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# Full-scale modal wind turbine tests: comparing shaker excitation with wind excitation

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## ABSTRACT

The test facilities at the National Wind Technology Center (NWTC) of the National Renewable Energy Laboratory (NREL) include a three-bladed Controls Advanced Research Turbine (CART3). The CART3 is used to test new control schemes and equipment for reducing loads on wind turbine components. As wind turbines become lighter and more flexible to reduce costs, novel control mechanisms are necessary to stop high winds from damaging the turbine. However, wind turbines must also be designed to capture the maximum amount of energy from the wind, so engineers must devise new ways of achieving this while controlling wind loads that would cause the turbines to fatigue quickly. New control mechanisms and computer codes can help the wind turbine shed some loads in extreme or very turbulent winds. The special configuration of the CART3 allows researchers to analyze these diverse control schemes.

This paper reports on the initial results of a major full-scale modal testing campaign to validate and refine simulation models. One model is a tailored multi-body dynamic simulation model that will be used to develop an advanced controller designed to optimize power and minimize structural loads. Researchers would also like to tune Finite Element Models of the blades, nacelle and tower assembly to predict the higher order rotating modes of the wind turbine for a range of inflow conditions.

The paper will discuss an Experimental Modal Analysis approach where the wind turbine in parked condition is excited by shakers connected with cables. This approach will be compared to Operational Modal Analysis where the same structure is subjected to wind excitation without the shakers activated. These tests and data analyses will provide experience and increase confidence in the approach used for future tests in rotating conditions.

## 1 INTRODUCTION

During operation, wind turbines, like many engineering structures, are subject to dynamic loads, because the aerodynamic and gravitational loads vary with time as the turbine rotates [1]. The optimal design of a turbine is dependent on reducing vibrations, which can be done by accurately identifying turbine's natural frequencies and mode shapes [2]. By determining the modes of the wind turbine, it can be ensured that the turbine's operational conditions preclude resonant frequencies, thereby minimizing dynamic loads and lengthening the life of the turbine [2][3]. In order to develop accurate structural models, extensive experimental testing must be done to refine and validate computer simulations. Modal testing, an experimental method for measuring the modes (natural frequencies, mode shapes, and damping ratios) of a structure, is a common method for validating a structural model as modes are readily computed from models and define the dynamics of the turbine over the operating speed of the rotor [1].

In addition, dynamic loads can be reduced through controls, which actively damp component vibration. Wind turbine control can increase power production while reducing fatigue on the turbine parts. In order to accurately design controls, the turbine system model must accurately predict the dynamics for a variety of operating

conditions [4][5]. Poorly designed controls can result in control loop interference, causing the turbine to vibrate unstably, introducing large damaging loads. Modal testing validates the inputs and plant parameters used to develop and test the controls model. The more insight a modal test provides, the more accurate a model becomes, and the better the final structure operates. This modal investigation will be used to validate an analytical structural model and the chosen inputs used for a turbine control model.



**Figure 1: (Left) Test facilities at NREL's National Wind Technology Center (NWTC), Boulder, Colorado. (Right) CART3 – Controls Advanced Research Turbine: 3-bladed – 600 kW Wind Turbine (40m rotor diameter & hub-height).**

In a traditional Experimental Modal Analysis (EMA), measured input forces are applied by shakers connected to cables which excite the structure [6][7][8][9]. In this method, both input and output signals are measured to collect Frequency Response Functions (FRFs). By fitting a parametric model through the FRFs, the modal parameters of the structure are extracted. This method provides an approximation of the static (i.e. parked rotor) system modes, which are used as a starting point for tuning the structural and control models. However, in this method, the experimentally determined vibrations are not the turbine's pure modes [9]. Mode shapes are changed by boundary conditions, and shaker excitation ignores the ambient loads acting on the turbine during operation: the wind. Wind excitation has sufficient energy to excite the fundamental system modes, however, the input forces are not measured and therefore, FRFs cannot be computed. In these cases the deterministic knowledge of the input is replaced by the assumption that the input is white noise and Operational Modal Analysis (OMA) techniques are applied to extract the modal parameters from output-only measurements. OMA allows the investigator to collect modes that are relevant to the operation of the machine. A review of OMA methods can be found in [10] and some recent developments are discussed in [11]. The original application of OMA (although that name was not in use at that time) to wind turbines can be found in [12][13].

Modal tests can be conducted with the turbine in a parked position or in rotational operation. Standstill modes are useful for validating and refining models, however there is no inclusion of the rotational effects that dominate modes [9]. Modal characteristics vary between a parked and rotating turbine, so including rotational effects is important for computing modal parameters [3][6]. Rotational testing has been conducted since the 1980s. Carne et al. [3][6][12][13] focused on vertical-axis wind turbines (VAWTs), which do not deal with the changes in gravitational forces acting on the rotating structure or the bending moments acting out of the plane of rotation – challenges that rotating horizontal axis turbines face. Horizontal-axis wind turbines (HAWTs) have been tested with a variety of excitation methods [9].

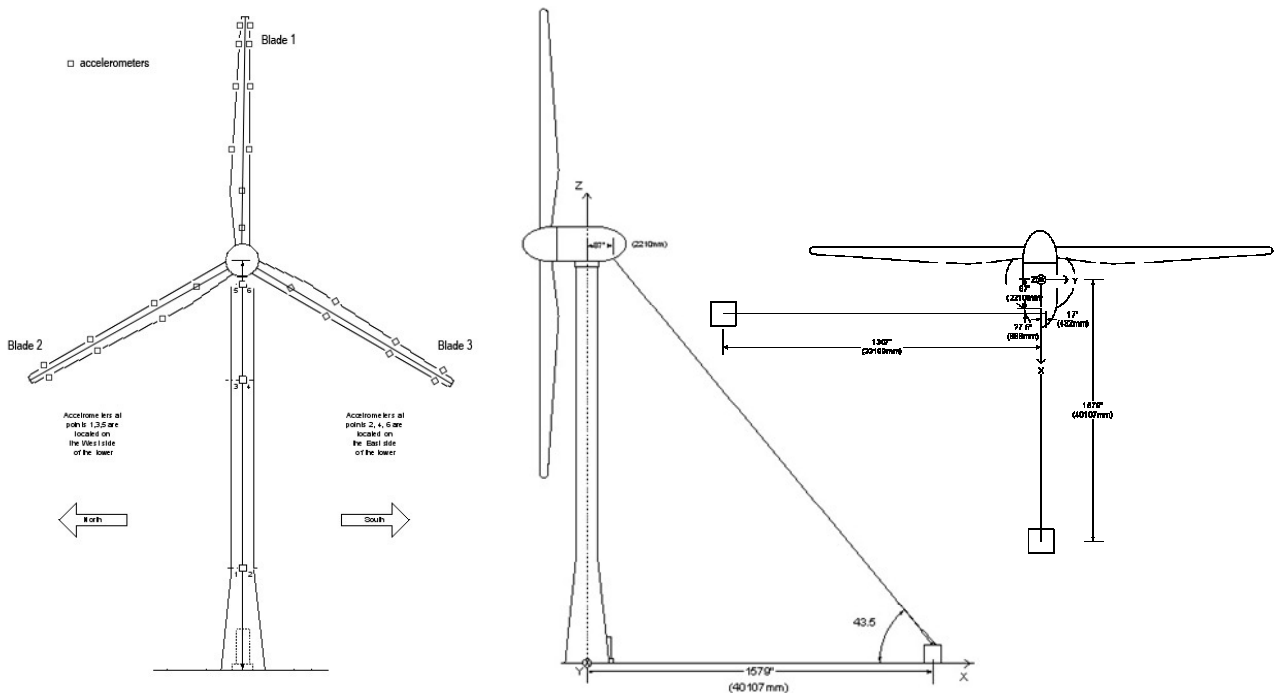
This investigation describes a series of full modal tests conducted on a highly modified Westinghouse model WWG-0600 wind turbine known as the CART3 (three-bladed Controls Advanced Research Turbine; see Figure 1). Standard EMA test methods are employed to identify the parked-rotor or static wind turbine modes of vibration; hydraulic shakers are used to excite the turbine in the fore-aft and side-to-side directions (i.e. along the

turbine's wind axis and across the turbine's wind axis). A secondary test using wind excitation to collect an ambient response is also conducted. OMA is applied to this wind excitation data. A new distributable data acquisition system (DAS) is used to collect both static and operational modes of vibration. This new next-generation DAS allowed for measurements to be collected in the wind turbine hub, nacelle, and tower in stark contrast with the old DAS, which was centrally located in the test shed at the base of the wind turbine tower.

## 2 STRUCTURE AND DATA ACQUISITION

### 2.1 Wind Turbine Description

The three-bladed rigid-hub upwind rotor of CART3 incorporates independent full span pitch of each blade to control aerodynamic torque. The drive train employs advanced power electronics which allows for variable speed operation unlike the original constant speed synchronous generator. The rotor diameter is approximately 41 m and operates over a speed range of 10-43 RPM, and produces rated power of 600 kW for a wind speed of 18 m/s. The hub height of the turbine is 36.6 m. The tower base has a diameter of 4.42 m, which linearly tapers to 2.18 m at a height of 9.20 m from ground. From 9.20 m to the nacelle, the tower has a constant 2.18 m diameter. The drive train has a low-speed brake located between the main shaft and gearbox and a high-speed brake at the end of the generator shaft. The instrumented CART3 turbine is shown in Figure 2 (Left). As shown in the figure, this test has blade one aligned vertically with the yaw axis and blades two and three oriented 120° about either side of it.



**Figure 2: (Left) CART-3 turbine node map. Accelerometers were located along the tower wall, blade edges, and nacelle for modal characterization. (Right) Hydraulic shaker set-up, side and top view. One steel cable was connected to the nacelle and another to the tower.**

### 2.2 Approach and Test Setup

The initial test was conducted similar to the CART2 test done in 2002 [7], using a multiple-input/multiple-output (MIMO) procedure to measure the modes. Like the CART2 [7], the turbine was excited in the fore-aft and side-to-side directions utilizing two independent hydraulic actuators. The fore-aft actuator was connected to the downwind side (East) of the nacelle while the side-to-side actuator was connected along the tower's South face, as shown in Figure 2 (Right). Tests were conducted running one actuator at a time. These different orientations were used to put energy into the nacelle pitching and yawing modes. The tower side-to-side excitation put energy in the rotor's in-plane and asymmetric bending modes. The tower fore-aft excitation put energy into the rotor's symmetric and

asymmetric out-of-plane bending modes. In order to determine modes excited at low energy, the turbine was parked in the wind, and excited by the ambient wind.

### 2.3 Test Equipment and Instrumentation

Figure 2 (Left) provides a schematic of the setup for the modal test. Force transducers were attached to the nacelle and tower to measure the force applied to the turbine by the hydraulic ram. These sensors were attached to the ram by a high-strength, tensioned steel-cable (Figure 3). Biaxial and triaxial accelerometers were configured from single-axis accelerometers. Twenty-four biaxial accelerometers (48 channels) were used to map the rotor's in-plane and out-of-plane bending modes. The blade accelerometers were located along the leading and trailing edges, as well as along the chord line of the blade near its root. One axis was aligned to measure flap (out-of-plane bending) and the other lag (in-plane bending) motion. Blade one was equipped with more accelerometers to determine higher-order mode shapes (Figure 4). Five triaxial accelerometers (15 channels) were used to map bending in the drive train (nacelle). The accelerometers were oriented to capture the fore-aft, side-to-side, and the pitching motion of the nacelle (Figure 5 – Right). Finally, six biaxial accelerometers (12-channels) were used to map the bending modes of the tower. These sensors were placed along the upwind and downwind sides; collecting data in the fore-aft and side-to-side directions (Figure 5 – Left). The coordinate systems and orientations comply with the International Electrotechnical Commission (IEC) convention for the geometry of the blades, tower, and nacelle, and are shown in Figure 4 and Figure 5.



Figure 3: CART3 excitation by hydraulic shaker.

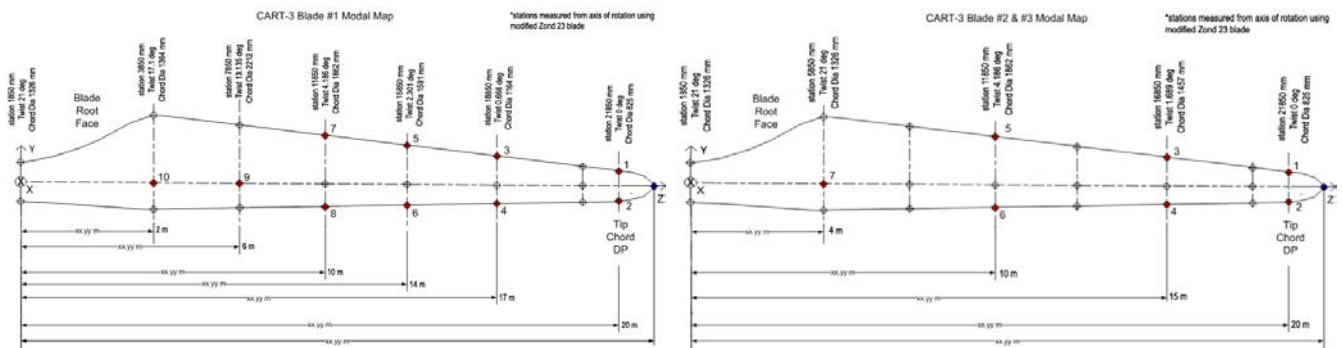
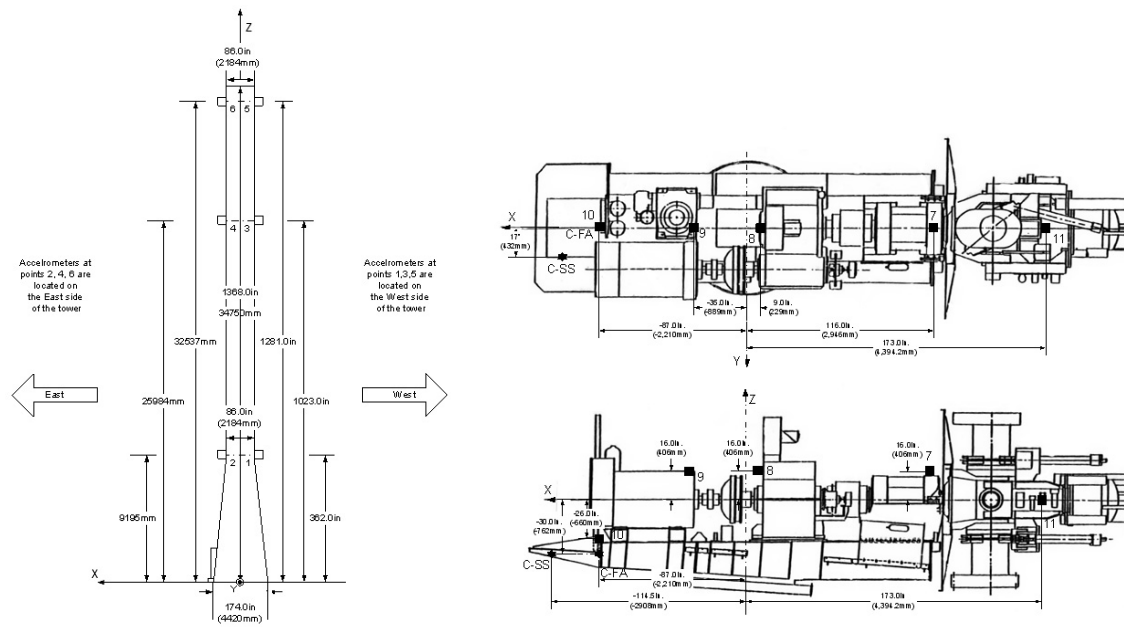


Figure 4: Blade one, two, and three node map. Blade one was instrumented with ten biaxial accelerometers, while two and three were instrumented with seven. Locations were approximated using the station information from a modified Zond 23 m blade design.



**Figure 5: (Left) Tower node map. Six biaxial accelerometers were located along the height of the turbine. (Right) Nacelle node map, top view and side views. Five triaxial accelerometers were located along the central axis.**

In total, seventy-five channels of acceleration were measured to map the modal response of the wind turbine system. In addition, two driving points were collected using four channels; two channels of force and two channels of acceleration. A driving point is the collocated measurement of the input excitation force and response acceleration in the direction of the applied force. Therefore, the modal test system collected 79 channels of data for the modal test. The LMS SCADAS Mobile (Figure 6) data acquisition system (DAS) was used to collect, reduce, and save the processed modal test data on a laptop PC running LMS Test.Lab software [14][15].

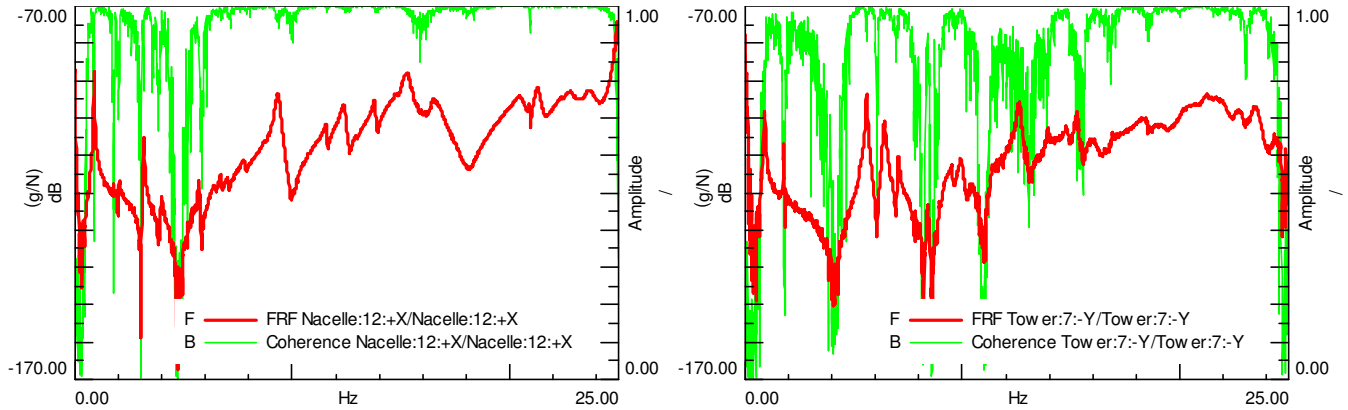


**Figure 6: (Left) LMS Scadas Mobile data acquisition system. (Right) blade accelerometers.**

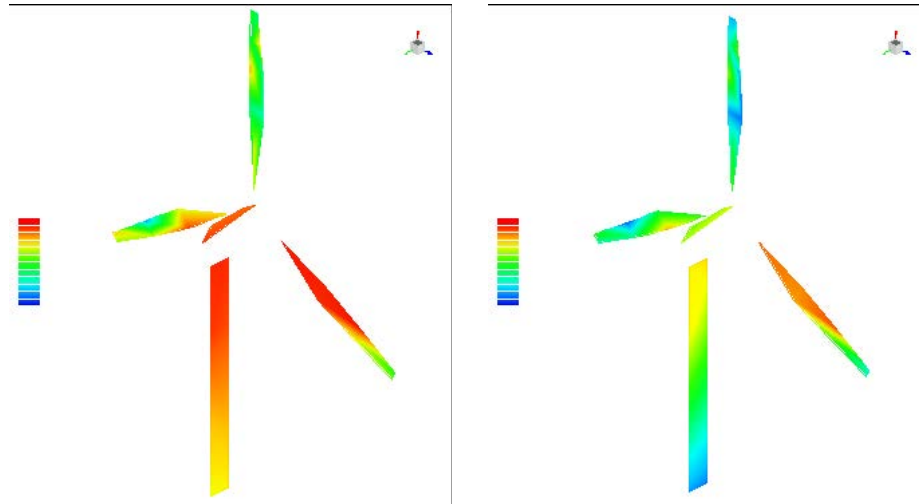
### 3 EXPERIMENTAL MODAL ANALYSIS – SHAKER EXCITATION

A random signal was sent to the shaker. A Hanning window was applied to the excitation and response signals. The sampling frequency was 50 Hz and the block size 4096, yielding a frequency resolution of the spectra, FRFs and coherences of 0.0122 Hz. Ten averages were used in the computation of the measurement functions. Therefore the measurement time was 819.2 s. The driving point FRFs and corresponding coherences are shown

in Figure 7. A quality indicator for the measurements is the coherence function that combines the effects of leakage, noise and non-linearity on the FRFs. When performing high channel count measurements, browsing through all coherence functions is rather cumbersome. A synthetic view is provided by averaging the coherences over certain frequency ranges and plotting these at each sensor location in a geometry display. A relative quality assessment can then be made: for instance, the coherences of close and structurally related locations should not differ too much. In Figure 8 the averaged coherences in the full frequency band (0.5 – 25 Hz) and a low-frequency band (0.5 – 5 Hz) are plotted. In general the sensors close to the force inputs have a better signal-to-noise ratio and hence a higher coherence. Apparently, the blade 2 coherences are higher than the other blades.



**Figure 7: Driving point FRFs and coherences. (Left) nacelle fore-aft excitation; (Right) tower side-to-side excitation.**



**Figure 8: Tower excitation averaged coherences in a geometry display. (Left) full frequency band 0.5 – 25 Hz; (Right) low-frequency band (0.5 – 5 Hz). The color scale varies between 0 (blue) and 1 (red).**

After the data validation, the PolyMAX modal parameter estimation algorithm [15][16] was applied to the measured FRFs. Both shaking tests were carried out sequentially, but processing both datasets simultaneously yielded the most complete list of modes. Despite the relatively noisy FRFs, the structural modes are clearly identified in the PolyMAX stabilization diagram (Figure 10 – Left). A stabilization diagram shows the modes that are independently calculated at different model orders in the system identification process. Experience on a very large range of problems shows that in such analysis, the pole values of the “physical” eigenmodes always appear at a nearly identical frequency, while mathematical poles tend to scatter around the frequency range. Fifteen modes were found below 10 Hz. Their frequencies and damping ratios are listed in Table 1. The first nine mode shapes are represented in Figure 9. The FRFs can be synthesized from the identified modes and compared with the measured FRFs. Figure 10 (Right) shows the comparison between the sum of the measured FRFs and the sum of the synthesized FRFs. The good agreement indicates that most important dynamics have been identified from the data.

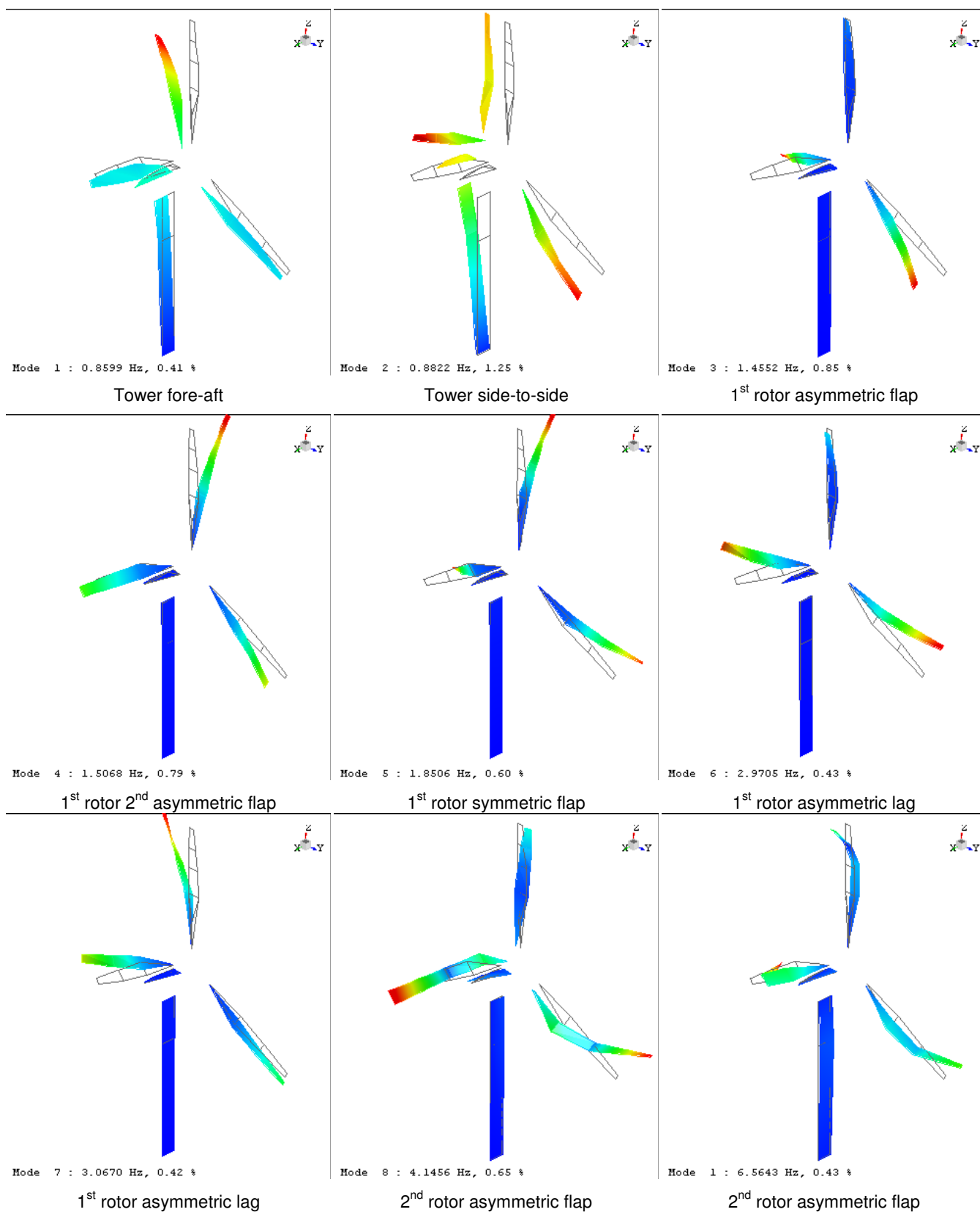


Figure 9: EMA-PolyMAX mode shapes. The color scale indicates the relative amplitudes.

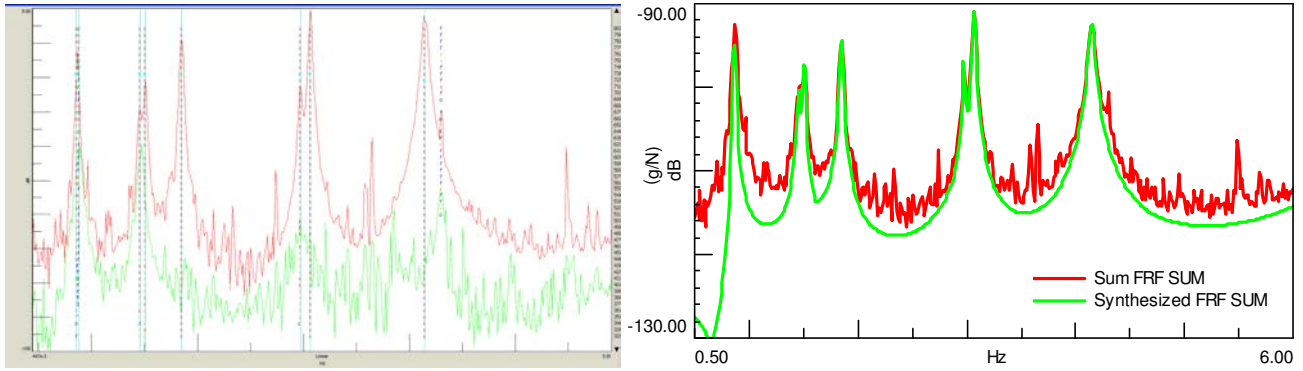


Figure 10: (Left) EMA-PolyMAX stabilization diagram; (Right) sum of measured and synthesized FRFs.

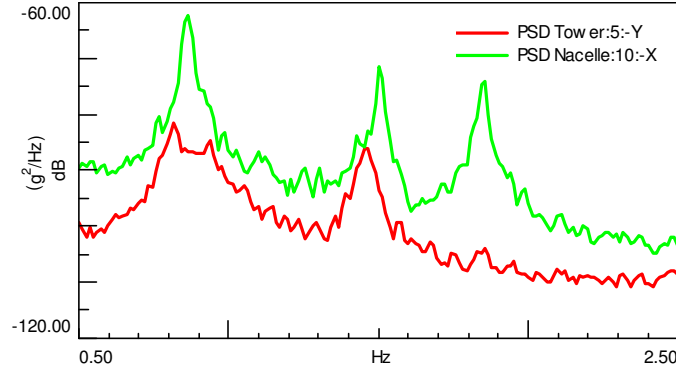
Table 1: Modes from hydraulic tests. 15 modes were found between 0.8-10 Hz. Combining both the fore-aft and side-to-side data sets for processing revealed the 15 modes given.

| Mode | Description                                           | Frequency | Damping  | Processing                                        |
|------|-------------------------------------------------------|-----------|----------|---------------------------------------------------|
| #    |                                                       | Hz        | %        | F-A: fore-aft<br>S-S: side-to-side<br>C: Combined |
| 1    | Tower fore-aft                                        | 0.86      | .24-.38  | F-A, S-S, C                                       |
| 2    | Tower side-to-side                                    | 0.88      | 1.12     | C                                                 |
| 3    | 1 <sup>st</sup> rotor asymmetric flap                 | 1.45      | 0.88     | C                                                 |
| 4    | 1 <sup>st</sup> rotor 2 <sup>nd</sup> asymmetric flap | 1.51      | .35-.61  | F-A, S-S, C                                       |
| 5    | 1 <sup>st</sup> rotor symmetric flap                  | 1.85      | .55-.57  | F-A, S-S, C                                       |
| 6    | 1 <sup>st</sup> rotor asymmetric lag                  | 2.97      | 0.33     | F-A, C                                            |
| 7    | 1 <sup>st</sup> rotor asymmetric lag                  | 3.06      | 0.38     | C                                                 |
| 8    | 2 <sup>nd</sup> rotor asymmetric flap                 | 4.14      | 0.65     | C                                                 |
| 9    | 2 <sup>nd</sup> rotor asymmetric flap                 | 6.44      | 0.25     | C                                                 |
| 10   | 2 <sup>nd</sup> rotor asymmetric flap                 | 6.56      | 0.53     | C                                                 |
| 11   | 2 <sup>nd</sup> rotor symmetric flap                  | 6.89      | .31-.46  | S-S, C                                            |
| 12   | 2 <sup>nd</sup> rotor asymmetric flap                 | 6.96      | 0.65     | C                                                 |
| 13   | 2 <sup>nd</sup> tower fore-aft                        | 7.16      | .51-.77  | F-A, S-S, C                                       |
| 14   | 3 <sup>rd</sup> asymmetric flap                       | 9.44      | .45-.87  | F-A, C                                            |
| 15   | 3 <sup>rd</sup> asymmetric flap                       | 9.89      | 1.1-1.21 | F-A, C                                            |

#### 4 OPERATIONAL MODAL ANALYSIS – WIND EXCITATION

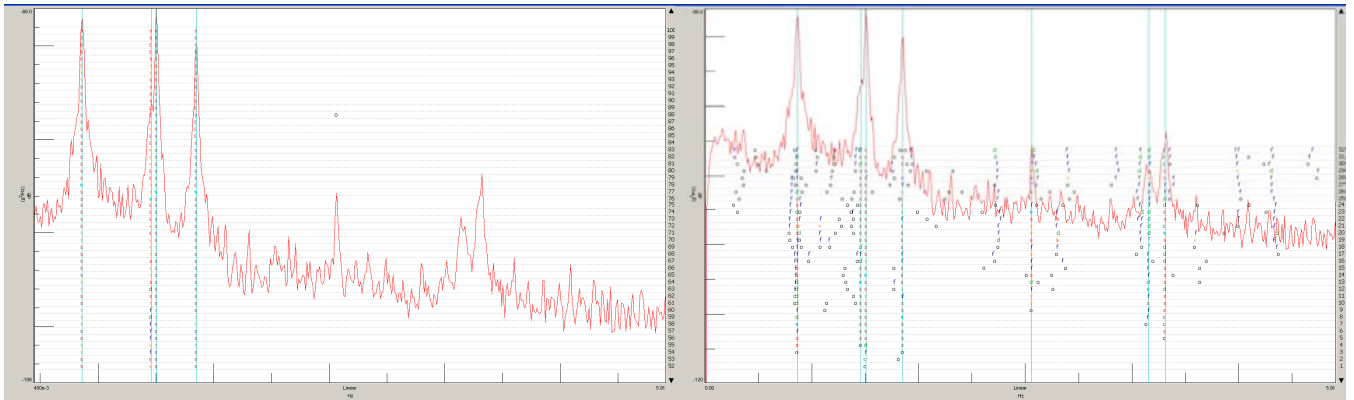
Wind turbines are ideal structures for the application of Operational Modal Analysis (OMA). They are large structures which makes it challenging to introduce sufficient artificial excitation in an EMA approach. To obtain high-quality FRFs, it is important that the applied and measured excitation generates response levels which are well above the ambient response and/or noise floor. As shown in this paper, large hydraulic shakers can be used in combination with cables so that the forces are introduced well above ground level. It is however clear that it is cumbersome to use shaker excitation in this case. Fortunately, wind excitation is typically often present at wind turbine locations and constitutes an excellent ambient excitation for the application of OMA. An additional reason for performing in-operation test is that the eigenfrequencies of wind turbines depend on the rotation speed of the blades due to centrifugal and Coriolis forces and the effect of tension stiffening. In this paper, OMA will be applied with the rotor in parked conditions, but at a later stage also rotating conditions will be experimentally investigated.

The wind excitation data used in this paper originates from a preliminary limited test in which power and cross spectra were acquired under wind excitation (Figure 11), but without storing the time histories. Therefore, the ideal leakage-free pre-processing of the time data into so-called half spectra could not be performed. Moreover, the use of an exponential window in the pre-processing stage is an excellent “de-noising” tool that could not be applied here. More details about the advantages of the use of half spectra (or correlograms) in comparison with the more classical spectra (or periodograms) can be found in [11]. In the near future, more extensive operational time-domain data will be acquired. Nevertheless, the present data already yielded very interesting results.



**Figure 11: Lower-frequency part of wind turbine response spectra due to wind loading.**

Figure 12 compares the stabilization diagrams of two modal parameter estimation methods available in Test.Lab [15]: Operational PolyMAX and Stochastic Subspace Identification (SSI). PolyMAX is known to yield very clear stabilization diagrams, but identifies modes which are heavily affected by noise with negative damping. These modes are not shown in the stabilization diagrams. Reason for the noise influence on the modes is the non-ideal pre-processing of the data. The SSI method does not suffer from this. From Figure 12 (Right), it is clear that more modes are identified though the diagram is less clear. Note that the SSI method is a time-domain method, but it has been implemented in such a way that also data reduced to power and cross spectra can be used. The MAC values in Figure 13 (Left) indicate that both Operational PolyMAX and SSI mode shapes are in excellent agreement. Also the correspondence between mode shapes identified using shaker excitation (EMA) and wind excitation (OMA) is excellent (Figure 13 – Right). The modes that were not identified using OMA (modes 2 and 6, see Figure 9 for the shapes) are indeed modes that are not well excited by the wind.



**Figure 12: Operational PolyMAX (Left) versus Stochastic Subspace Identification (Right) stabilization diagram.**

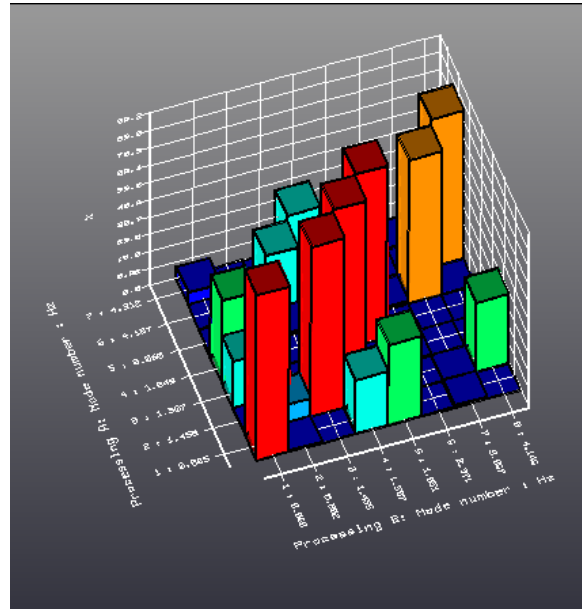
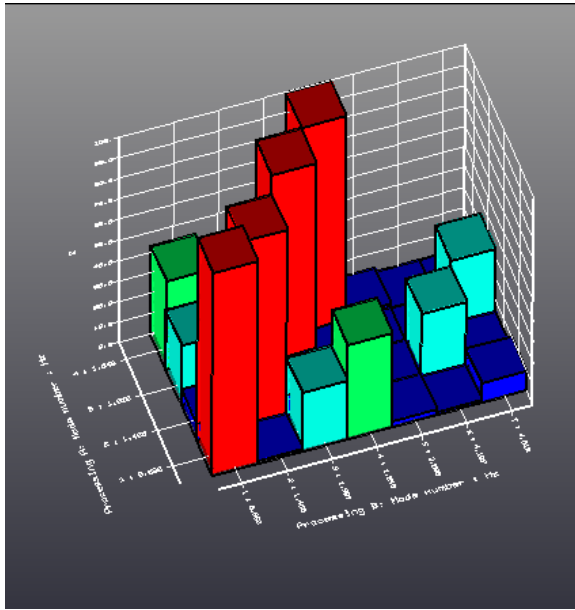


Figure 13: (Left) MAC values between Operational PolyMAX and SSI mode shapes. (Right) MAC values between shaker excitation (EMA / PolyMAX) and wind excitation (OMA / SSI) mode shapes.

## 5 FURTHER USE OF EXPERIMENTAL MODAL PARAMETERS

The Campbell diagram given in Figure 14 (Left) compares the experimentally measured modes (See Figure 9 and Table 1) to the CART3 operating frequencies. The harmonics of the rotor frequency for different rotor speeds are given by the black lines, 1P, 2P, 3P, etc. where P is the given rotor frequency. At the rated operation, the rotor speed is 41.7 rpm. From this figure, it can be observed that the flap modes and second tower fore-aft modes coincide with the rotor harmonics of 2P, 6P and 10P at this rated speed. However, since the modal test does not include aerodynamic damping, operating data is needed to determine if the indicated natural frequencies pose a resonance hazard to the operation of the turbine. Note that in reality, all rotor frequencies should show a slight increase as the rotor speed is increased. This is not shown in Figure 14 because here the idea is to get a preliminary estimate of the cross-over frequencies and hence possible resonance points just based on measured frequencies at zero rotor speed conditions.

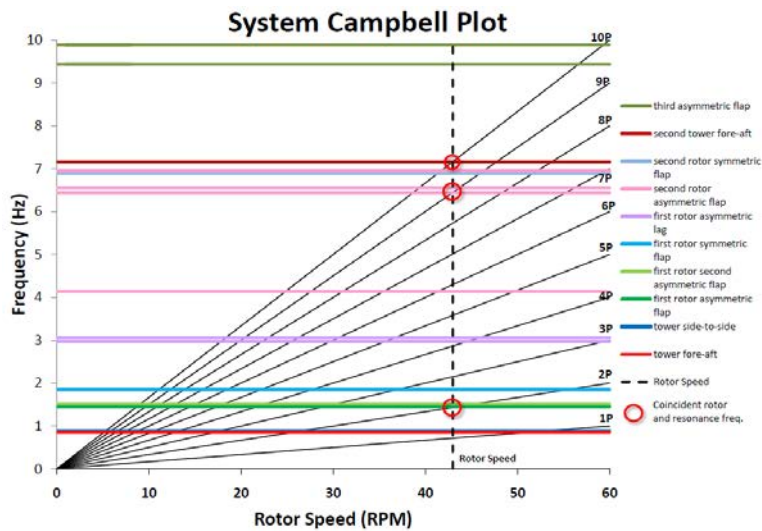


Figure 14: (Left) Campbell plot for comparing rotor harmonics (black lines) with structural resonances (colored lines). (Right) Examples of simulated full turbine system modes for a 3-bladed parked rotor wind turbine.

The modes determined from the hydraulic test differed significantly from the modes computed from the initial multi-body dynamic simulation (MBS) model of CART3 (Figure 14 – Right). Therefore, the experimental results proved useful in tuning the numerical model. The results of the comparison between test and tuned model can be found in Table 2. The first and second fundamental bending modes and the first rotor flap mode extracted from the modal data were determined within 1% of the model's prediction. Modes 4 and 5 representing the other first rotor flap modes were predicted within 3-4% of the experimentally found modes. The MBS model, mode 6, with drive-train twist and first symmetric lag, was not found experimentally. This is likely due to the location of tower side-to-side excitation not being sufficiently offset from the tower and nacelle yaw axis. Model modes 7 and 8, the first rotor asymmetric lag responses, were predicted within 3% of the experimental results. Model mode 9, the first emergent second rotor asymmetric flap mode, is predicted within a 4% difference with the experiment. Model mode 10 however was predicted with a 30.8% difference. This is likely due to a wrong pairing of experimental and model modes. Experimental modes 9-11, second rotor asymmetric flap shapes, were not found by the model. Likewise, model mode 11, a first tower torsion mode at 5.867 Hz was not found from the modal data extraction.

**Table 2: Correlation of experimental modes with model predicted modes. The low fundamental modes yield a percent difference of < 5%, indicating high correlation. The experiment does not determine torsion or twist modes. The last mode predicted by the model shows a 30% difference from the experiment.**

| Exper. Mode | Exper. Nat Freq [Hz] | Model Mode | Model Nat Freq [Hz] | % Diff | Shape                                                 |
|-------------|----------------------|------------|---------------------|--------|-------------------------------------------------------|
| 1           | 0.86                 | 1          | 0.868               | 0.930  | Tower fore-aft                                        |
| 2           | 0.88                 | 2          | 0.882               | 0.227  | Tower side-to-side                                    |
| 3           | 1.45                 | 3          | 1.45                | 0.000  | 1 <sup>st</sup> rotor asymmetric flap                 |
| 4           | 1.51                 | 4          | 1.556               | 3.046  | 1 <sup>st</sup> rotor 2 <sup>nd</sup> asymmetric flap |
| 5           | 1.85                 | 5          | 1.931               | 4.378  | 1 <sup>st</sup> rotor symmetric flap                  |
| 6           | 2.97                 | 7          | 3.047               | 2.593  | 1 <sup>st</sup> rotor asymmetric lag                  |
| 7           | 3.06                 | 8          | 3.113               | 1.732  | 1 <sup>st</sup> rotor asymmetric lag                  |
| 8           | 4.14                 | 9          | 3.98                | 3.865  | 2 <sup>nd</sup> rotor asymmetric flap                 |
| 12          | 6.96                 | 10         | 4.813               | 30.848 | 2 <sup>nd</sup> rotor asymmetric flap                 |

## 6 DISCUSSION AND CONCLUSION

This paper reported on the initial results of a major full-scale modal testing campaign on the CART3 wind turbine to validate and refine simulation models. The paper compared an Experimental Modal Analysis approach, where the wind turbine in parked condition is excited by shakers connected with cables, with an Operational Modal Analysis approach, where the same structure is subjected to wind excitation. Hydraulic testing with fore-aft and side-to-side shaking revealed 15 modes below 10 Hz. Both shaking tests were carried out sequentially, but processing both datasets simultaneously yielded the most complete list of modes. Simultaneously exciting side-to-side and fore-aft in hydraulic testing seems more ideal. This is considered as an option in future.

Though, the use of wind excitation data in an OMA approach was only preliminary investigated, already very promising results were obtained: the identified OMA modes agreed very well with their EMA counterparts. In near future, more extensive wind excitation time-domain data will be acquired, allowing a more detailed OMA. Also, higher levels of wind excitation data are desired. The observed low amplitude of mean wind speed recorded indicated that there was insufficient low frequency excitation of the wind turbine structure. Usually wind excitation provides greater amounts of low frequency excitation when compared to the available low frequency excitation energy supplied by the hydraulic test system.

Comparing the multi-body simulation model with the shaker experiment modes revealed close agreement for many modes. The lower modes were predicted very well by the model. Due to limitations in resolution afforded by the modal map, the test was unable to identify the rotor twist and torsion modes of the blade, and there was poor

correlation for higher second rotor flap modes. In order to interpret higher order system modes, more measurement points will be required.

The next step for this test is to conduct the experiment with the turbine operating. Rotation of the rotor changes the loads and damping forces acting on the turbine, thus changing the boundary conditions on the turbine. Identifying modes from operating conditions allows a further refinement of the structural and control models of the turbine. This test however, has provided the opportunity to develop test procedures and data analysis methods which will be used to plan modal surveys for multi-megawatt large-scale wind turbines. Two of which (a 1.5 and 2.3 MW turbine) are currently being installed at the National Wind Technology Center.

## ACKNOWLEDGEMENTS

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## REFERENCES

- [1] D.T. Griffith, T.G. Carne, and J.A. Paquette. Modal testing for validation of blade models, *Wind Engineering*, **32**(2):91-102, 2008.
- [2] G. Bir and J. Jonkman. Modal dynamics of large wind turbines with different support structures, *Proc. of the ASME 27<sup>th</sup> International Conference on Offshore Mechanics and Arctic Engineering*, Estoril, Portugal, Jun. 15-20, 2008.
- [3] T.G. Carne, D.W. Lobitz, A.R. Nord, and R.A. Watson. *Finite Element Analysis and Modal Testing of a Rotating Wind Turbine*, SAND82-0345, Sandia National Laboratories, Albuquerque, NM, Oct. 1982.
- [4] A.D. Wright, L.J. Fingersh, and K.A. Stol. Designing and Testing Controls to Mitigate Tower Dynamic Loads in the Controls Advanced Research Turbine, *Proc. of the 45<sup>th</sup> AIAA Aerospace Sciences Meeting and Exhibit, Wind Energy Symposium*, Reno, Nevada, Jan 8-11, 2007.
- [5] A. Wright and M. Balas. Design of state-space-based control algorithms for wind turbine speed regulation, *Proc. of the 21<sup>st</sup> American Society of Mechanical Engineers (ASME) Wind Energy Symposium*, Reno, Nevada, Jan. 14-17, 2002.
- [6] J.P. Lauffer, T.G. Carne, and T.D. Ashwill. *Modal Testing in the Design Evaluation of Wind Turbines*, SAND87-2461, Sandia National Laboratories, Albuquerque, NM, Apr. 1988.
- [7] R.M. Osgood. *Dynamic Characterization Testing of Wind Turbines*, NREL Technical report NREL/TP-500-30070, May 2001.
- [8] R.M. Osgood, H. G. McFarland, and G.L. Johnson. *Full System Modal Survey Test Results of the Controls Advanced Research Turbine (CART)*, NREL Internal Letter Report, National Renewable Energy Laboratory, Golden, CO, Feb. 2002.
- [9] M.H. Hansen, K. Thomsen, and P. Fuglsang. Two methods for estimating aeroelastic damping of operational wind turbine modes from experiments, *Wind Energy*, **9**:179-191, Jan. 2006.
- [10] B. Peeters and G. De Roeck. Stochastic system identification for operational modal analysis: a review, *ASME Journal of Dynamic Systems, Measurement, and Control*, **123**(4):659-667, 2001.
- [11] B. Peeters, H. Van der Auweraer, F. Vanhollebeke, and P. Guillaume. Operational modal analysis for estimating the dynamic properties of a stadium structure during a football game, *Shock and Vibration*, **14**(4):283-303, 2007.
- [12] G.H. James, T.G. Carne, and P.S. Veers. Damping measurements using operational data, *Transactions of the ASME*, **118**:190-193, Aug. 1996.
- [13] G.H. James, T.G. Carne, J.P. Lauffer, and A.R. Nord. Modal testing using natural excitation, *Proc. of the 10<sup>th</sup> International Modal Analysis Conference*, San Diego, CA, 1992.
- [14] LMS International. *LMS SCADAS Mobile Data Acquisition Front-end*, Breda, The Netherlands, [www.lmsintl.com](http://www.lmsintl.com), 2009.
- [15] LMS International. *LMS Test.Lab Structures*, Leuven, Belgium, [www.lmsintl.com](http://www.lmsintl.com), 2009.
- [16] B. Peeters, H. Van der Auweraer, P. Guillaume, and J. Leuridan. The PolyMAX frequency-domain method: a new standard for modal parameter estimation? *Shock and Vibration*, **11**:395-409, 2004.