Effect of linear and non-linear blade modelling techniques on simulated fatigue and extreme loads using Bladed

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Abstract. There is a trend in the wind industry towards ever larger and more flexible turbine blades. Blade tip deflections in modern blades now commonly exceed 10% of blade length. Historically, the dynamic response of wind turbine blades has been analysed using linear models of blade deflection which include the assumption of small deflections. For modern flexible blades, this assumption is becoming less valid. In order to continue to simulate dynamic turbine performance accurately, routine use of non-linear models of blade deflection may be required. This can be achieved by representing the blade as a connected series of individual flexible linear bodies – referred to in this paper as the multi-part approach. In this paper, Bladed is used to compare load predictions using single-part and multi-part blade models for several turbines. The study examines the impact on fatigue and extreme loads and blade deflection through reduced sets of load calculations based on IEC 61400-1 ed. 3. Damage equivalent load changes of up to 16% and extreme load changes of up to 29% are observed at some turbine load locations. It is found that there is no general pattern in the loading differences observed between single-part and multi-part blade models. Rather, changes in fatigue and extreme loads with a multi-part blade model depend on the characteristics of the individual turbine and blade. Key underlying causes of damage equivalent load change are identified as differences in edgewise-torsional coupling between the multi-part and single-part models, and increased edgewise rotor mode damping in the multi-part model. Similarly, a causal link is identified between torsional blade dynamics and changes in ultimate load results.

1. Introduction

It has been demonstrated [1], [2] that linear and non-linear blade models can show significantly different blade response, particularly for blade torsional deflection. Differences in bending-torsion coupling and edgewise damping between linear and non-linear models have also previously been identified [2],[3],[4],[5]. These differences can be expected to lead to significant variations in simulated turbine loads. This paper explores the implications of these modelling differences for extreme and fatigue loads for a range of representative modern turbine designs.

The paper by Manolas et al. [2] looks at the NREL 5MW turbine [6] and compares a linear blade model with two different non-linear blade formulations. Differences in bending-torsion coupling and a frequency shift of the 2nd edgewise modes are noted. The aeroelastic code used was the NTUA code, GAST. Larsen et al. [3] point to differences in edgewise-torsional coupling between linear and non-linear blade representations. Clear torsional differences in static calculations are noted and the change to rotor area is identified as a major factor affecting loads and power predictions between linear and non-linear blade models. The code used was HAWC. Kallesoe [4] describes variation in edgewise mode damping across all operational wind speeds, and notes both lower first edgewise and higher second edgewise blade mode damping around rated wind speed in a non-linear model of the NREL 5MW turbine. HAWC2 was used [7].

This paper builds on the work referenced above in the following areas: sophistication of turbine models, number of turbine models studied, and number and variety of simulations performed. A full range of fatigue and extreme load cases are considered, and turbine models nos. 1 to 3 are representative of the current state-of-the-art in terms of structural definitions and controller sophistication.

2. Method

2.1. Outline of multi-part (non-linear) blade modelling

A non-linear blade deflection model can be defined in Bladed by splitting the blade into several linear bodies, as illustrated in Figure 1 for a two-part blade. Each blade section can undergo rotations with respect to neighbouring sections enabling the deflections within each modal body to remain small. An accurate model of non-linear deflection is thereby achieved provided sufficient blade parts are defined.

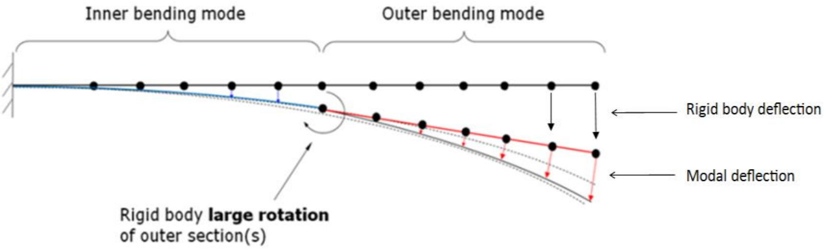

Figure 1. Non-linear 'multi-part' blade modelling approach

2.2. Preparation of turbine models and load simulations

Bladed version 4.7 has been used for this work [8]. Bladed is a well known aeroelastic code that employs the Craig-Bampton modal reduction method [9] for each blade part and for the tower. Four Bladed turbine models were studied, covering commercial turbine sizes from 3-6MW. Two versions of each model were prepared: one with a conventional single-part (linear) blade definition, and a second with a multi-part (non-linear) blade definition. The multi-part blade models used five linear blade bodies per blade. Five blade parts was found to be sufficient to give convergence in blade loads and deflection for the blades in this study. The single-part models included a non-linear correction term to account for centrifugal stiffening, via a model of geometric stiffness that captures the effect of axial blade element loads on structural response. The multi-part model included a further geometric stiffness correction to include the effect of shear loads, to more accurately capture the blade torsional response. For a more complete discussion of geometric stiffness, see [10].

A brief description of the four turbines studied is as follows

- Turbine 1: 6MW turbine with particularly flexible blade (≈ 75m blade)
- Turbine 2: 6MW turbine with typical blade stiffness characteristics (≈ 80m blade)
- Turbine 3: 3.5MW turbine with typical blade stiffness characteristics (≈ 65m blade)
- Turbine 4: NREL 5MW turbine [6]. Well known publically available research model (≈ 61.5m blade)

For each model, an identical set of fatigue and extreme load cases were run. The load cases were a reduced set of IEC 61400-1 Ed 3 load simulations (109 cases in total). All load case groups are represented (see Table 1 below). For most load case groups, simulations were run at rated and cut out wind speeds only. A single wind speed seed was included for each wind speed bin within the dlc1.2 power production cases. A different seed was used for each wind speed.

Table 1. Description of test load cases

Load case	Description	Turbulent / Steady wind	Fatigue / Extreme	No. of simulations
Dlc1.2	Power production	T	F	11
Dlc2.4	Fault: extreme yaw operation	T	F	4
Dlc4.1	Normal Stop	S	F	1
Dlc6.4	Idling	T	F	2
Dlc1.3	Power production – extreme turbulence	T	E	2
Dlc1.3 c=2.3	50-year extreme simulation (extrapolation)	T	E	2
Dlc1.4	Gust and direction change	S	E	18

Dlc1.5	Extreme wind shear	S	E	8
Dlc2.1	Faults: n4 overspeed, individual pitch runaways	T	E	11
Dlc2.2	Faults: nA overspeed, collective pitch runaway towards fine, single blade pitch seizure	T	E	10
Dlc2.3	Faults. Grid loss and gust	S	E	8
Dlc4.2	Normal stop and gust	S	E	4
Dlc5.1	Emergency Stop	T	E	2
Dlc6.2	Idling at V50 wind speed with yaw failure	T	E	12
Dlc6.3	Idling at V1 with extreme yaw misalignment	T	E	1
Dlc7.1	Idling at V1 with pitch failure in one blade. Parked with brake applied	T	E	3
Dlc8.1	Maintenance cases. Yaw system locked & Pitch system maintenance at 90deg yaw	T	E	4
Dlc8.2	Quasistatic maintenance cases	-	E	5

All variants of each turbine model were run with a controller tuned for the baseline single-part blade modelling configuration. A high robustness controller was used in order to reduce the sensitivity of results to changes in the blade structural dynamic model. Ultimate load and damage equivalent load (DEL) results were generated in accordance with standard IEC Ed3 guidelines.

2.3. Initial model characterisation

Figure 2 shows steady flapwise and mean torsional blade deflections at the rated wind speed for each turbine, plotted against the normalised blade radius. These are steady state calculations in which upflow, wind shear and tower shadow are ignored.

For each of the turbines tested, the multi-part and single-part versions showed virtually identical steady state flapwise blade deflection. Therefore only the multi-part deflections are shown in the figure. Mean torsional deflection showed clear differences between multi-part and single-part for turbines 1 & 2. Turbines 3 & 4 showed a reasonably good match between multi-part and single-part. The match between multi-part and single-part for mean steady state torsional deflection was not significantly improved by including additional modes to the single-part model.

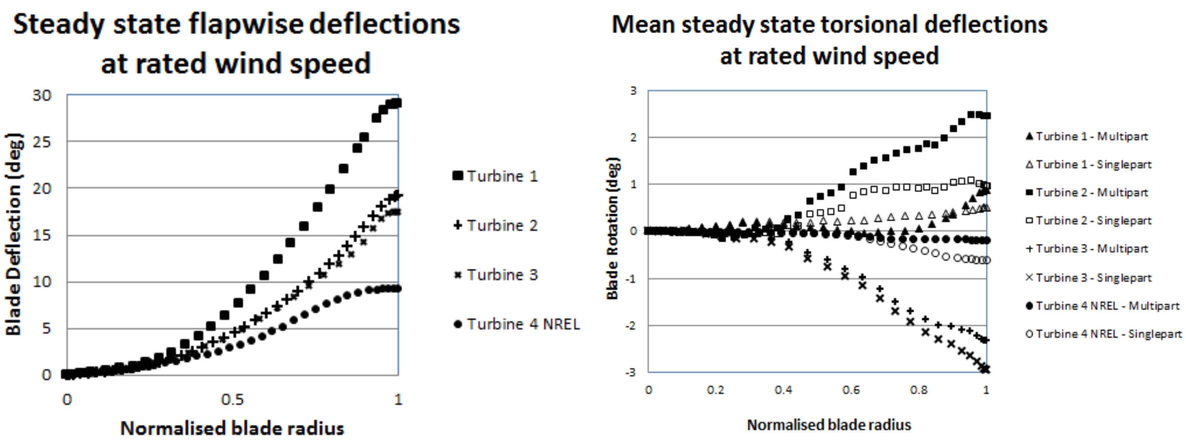

Figure 2. Steady operational loads flapwise and torsional blade deflections

Figure 3 illustrates torsional blade tip rotation in steady wind (no turbulence) at rated wind speed for turbines 1 to 4. Wind shear, rotor tilt, upflow and gravity loads were all enabled.

In all four turbines, the amplitude and mean value of the periodic torsional response differs between single-part and multi-part. Turbines 1 and 3 also show a phase difference. This phase difference is not due to a difference in azimuth angle. For turbine 3 the single-part response shows clear higher frequency content at around 5P, corresponding to the first edgewise rotor mode.

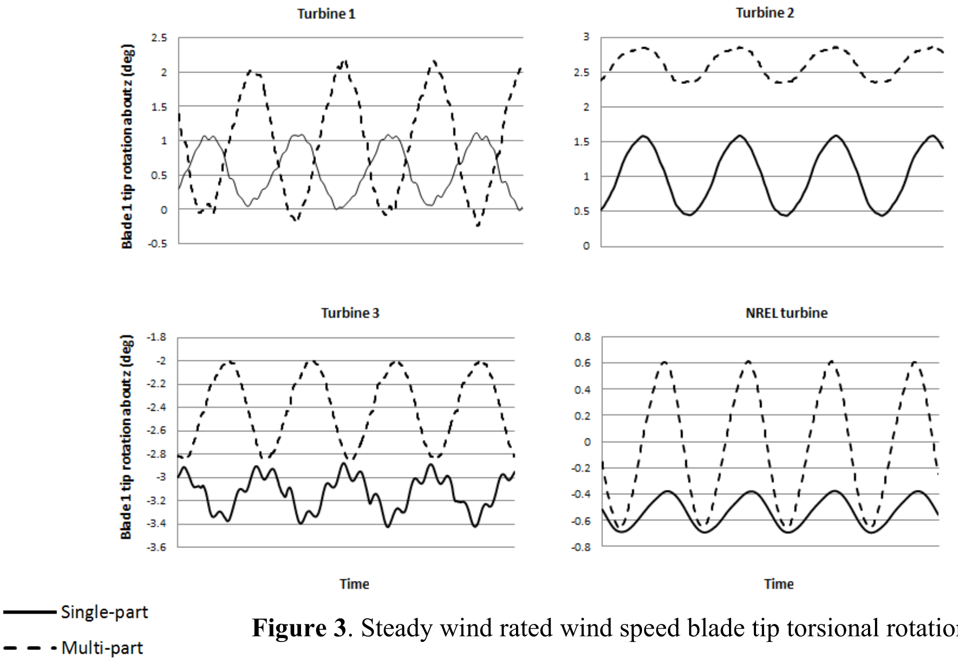

Figure 3. Steady wind rated wind speed blade tip torsional rotation

3. Results

Fatigue and extreme loads are presented in sections 3.1. and 3.3. respectively. A case study on a generator short-circuit load case is presented in section 3.4.

3.1. Damage equivalent loads (DELs)

Table 2 presents DELs for the multi-part blade simulations as percentage differences from the baseline linear blade load results. Changes in damage equivalent load of up to 16.7% are observed.

Turbine 1 shows the largest percentage changes, which was expected as this turbine has the most flexible blade. Turbine 2 generally shows slightly lower load changes except at the tower base. Turbine 3 shows smaller load differences, as would be expected for a shorter stiffer blade. Turbine 4, the NREL 5 MW, shows the smallest loads differences. No general trend in load increase or decrease is established.

Table 2. Percentage change in DELs between single-part and multi-part blade simulations

DEL results (Blade root = SN 10, all other components = SN 4)	% change of multi-part results from linear results			
Load Component	**Turbine 1**	**Turbine 2**	**Turbine 3**	**NREL 5MW**
Blade root edgewise moment Mx	-3.9	+0.0	**-6.0**	+0.4
Blade root flapwise moment My	-1.8	+4.7	+0.4	-0.9
Blade root pitching moment Mz	**-16.7**	+3.3	-1.5	-0.3
Hub Mx (shaft torque)	-1.2	+0.3	-0.5	+3.0
Hub (rotating frame) My	**-12.9**	-1.6	-2.0	-1.2
Hub (rotating frame) Mz	**-13.8**	-1.2	-2.2	-1.3
Hub (fixed frame) My	**-6.4**	**+5.8**	-0.3	+2.4
Hub (fixed frame) Mz	**-6.0**	**+6.5**	+0.3	+2.2
Yaw Bearing Mx	-2.4	+1.2	-1.8	+2.3
Yaw Bearing My	**-6.5**	**+5.8**	-0.6	+2.3
Yaw Bearing Mz	**-5.0**	**+6.4**	-0.5	+2.8
Tower base side-side moment Mx	+0.8	+4.2	+4.2	+3.3
Tower base fore-aft moment My	-0.1	+3.5	+2.0	+3.7
Tower base Yawing moment Mz	-4.9	**+5.9**	-0.4	+2.8

3.2. DELs analysis

This section contains selected brief commentary on the DELs load changes observed for turbines 1-4.

Turbine 1

Figure 4 shows DELs contribution histograms for the load components with the largest differences in DELs between single-part and multi-part for turbine 1. The load cases causing higher DELs for the single-part model are power production simulations at around rated wind speed (dlc1.2ca1, da1, ea1).

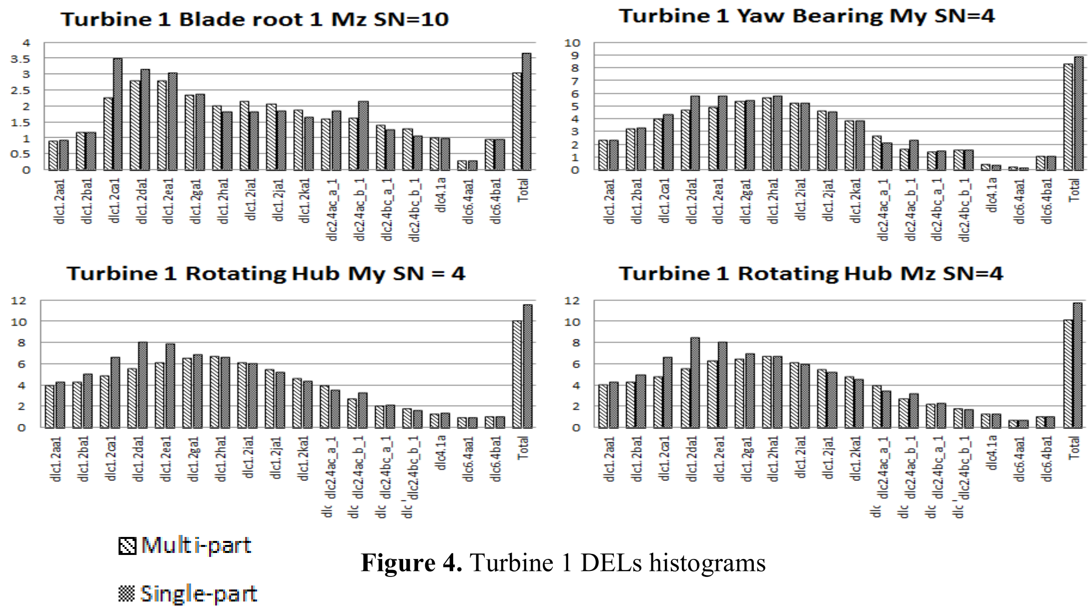

Figure 4. Turbine 1 DELs histograms

Figure 5 is an auto spectral density plot of blade root Mz for load case dlc1.2ca1. This is a turbulent wind power production simulation with a mean wind speed of 8m/s. The solid line and heavy dashed line are the single-part and multi-part runs used in the study.

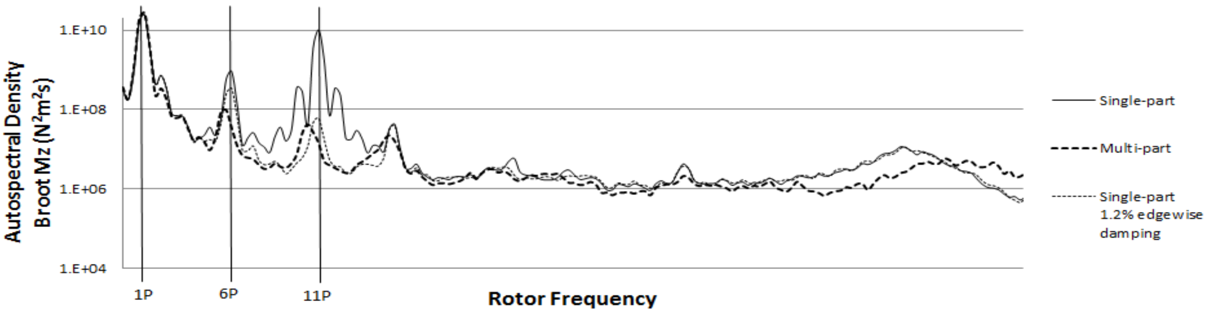

Figure 5. Autospectrum of blade root Mz - turbine 1

The single-part auto spectral plot shows much higher energy content compared to multi-part at 11P, corresponding to the first edgewise collective rotor mode. For the single-part model, analysis using the Bladed Campbell diagram calculation showed that the first edgewise collective rotor mode has a damping ratio of 0.0009 at rated wind speed. For the multi-part model, the damping ratio is 0.011 – a factor of 10 higher, corresponding to an additional 1% damping for the multi-part blade. Note that both models originally had the same structural damping of 0.2% as input for the 1st edgewise mode. The thin dashed line on Figure 5 shows the effect of increasing the uncoupled first edgewise blade mode structural damping on the single-part model from 0.2% to 1.2%. Increasing the damping in the single-part blade model brings the auto spectral plot closer to that from the multi-part blade model. This suggests that additional edgewise damping generated by changing from a single-part to multi-part blade model is responsible for the difference between the two models. For this particular blade, a similar dynamic response to the multi-part blade model can be achieved simply by adding ~1% damping to the first edgewise mode in the single-part blade model.

Figure 6 shows time domain plots of blade tip deflection and blade root Mz for the same three dlc1.2ca1 simulations. The 11P frequency component is clearly visible as the high frequency variation in the original single-part plots. This frequency component does not appear strongly in the multi-part plot or in the single-part plot in which blade edgewise structural damping had been increased.

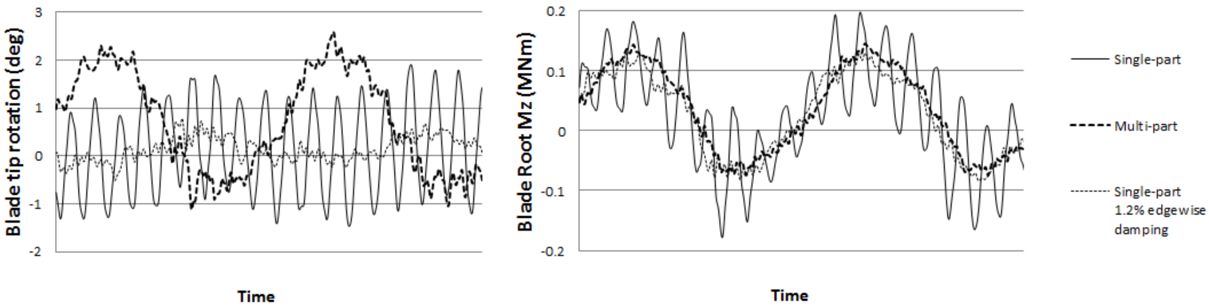

Figure 6. Turbine 1, Blade tip rotation and blade root Mz, dlc1.2ca1

Figure 5 and Figure 6 illustrate that damping on the edgewise blade motion is the cause of the different response between the single-part and multi-part blade models. The additional damping captured by multi-part (non-linear) blade modelling is likely to correspond to aerodynamic damping associated with flapwise motion of the blades, which is more strongly coupled to blade edgewise motion for the multi-part blade. The higher damping of the first edgewise rotor mode in multi-part has a direct impact on the reduction of blade root Mz DELs for turbine 1.

Turbine 2

For turbine 2, DELs are slightly higher for the multi-part model than for single-part for most load components. However, the magnitude of DEL changes is smaller than those observed for turbine 1, which has a more flexible blade of similar length. Figure 7 shows DELs contributions for the load components with the largest DELs differences.

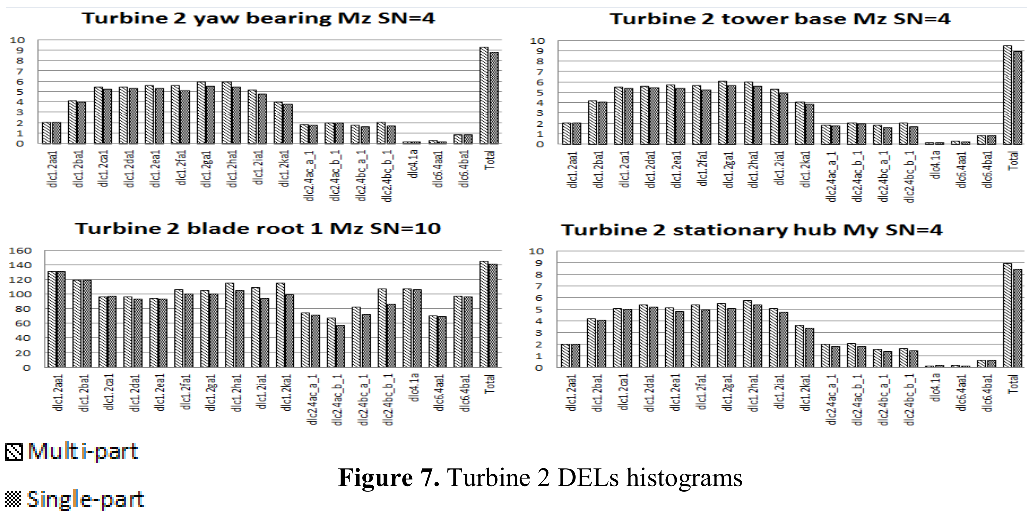

Figure 7. Turbine 2 DELs histograms

Increased DELs with multi-part are seen across the majority of fatigue cases, but in particular for above rated power production runs, and for dlc2.4 in the case of blade root Mz loads.

It was found that the blade set angle imbalance, which introduces a pitch error of +-0.3deg for blades 2 and 3, made a significant contribution to the DELs difference between single-part and multi-part simulations. The set angle imbalance produced up to 50% larger 1P load responses in multi-part than in single-part. By removing this imbalance, DELs differences for yaw bearing & tower base Mx & Mz were reduced by a third, differences for blade root & fixed frame hub Mz were halved, and rotating hub My & Mz loads differences were removed almost entirely. Without set angle imbalance, DELs differences for all components were typically less than 5%.

Turbine 3

For turbine 3, the difference in DELs between single-part and multi-part is generally smaller than for turbines 1 and 2. This is expected as the blade is shorter and stiffer. However Figure 3 shows clear higher frequency content at around 5P in the torsional blade tip rotation for single-part that is not present with the multi-part model. Campbell diagram analysis shows that this frequency corresponds to the first edgewise collective rotor mode frequency, with a damping of 0.0087 and 0.015 at rated rotor speed for the single-part and multi-part models respectively. The higher damping of the edgewise rotor mode in multi-part accounts for the 6% reduction in blade root Mx with the multi-part model.

The difference in blade root Mx DELs with turbine 3 is a further example of a difference in edgewise-torsional coupling between single-part and multi-part. However unlike turbine 1, the increased damping of the edgewise mode in multi-part for turbine 3 does not lead to significant DELs reductions for blade root Mz or other load components. This is perhaps because the damping in the single-part blade edgewise collective mode is higher in turbine 3 than the same mode in turbine 1.

Turbine 4 – NREL 5MW

For the NREL turbine, blade root DELs differences were < 1%. Slightly higher differences were seen for other components, but none were larger than 3.7%. Given the general complexity of the modelling, and the impact of controller dynamics, loads differences of this magnitude can be considered as minor.

3.3. Ultimate loads

Table 3 presents ultimate loads for turbines 1 to 3. The multi-part blade loads are given as percentage differences from the linear blade load results. Ultimate load changes of up to 29% are observed.

Table 3. Percentage change in ultimate load predictions between single-part and multi-part simulations

Ultimate load results	Driving load case group		**Turbine 1**	Driving load case group		**Turbine 2**	Driving load case group		**Turbine 3**
Load Component	single-part	multi-part	% change	single-part	multi-part	% change	single-part	multi-part	% change
Blade root edgewise moment Mx	dlc8.2	dlc1.3	**+6.9**	dlc8.2	dlc8.2	-0.2	dlc1.3	dlc1.3	-4.4
Blade root flapwise moment My	dlc1.3	dlc1.4	**+6.5**	dlc1.4	dlc1.4	**+13.9**	dlc2.2	dlc2.2	-0.6
Blade root pitching moment Mz	dlc1.3	dlc1.4	**-11.5**	dlc8.2	dlc8.2	+0.6	dlc2.2	dlc2.2	**+8.0**
Hub Mx (shaft torque)	dlc1.3	dlc1.3	+0.1	dlc1.3	dlc2.3	+0.3	dlc1.3	dlc1.3	-0.2
Hub (rotating frame) My	dlc1.4	dlc1.4	-4.3	dlc1.4	dlc1.4	**+28.9**	dlc2.2	dlc2.2	+3.1
Hub (rotating frame) Mz	dlc1.4	dlc1.4	**-12.9**	dlc1.4	dlc1.4	**+26.9**	dlc1.3	dlc1.3	+0.8
Hub (fixed frame) My	dlc1.4	dlc1.4	-6.1	dlc1.4	dlc1.4	**+26.1**	dlc1.3	dlc1.3	-2.2
Hub (fixed frame) Mz	dlc1.3	dlc1.3	**-10.6**	dlc1.3	dlc1.4	**+14.4**	dlc2.2	dlc2.2	-0.5
Yaw Bearing Mx	dlc1.3	dlc1.3	-2.0	dlc1.3	dlc1.4	+3.2	dlc1.3	dlc1.3	-0.4
Yaw Bearing My	dlc1.4	dlc1.4	-2.5	dlc1.4	dlc1.4	**+14.4**	dlc1.3	dlc2.2	-2.0
Yaw Bearing Mz	dlc1.4	dlc1.3	**-9.8**	dlc1.3	dlc1.4	**+7.2**	dlc2.2	dlc2.2	-1.5
Tower base side-side moment Mx	dlc6.2	dlc6.2	-0.1	dlc6.2	dlc6.2	-0.0	dlc6.2	dlc6.2	+4.4
Tower base fore-aft moment My	dlc6.2	dlc6.2	+5.2	dlc2.1	dlc2.1	**+11.0**	dlc4.2	dlc2.2	+4.0
Tower base Yawing moment Mz	dlc1.4	dlc1.3	**-9.3**	dlc1.3	dlc1.4	**+7.0**	dlc2.2	dlc2.2	-1.6

Figure 8 compares the turbine 2 driving load case for blade root flapwise bending moment (My). This is a dlc1.4 simulation (steady wind) which includes an extreme coherent gust with direction change. The differing blade torsional deflection response to the wind gust is thought to be responsible for the difference in the blade root flapwise load between the two simulations.

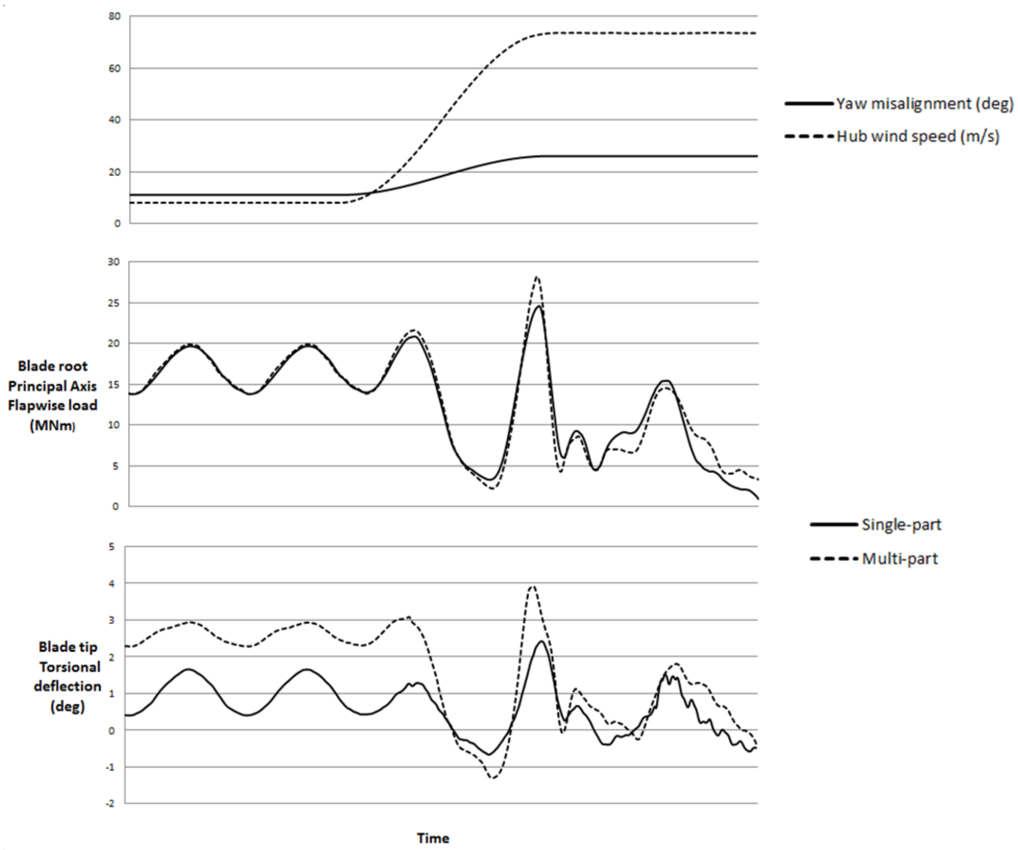

Figure 8. Turbine 2 blade root My driving load case

The causal link between blade torsion and the change in flapwise load is confirmed by comparing single-part and multi-part blade results with the blade torsion disabled, as shown in Figure 9 below.

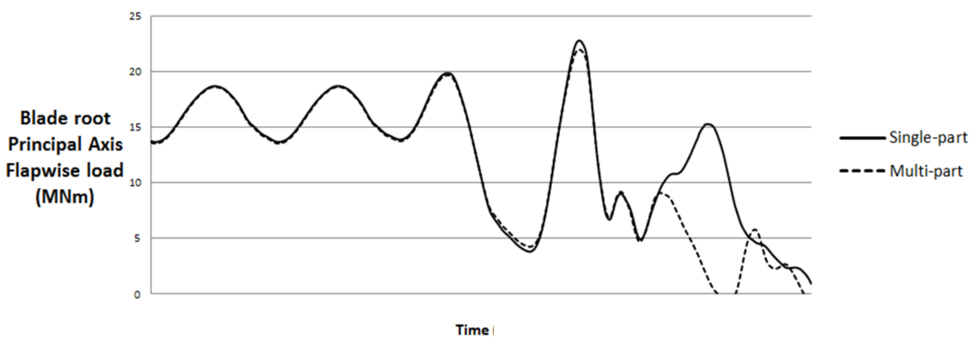

Figure 9. Turbine 2 blade root My driving load case with no torsional degree of freedom

3.4. Generator Short Circuit simulations

Generator short circuit events are characterised by very large, rapidly fluctuating torque loads acting directly on the drivetrain. Short circuit events can often lead to driving extreme loads for hub Mx and for blade Mxy towards the blade tip. The response of multi-part and single-part models to such load cases was compared. The turbine model was a variant of turbine 1.

Figure 10 shows the applied generator torque time history, and the blade Mx load response at a radius = 80% of the blade length. The total time period shown is around 2 seconds. Loads are shown for three structural blade models: a 5 part multi-part model, a conventional single part model with 9 blade modes, and a further single part model with 36 blade modes (to match the number of blade modes captured with the multi-part model).

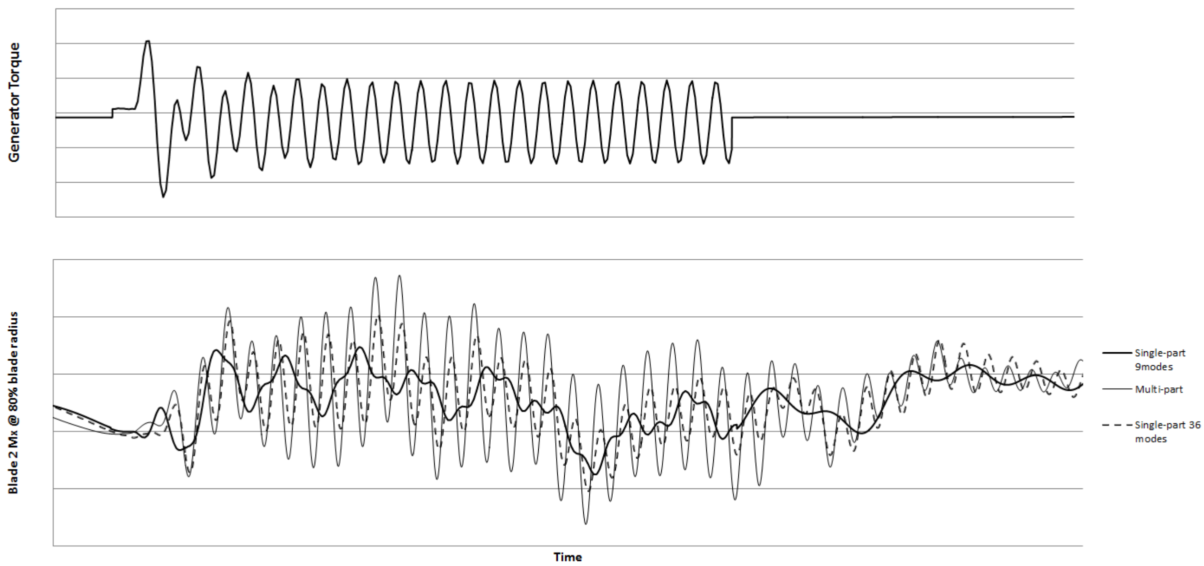

Figure 10. Turbine 1, generator short circuit simulation. Blade Mx at 80% radius

For this type of simulation, it is found that multi-part modelling predicts considerably larger blade loads than conventional single-part modelling. The multi-part blade shows significant vibrational response at the frequency of the applied generator torque, but the single-part blade with 9 blade modes does not. The dynamic response of the single-part blade can be improved by including additional vibration modes, as illustrated by the dashed line in Figure 10.

However loads differences between multi-part and single-part remain even when the total number of modes is increased on the single-part model. These remaining loads differences can be attributed directly to the difference in the structural formulation of the two blade models.

These observations suggest that for generator short circuit cases, and for other simulations involving high amplitude, high frequency external conditions, single-part simulations with typical number of vibration modes used for load analysis are insufficient to adequately capture the blade responses. Such simulations should either be run with a multi-part blade, or failing that, with single-part that includes considerably higher numbers of blade modes than are conventionally used.

4. Conclusions

The choice between linear and non-linear models of blade deflection in Bladed has been shown to have a significant impact on both fatigue and extreme loads. Generally, for more flexible blades larger load changes are observed. However, exact loads differences are highly blade specific, and cannot be easily predicted by examination of the blade definition.

For the studied turbines, damage equivalent load changes of up to 16.7% were observed. For two of the turbine models tested, it was found that increased edgewise rotor mode damping with a multi-part blade played a significant role in differences in DELs between multi-part and single-part blade models. The torsional dynamic response of the single-part blade model could be made similar to that of the multi-part blade model by adding damping to the blade first edgewise mode.

Ultimate load changes of up to 29% were observed. The differing blade torsional response is thought to be a key driver of the load changes. It was shown that significant changes in extreme loads between single-part and multi-part blade models can be caused by the differing torsional deflection between the two models.

The study also showed that for the NREL 5MW turbine, changes in DELs between single-part and multi-part blade modelling were smaller than for the other turbines. The largest difference in DELs for this turbine model was 3.7%. This suggests that the blade model for the NREL 5MW turbine should no longer be considered as representative of the latest generation of commercial blade designs in terms of length, flexibility, and impact on loading predictions of non-linear blade modelling techniques.

For certain types of calculations, for example generator short circuit simulations, the multi-part blade is shown to respond more appropriately to the applied high frequency generator load. Typical single-part blade definitions used in industry with just enough modes to include the first torsional mode do not accurately capture the blade response in such cases.

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