Modelling and Analysis of Acoustic Emissions and Structural Vibration in a Wind Turbine

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Abstract: The onshore wind turbine industry must overcome many technical, commercial, and environmental difficulties. A significant element for planning consent is operational noise. Acoustic limits are strictly enforced and can lead to near-neighbour complaints as well as contractual disputes. Reactec have diagnosed, analyzed and solved a problematic tonal resonance using COMSOL Multiphysics. This paper presents the measurement, analysis, modelling and solution simulation process when tackling a tonal noise issue in a megawatt scale wind turbine. Extensive modelling and simulation of the entire turbine system allows an in-depth diagnosis as well as a virtual test bed for solution design. The analysis reveals a complex interaction between gearbox resonances, mounting system performance, and tower design. A mitigation measure is developed and simulated in order to provide sufficient tonal reduction.

Keywords: wind turbine, tonal noise, eigenfrequency analysis, acoustic-structural interaction.

1. Introduction

Noise produced by wind turbines can be problematic when wind farms are built close to urban environments and can lead to annoyance and sleep disturbance, which may in turn be linked with stress-related health issues [1, 2]. Noise from wind turbines comes in two forms: the first is aerodynamic noise from the blades slicing though the air leading to the characteristic swish-swish noise; the second is mechanical noise associated with machinery housed in the nacelle of the turbine. This mechanical noise tends to be tonal as it is created by rotating machinery (e.g. gearboxes, drive shafts, generators, see Figure 1) which have discrete rotational frequencies [3]. Tonal noises are particularly annoying to neighbouring residents and therefore incurs heavy regulatory penalties throughout Europe and North America [4].

A set of megawatt scale wind turbines were found to be emitting excessive tonal noise in the 570-610 Hz frequency band. The manufacturer identified the likely source to be the 575 Hz frequency at which the gear teeth mesh during the last step-up stage within the gearbox; the *gear-meshing frequency*. All conventional gearboxes in wind turbines have gear-meshing frequencies but do not cause problematic noise indicating that the vibration was being amplified structurally, before becoming air-borne and sensed as noise.

An extensive vibration survey was carried out using a set of accelerometers to identify the vibration pathway between the gearbox, nacelle and tower walls. A structural resonance involving the gearbox and its mounts was found in the 530 to 610 Hz band, as were a suite of resonances in the tower wall. A complex dynamic relationship between the gear teeth striking each other, the gearbox mount and the tower wall led to the amplification of the gear-mesh frequency and the resultant tonal noise.

Here we show how COMSOL Multiphysics was used as an applied engineering tool to identify the modal shapes of each vibrating element using eigenfrequency and frequency response models. A structural-acoustic interaction model was then developed to design and optimise a solution to the tonal noise problem.
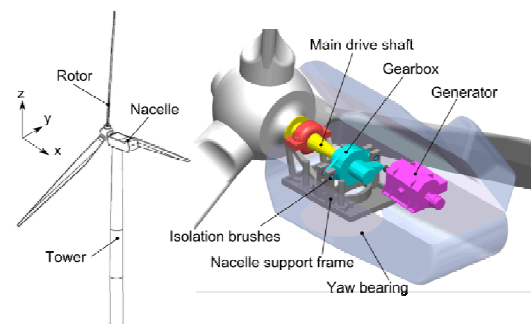
Figure 1. Schematic diagram of a wind turbine showing the main structural and mechanical components.
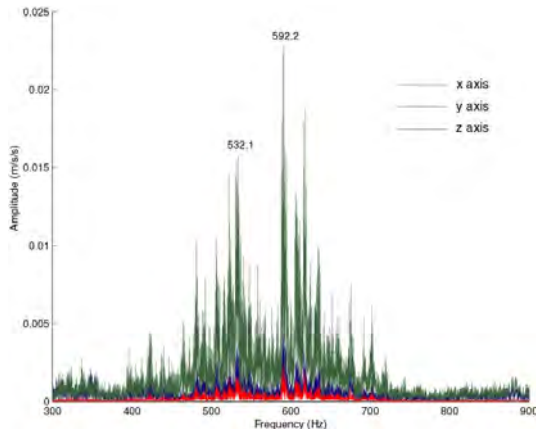
Figure 2. Gearbox vibration following the yaw test indicates a structural resonance at 592.2 Hz.
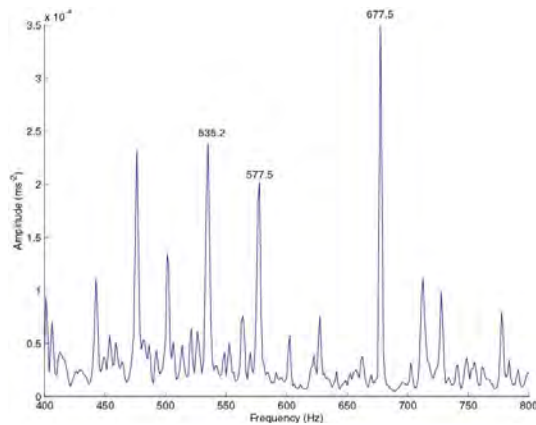
Figure 3. Tower wall vibration following the yaw test indicates a suite of resonances between 535 and 677 Hz.

2. Vibration Survey

The vibration survey was carried out using Brüel & Kjær tri-axial accelerometers which were placed on the gearbox, generator, yaw bearing (bearing between top of the tower and the nacelle) and the tower wall. The data was logged at 3000 Hz and all accelerometers were aligned with the z-axis vertically upward and the x-axis parallel to the main drive shaft (wind direction when the turbine is operational, see Figure 1).

2.1. Identifying structural resonances

Initially a yaw test was carried out to excite all the resonant frequencies in the structure. The yaw test is akin to a bump test where a structure is struck with an acoustic hammer and the response recorded. However, wind turbines are 100's of tons in mass and tens of meters high so an acoustic hammer is insufficient to excite the structure. Instead, the nacelle is yawed (rotated around the vertical axis) by 90° then stopped suddenly and the response recorded. A strong vibration was recorded in the gearbox with a broad band between 500 and 650 Hz with peaks at 532.1 and 592.2 Hz (Figure 2) acting in the y-direction (transverse to the main drive shaft). This is indicative of a structural resonance involving the gearbox and its isolation mounts. A suite of resonances were also identified in the tower wall, with the tower skin vibrating at 535.2, 577.5, 600.9 and 629.1 Hz (Figure 3).

2.2 Identifying forced frequencies

An active test where the turbine is spun up to its normal operational speed was conducted to determine how the *forced frequencies* interact with the structural resonances in the turbine. Here we refer to forced frequencies as those that are the result of some periodic motion, normally mechanical. Forced frequencies include rotation of the rotor, gear-meshing and bearing noise. Waterfall diagrams are used to separate forced frequencies from the structural harmonics of a system (Figure 4). These diagrams show the variation of vibration amplitude with frequency (x-axis) and time (y-axis). The frequency of gear-meshing, the rotation of bearings and the drive train all depend on the rotation speed of the turbine. As the turbine speeds up, any forced frequencies will increase over time and their associated data lines will travel across the waterfall diagram. Structural resonances do not change their frequency and therefore always appear as vertical lines on waterfall diagrams.

Figure 4 shows the waterfall diagram for the gearbox in the y-axis, with time increasing down the diagram. Initially the diagram shows the vibrations due to the nacelle yawing (1200 – 1500 seconds). The peak amplitudes ~ 600 Hz are the structural resonances of the gearbox discussed above. The turbine starts at 1508 seconds and several peaks are shown moving to the right (higher frequency) for ~ 30 seconds: these peaks are forced frequencies. Once the turbine is at full speed these forced frequencies settle around ~ 600 Hz and are thus magnified by

the structural resonance of the system. This magnification leads to two very strong vibrations at 576.5 Hz and 678.5 Hz.

3. Modal Shapes of Structural Resonance

High order structural resonances tend to have complex modal shapes that have wavelengths orders of magnitude less than the scale of the entire dynamic structure. While it is possible to interpret the first one or two bending modes and their modal shapes based on data from accelerometers, it becomes increasingly difficult with increasing orders as more accelerometers are required. Furthermore, the higher order modes have higher frequencies therefore making modal analysis based on phase relationships impossible as the timing errors in data logging propagate through the calculations. Structural mechanics models were made in COMSOL Multiphysics to identify the modal shapes. The models were based on geometries and material properties supplied by the turbine manufacturer and were calibrated using experimental data.

3.1. Gearbox resonance

The gearbox and its mounting brushes (Figure 1) were modelled using solid elements. The mass of the gearbox is 11,000 kg and it has a moment of inertia about an axis co-linear with the main drive shaft of 8.1 kg m^2 (Figure 5a). The rubber brushes that isolate the gearbox have a spring constant of 160 kN/mm, which given their geometry equates to a Young's Modulus of 43 MPa. The outer boundaries of the rubber brushes where given fixed boundary conditions. The main drive shaft connects to the front of the gearbox prohibiting vertical and horizontal movement but allows rotation, so roller boundary conditions were applied to the front of the gearbox (Figure 5a). An eigenfrequency analysis found a resonance at 602.3 Hz that involves rotation about an axis co-axial with the main drive shaft (Figure 5b). Subsequent frequency response analysis showed that the resonance can be excited by a broad range of frequencies from 520 to 640 Hz. The modal shape and broad frequency range that excites the resonance is consistent with the data recorded
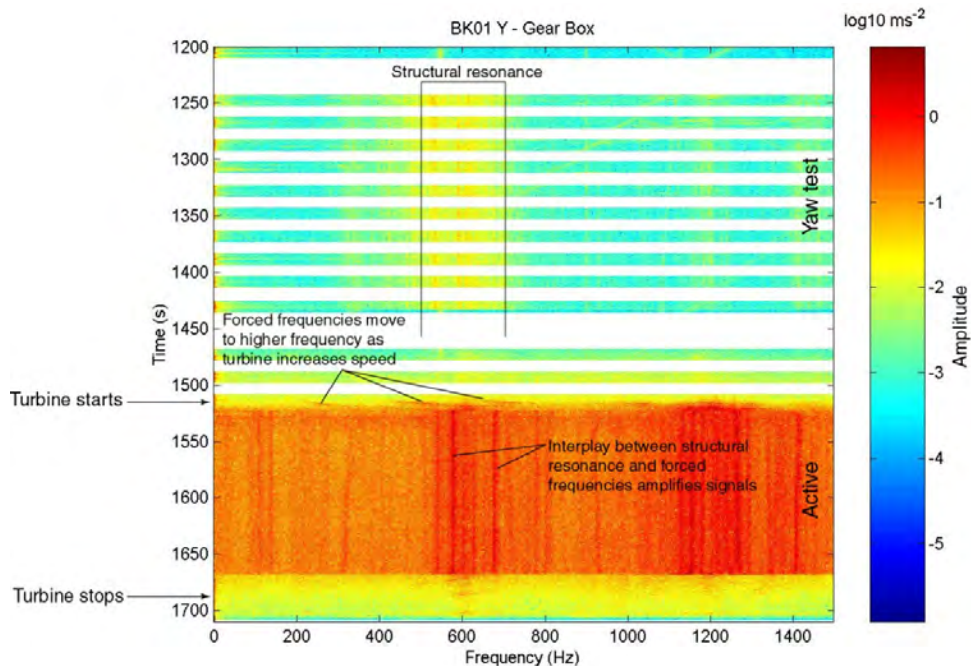
Figure 4. Waterfall diagram for the gearbox accelerometer's y-axis. Time increases downwards. The bands from 1200 seconds to 1500 seconds are individual yaw tests. The turbine starts at 1508 seconds and several red/orange lines can be seen moving diagonally indicating they are forced frequencies. These are amplified by the structural resonances between 530 and 630 Hz.
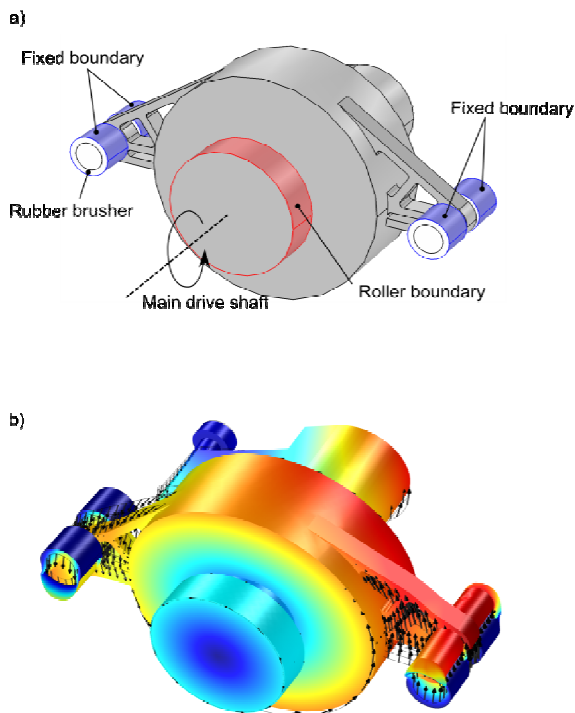
Figure 5. a) Geometry and boundary conditions used in eigenfrequency study. b) The modal shape of the 602 Hz resonance where the colour scale shows unitless displacement with red being the maximum.
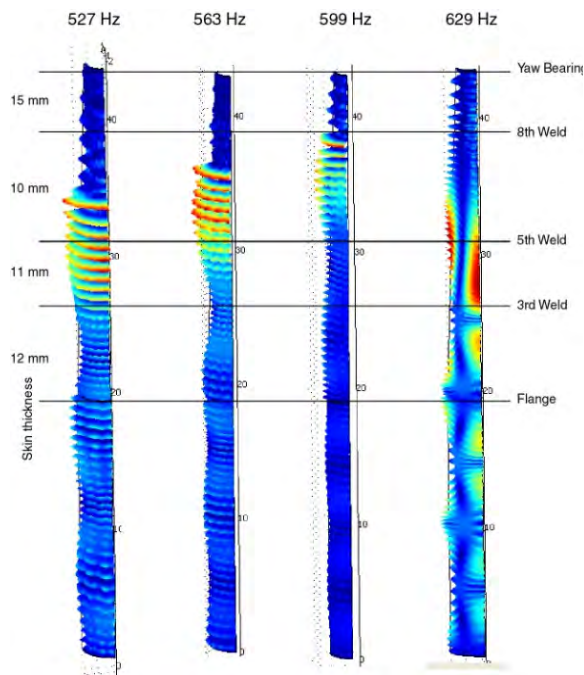
Figure 6. Tower modal shapes for the most significant eigenfrequencies between 500 and 700 Hz.

during yaw tests (Figure 2). This resonance involves displacement across the rubber brushes. The involvement of the rubber brushes means that their isolating properties are ineffectual in this frequency range and gear-meshing at 575 Hz can pass through them and into nacelle's support frame (Figure 1), and from there into the tower.

3.2 Tower skin resonance

Sheet metal has what are referred to as *skin frequencies*. These are the high order bending modes that are excited when a piece of metal is stuck with a hammer and are responsible for the ringing note that is heard. The skin frequencies can be high order resonances with 10s or 100s of different frequencies and modal shapes. Often important skin frequencies which produce the greatest overall displacement cluster around one frequency and combine to produce a distinctive note.

The turbine studied here is supported by a tubular steel tower that was a sheet thickness of 21 mm at the base and 10 mm close to the top of the tower. A three-dimensional eigenfrequency analysis of the turbine tower was performed using shell elements in COMSOL Multiphysics. There are two planes of symmetry about the vertical axis of the tower, so the tower was quartered and symmetrical boundary conditions used on appropriate edges. The lower edge was assigned a fixed boundary condition. The effective modal mass was determined for each of the 781 eigenfrequency solutions returned between 500-700Hz to determine those with the highest surface participation. Four key skin frequencies were found with high participation rates: 527 Hz, 563 Hz, 599 Hz and 629 Hz (Figure 6). These skin frequencies excite 10's of square meter of surface area close to the top of the tower, thus providing a very effective mechanism to amplify the gear-meshing vibration and transmit it to the surrounding air as noise.

4. Acoustic-structural Interaction Model

To better understand the transmission of vibration energy from the tower to the surrounding environment as air-borne noise, an acoustic-structural interaction model of the tower walls and air both inside and outside of the tower was developed. The model combined two
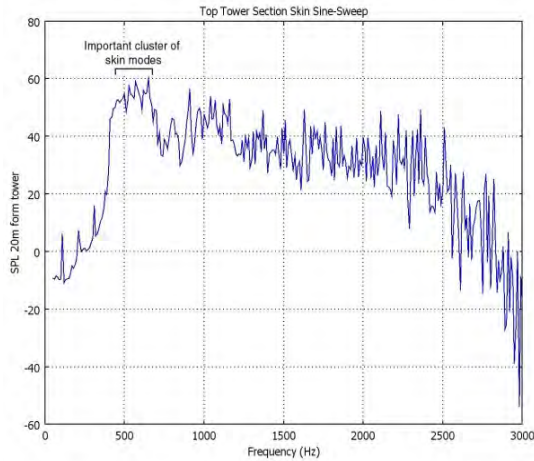
Figure 7. Sound pressure level 20 m from the base of the tower that is stimulated with a constant amplitude sine swept vibration.
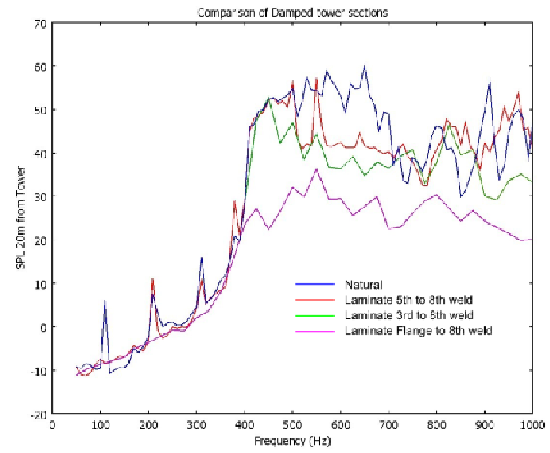
Figure 8. Sound pressure level 20 m from the base of the tower for different amounts of coverage with a damping laminate. The natural model represents an unmodified turbine.

Table 1: Summary of boundary conditions used in acoustic-structural interaction model.

	Air inside tower	Tower wall	Air outside tower
Model suffix	aco_in	smaxi	aco_out
	Normal acceleration	Load	Normal acceleration
Inner wall	nr_aco_in * uaxi_tt_smaxi + nz_aco_in * w_tt_smaxi	-p_in * nu_smaxi	Not applicable
Outer wall	Not applicable	-p_out * nu_smaxi	nr_aco_out * uaxi_tt_smaxi + nz_aco_out * w_tt_smaxi

acoustic time-harmonic analyses representing internal and external air, which are separated geometrically by an axial-symmetric structural-mechanics harmonic response analysis that represents the skin of the tower wall. The model was axially symmetric about the vertical axis of the wind turbine tower. The boundary condition at the interface between each model is summarised in Table 1. The lower boundary condition for the acoustic model inside the tower was a sound hard wall (the base is a concrete slab) and for the acoustic model outside the tower it was a sound soft condition. The acoustic model representing air inside the tower was excited at the upper boundary (bounding the nacelle) with a constant non-zero pressure source and the energy allowed to propagate through all three models. Figure 7 shows the

resultant sound pressure level 20 m from the base of the tower between 1 and 3000 Hz with peaks between 500 and 700 in good agreement with the vibration survey at site and the eigenfrequency analysis of the tower skin.

5. Solution Design

The solution design was focused on breaking the vibration pathway between the source in the gearbox and the tower wall where energy becomes air-borne noise. The simplest method would be to modify the elastic properties of the rubber brushes used for gearbox isolation such that they no longer participate in any resonances close to 575 Hz and thereby cease to allow passage of the gear-mesh vibration. However, due to other engineering constraints, such as the

required stroke length, it proved to be impractical to either sufficiently stiffen or soften the brushes. Instead, the resonance in the tower skin was modified.

The tower skin was a very high Q-factor (~ 200, equivalent to a loss factor of 0.0025) which is why it rings so effectively. To reduce the amplification effect of the tower skin in the 530 to 670 Hz range, a thin composite laminate material with high loss factor damping (0.09) was selected to be adhered to the turbine walls. This material is expensive and height accredited rope access engineers are required to fix it in place. There was, therefore, a significant emphasis on optimising the solution to reduce costs. The eigenfrequency models of the tower skin were used to identify hotspots where significant amplification occurs. The important hotspots all fall between the 3rd and 8th weld close to the top of the tower, but not at the very top (Figure 6).

The acoustic-structural interaction model was modified to examine the effects of covering different parts of the tower in the damping laminate. A 3mm thick section of material with appropriate elastic parameters and a damping factor of 0.09 was placed on the tower wall which has a damping factor of 0.0025. Figure 8 shows the sound pressure level 20 m from the base of the tower for models with different amounts of damping laminate coverage. These results provided sufficient information for the turbine manufacturer to conduct a cost-benefit analysis and coverage from the 3rd to 8th weld was selected.

6. Conclusion

This project demonstrated several uses of COMSOL Multiphysics in a commercial engineering setting. Eigenfrequency analysis was used to compliment data collected in a vibration study and allowed the determination of complex modal shapes of high order resonances in a wind turbine. The models showed that vibrations caused by gear teeth meshing were moving from the gearbox, through the gearbox isolators and into the tower wall. The tower walls are easy to excite close to the gear-meshing frequency, so these vibrations were then amplified before being transferred to the air as problematic tonal noise.

Once the models had allowed an understanding of the modal shapes of the amplifying skin frequencies, it became possible to target areas close to the top of the tower with a damping laminate. Finally, COMSOL Multiphysics was used to develop an acoustic-structural interaction model that showed how different amounts of coverage with the damping laminate affected the sound level outside the tower. This model allowed the turbine manufacturer to select a cost-effective solution which satisfied regulatory requirements and allowed the continued operation of the wind turbine sites without impinging on residential communities.

7. References

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