Prediction of Dynamic Loads and Induced Vibrations in Stall STALLVIB

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Abstract The results from a Joule-III research project are presented. The objectives of the project have been improvement of design methods for stall regulated wind turbines related to stall induced vibrations and dynamic stall. The primary concern is the edgewise vibrations in the fundamental blade natural mode shape, which have caused trouble on modern wind turbines of the 500 kW size, corresponding to a rotor diameter of approximately 40 m. A theoretical study, based on quasi-steady aerodynamics, confirms that the basic source driving the vibrations is energy supplied from the aerodynamic forces to the vibration during stalled operation. The phenomenon can be described as negative aerodynamic damping. The theoretical approach identifies the main parameters controlling the phenomenon. They are related to the steady and the dynamic airfoil characteristics, the overall aerodynamic layout of the blade, e.g. chord length and twist, the structural properties of the blade, e.g. structural damping and properties controlling the resulting vibration direction. Furthermore, full aeroelastic calculations and comparison with measurements show that the properties of the supporting structure, i.e. the nacelle and the tower, are important, as this part of the structure may either resist the vibration by supplying damping or support the vibration, when the frequencies of the coupled rotor and tilt-yaw mode shapes in the rotating frame of reference -- usually denoted the backwards and the forwards whirling frequencies -- are close to the blade natural frequency. It is confirmed that qualified aeroelastic calculations can be used for determining the influence of changing the primary design parameters. The design guidelines therefore builds on both the simple quasi-steady models, which can be used for the preliminary choice of the design variables mentioned above, and full aeroelastic calculations. The full aeroelastic calculations refine the design basis and should be used for choosing the final design variables and for final verification of the design. Through this design procedure it is possible to assess the required safety margin against stall induced vibrations.

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Contents

Preface

The project has been carried out by the following persons from different European research institutes, universities and a wind turbine manufacturer:

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The present report is the publishable, final report for the project where a summary of the work is presented. The other final reports of the project are:

Petersen, J.T. et al. "Prediction of Dynamic Loads and Induced Vibrations in Stall". Final report on contract JOR3-CT95-0047, period 01.01.1996-31.12.1997. Confidential.

Petersen, J.T. et al. "Prediction of Dynamic Loads and Induced Vibrations in Stall". Risø-R-1045(EN). Risø National Laboratory, Roskilde, May 1998.

Madsen, H. A. et al. "Field Rotor Measurements – Data Sets Prepared for Analysis of Stall Hysteresis". Risø-R-1046(EN), Risø National Laboratory, May 1998.

1 Objectives

The objectives of the project are related to improvement of design methods in the area of dynamic stall, 3-D flow effects and stall induced vibrations. The primary concern is the edgewise vibrations in the fundamental blade natural mode shape, which have caused trouble on modern wind turbines of the 500 kW size, corresponding to a rotor diameter of approximately 40 m. Specifically, the work aims at supporting and improving the design phase, where appropriate safety margins must be obtained.

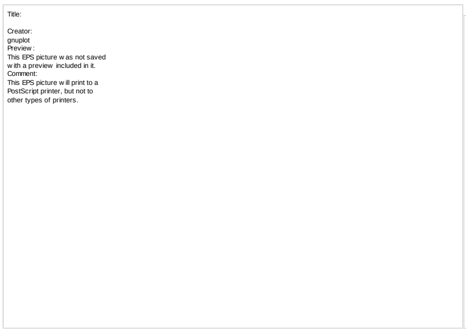

Figure 1-1 Edgewise blade root bending moment for a 500 kW turbine during operation in stall. Damped edgewise vibrations.

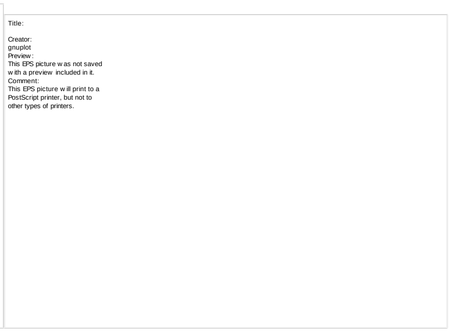

Figure 1-2 Edgewise blade root bending moment for a 500 kW turbine during operation in stall. Undamped edgewise vibrations at the edgewise natural frequency have started.

The area of work is illustrated in Figure 1-1 and Figure 1-2 which shows an example of stall induced edgewise vibrations measured on a prototype blade. The time trace in Figure 1-1 shows the quite normal moment dominated by gravity before development of the vibrations, whereas the trace in Figure 1-2 shows the moment after the dominating vibrations at the edgewise natural frequency have started. The objectives of the present project are to provide improved design and analysis tools for the stall controlled horizontal axis wind turbine (HAWT) rotor. The work supports the design of large-scale stall-regulated rotors, bringing together and exploiting the knowledge gained from research carried out under JOULE I and JOULE II. The overall objectives of the project are:

- Improvement of the prediction capabilities with respect to dynamic loads in stall and stall induced vibrations
- Establishment of guidelines aiming at achieving safety margins against stall induced vibrations.

The outcome of the project is improved and validated aeroelastic models with updated sub models for stall hysteresis and 3-D flow effects. In particular improvement of prediction capabilities together with derived design guidelines for safety margins against stall induced vibrations are made available for the industry.

2 Technical description

2.1 The principle of stall induced vibrations

When a wind turbine blade vibrates in a natural mode, the aerodynamic force on a section of the blade might have a component, which is proportional to the vibration velocity of the blade section. Such a force is experienced by the blade as a viscous damping force. In the present context we are interested in force terms arising in connection with velocities superimposed on the dominating velocity associated with the rigid body rotation of the blade due to rotor rotation. This principal velocity component together with the free wind velocity gives the main contribution to the force on the blade section, which results in the driving torque and the axial force. Velocities superimposed on the principal velocity contribute with a force, which with good approximation can be considered proportional to the superimposed velocity, and therefore act as a damping force - usually denoted the aerodynamic damping.

Under special operational conditions - primarily when the turbine is operating in stall - the aerodynamic damping force can be negative, meaning that the force and thus the flow supplies energy to the turbine structure. This results in a potentially self exciting system. For example, if the blade velocity originates from a vibration in a natural mode shape, a corresponding negative damping force will supply energy to the vibration, which increases, if the energy is not removed, e.g. through structural damping. This is the basic principle of stall induced vibrations.

2.2 Structure of the work in the project

The structure of the work in the project [1], [2], [3] has been first to identify the most important parameters for stall induced vibrations using a simple quasi-steady aerodynamic model for a two-dimensional blade section. This part of the project showed the importance of:

- the quasi steady C_L and C_D characteristics
- the direction of vibration of the blade section

Next, the analysis was extended to a whole blade using a simple mode shape representation of the blade. This part of the work showed the importance of:

- the mode shape of the blade
- the mode shape vibrational direction

Now, the aerodynamic model was extended to include stall hysteresis. A comprehensive analysis of a number of data sets from field rotor experiments at Delft University, ECN, Imperial College and Risø National Laboratory was carried out to see if an update and tuning of the different stall hysteresis models could be performed using these data sets. Only limited use of the data was possible due to problems with the measured angle of attack. However, extensions of the stall hysteresis models were developed with focus on stall hysteresis related to the edgewise movement of a blade section and hysteresis on C_D. This part of the work showed the importance of:

- parameters in the stall hysteresis models

- hysteresis on C_D and hysteresis from the edgewise movement of the blade (shed wake effects and variation of the relative velocity)

After these investigations with relative simple structural and aerodynamic models, full aeroelastic simulations with different models were carried out for a 500 kW turbine and comparisons with measurements were performed. This part of the work showed the importance of:

- the coupling between the blade vibrations and the dynamics of the whole wind turbine structure – local blade whirl and global rotor whirl
- the dynamic characteristics of the turbine (the relative position of the different eigenfrequencies)
- the structural damping

Finally, on basis of all the investigations a set of design guidelines were derived in order to achieve safety margins against stall induced vibrations.

Below, a few selected parts of the described work within the project will be summarized but for details the final reports [2], [3] should be used. After the summary sections follows a paragraph with the results in form of the guidelines.

2.3 Local aerodynamic damping on a blade section

A section of the blade as shown in Figure 2-1 is considered, and with the notation as in the figure a linearized expression for the aerodynamic damping is derived by use of quasi-steady, 2-D aerodynamics. In order to simplify the problem the influence of induction is neglected. This does not affect the fundamental conclusions, which can be drawn from the results.

The following matrix equation for the aerodynamic forces per unit length F_x^R and F_y^R can be derived [2]:

$$\begin{Bmatrix} F_x^R \\ F_y^R \end{Bmatrix} \cong \begin{Bmatrix} F_{x0}^R \\ F_{y0}^R \end{Bmatrix} + \begin{bmatrix} c_{xx}^R & c_{xy}^R \\ c_{yx}^R & c_{yy}^R \end{bmatrix} \begin{Bmatrix} \dot{x}_R \\ \dot{y}_R \end{Bmatrix} \qquad (2\text{-}1)$$

where the single matrix terms representing the damping coefficients are:

$$c_{xx}^R(r,V) = \frac{1}{2} c r \frac{r\Omega}{W}\left[\left(\frac{2r^2\Omega^2 + V^2}{r\Omega}\right) C_D - V\frac{\partial C_D}{\partial a} - VC_L + \frac{V^2}{r\Omega}\frac{\partial C_L}{\partial a}\right] \qquad (2\text{-}2)$$

$$c_{xy}^R(r,V) = \frac{1}{2} c r \frac{r\Omega}{W}\left[-VC_D - r\Omega\frac{\partial C_D}{\partial a} + \left(\frac{2V^2 + r^2\Omega^2}{r\Omega}\right) C_L + V\frac{\partial C_L}{\partial a}\right]$$

(2-3)

$$c_{yx}^R(r,V) = \frac{1}{2} c r \frac{r\Omega}{W}\left[-VC_D + \frac{V^2}{r\Omega}\frac{\partial C_D}{\partial a} - \left(\frac{2r^2\Omega^2 + V^2}{r\Omega}\right) C_L + V\frac{\partial C_L}{\partial a}\right]$$

(2-4)

$$c_{yy}^{R}(r,V)=\frac{1}{2}cr\,\frac{r\Omega}{W}\left[\left(\frac{2V^{2}+r^{2}\Omega^{2}}{r\Omega}\right)C_{D}+V\,\frac{\partial C_{D}}{\partial a}+VC_{L}+r\Omega\,\frac{\partial C_{L}}{\partial a}\right]$$

(2-5)

where c is chord length, ρ is density, Ω is rotor rotational speed and r is radius.

The c_{xx}^{R} term corresponds to pure in plane blade movement, i.e. movement parallel with the x_R axis and mainly edgewise blade movement, and the c_{yy}^{R} term corresponds to out of plane blade movement, i.e. movement parallel with the y_R axis and mainly flapwise blade movement. Usually, the blade movement direction will describe an angle relative to the rotor plane due to the twist of the blade principal bending axe, meaning that both x_R- and y_R- components are present. In this case both columns in the damping matrix are involved in the damping expression.

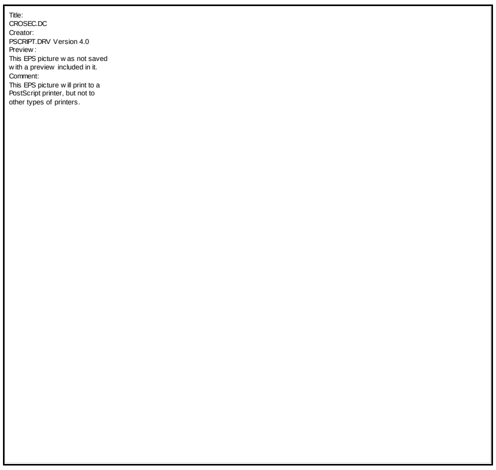

Figure 2-1 Flow velocities and forces at a blade section.

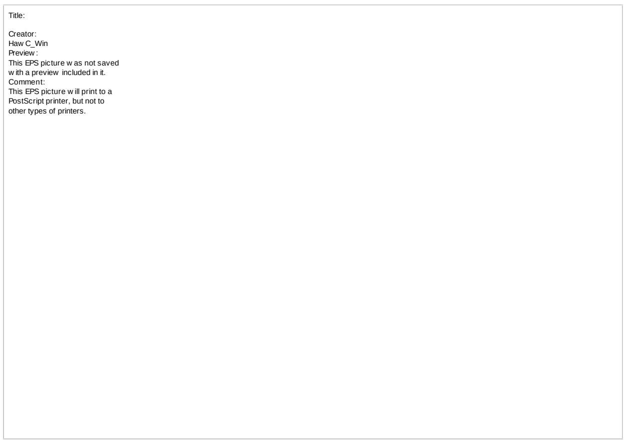

Figure 2-2 Example of airfoil data for a 18% section.

In order to illustrate the relative influence of the airfoil characteristics C_D, $\partial C_D/\partial\alpha$, C_L, $\partial C_L/\partial\alpha$, in the expressions for damping in eq. (2-2) to (2-5), typical airfoil data are chosen and the values of the single terms are calculated.

Figure 2-3 The in-plane damping coefficient c_{xx}^{R} *and its components (the individual terms in eq. 2-2) with reference to the R-co-ordinate system*

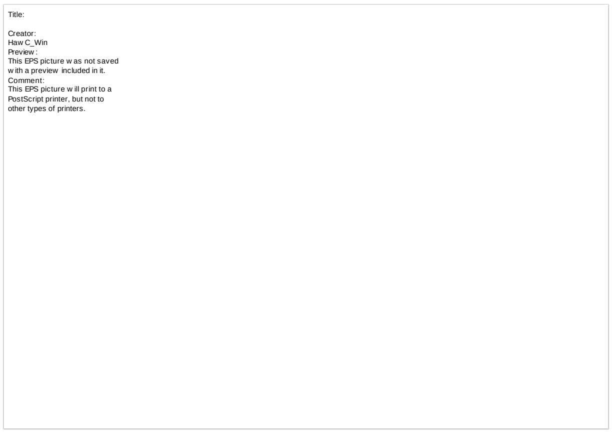

Figure 2-4 The out of plane damping coefficient c_{yy}^{R} *and its components (the individual terms in eq. 2-5) with reference to the R-co-ordinate system*

The actual airfoil data are shown in *Figure 2-2* and the main data for this example are:

c	=	1.06 m	(chord length)
ρ	=	1.23 kg/m	(air density)
r	=	14.00 m	(actual radius)
Ω	=	3.04 rad/s	(rotor angular velocity)
θ	=	2.00 deg.	(blade twist)

The results for the diagonal terms in the damping matrix c_{xx}^{R}, c_{yy}^{R} are shown in Figure 2-3 and Figure 2-4, respectively. In it is observed that for c_{xx}^{R} (edgewise direction) the resulting damping coefficient is negative in the shown wind speed range. All terms, except the one connected to the drag, give negative contributions. For c_{yy}^{R} all terms are positive, except the $\partial C_L/\partial\alpha$ term and the resulting damping is positive except for a wind speed range from 13-20 m/s. Three important conclusions can be drawn on basis of these results:

- the damping coefficient is strongly dependent on the vibration direction
- the damping is closely related to the airfoil coefficients

- the damping is dependent on the wind speed and for the edgewise direction the damping is most negative in the interval from 15-25 m/s (stall region)

The influence of the vibration direction on the damping was further investigated, and the following equation for the damping in an arbitrary direction, defined by the angle θ_{RB} in clockwise direction from the x_R axis, Figure 2-1, can be derived on basis of the equations (2-1) to (2-5):

$$c_{xx}^{B}(\theta_{RB}) = \cos^2(\theta_{RB})c_{xx}^{R} + \cos(\theta_{RB})\sin(\theta_{RB})(c_{xy}^{R} + c_{yx}^{R}) + \sin^2(\theta_{RB})c_{yy}^{R}$$

(2-6)

The damping as function of the vibration direction θ_{RB} is illustrated in Figure 2-5 for three different wind speeds. A real blade will vibrate both in the flapwise and edgewise directions and normally they will be almost perpendicular to each other. At 20 m/s it is not possible to find two directions with 90 deg. offset (a pair of flapwise and edgewise directions), where the damping is positive. However it is clear that an optimal pair can be found, where the damping has the least negative value for both directions (about –7 deg. for edge and 83 for flap). It is thus important to adjust the structural design of the blade to achieve less negative damping, both for the edgewise and flapwise direction.

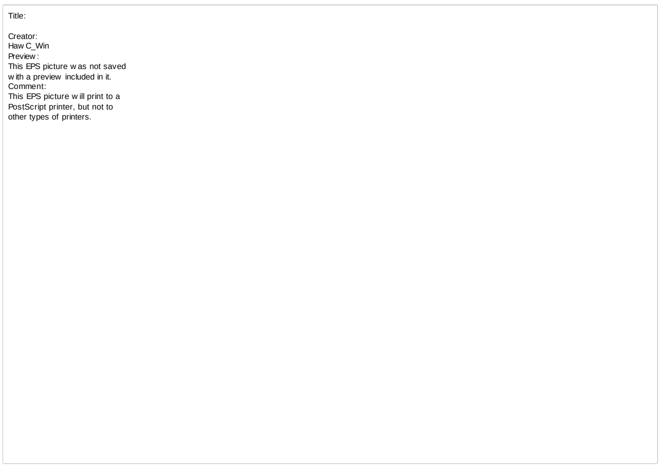

Figure 2-5 The damping coefficient c_{xx}^{B} as function of vibration direction (0 is edge and 90 is flapwise direction) with the wind speed as parameter.

It was concluded above that the damping depends strongly of the airfoil characteristics. For example a high drag coefficient will give a positive contribution to the damping in edgewise direction, Figure 2-3. However, such a design would collide with the requirement of good power production. Therefore it is important also to relate the damping coefficient to an expression for the rotor power P_u, which e.g. for the in-plane damping can be worked out as [2]:

$$c_{xx}^{R}(r,V) = -\frac{2}{r^2\,\Omega^2}P_u(r,V) + \frac{V}{r^2\,\Omega^2}\frac{\partial P_u(r,V)}{\partial V} \qquad (2\text{-}7)$$

This equation shows the important result that for constant power output (stall regulation) the in-plane damping coefficient will always be negative. However, with a combination of proper design of the vibration directions of the blade and optimal airfoil characteristics (e.g. no negative slope on the C_L curve) it is still possible to obtain positive aerodynamic damping for both vibration directions at constant power.

2.4 Modal aerodynamic damping for a blade

So far only the damping for a blade section has been considered. Extending the concept to a whole blade it is convenient to describe the damping in relation to the mode shapes of the blade, because the vibration in a specific mode is the structural response, which is most likely to be involved in the energy exchange related to the aerodynamic damping. The modal damping per unit blade length can be derived as [2] :

$$c_n(r) = c_{xx}^{Bn}(r)\, j_{n0}^{2}(r) \qquad (2\text{-}8)$$

where $j_{n0}(r)$ is the local amplitude of the nth mode shape of the blade, and $c_{xx}^{Bn}(r)$ is the damping coefficient in the local direction of vibration for mode shape n. This equation shows the importance of the mode shape for the total damping of the blade, as the local damping coefficient is weighted with the amplitude of the mode shape squared. It also improves the possibility for design of a blade with both good aerodynamic performance and damping properties. For example local airfoil coefficients with good damping properties (low C_{Lmax}) should be used, where the amplitude of the mode shape is high (at the tip of the blade), whereas airfoils with high C_{Lmax} could be used on the inner part of the blade, because the associated negative damping will not contribute much to the total damping, as the mode shape amplitude is small here.

2.5 Influence of dynamic stall on aerodynamic damping

Quasi steady aerodynamics has been used so far as this simplifies the equations considerably. However, the importance of dynamic stall and stall hysteresis for the dynamic loading of stall regulated wind turbines operating at high wind is well recognized.

Different stall hysteresis models already used by the participants were extended and updated with focus on stall hysteresis related to the edgewise movement of a blade section and hysteresis on C_D. The development of the classical engineering stall hysteresis models, such as the Beddoes model, has been based primarily on 2D wind tunnel test. However, only sparse wind tunnel experiments with varying inflow velocity exist. A comprehensive analysis of a number of data sets from field rotor experiments at Delft University, ECN, Imperial College and Risø National Laboratory was therefor carried out to see if an update and tuning of the different stall hysteresis models could be performed using these data sets. Only limited use of the data was possible due to problems with the measured angle of attack. However, analysis of one of the data sets indicated that the variation of the velocity w relative to a blade section on a rotor due to e.g. instantaneous yaw error has some influence on the stall hysteresis, [3]. Using the fgh model [4], [5], with parameters optimized to fit the Risø field rotor measurements, simulations of the normal force coefficient C_N with harmonic variations of angle of attack α and the relative velocity w were performed.

Considerable difference between the hysteresis loops are observed depending on the phase between α and w, in-phase in Figure 2-6 and counter-phase in Figure 2-7. This indicates some influence of w on the hysteresis loops, and one

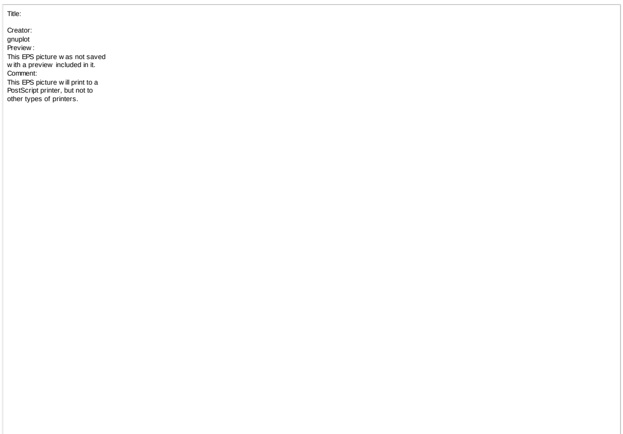

Figure 2-6 Simulations with the fgh stall hysteresis model with parameters from the Risø field rotor measurements. Variations of angle of attack and relative velocity are in phase.

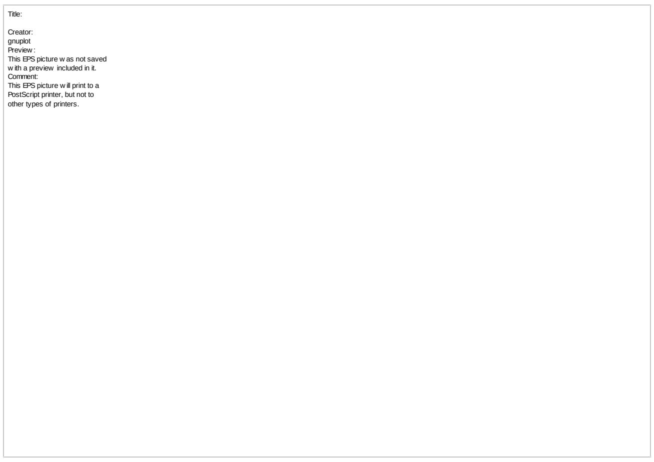

Figure 2-7 Simulations with the fgh stall hysteresis model with parameters from the Risø field rotor measurements. Variations of angle of attack and relative velocity are in counter-phase.

Figure 2-8 Damping work calculated with an extension of the standard Beddoes model to include influence from variation of the inflow velocity w (shed wake effects and velocity effects). Compared with the standard Beddoes model (legend "no shed wake etc.") and quasi steady aerodynamics. Edgewise oscillations.

of the participants made an extension to the Beddoes model to simulate this effect.
The result for a blade section undergoing edgewise vibrations is shown in Figure 2-8 where the computed parameter is normalized work per cycle. Including the effect of w variations makes the damping (work coefficient) slightly less negative. However, the change is small compared with the considerable reduction in negative damping, when stall hysteresis is included, compared with quasi steady aerodynamics.

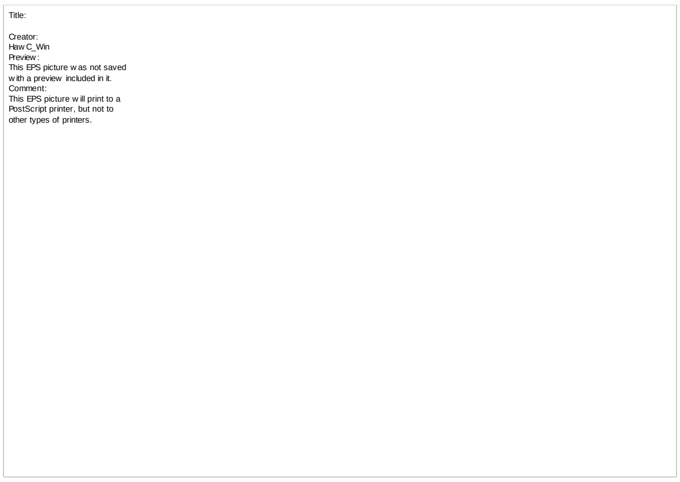

Figure 2-9 Aerodynamic damping (logarithmic decrement) for a blade vibrating in its 1st edgewise mode shape. With and without dynamic stall on C_L and C_D.

Also the influence of hysteresis on C_D using the approach formulated by Leishman and Beddoes [6] has been investigated. The simulation was performed for a blade in its first edgewise mode and the logarithmic decrement decreases slightly when hysteresis on C_D is included, Figure 2-9. This means that a dynamic stall model without hysteresis on C_D can underestimate the vibrational phenomenon.

2.6 Simulations with a full aeroelastic model

In order to generalize the findings based on the simplified models as described above, but also to investigate aspects of damping which cannot be simulated with simple models, such as coupling between blade vibrations and the turbine structure, full aeroelastic simulations were performed by different participants. The simulations were performed on a Bonus 500 kW prototype wind turbine, where detailed measurements were available. As an example of computed response, the edgewise blade bending moment during operation at 23 m/s is shown in Figure 2-10. At start the moment corresponds almost to the moment from the gravity of the blade but soon severe vibration at the edgewise frequency is seen to build up. This corresponds well with measurements, where edgewise vibrations have been observed at high wind. The cause for strong vibrations is that the aerodynamic damping is less than the structural damping. Using the full aeroelastic model the logarithmic decrement for the aerodynamic damping of the blade in its 1st edgewise mode was calculated. With dynamic stall included it shows a minimum aerodynamic damping of almost –4% [log.

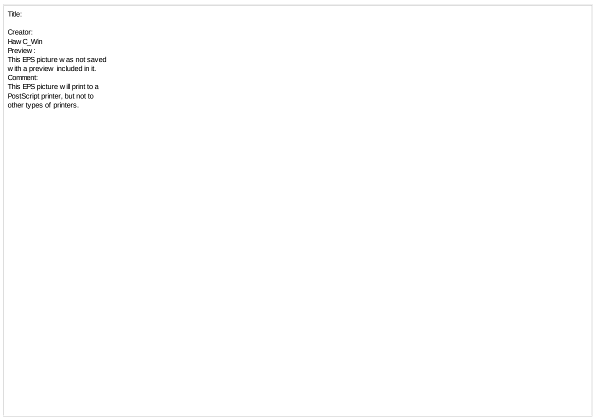

Figure 2-10 Simulated blade root bending moment

decrement] at 23 m/s. As the structural damping has been estimated to +3% [log. decrement] the total damping is negative in a wind speed interval around 23 m/s, and un-damped vibrations will occur. This example illustrates the importance of the structural damping and increasing this parameter from changes in design for example by including damping material in the blade will reduce the problem of induced vibrations.

2.7 Coupled rotor modes and edgewise vibrations - whirling

During the project detailed investigations of the coupling of the blade vibrations with the rotor support – the main shaft, the nacelle frame and the tower were performed. In order to clarify the basic mechanisms of the coupling, two simple structural models were developed; 1) for analysis of local blade whirling and 2) for analysis of global rotor whirling.

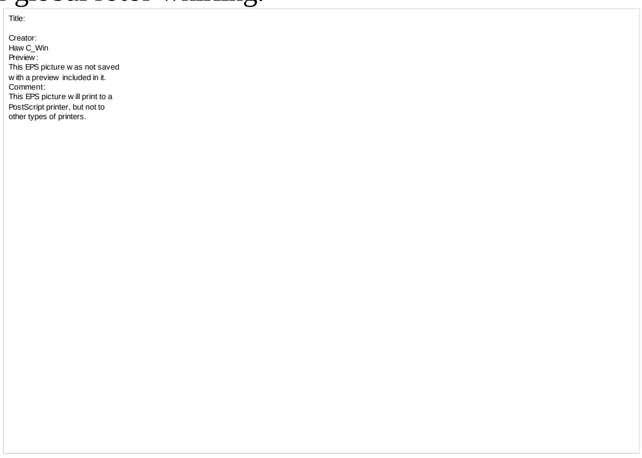

Figure 2-11 Simulated blade root edgewise bending moment with increased shaft stiffness which can be compared with the standard configuration in Figure 2-10.

In the simplified model for global rotor whirling the rotor is considered stiff and the main degrees of freedom are the 1st and 2nd tilt and yaw modes due to rotations in the tower top and the bending of the shaft. Due to gyroscopic coupling the associated frequencies will vary with the rotational speed. The center of the shaft at the hub will typically describe an elliptical path, which can be turning in the rotational direction (forward whirl) or in the opposite direction of rotation (backward whirl).

Local blade whirl is associated with the in-plane blade edgewise vibrations. Although the edgewise mode shapes are dynamically in balance within the rotor with respect to the shaft torque, a resulting in-plane inertia force will occur, varying with the blade edgewise frequency, and the force will rotate with the edgewise frequency within the rotor plane corresponding to a whirl of the modes.

Now, this in-plane inertia force can interact with the global whirl modes and with increasing effect the closer the blade edgewise natural frequency is to one of the global whirl frequencies ± 1P.

As an example, a simulation with increased shaft stiffness for the Bonus turbine was carried out. In this simulation the global whirl frequencies are separated from the edgewise frequency. The resulting time trace for the edgewise blade bending moment is shown in Figure 2-11 and comparing with the corresponding time trace for the standard configuration, Figure 2-10, it is seen that the un-damped vibrations have almost disappeared.

3 Results and conclusions

Basically, the results confirm that the edgewise vibrations are caused by negative aerodynamic damping, i.e. energy supplied to the vibrating blade by the aerodynamic forces. It is shown that the occurrence of stall induced vibrations are controlled primarily by properties related to 4 principal areas, which influence partly the aerodynamic damping, partly the structural damping and partly the properties of the complete structure with respect to either damping or amplification of an existing edgewise vibration.

The results from the investigations performed during the present project have been used for formulation of a set of design guidelines and recommendations, particularly aiming at supporting the design of wind turbines with an adequate safety margin against stall induced vibrations. The majority of guidelines and recommendations are general, but some are based on investigations on the typical European 3-bladed concept with fixed blades and rather stiff support of both individual blades and rotor, and these guidelines might therefore be specific to this turbine type. The relevance of specific design guidelines depends on the actual design situation and in particular on the freedom the designer has with respect to choosing design parameters. For example, it could be the design of a new turbine, where a considerable freedom to choose design parameters should be expected. Or it could be a situation, where re-design of an existing turbine is considered, for instance due to detected problems with stall induced vibrations.

3.1 Design guidelines

The design guidelines are related to four principal areas:

- aerodynamic characteristics of the blade, represented by the quasi-steady and the dynamic lift and drag coefficients.
- The structural characteristics of the blade, which influence the mode shape and the natural frequency. Especially the orientation of the principal bending axes are of great importance, because the aerodynamic damping depends heavily on the resulting direction of the vibration.
- The material and structural properties, which influence the structural damping of the blade.

- The properties of the supporting structure, i.e. the nacelle and the tower, which partly modify the mode shape of the blade and partly contribute to the resulting damping of the structure.

Aerodynamic characteristics

Within this area the most important design parameters are:

- The airfoil type and the associated characteristics,
- the variation of the airfoil type along the blade and
- the blade planform.

The present design guidelines impart a recommendation to the designer, which might be new compared with previous common design approaches, namely, that the requirements on aerodynamic damping characteristics should be taken into account already when initiating the design of a new blade in parallel with requirements on rotor performance. The expressions for aerodynamic damping for a section as well as for the whole blade using quasi-steady aerodynamics are simple equations and they can easily be programmed and computed. In fact, they are well suited to be used in a numerical optimization algorithm.

Results from examples in the present work suggest some recommendations with respect to choice of airfoil characteristics and distribution of properties along the blade but a specific computation of aerodynamic damping should be included in a design situation. To minimize the negative aerodynamic damping, low lift airfoils should be preferred on the outboard part of the blade in order to reduce the resulting contributions from negative lift curve slope to both flapwise and edgewise aerodynamic damping. However, in order to obtain a favorable power production and a slender blade, it might be desirable with high lift airfoils on the inboard part of the blade, e.g. obtained by vortex generators. This differentiation of outboard and inboard properties with respect to damping and power production is favorable due to different weighting along the blade of the local properties, when the integrated power and modal damping is determined. The weighting of the local damping is the actual mode shape squared. This means for instance that considerable negative damping, associated with high lift on the inner part of the blade, will contribute only marginally to the integrated damping for the blade

Usually, a high camber is desirable in order to achieve a high lift coefficient at zero angle of attack, which makes it possible to obtain a more favorable vibration direction along the blade, thus reducing the negative damping in edgewise direction. Further, airfoil data with smooth stall characteristics are preferable in order to avoid local peaks in the damping characteristics as function of wind speed.

Structural characteristics of the blade

Within this area the most important design parameters are:

- the local structural pitch, controlling the vibration direction
- the modal mass and stiffness of the blade
- the structural damping

The local vibration direction or the structural pitch is a very important parameter for the damping. Due to structural design limits this parameter cannot be chosen arbitrarily, but it is important to aim at pitch values, which are favourable with respect to damping. Usually the structural pitch should be chosen as negative as possible. But as a change of vibration direction mainly redistributes the damping between edgewise and flapwise mode shapes, a proper damping of both the edgewise and flapwise vibrations must be considered. With respect to mass and stiffness, the blade structure should be designed for the highest possible edgewise stiffness even at the expense of added weight. Blades with relatively large blade chords at the inner and middle part of the blade will have improved edgewise stability compared to more slender blades as a result of increased structural stiffness and damping in the edgewise direction. Finally, the design of the blade and the whole turbine structure should aim at achieving as high structural damping as possible.

Properties of the supporting structure

If any of the rotor yaw and/or tilt frequencies gets close to the frequency range of the edgewise natural frequency +- 1P, careful analysis of the dynamics of the complete turbine is needed in order to predict the damping properties of the edgewise blade vibrations, as significant coupling might take place. The investigations in the present work of this coupling are specific to the typical European 3-bladed wind turbine concept. The conclusion from these investigations are that the stiffness of the rotor support in the tilt and yaw direction (resulting from bending stiffness of the shaft, nacelle frame stiffness, tower top torsion and bending stiffness) should be chosen as high as possible.

4 Exploitation plans and anticipated benefits

The outcome of the project is: 1) improved aeroelastic models with updated stall hysteresis models and; 2) a set of design guidelines for achievement of safety margins against stall induced vibrations. The results are intended to be used by the wind turbine industry and different ways of transferring information from the project to the industry is possible (see also the implementation plan for the project):

1) The design guidelines are contained in one of the publishable reports from the project [2].
2) Most of the participants in the project have direct, close corporation with the industry.
3) Some of the participants can offer consultant assistance to the industry, e.g. in the form of aeroelastic simulations, on a payment basis.
4) Key results from the project are published in a concentrated format as so-called “fact sheets” [7].

Considerable benefits can be expected if the information gathered through the project is incorporated in new designs as well as used for limiting the problem on existing blades.

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