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Experimental Verification of the Implementation of Bend-Twist Coupling in a Wind Turbine Blade

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Abstract
This paper presents some results and aspects of the multidisciplinary and interdisciplinary research oriented for the experimental and numerical study in static and dynamic domains on the bend-twist coupling in the full scale section of a wind turbine blade structure. The main goal of the conducted research is to confirm experimentally the numerical prediction of modification of the dynamic and static properties of the wind turbine blade. Bend-twist coupling was implemented by adding angled UD layers on the suction and pressure side of the blade. Static and dynamic tests were performed on a section of a full scale wind turbine blade provided by Vestas Wind Systems A/S. The results are presented and compared to the original and modified blade measurements. The second part of the paper presents the structural dynamics identification, which was performed by means of experimental modal analysis. A finite element method model was developed, updated and validated against the static measurement results. Based on the validated model the modified design of existing blade was studied. A baseline concept of the modification was implementation for bend-twist coupling by means of application of additional UD1200 composite material layers. The original blade section was modified with four layers of UD1200, which were laminated on the pressure and suction side of the blade, with an angle of 25° to create a measurable flapwise bend-twist coupling.

2. Static investigation of original blade section
Static loads in bending, torsion and combination of bending and torsion were introduced with different loading force levels [1-3]. The coupling causes the feathering blade to twist under the bending load and as a result decreases the angle of attack. This paper presents the progress and results of a comprehensive long-term scientific research focused on the bend-twist coupling analysis, design and implementation in a wind turbine blade made from composite materials. First three parts of the paper briefly recall the research activity carried out [1-3] while the main focus is put on measuring and modeling the dynamic behavior described within the fourth part. The first part of the paper reports on the experimental and numerical studies of a standard wind turbine blade section. The wind turbine blade section made of composite material was statically tested and modeled. Different load configurations were applied at the tip of the blade section to assess the twist and bend behavior [1].

The second part of the paper presents the structural dynamics identification, which was performed by means of experimental modal analysis. A finite element method model was developed, updated and validated against the static measurement results. Based on the validated model the modified design of existing blade was studied. A baseline concept of the modification was implementation for bend-twist coupling by means of application of additional UD1200 composite material layers. The original blade section was modified with four layers of UD1200, which were laminated on the pressure and suction side of the blade, with an angle of 25° to create a measurable flapwise bend-twist coupling.

In the third part of the paper the experimental and numerical analysis on the modified blade section is exposed to verify the design correctness of the bend-twist coupling. Finally in the fourth part the dynamic behavior of the modified blade section is experimentally identified with the assessment of the coupling. The main goal of the conducted research is to study, design and implement desired bend-twist coupled behavior. The original blade section was modified with four layers of UD1200, which were laminated on the pressure and suction side of the blade with the fibers angle of 25° to create a measurable flapwise bend-twist...
For adequate identification of the blade dynamic displacement, accelerations of the vibrations were measured in 130 points. Thirteen equidistant measurement cross sections were defined along the spanwise direction (Z) every 0.5 [m]. Each cross section contains five measurement points in which accelerations were acquired along the flapwise (X) and edgewise (Y) direction. These points are located at the leading edge, trailing edge, on the line of airfoil maximum thickness and in the mid-points between the previous three. Measurement directions were precisely defined based on the CAD geometry of the blade section.

During the processing of the data, some significant noise was observed in the acquired FRFs in the low frequency region. The driving point coherence functions show a small drop in this region meaning a non-ideal excitation (Figure 7).

The modal parameter identification technique was not able to clearly stabilize modes in this region, possibly resulting in some local errors in the mode shapes below 7 Hz. The estimation provided natural frequencies, mode shapes and corresponding damping ratios in the frequency bandwidth 0-60 Hz. First five out of 12 identified mode shapes are provided on Figure 8.

MAC (Modal Assurance Criterion) can be used to compare two modal models. The MAC between two mode shape vectors, \( \phi_i \) and \( \phi_j(s) \), is defined as:

\[
MAC(s) = \frac{\langle \phi_i(s), \phi_j(s) \rangle}{\langle \phi_i(s), \phi_i(s) \rangle^{0.5} \langle \phi_j(s), \phi_j(s) \rangle^{0.5}}
\]  

If a linear relationship exists between the two complex vectors, \( \phi_i \) and \( \phi_j(s) \), the MAC value will be near to 100. If they are linearly independent, the MAC value will be small (near zero). Figure 9 shows a comparison between the AutoMAC of the modal model obtained by considering only the sensors on the blade and the one where also the response of the supporting structure is included.

Low valued off-diagonal terms for the blade only model ensure linear independence of estimated modal vectors. The correlation between off-diagonal terms is increased when including the supporting structure in the analysis. This is due to the fact that the clamping is not perfectly rigid and the support has its own dynamic behavior which influences the measured response of the blade.
related to dynamic properties of the supporting structure.

5.1 Correlation analysis for the simulation and test results

Based on the estimated experimental modal model and two developed FEM models (modelling the original and modified blade), the correlation analysis can be applied. The FE model should be characterized by good consistency of the natural frequency values and mode shapes obtained from measurement. Modal Assurance Criterion is used as the original-modified blade simulation and also test-simulation correlation metrics.

The global axis system used to define the test model differs from that used for the FE model. In order to make the models match it is necessary to apply geometric correlation by translation and rotation of the test model (Figure 10). Next step is node mapping. The number of measurement nodes is much less than the FE nodes. Modal vectors are compared only for the nodes from FE which are located closest to the measurement points. Only the portion of the blade after the clamp is considered.

For tests described in section 2 to 4, the support structure was not taken into account and measured data reduced to obtain a perfectly rigid boundary condition on the clamped section [1-5]. Since the FEM model of the blade is modeled using the same assumption, some differences in frequency values and mode shapes from experimental results for the modified blade are expected. The best test and simulation modal vectors consistency can be observed for the 2nd flapwise bending. The consistency of the results can be recognized as satisfactory, however the present differences need to be further investigated. Observing the values of the MAC criterion between test and simulation modes (Figure 9), differences can be notified. They are caused by the influence of the support structure and not perfectly excited 1st bending mode. Comparison of natural frequencies for experimental and simulation for the original and modified blade is presented in Table 1.

The blade model was solved to compute mode shapes in the 0-60 Hz frequency bandwidth and computations were performed on a 50Tflop cluster.

First Modal Assurance Criterions were calculated for the corresponding modes in order to associate the closest numerical and experimental modes shapes (Figure 11). The procedure accounted for both natural frequency value and the mode shape consistency. The difference between the Test and FE frequencies can be explained by the modeling of the boundary condition as rigid in FE. Moreover, further differences in the frequency values between original and modified blade results are introduced by additional mass and stiffness implemented from angled UD layers on the suction and pressure side of the blade. These UD layers introduce measurable bend-twist couplings, which the original blade did not have both in terms of static and dynamic response of the investigated section of the blade. While in static response there was clear indication of the coupling, the modification of dynamic stiffness is not fully recognized which can be observed from the comparison of the simulation of original and modified blade structural dynamics (Figure 12).

Figure 10 Test and FE geometry correlation with node mapping.

Figure 11 MAC matrix for test and FE simulation modal vectors of modified blade.

Figure 12 MAC matrix for original and modified blade Finite Element models

Table 1 Comparison of the natural frequencies for the experimental and numerical results obtained for the original and modified blade.

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Original blade FE</th>
<th>Original blade Test</th>
<th>Modified blade FE</th>
<th>Modified blade Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.75</td>
<td>4.5 Hz</td>
<td>5.0 Hz</td>
<td>4.48 Hz</td>
<td></td>
</tr>
<tr>
<td>10.85</td>
<td>8.7 Hz</td>
<td>12.5 Hz</td>
<td>12.08 Hz</td>
<td></td>
</tr>
<tr>
<td>15.56</td>
<td>18.9 Hz</td>
<td>20.03 Hz</td>
<td>19.24 Hz</td>
<td></td>
</tr>
<tr>
<td>42.99</td>
<td>39.5 Hz</td>
<td>43.75 Hz</td>
<td>40.92 Hz</td>
<td></td>
</tr>
</tbody>
</table>
airfoil for each section are considered. By considering two consecutive cross-sections, relative bending is evaluated as the difference between the modal displacements in the x direction (see Figure 1). The tangent of the relative angle for the section is then computed by dividing the difference for the length of the section. On each cross-section, the global bending angle is computed by summing all the relative angles for the previous cross-sections. Computed angles are modal angles, so they depend on the scaling applied to the modal vectors. Moreover, torsional angles are computed assuming that the mass and shear centers are in the same locations.

5.3 Numerical and experimental twist and bend angles

The methodology described in section 5.2 is applied both to numerical and experimental modal analysis results. In the present paper only a limited part of the experimental results are presented in details. The results for first and second flap modes will be discussed.

Original blade simulation flap bending translation values (Figure 13 a, e) are plotted with the dark blue, edge bending with red and rotation with light blue.

Original blade experimental results are presented on Figure 13 a, g. Deflection in the flap direction is plotted with the red line, edge-wise direction with blue and rotation around the blade axis is marked with green. The angles values are assumed to be 0 in the clamped section.

For the modified blade section bending angle values are plotted in dark blue and twist in red. To have an overview of the overall behavior, the absolute angles are presented. Results are obtained by processing the fitted mode shapes.

5.4 Bend-twist coupling index

One of the main objectives is to investigate the amount of twisting introduced by the additional UD layers implemented on the modified blade. To obtain a quantitative measure of the coupling between twisting and bending angle, an index is introduced. For each considered blade cross-section, the ratio between the computed relative twisting and bending angles is evaluated. A coupling index value close to zero means that twisting is dominant. If the index is close to one, twisting and bending are of the same order of magnitude. Figure 14 shows the computed coupling index for the 1st and 2nd flapwise modes both for experimental and numerical results.

Figure 14: bend-twist coupling index for 1st (top) and 2nd (bottom) flapwise mode
Comprising the numerical and experimental results, some difference can be observed. These differences are directly related to the observation made for Figure 11. For the first mode shape the MAC value between experimental and numerical model is 69.5. Explanation of the lower MAC value for the 1st mode can be found in the coherence plot presented on Figure 7. It is a ratio of the maximum energy in a combined output signal due to its various components, and the total amount of energy in the output signal. Coherence is used as a measure of the power of the output signal that one can use to obtain the accuracy of the frequency response function measurements. The coherence function can take values that range between 0 and 1. A high value (near 1) indicates that the output is due almost entirely to the input and one can feel confident in the frequency response function measurements. A low value (near 0) indicates problems such as extraneous input signals not being measured, noise, nonlinearities or time delays in the system. The 1st mode is located in the frequency range of relatively poor coherence leading to decreased quality of its estimation. Shakers which were used in the measurement have a low frequency limit around 2 Hz. The excitation signal was random that provides homogeneous distribution of injected energy over the excited bandwidth. This led to the insufficient energy exciting the 1st mode. Moreover, shakers were hung from support cables. In such case at very low frequency in the sub-band 10 Hz range there is a problem to more inertia to push against the structure being excited for improved performance. To improve the excitation of the 1st mode shape additional mass was applied to increase the inertia of the shakers apparently bringing not much improvement. In the higher frequency where the displacements become lower as is the case for the 2nd mode the consistency between test and simulation is much better since the MAC value is 89.5. Bend and twist coupling index calculations are based on the experimental and numerical modal vectors. The difference between experimental and numerical models bend-twist indexes is caused by the relatively weak excitation of the 1st mode due to the abovementioned reasons. 

6. Conclusions 

This paper presents some results and aspects of the multidisciplinary and interdisciplinary research oriented for the experimental and numerical study in static and dynamic domains on the wind turbine blade modal behavior and the effect of excitation load on the blade structure. An extensive test campaign performed on the original and modified wind turbine blades section was presented. Test setups included different load configurations, excitation and measurement techniques of contact and non-contact type. Experimental test data examples were shown and used for two purposes. Firstly, to evaluate the ability of different test method to measure the bend-twist coupling and secondly to use the test results for FE-model updating. The common observation from displayed comparisons is that the original blade section did not have a measurable bend-twist coupling. Because one of the primary aims of this work is to develop and use a FE-model capable of modeling correct bend-twist coupling behavior, the original blade section was modified. For this purpose four UD layers were laminated on the pressure and suction side of the blade section to introduce a measurable flapwise bend-twisting. 

The successful implementation of the bend-twist coupling was confirmed by extensive static and dynamic measurement campaigns. In both experimental methods the comparison of original and modified blade properties clearly show the presence of the bend-twist coupling. Further research should introduce two damping in the FE model in order to get a realistic structural behavior of the wind turbine blade. Use of plywood plates and steel profiles should be included and contact elements should be applied to model the contact between shanks and blades. It is expected that the more sophisticated support structure FE representation will improve the consistency in between test and simulations. 

7. Acknowledgements 

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