Proof of concept: elliptical biaxial rotor blade fatigue test with resonant excitation

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Abstract. Rotor blades of wind turbines are subjected to high-cycle fatigue during their lifetime of 20-25 years. To show that a blade design fulfils the normative requirements, the test campaign usually consists of two consecutive cyclic tests, i.e., in the flapwise and lead-lag directions. The fatigue test excites the blade close to its corresponding natural frequency. Combining the two uniaxial tests into one biaxial test excites the blade in both directions simultaneously. Using the same excitation frequency for both directions results in the blade cross-sections describing an elliptical deflection path. To realize such a test while still exciting close to resonance, the natural frequencies for the two directions need to be equalized with the aid of decoupled masses and stiffness elements. This approach reduces the testing time, and induces a more realistic loading which is comparable with field conditions while keeping the energy consumption of the hydraulic actuation low. This work describes the concept of the elliptical biaxial rotor blade fatigue test with resonant excitation using a commercial blade design. To this end, the design model, which uses both a transient and a harmonic simulation, is validated with the experimental results of the test. The simulation model and the experiment agree well with each other in terms of displacements and loads along the blade.

1. Introduction
Rotor blades of wind turbines are subjected to high-cycle fatigue during their expected lifetime of 20-25 years. As proof that a rotor blade design is able to withstand the fatigue loads occurring, experimental validation tests are necessary, as described in IEC 61400-23 [1]. The test campaign usually consists of several static extreme load tests and two consecutive cyclic fatigue tests, i.e., in the flapwise and lead-lag directions. To achieve certification, these tests require the target bending moment distributions to be exceeded within a specific area of interest (AoI) along the blade without excessive overloading, which could damage the blade prematurely. The fatigue tests are time consuming since they require two to seven million load cycles. During the fatigue tests, the blade is excited close to its corresponding natural frequency to reduce the amount of energy which has to be fed in. The associated bending moments over the blade length are realized so as to be damage-equivalent to field loads. This represents a major drawback of these unidirectional tests: The loads introduced into the blade do not necessarily represent the realistic load conditions, as shown by Rosemeier et al. [2].

One approach to realizing more realistic testing is to combine the two unidirectional tests into one so-called biaxial test [3]. Some attempts at a possible realization can be found in the
literature [3–7]. Conducting the biaxial fatigue test is much more challenging, however, as the unidirectional frequencies and the phase angle between the two displacement excitations have to be chosen carefully so as to apply the desired combined loading.

Several attempts at performing a biaxial fatigue test with random phase resonant excitation can be found [3, 6, 7]. The concepts here involve summing the damage introduced for the fatigue resistance evaluation using complex computations. A concept with more predictable loads requires the excitation frequencies in both directions to be at a fixed integer ratio, leading to Lissajous motion of the blade tip [4, 5, 8].

Recently, we developed and realized a biaxial test concept with elliptical resonant excitation which has a frequency ratio of 1:1. The concept requires the near equalization of the flapwise and lead-lag natural frequencies using masses and spring elements, as described by Melcher et al. [9]. Decoupled load elements modify the blade properties independently in each direction, as introduced by Post et al. [10].

The approach requires a precise definition of the set-up to match the frequencies and achieve the target bending moments. In practice, this precision can only be provided by an optimization-based computational process. Lee and Park [11] used an algorithm to optimize the overloading by determining the optimal mass distribution, actuator position, and excitation frequencies in unidirectional set-ups. A different optimization algorithm was evaluated by Zhang et al. [12], which was especially used to determine the position of blade tip cut off. Melcher et al. [9] presented an optimization-based tool to determine several test objectives and variables such as positions and values of decoupled masses and stiffnesses as required for an elliptical biaxial fatigue test. Of specific relevance for the elliptical test is the adequate determination of the excitation phase angle. It has a very high impact on the desired bending moments in the combined regions, the so-called off-axis directions, as investigated in our earlier work [13].

This work presents the experimental validation of the elliptical biaxial fatigue test approach with resonant excitation. The aim is to demonstrate the feasibility of the approach, and to quantify the test prediction accuracy by comparison of simulation and experiment, as shown in figure 1.

![Figure 1. Simulation model (a) and experimental test setup (b) for elliptical biaxial rotor blade fatigue test.](image-url)
2. Method

To validate the biaxial fatigue test concept, numerical simulations and a corresponding experiment are performed on a state-of-the-art commercial rotor blade, which is between 60 m and 70 m long\(^1\). To reduce the complexity of this feasibility study, a reduced AoI between 14\(\%\) and 37\(\%\) of the blade length is chosen. The test setup selected is optimized accordingly. This includes the position and required impact of the spring and mass elements.

2.1. Numerical setup definition

The biaxial test setup is defined using the optimization approach of Melcher et al. [9, 13] which is based on a preloaded harmonic simulation of a parametrized finite element (FE) model implemented in ANSYS APDL 18.1 [14]. In the optimization, the magnitude of the spring stiffnesses and the masses, and their positions, and the excitation frequency and phase angle are used as design parameters to find a feasible fatigue test setup which matches the target bending moment distributions as closely as possible, and equalizes the natural frequencies for the flapwise and lead-lag directions to allow for elliptical biaxial resonant excitation.

The final FE model of the test setup is shown in figure 1(a). The blade shell displayed serves only visualization purposes, as the blade is actually represented by 120 BEAM188-elements in the model using preintegrated composite beam cross-sections. The load-elements are modeled using a combination of MPC184-rigid-beam-elements for rigid connections, MPC184-rigid-link-elements for hinged pushrods, MASS21-elements for masses, BEAM188-elements for mass-levers and spring-beams, and LINK11-elements for actuators.

Once the test setup is defined, another approach using transient simulation evaluates the test setup. The same FE model is used, but unlike the harmonic simulation, this approach considers nonlinear effects. In this simulation, the actuator elements are excited using sinusoidal stroke input signals with frequency, phase angle and amplitudes taken from the preceding harmonic simulation. This transient approach is deemed more precise, but also requires significantly more computational power, and is less stable than the harmonic simulation and therefore unsuitable for use in the optimization.

2.2. Experimental setup

For the fatigue test, the blade root is attached to an inclined test block, tilting the blade pitch axis 7.5\(^\circ\) upwards from the horizontal. Moreover, the blade is rotated through 15\(^\circ\) about the pitch axis in order for the vertical and horizontal load elements to better align with the actual mode shapes of the blade, which helps to influence the natural frequencies independently. Although this means the load elements are not aligned exactly with the flapwise and lead-lag directions of the blade, they will nevertheless be referred to as such in the following. The load elements attached in the validation setup consist of the following, as shown in figure 2:

\(\begin{align*}
(a) & & \text{One spring element to increase the flapwise natural frequency of the test setup} \\
(b) & & \text{Two decoupled mass elements to decrease the lead-lag natural frequency of the test setup} \\
(c) & & \text{Two hydraulic actuators to excite the test setup close to its natural frequencies}
\end{align*}\)

To introduce the loads into the test blade, we adapt the loadframe design proven at the Fraunhofer IWES test facility, which consists of a steel frame with wood inserts clamped to the blade. A total of three loadframes are mounted to allow the load elements to be added at 38\(\%\), 48\(\%\) and 61\(\%\) blade length, as shown in figure 1.

The passive elements (masses and spring) are connected to the loadframes using pretensioned steel pipes (40 kg m\(^{-1}\)) as pushrods with swivel bearings at both ends. To minimize the effect

\(^1\) Detailed information on the blade cannot be provided due to confidentiality constraints.
of the decoupled masses on the flapwise direction, the pushrod’s angle of incidence on the blade must change as little as possible. This is achieved by increasing the length of the horizontal pushrods up to 6.7 m. Hence, the blade pitch axis is tilted by 2.5° to the side for optimal utilization of the available hall space.

The first hydraulic actuator, which excites the blade in the flapwise direction, is directly attached to the loadframe at 48 % blade length. The second, smaller hydraulic actuator, which excites the blade in the lead-lag direction, is connected to the same loadframe with a pushrod and a lever to reach higher displacement amplitudes at the blade than the actuator itself is capable of. Also, the actuator has to be decoupled from the flapwise movement, as this is much larger than the lead-lag movement. The lever with added mass of 3.6 t at the top further acts as a third decoupled mass and reduces the lead-lag natural frequency.

The sinusoidal strokes of the two actuators are controlled independently, only the phase angle is fixed. Apart from this synchronization, the actuators are controlled as in a uniaxial fatigue test, by applying their sinusoidal strokes to the rotor blade and observing the resulting loads.

Adding the decoupled masses to the other two loadframes also calls for a vertical lever construction, supported at a pivot in the center, with the required mass divided into two separate steel blocks at the top and bottom of the lever, and the pushrod transferring the load to the blade connected to the top. The total inertia of these constructions acting on the blade corresponds to that of masses of 4.0 t at 38 % blade length and 3.8 t at 61 % blade length.

To provide the required stiffness of 31 kN m−1 with a maximum deflection of 300 mm, the spring at 38 % blade length is designed as a beam 6 m in length made of glass-fiber reinforced epoxy.

2.3. Measurement data acquisition
The measurement data are sampled at 50 Hz and collected in a single measurement system.

All load elements are equipped with load cells to measure the force at the pushrod. The hydraulic differential forces of the actuators are also recorded.

To track the displacement of the blade by means of a three-dimensional camera system, two reflectors are attached to each loadframe and one to the tip of the blade, as shown in figure 3(a).

To determine the bending moments along the blade, the method described by Lee et al. [15] is used. By measuring the strain at multiple positions along the circumference of selected cross-
sections, the flapwise and lead-lag bending moments can be computed on the basis of preceding calibration tests. In the present case, this was used to derive four values for both the flapwise and lead-lag bending moments at every cross-section selected by using different strain gauge combinations. The mean values for each cross-section are used for further computations and are shown in the results below. The off-axis bending moments $M_\alpha$ in the $30^\circ$, $60^\circ$, $120^\circ$ and $150^\circ$ directions are derived from the flapwise moment $M_F (0^\circ)$ and the lead-lag moment $M_L (90^\circ)$ using equation (1).

$$M_\alpha = M_F \cdot \cos \alpha + M_L \cdot \sin \alpha \tag{1}$$

3. Results and discussion

The experiment and the corresponding simulations are performed at different load levels up to 90% target load. A higher load is not applied so as not to damage the specimen unnecessarily. Furthermore, different frequencies and phase angles are tested to evaluate their influence on the fatigue test. These parameter studies show good correlation between experiment and simulation model but exceed the scope of this work.

The results presented in the following use a combination of a frequency of 0.548 Hz and a phase angle of 78° at 90% load, which results in the best match for the target bending moments.

3.1. Displacement

Figure 3(a) shows the position of the numbered points on the blade whose motion was tracked during the experiment. The corresponding measured motion trajectories (red lines) are presented in figure 3(b). These data are in the test hall coordinate system, whose origin is below the blade root. Since the pitch axis is tilted, the trajectories are offset correspondingly.

![Figure 3](image.png)

**Figure 3.** Positions of motion-tracked reflectors using a three dimensional camera system (a) and corresponding measured and simulated displacement trajectories (b).

The black dotted and dashed lines in figure 3(b) represent the results from the harmonic and transient simulation, respectively. The correlation between measurement and transient simulation is good, especially for point 1 at the tip. The mean value of the harmonic simulation deviates slightly from the experiment, since the geometric nonlinear effects are neglected in this simulation. The other points exhibit less displacement, leading to smaller differences between the two simulation methods. Moreover, the shape of their motion in the measurement agrees well with that in the simulation. The offset between measurement and simulation at points 2, 6 and...
7 may be due to measurement errors while finding the undeformed position of the corresponding FE nodes.

3.2. Bending moments
The test to target bending moment ratios are shown in figures 4 and 5. Figure 4 shows the moment envelopes for three exemplary blade cross-sections at 14%, 29% and 43% blade length. The axis scale ratio of the plots conforms to the actual flapwise to lead-lag moment ratio. The positions of the corresponding cross-sections are also indicated by black dashed lines in figure 5, which displays the moment ratio distribution along the blade for the flapwise and lead-lag directions (figure 5(a)) and for the off-axis direction (figure 5(b)). The legend describes how color and graph type are combined for the individual data plots.

Figure 4. Target, measured and simulated flapwise moment over lead-lag moment envelope of cross-sections at 14% (a), 29% (b) and 43% (c) of blade length.

Figure 5. Measured and simulated test to target moment ratio along blade length for flapwise and lead-lag direction (a) and for off-axis directions (b). Color and graph type combination describe the case.
In general, all simulated load directions show good agreement with the measurements. The lead-lag moments only exhibit noteworthy differences outboard of 60% blade length. Hence, in figure 5(b), the harmonic simulation results are omitted for reasons of clarity.

When only the flapwise and lead-lag moments are considered (figure 5(a)), the simulation results are in good agreement with the measurement data. Within the AoI, in particular, the deviation between measurement and simulation is less than 3%. Off-axis directions deviate by less than 5% within the AoI. In addition, the deviation between off-axis simulation and measurement data seems to change along the blade. At 14% blade length (figure 4(a)), the simulation amplitude is higher than the measurement amplitude in the 150° direction and lower in the 60° direction, whereas at 43% blade length (figure 4(c)), the opposite is the case. This behavior may be caused by inaccuracies in the beam element properties representing the mass and stiffness distributions of the blade.

Even though the area of interest was chosen to be between 14% and 37% blade length, the data for the flapwise and lead-lag directions indicate it could be extended to 43% blade length. Only the 30° and 60° off-axis moments are slightly below the target values in the extended area.

### 3.3. External loads

Figure 6 shows the loads which are generated by the different load elements.

The actuator strokes over time are shown in figure 6(a). Here, good agreement between simulation and experimental results is observed. Only the simulated amplitude of the flapwise actuator is almost 4% higher than in the measurement.

Figure 6(b) shows the actuator forces. These values differ from sinusoidal curves, which are assumed in the harmonic simulation. Also, the deviation between transient simulation and measurement is large, especially for the lead-lag actuator. The cause of this deviation might lie within the hydraulic system and/or the friction in the actuators, neither of which is taken into account in the simulation.

Finally, the forces at the pushrods of the spring element and the decoupled masses are displayed in figure 6(c). The force of the spring shows good agreement. The force of the
decoupled mass 1, at 38% blade length, also exhibits good agreement apart from a superimposed signal at higher frequency. This signal might result from an unknown resonance within the load element structure itself, which is not captured by the simulation. The force of decoupled mass 2, at 61% blade length, reveals a nonlinear behavior which is caused by the changes in the pushrod’s angle of inclination over a range of 22°. This behavior is largely reproduced by the transient simulation.

4. Conclusion
Overall, we have shown that a biaxial fatigue test with elliptical resonant excitation can be realized by equalizing both main excitation frequencies in a 1:1 ratio. This test concept is therefore feasible. In addition, the optimization-based test setup definition provided the accuracy required to match the target bending moments. Moreover, the transient simulation was in good agreement with the experimental displacements, bending moments, and forces, with only negligible deviations, and is therefore validated.

However, for demonstration purposes, the AoI considered in this study was reduced to 23% of the blade length. For certification, a larger AoI is necessary (about 55%), requiring more load elements. Furthermore, the time-saving potential of the approach presented and insights into the increased structural reliability of the blade need to be investigated.

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