![Figure 12 Stiffness Ratio](image)

The displacement of the blade is based upon its mode shape. The target loads were identified then the mode shapes in the flap and edge directions were scaled to meet these target loads. Figures 13 & 14 show the corresponding mode shapes for the first and second natural frequencies [8].

![Figure 13 Mode Shapes for First Natural Frequency](image)

![Figure 14 Mode Shapes for Second Natural Frequency](image)

The results from this system were promising when compared to previous test methods. However, a scaling study showed that the amount of stiffness required to match the flap and edge natural frequencies for a 44.5-meter blade was very large. This would effectively hold the blade stationary at the location of the stiffening element. Phase two of the design process took this in mind. For detailed information on this phase of the testing development, see [8].

**Phase II**

In phase two of the PhLEX test development, the blade was modeled as a dynamic lumped mass-spring-damper system. This allowed for the system to be modeled taking parameters such as blade damping and the dynamics of larger blades into account. This analysis modeled the six degrees of freedom discussed earlier, with ten elements. In this phase, the number of elements was reduced in order to lessen the computational complexity and processor speeds required to perform the computations for the proof of concept. Future works will consider the inaccuracies caused by the current number of elements used to model the system.

First, the equations of motion were derived using the formula shown below. The left hand side of the equation represented the actual blade, while the right hand side represented the input forcing function. This forcing function was derived based on the flap and edge target bending moments. Because of the extensive processor requirements to perform such calculations, the number of elements was reduced to ten.
\[ m \ddot{x} + c \dot{x} + kx = F(\text{flap, edge}) \omega t \]

The differential equations describing the motion of the system were then solved for:

\[
\begin{align*}
m_t x(t)_1 &= u - k_t(x(t)_1 - x(t)_2) - c_t(x(t)_1 - x(t)_2) \\
m_t \ddot{x}(t)_1 + c_t \dot{x}(t)_1 + k_t x(t)_1 &= u + k_t x(t)_2 + c_t x(t)_2
\end{align*}
\]

The accelerations of both masses are denoted as \( x(t)_1 \) and \( x(t)_2 \) respectively.

\[
\begin{align*}
m_2 x(t)_2 &= k_2(x(t)_1 - x(t)_2) + c_2(x(t)_1 - x(t)_2) - k_2 x(t)_2 - c_2 x(t)_2 \\
m_2 \ddot{x}(t)_2 + (k_1 + k_2)x(t)_2 + (c_1 + c_2) x(t)_2 &= k_1 x(t)_1 + k_2 x(t)_2
\end{align*}
\]

After the equations were derived, the model was put in State Space format. The state variables flap/edge displacements and velocities are as follows:

\[
\begin{align*}
x_1 &= x(t)_1 \\
x_2 &= x(t)_2 \\
x_3 &= \dot{x}(t)_1 \\
x_4 &= \dot{x}(t)_2
\end{align*}
\]

The resulting system in state space format becomes:

\[
\dot{x} = Ax + Bu
\]

\[
x = \begin{pmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{pmatrix} = \begin{pmatrix} x(t)_1 \\ x(t)_2 \\ \dot{x}(t)_1 \\ \dot{x}(t)_2 \end{pmatrix}, \quad B = \begin{pmatrix} 0 \\ 0 \\ \frac{1}{m_1} \\ 0 \end{pmatrix}
\]

\[
A = \begin{pmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ \frac{k_1}{m_1} & \frac{k_1}{m_1} & \frac{b_1}{m_1} & \frac{b_1}{m_1} \\ \frac{k_2}{m_2} & \frac{k_1 + k_2}{m_2} & \frac{b_1}{m_2} & \frac{k_1 + k_2}{m_2} \end{pmatrix}
\]

Using this format allowed for the system to be analyzed as a Multi Input Multi Output (MIMO) model. The inputs were flap and edge forcing function, which resulted in displacements and velocities as outputs. A step response along with standard Eigen analysis was initially used to find the eigenvectors and eigenvalues. Since the objective of the test was to excite the blade in the edge direction, while forcing the flap at the same natural frequency, only the lowest eigenvalues were of interest. Recall, the natural frequency is related to the eigenvalue through the following equation:

\[
\omega_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}}
\]

The next step involved matching the bending moments specified by the manufacturer. This was accomplished by applying a 1-Newton of force to the input forcing function then calculating the resulting output bending moment at the root of the blade. This resulting bending moment was compared to the manufacturer’s target loads and the resulting required force was obtained. After the forcing function was derived, it was applied to the model and the response was obtained. Figure 15 below shows the normalized target bending moment in the flap and edge direction and the matching simulated moments.

![Figure 15 Target Loads](image)

Interestingly, it was found that the response of the blade in the edge direction was greatly affected by the flap input forcing function. This resulted in a very large edge response caused by the flap input forcing function. The reason for the codependency between the flap and edge responses of the blade is mainly due to the shape and the varying angular twist spanning the length of the blade. To solve this issue, the forcing actuator was positioned at an angle in order to reduce the influence of the flap input on the response of the edge. Furthermore, an optimization routine was used to find this minimized angle.
PhLEX Control System

Phase I
In this phase an adaptive PID controller was used to implicitly control four actuators by controlling the phase angle between the edge and flap motions. Field testing showed that this angle occurs at 72 degrees. The adaptively of the controller is due to its ability to handle disturbances such as blade softening or heating of the blade during testing. This changes the material properties of the blade, which leads to variation in the system. The phase angle is detected by means of a peak detection algorithm that utilizes the flap and edge displacements. The following figure 16 illustrates the response of a 9 meter blade to the discussed control strategy.

The PID controller is used to force the phase angle to 72 degrees. This algorithm subtracts the current phase angle from the desired phase angle of 72 degrees to get an error signal, which in turn sends a command signal to the actuator [13].

Phase II
Again, the objective of this phase was to excite the blade in the edge direction while forcing it in the flap direction at the edge natural frequency. This is theoretically possible, however, the response will be very noisy.

Figure 17 Uncontrolled Flap & Edge Response
This is because the blade is still affected in the flap direction by its flap natural frequency. In order to smooth out the response, a filter was designed using pole placement. Figure 18 shows the unfiltered Normalized Flap vs. Edge Displacements, where as Figure 19 shows the filtered Normalized Flap vs. Edge Displacements.

Figure 16 Phase Angle vs. Time

Figure 18 Normalized Unfiltered Flap vs. Edge Displacement
After the flap and edge natural frequencies were identified, a transfer function containing the edge natural frequency in the numerator and the flap frequency in the denominator. This effectively cancelled the flap frequency and replaced it with the edge frequency, which resulted in a smooth response to any impulse.

Figure 20 Controlled Flap & Edge Response

Figure 20 above illustrates the response of the system after the flap natural frequency was filtered. This method was effective in smoothing the response of the blade, however, the force requirement to meet the flap and edge target loads increased significantly. Furthermore, the power analysis (discussed in the following section) performed showed that the energy required to perform the task outlined above was very power costly. Also, the hardware control capabilities at NREL do not permit for such filtering methods. Figure 21 below illustrates the response of the blade at every element to the flap in edge forcing inputs. Note: These deflections correspond to the input forces calculated with the implementation of the pole-placement filtering method.

**ENERGY ANALYSIS**

**Phase I**

The power requirements of the system in phase I were found by taking the flow rate and multiplying it by the hydraulic pressure. The flow rate was found using the following equation

\[
\text{FlowRate} = \text{AreaPiston} \times \text{StrokeActuator} \times \text{ExitationFrequency}
\]

The hydraulic pump was rated at 3000 psi. The sum of the flow rate at each cycle was multiplied by the pump pressure. Results show that the PhLEX system will take approximately 1.62 times more power to conduct the test than previous test methods. However, the time to reach the number of cycles will be decreased by a factor of 2.5. This is because the test is conducted at a higher natural frequency. Figure 22 below illustrates the power requirements from phase I, for the PhLEX test. The maximum power required to test a 9-meter blade using the PhLEX method was roughly 60 hp [8].
Phase II
In this phase of the analysis, the power was calculated from data internal to the model. Since the forcing functions were derived from the target loads, the resulting velocities the blade underwent during the simulation were used to obtain the power requirements of the system. The resulting power needed for a full scale 45-meter blade test was in the order of 120 horse power. Figure 23 greatly increased to more than twice this amount after the rudimental filtering method was applied.

Future Work
Future work takes into account the inaccuracies caused by the current number of elements used to model the system. This can be solved by increasing the number of elements for more accurate results. As discussed in the power analysis section of this paper, the current model from phase II gives realistic results for full-scale testing of a 45-meter blade. However, it was found that the power required to perform the test with the current control algorithm will be very energy costly. Future work also exists in upgrading the control system to reduce the overall power requirement. Also, in the coming months, the research team will be performing a proof of concept of the testing methods outlined in this paper at the National Renewable Energy Laboratories in Boulder, Colorado. This will give the research team a better idea of the control system options based on the existing hardware at the testing facility. Until then, improvements will be made to the overall accuracy of the model.

CONCLUSIONS AND FUTURE WORK
The analysis herein outlines the design of a fatigue test for wind turbines using a newly developed Phase-Locked Excitation testing method. The test takes advantage of resonance testing in order to decrease the overall energy requirements of full-scale wind turbine blade while simultaneously decreasing test duration. Theoretically, the results from both phases of the PhLEX design process indicated that both systems simulated a dual-axis resonant fatigue test with a good deal of certainty. Towards the end of phase one, a scaling analysis was performed on the PhLEX model, which showed that the simulation method was sufficient for smaller blades. The dynamics of larger blades affected the results of this model. In phase two of the design, the model was changed to accommodate for the testing of larger blades. The results from this model are promising. The Power analysis performed on the system showed a weakness in the control system methodology, which lead to large power requirements.

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REFERENCES
[5] Website of the National Renewable Energy Laboratory blade testing facility,
http://www.duwind.tudelft.nl/