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Analysis of a Two-Bladed, Teetering-Hub Turbine Using the ADAMS®[1] Software

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Abstract

A major goal of the federal wind energy program is the rapid development and validation of structural models to determine loads and reponse for a wide variety of different wind turbine configurations operating under extreme conditions. Such codes are crucial to the successful design of future advanced wind turbines. In cooperation with R. Lynette & Associates the Wind Technology Division at NREL has developed a full system dynamics model of the AWT-26 P1 machine, using the Automatic Dynamic Analysis of Mechanical Systems (ADAMS) software from Mechanical Dynamics, Inc. In this paper, we show validation of sub-models by comparisons with modal test data. We describe the most important system modes involved in this structure and demonstrate how the ADAMS model can be used to tune the structure to avoid interactions. We also identify questions that remain unanswered by ADAMS in modeling this turbine and recommend future directions that DOE code development activities should take.

Introduction

Previous codes developed under the DOE Wind Program and in Europe have had restrictive assumptions. Reference 1 presented a list of codes that have been developed both in the United States and Europe as well as their limitations. Because of these limitations, NREL decided in 1991 to use an existing multibody dynamics code (ADAMS) to develop system dynamics models for wind turbines. The code develops numerical equations of motion automatically at run time, removing the major modeling task of developing and validating complicated sets of equations of motion. Other advantages of using this approach are the lack of limitations on the type or magnitude of displacements and rotations, and the ability of ADAMS to be linked with user-written subroutines that describe the aerodynamic loads. Through developing and validating models using this method, the U.S. wind industry now has a tool to assist in the understanding of complex interactions. This tool can also be used to determine turbine design changes needed to minimize such interactions once they have been identified.

The P1 (pre-prototype) version of the AWT-26, a two-bladed, teetering-hub, horizontal-axis wind turbine, developed by R. Lynette and Associates under NREL's Advanced Wind Turbine Program, has shown response at normal operating speed, involving blade symmetric edge-bending and nacelle and tower top tilt motion. This interaction is seen as a 7-per-revolution response in the blade root-edgewise bending moments, and as a 6- and 8-per-revolution response in the nacelle and tower. In cooperation with R. Lynette & Associates the Wind Technology Division at NREL has developed a full-system dynamics model of the AWT-26 pre-prototype machine (P1), using ADAMS.

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[1] ADAMS is a registered trademark of Mechanical Dynamics, Inc.

First, we developed an ADAMS blade submodel and compared predicted blade modes and frequencies to isolated blade modal test data. We used a submodule of ADAMS named ADAMS/Linear to give us predicted blade modal information. Of great importance in this blade modeling was the inclusion of significant coupling between blade flap and edge degrees of freedom caused by the blade's pretwist and principal axis orientation.

We then expanded the blade model to include the rotor hub, drive-train shaft bending, nacelle yaw, and tower subassemblies. Validation of this model was done by comparing ADAMS/Linear results for the static machine to modal test results. In addition, various operating machine loads measurements were compared to model predictions, such as blade root flap-, edge-bending moments, and nacelle pitching accelerations.

We found the comparison of ADAMS/Linear predictions to modal test data crucial in helping to identify the cause of this machine interaction. Early in the modeling process we did not have accurate blade or system modal information from which to check the accuracy of our model. We made numerous ADAMS runs, only to find that the amount of 7-per-revolution response in the blade root-edgewise bending moment was greatly underpredicted. Identifying the important modes involved in this interaction and validating them with modal test data greatly accelerated our progress, although some uncertainty still exists regarding correct tower-top and machine bedplate properties, as will be described later in this paper.

Machine Interaction Description

Figure 1 shows an ADAMS depiction of the P1 version of the AWT-26. It is a free-yaw, downwind machine. The 12.1 m (39.7 ft) fixed-pitch blades have a 5.5-degree pretwist with a maximum chord of 1.2 m (3.8 ft). They use the NREL thick airfoil family (S809, S810, and S815[2]) designed for 12-m (40-ft) blades. The rotor diameter is 26.2 m (86 ft) with a 7-degree precone. It sits on top of a free-standing truss tower and the hub height is 24.4 m (80 ft). The turbine rotates at 57 revolutions per minute (rpm) and generates 275 kW of power at rated wind speed (18 m/s, 40 mph). Each blade has a tip brake, which weighs approximately 11.3 kg (25 lbm).

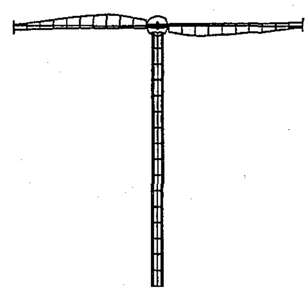

FIGURE 1. AWT-26

Figure 2 shows a plot of blade root edgewise bending moment and nacelle vertical (or pitch) acceleration taken from the rotor enhancement project (REP) machine. The P1 machine exhibits very similiar behavior.

Positive root edgewise bending occurs when the trailing edge of the blade is in tension (leading edge in compression). One can gain an understanding of this convention by referring to Figure 1. If in that figure the wind is coming out of the page, an observer standing downwind looking upwind toward the turbine sees the rotor rotate counterclockwise, with zero azimuth position being blade #1 straight up. Blade #1 (at the left) is now at the 90 degree azimuth position and its trailing edge is in tension due to gravity. The root edgewise-bending moment, shown in Figure 2 for blade #1, can be seen to increase due to gravity as the blade moves from 0 to 90 degrees azimuth and then to decrease as the blade goes to 180 degrees azimuth.

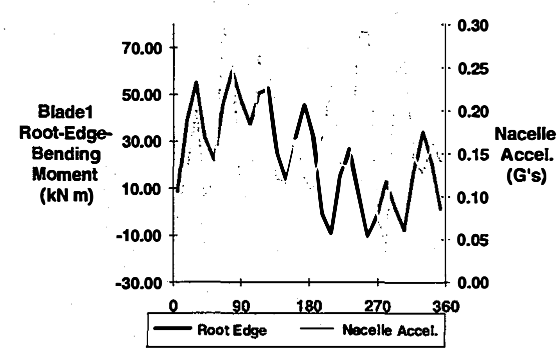

FIGURE 2. BLADE ROOT-EDGE-BENDING MOMENT AND NACELLE PITCH ACCELERATION

[2] S809, S810, and S815 are trademarks of the National Renewable Energy Laboratory.

The nacelle vertical acceleration (also called pitch acceleration) was measured by an accelerometer located on the nacelle bedplate upwind of the yaw-axis. Since displacement and acceleration are 180 degrees out of phase, the maximum positive acceleration occurs when the upwind end of the nacelle has reached its lowest point of displacement. One can see from Figure 2 the high degree of coupling between blade edge-bending moment and nacelle vertical acceleration. This motion corresponds to a mode in which, when the blades are horizontal, the blade tips move in phase with the upwind end of the nacelle, or 180 degrees out of phase with the hub end of the nacelle. We tried to identify static machine modes involving symmetric edgewise bending motion and nacelle vertical acceleration. We now show results of this investigation and modeling work.

MODEL DEVELOPMENT AND VALIDATION

Our main method of developing an ADAMS turbine model was to first develop an accurate submodel of the blade, because this is the most important component that generates the system loads. We performed a blade modal test at the National Wind Technology Center (NWTC) in May 1993, which proved crucial for verifying the blade's mass and stiffness characteristics. We removed the blade from the machine at the point it was attached to the root adapters and attached it to a rigid test fixture. We obtained test data for three different tip mass values: 0 kg, 11.3 kg and 15.4 kg.

We modeled the blade with several PARTS [4] and connected them with uniform ADAMS BEAMS [4]. We found it necessary to use about 8 to 11 BEAMS, to capture the characteristics of the modeshapes, including node point location and important coupling effects.

From the modal test results, we found that there was significant coupling between the blade's first edgewise bending shape and its second flapwise bending shape. The only way to distinguish between these two natural modes was to analyze the phase of the edge and flap components, the amount of participation between flap and edge being about equal for both modes. These couplings were due to blade geometric pretwist. For most sections, especially from the blade root to about 40% span, the principal axes were not aligned with the section chordline. To account for these effects, each blade BEAM was oriented separately by rotating its MARKERS by an angle corresponding to the orientation of the principal axes.

We also added to our ADAMS model the effects of blade section mass and elastic axis offsets (from the beam's neutral axes). The effects of adding these offsets into the ADAMS model were to add a small amount of flap/pitch and edge/pitch coupling. Compared to the large amount of coupling between flap and edge due to pretwist and principal axis orientation, these effects were negligible. We also investigated the effects of moving the blade's tip vane center of gravity location with respect to the blade tip elastic axis location. These effects were not important to predicting the system interaction observed in this machine.

Another parameter of importance was blade modal damping. ADAMS accepts a damping input for each BEAM in the form of a BEAM structural damping ratio (CRATIO, see [4]). We obtained corresponding modal damping values from ADAMS/Linear. In general, we kept the modal damping very low for the higher modes so that machine cyclic response at these higher frequencies (7-10 Hz) would not be overdamped. This conclusion is probably subjective, we feel that aerodynamic damping is probably more pronounced for the first flap-bending modes and small for the higher modes.

Table 1 shows comparisons of ADAMS/Linear predictions to test results for a tip mass of 15.4 kg. The model accurately predicts the frequencies for each of these modes within 4% accuracy. Results of comparisons for other tip masses show similiar error bounds and are not shown here. We did not compare ADAMS/Linear predicted modal damping values to test data. Such a comparison is the topic for future work.

Figures 3-4 show the blade's first edge and second flap modes (the first flap mode is not shown). Each figure shows both the flap and edge contributions to that mode. One can see from Figures 3 and 4 that there is much coupling between flap and edge motion, with almost as much flap and edge participation in each mode. The large amount of flap/edge coupling is due to the effects of structural pretwist and the principal axis orientation. The blade tip mass contributes to some of this coupling, by acting to lower the second flapwise frequency and moving it close to the blade's first edgewise mode (within 1 Hz). From the close agreement between ADAMS blade predictions and test data, we felt confident in proceeding with the full systems model.

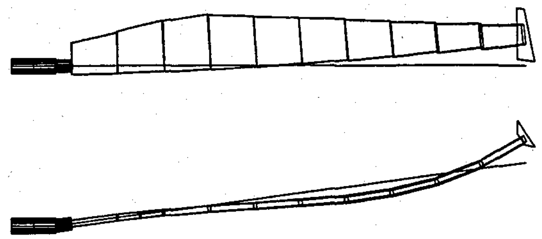

FIGURE 3. FIRST EDGE BENDING MODE

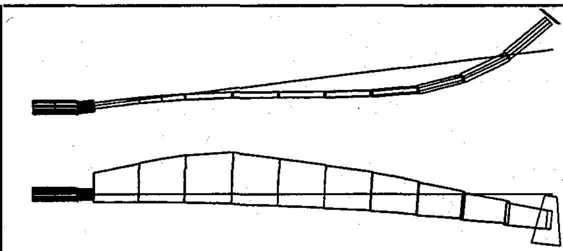

FIGURE 4. SECOND FLAP BENDING MODE

TABLE 1. COMPARISON OF PREDICTED BLADE FREQUENCIES TO TEST DATA FOR A 15.4 kg TIP MASS.

Mode	ADAMS/Linear	Test Data	Error
First Flap	2.33 Hz.	2.29 Hz.	1.8%
First Edge	6.59 Hz.	6.38 Hz.	3.3%
Second Flap	7.74 Hz.	7.47Hz.	3.6%

To model the complete rotor, we added a hub, modeled with three ADAMS PARTS: one located at the hub center and the other two at the ends of each hub arm. BEAMS then connected the hub arms to the center. We then attached two blades with identical properties to the end of each hub arm. The hub center PART was then constrained to the downwind end of the low-speed shaft via an ADAMS revolute JOINT [4] to simulate the teeter hinge. This hinge was placed an extra distance downwind of the hub center to model the effects of rotor undersling.

We modeled the low-speed shaft with PARTS and BEAMS and determined their properties from the shaft weight and dimensions. Including the shaft flexibility was very important in predicting the machine's symmetric edge modes. We did not model the high-speed shaft flexibility, even though adding this feature with ADAMS BEAMS might improve the results.

We modeled the nacelle simply as one PART which included the mass and inertia of the generator, gearbox, and high-speed shaft. We attached it to the tower top via another revolute joint for the yaw bearing. We neglected yaw bearing friction until better estimates of this parameter are obtained. We applied a torque to the generator end of the low-speed shaft to simulate the steady-state behavior of the generator as based on Thevinin's Theorem [3]. In addition, we included some of the effects of nacelle bedplate flexing by adding a couple of BEAMS into the nacelle. It appeared that including these effects was also important in predicting the machine's rotor edgewise responses.

We modeled the three-legged truss tower with the same lumped-mass/flexible-beam approach just mentioned. By observing system modal test results, we could see that some tower modes contained coupling between tower bending and torsion. It was almost impossible to include the effects of this bending/torsion coupling with the simple uniform beam elements in ADAMS; however, including this coupling for predicting the machine's interaction was not deemed critical. We obtained good machine response predictions without including this effect.

We performed full system modal tests of the machine in November, 1993, in Tehachapi, California. The nonrotating machine was excited by a hydraulic shaker. The frequency response measurements (FRFs) included 18 for the tower, 15 for the nacelle and 15 for each of the two blades. Each FRF was computed from a transfer function measured between the input degree of freedom (DOF) at the shaker and an output response DOF measured by a triaxial accelerometer.

The nonrotating system modal information proved crucial to the modeling of the complete turbine using ADAMS. Table 2 shows comparisons between ADAMS/Linear predictions and test data for modes involving rotor symmetric edgewise motion and symmetric flapwise response for the blades both vertical and horizontal. Except for the first one, these modes were all in the 6-8 Hz category. In this table, the plus sign indicates that the two components of this mode are in phase, while the minus sign indicates that they are 180 degrees out of phase.

TABLE 2. COMPARISON OF INITIAL PREDICTED MACHINE FREQUENCIES TO MODAL TEST DATA

	Blades Horizontal		Blades Vertical	
Mode Number and Shape	Measured (Hz)	Predicted (Hz)	Measured (Hz)	Predicted (Hz)
1. edge symm. + nacelle pitch	4.3	4.8	n/a	n/a
2. edge symm. + yaw	n/a	n/a	6.6	6.6
3. flap symm. + (tower long.)	7.2	7.2	7.2	7.3
4. edge symm. - nacelle pitch	7.4	8.1	n/a	n/a

Mode 1, shown in Figure 6, is dominated by rotor symmetric edgewise bending and low-speed shaftbending. In this mode, the hub end of the nacelle moves in phase with the blade tips. Figure 7 shows mode 2. The rotor's symmetric edge mode is now reacting against the machine's yaw inertia instead of the shaft and tower top, showing the difference in frequency of this mode compared to the previous one. Mode 3 (not shown) is the same for the blades vertical and horizontal. We found very little tower or nacelle participation in this mode; however, it's proximity to mode 4 is probably important. Mode 4, shown in Figure 8, involves nacelle participation in the opposite sense to mode 1. When the blade tips move up, the hub end of the nacelle moves down, 180 degrees out of phase with the blade tips. This mode seems to be the key for tuning our ADAMS model to get good agreement with test data.

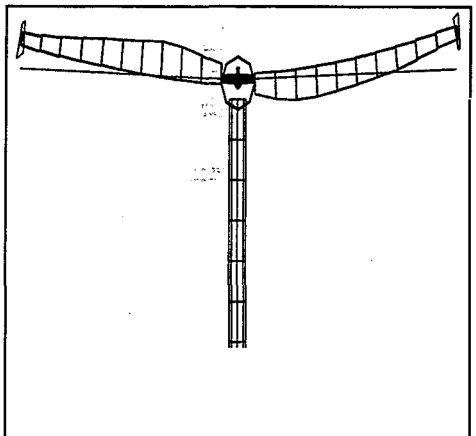

FIGURE 6. MODE #1

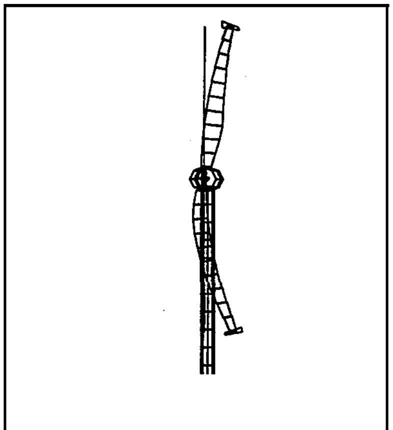

FIGURE 7. MODE #2

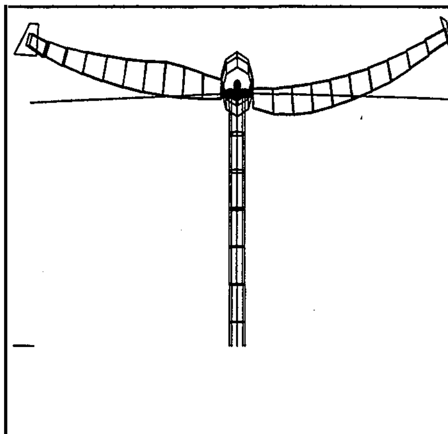

FIGURE 8. MODE #4

Input of our initial tower properties resulted in a prediction of 8.1 Hz for this mode, as seen in Table 2. The cause of this overprediction is still unknown, but we had to arbitrarily soften the upper tower elements by at least 50% to lower the frequency of this mode to it's measured value of 7.4 Hz, which is critical to predicting the response of this system. This "tuning" of the inputs was necessary to obtain predictions which were realistic as we now show.

Operational Loads Comparisons

We processed a 10-minute set of field data consisting of blade root flap and edge-bending moments, as well as nacelle vertical acceleration time series. We determined the mean wind speed and wind shear distribution from anemometer data at three levels on a tower. We then performed azimuth averaging of the loads data to remove the effects of stochastic wind fluctuations and find the machine response due to just the deterministic inputs of wind shear, tower shadow, and gravity. We ran the ADAMS code with aerodynamic subroutines supplied by the University of Utah [2]. We input tower shadows with a 50% deficit spread over a rectangular cross-section 3.05 m (10 ft) in width. These values were subjective but were thought to be about right for the type of truss tower we were analyzing.

In Figure 9 we compare ADAMS predicted loads versus measured results for the blade root edgewise bending moments. We show results for both our original tower top properties ("stiff tower top") in which the frequency of mode 4 was 8.1 Hz as well as the modified one ("soft tower top"), in which that frequency was tuned to 7.4 Hz. The difference in predicted response is readily seen. Tuning mode 4 to its measured value was very important to obtaining realistic predictions.

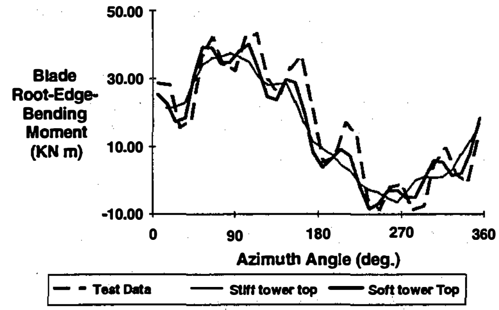

FIGURE 9. BLADE ROOT-EDGEWISE-BENDING MOMENT

We show similiar results for the nacelle pitch acceleration in Figure 10. The nacelle acceleration was also very sensitive to the frequency of mode 4. For the stiff tower case, the predictions didn't even begin to match the test data. The soft tower case matched better, although there was some underprediction of the peaks in the response.

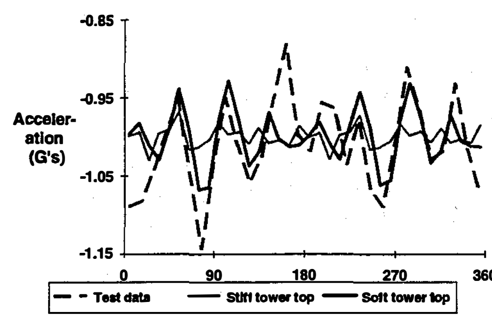

FIGURE 10. NACELLE VERTICAL ACCELERATION

Figure 11 shows the blade's root flapwise bending moment comparison. This parameter was not as sensitive to the frequency of Mode 4. The cyclic 8 per revolution response is somewhat overpredicted.

We investigated the effects of varying turbine and aerodynamic input parameters on the predicted behavior of these loads. One parameter we found to be of particuliar importance was tower shadow velocity deficit. We ran cases where we varied the amount of deficit from 10% to 50%. Figure 12 shows these results. It seems that the tower shadow, felt by the rotor to some extent during every rotor revolution, acts as a triggering mechanism for this response.

The fact that this response is sensitive to the placement of the system modes, especially mode 4, supports the claim that this response is due to a system interaction. In Figure 13, we see the effect of shifting mode 4 from 8.1 Hz to 6.9 Hz It is clearly seen that shifting this mode greatly impacts the predictions. We were careful to keep the tower shadow velocity deficit constant in these runs, as well as the frequencies of modes 2 and 3 by changing only the tower-top stiffness. It must also be mentioned that mode 4 can be modified by changing the nacelle bedplate stiffnesses. We did see some evidence of bedplate flexing in our observations of modal test animations.

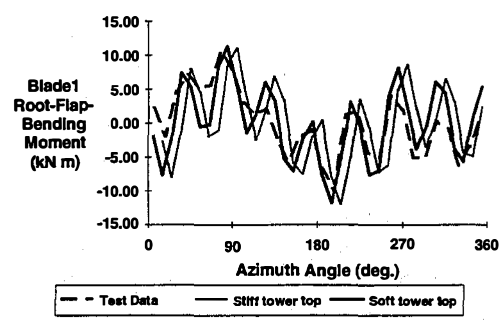

FIGURE 11. BLADE ROOT-FLAPWISE-BENDING MOMENT

One of the main uncertainties we still face is why mode 4 lies at 7.4 Hz instead of our original estimate of 8.1 Hz We carefully input the tower top, bedplate, and shaft properties from our best available knowledge of the turbine's properties. Somewhere in this system there is extra compliance that causes this mode to be lower than previously thought. Tuning the ADAMS turbine model to agree with these stationary modes does give us good results. It is possible that compliance could be added in other locations and similar results obtained.

The ADAMS tool can be used, once this discrepancy in inputs is resolved, to miminize such an interaction as seen here. We ran many different cases to determine the sensitivity of predicted blade root-edgewise-, flapwise-bending moments and nacelle pitch acceleration to different parameters, such as blade pretwist, principal axis orientation, blade stiffness, etc. Some of these parameters had a large effect on the predicted response of this machine, mainly because they changed the frequency of mode 4. Determining those parameters which can be most readily changed in the design and which also increase the frequency of mode 4 above 7.4 Hz to some acceptable value seems important. In addition, ways should be found to reduce the tower shadow velocity deficit, since it acts as a triggering mechanism for this response.

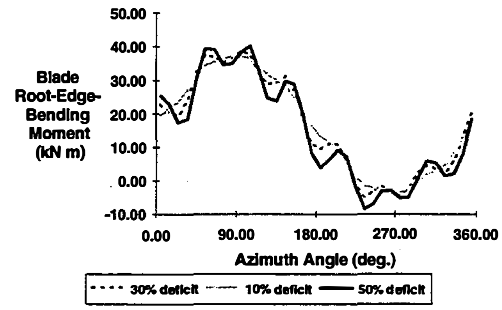

FIGURE 12. EFFECTS OF DIFFERENT TOWER SHADOW DEFICITS

CONCLUSIONS

We have shown the steps we took to develop a system dynamics model for this machine using ADAMS. We have also described a system dynamics interaction problem and shown how we have gained an understanding of this problem through the use of ADAMS in conjunction with modal test data.

We found the use of modal test data to be crucial in helping identify the cause of this interaction. Our initial ADAMS turbine model overpredicted the frequency of mode 4, which is highly involved in this interaction. When this mode was tuned in the ADAMS model to correspond to modal test data, we obtained good agreement between the field operating test data and ADAMS predictions. We still do not know the cause of the overprediction of this mode. Some extra compliance exists in the tower top or the nacelle bedplate that is hard to identify from turbine properties and drawings.

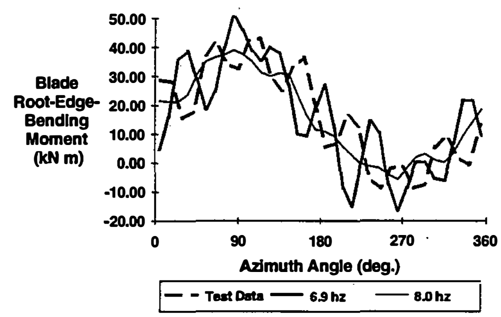

FIGURE 13. EFFECTS OF DIFFERENT FREQUENCIES OF MODE 4

With the ADAMS model tuned to obtain agreement, this tool should now be useful to designers to determine which turbine parameters can be controlled to reduce this machine interaction. The tool is useful not only for research into the cause of such an interaction but also for design considerations.

FUTURE WORK

We plan to investigate further the cause of the discrepancy in tower-top or bedplate stiffness input parameters. In addition, we plan to obtain the turbine's operating modeshapes, natural frequencies, and damping, both through analysis of turbine operating data and specialized modal tests. We also plan to expand our code predictive capabilities to obtain the machine's operating modes and to perform Floquet analysis to predict a periodic system's natural modes and stability characteristics. We also plan to run long model simulations with input of 3-D turbulence, produced with the VEER's 3-D wind simulation code and modified by NREL (5).

ACKNOWLEDGMENTS

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