Tonal characteristics of wind turbine drive trains

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ABSTRACT

Wind turbines use a drive train incorporating a rotor, gearbox and generator to harness power from turning of the turbine blades. Modern wind turbine drive trains utilise a multi-stage gearbox, typically involving a planetary gear stage (low speed shaft) and two parallel stages (mid and high speed shafts). These gearboxes have natural frequencies and mode-shapes dependent upon the stiffness of gear shafts and tooth pairs, which can interact with tonal sources such as gear meshing frequencies to result in audible tones, and greatly increase the annoyance factor of wind farms and adversely impact on human health. This paper examines the drive train of a wind turbine, the impact on environmental noise emissions from the turbine, and how tonal characteristic issues associated with its operation may be avoided.

Keywords: Wind Farms, Drive Trains I-INCE Classification of Subjects Number(s): 11.1.5, 14.5.4

1. INTRODUCTION

Assessment of noise from wind farm developments is critical in ensuring the long-term viability of such developments and maintaining the health and amenity of residences exposed to such noise. Tonal noise characteristics at wind farms can sometimes occur due to both aerodynamic noise sources from the turbine blades (e.g. laminar-boundary-layer-vortex-shedding), as well as mechanical noise sources within the nacelle (e.g. gearbox, yaw drive, etc). Tonal noise characteristics are not readily masked by broadband noise from wind blowing through trees, and can result in increased annoyance at receiver locations in the vicinity of the wind farm if left untreated. This paper examines the role of tonal noise emissions from the multi-stage gearbox of wind turbines (in particular excitation at gear mesh frequencies), and interaction with the gearbox mode shapes which can result in adverse tonal noise emissions to the environment.

2. TONAL NOISE FROM WIND FARMS

Wind turbine generators create noise through several sources, including mechanical and aerodynamic. While aerodynamic sources can result in special audible characteristics from wind turbines (such as tonal noise from laminar-boundary-layer-vortex-shedding) and is an important part of wind farm noise assessments, this paper focusses only on noise from mechanical sources.

2.1 Human Response to Tonal Noise

Special audible characteristics such as impulsive noise, low frequency noise, modulating or tonal noise is often perceived as more annoying than constant, broad and steady noise (2). In particular, a noise source with a tonal characteristic will often be described as something with a pitch or sharply defined note that is clearly audible above everything else. Research has found that the presence of a single tone in audible noise spectra both increases annoyance ratings and decreases human performance (1), with the effects classified into the following three general categories (17):

- Subjective effects including annoyance, nuisance, dissatisfaction
- Interference with activities such as speech, sleeping, and learning
- Physiological effects such as anxiety, tinnitus, or hearing loss (Rogers, Manwell & Wright, 2006)

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2.2 Assessment of Tonal Noise Characteristics

To quantify the impact of tonal noise characteristics on human response, Australian and International Standards include a noise characteristic penalty, whereby a measured source level with a tone present is adjusted by arithmetically adding to the overall level, effectively making the assessment criteria more stringent than if no tone was present in the source noise. Assessment methodologies undertaken in Australia include the South Australia Wind farms environmental noise guidelines (2) where a +5dB(A) penalty is applied where tonal audibility at a relevant receiver is greater than zero, and the New Zealand Standard NZS 6808:2010 'Acoustics – Wind Farm Noise' (14) (which is also referenced within the Policy and planning guidelines for development of wind energy facilities in Victoria (16)), where an adjustment of up to +6dB(A) is added to account for the adverse subjective response caused where tonal characteristics are audible.

The adjustment for tonal penalty will potentially limit the flexibility of wind turbine placement / number of wind turbines for wind farm developments, and tonal noise characteristics are therefore generally avoided through proper design and selection of wind turbines wherever possible.

3. WIND TURBINE DRIVE TRAINS

The gearbox in a wind turbine is used to transfer torque loads from the wind to a rotary output for the generation of power. Generally the drive train arrangement is governed by the low-speed shaft powered by the wind, and the high-speed shaft of the generator as shown in Figure 1.

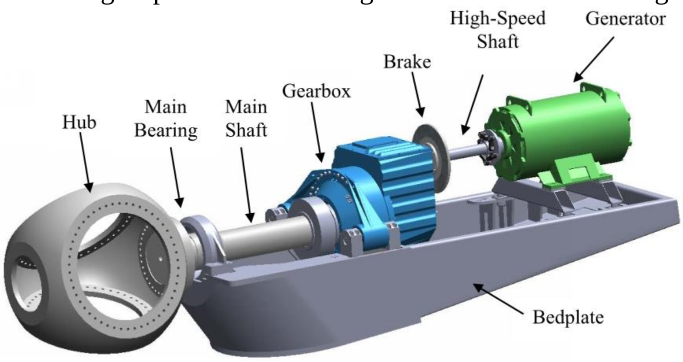

Figure 1 – Drivetrain configuration of a wind turbine (4)

The multi-stage gearbox of the wind turbine has three stages (a low speed planetary gear stage, and two higher-speed parallel gear stages, arranged as shown in in the schematic design in Figure 2) discussed in the following sections.

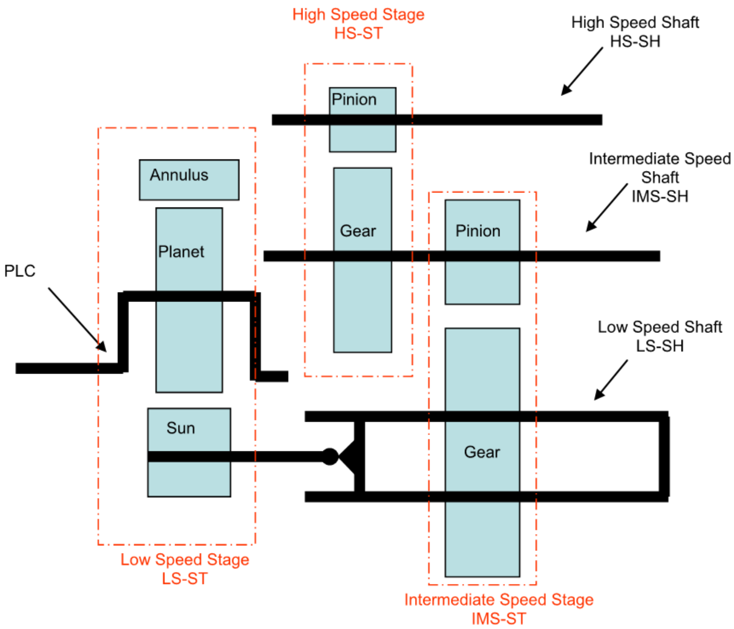

Figure 2 – Multi-stage gearbox arrangement (13)

3.1 Planetary Stage

The low speed planetary stage of the gearbox consists of three identical planet gear orbiting around a smaller sun gear, within a fixed ring gear housing (also called annulus gear, which is fixed to the gearbox housing), as shown in Figure 3.

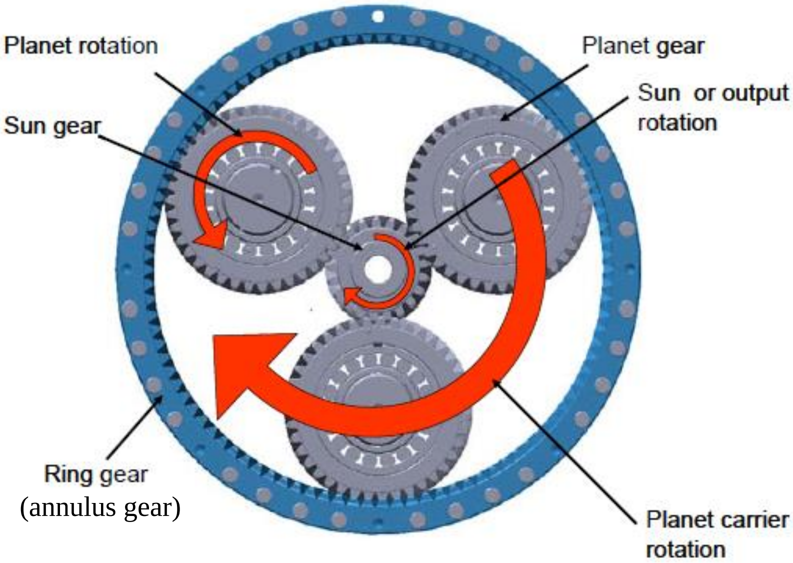

Figure 3 – Planetary stage of the gearbox and rotational directions (4)

The three planet gears are carried by the planet carrier of the main shaft which passes through the main bearing and is driven by the wind turbine rotor. Rotation of the main shaft causes rotation of the sun gear and the connected Low Speed Shaft (LSS) in the same direction. All gears in the planetary stage of the gearbox have a helix angle (teeth cut at an angle to the axis).

3.2 Parallel Stages

Rotation of the sun gear and LSS from the planetary stage of the gearbox is transferred through two parallel stages involving the Intermediate Speed Shaft (ISS) before reaching the generator through the High Speed Shaft (HSS). Each parallel stage consists of helical gears in a wheel and pinion arrangement.

4. GEAR MESH FREQUENCIES

Noise from gearboxes is generally tonal in nature, where noise occurs at discrete frequencies above the background noise level. One of the key frequencies which occur in all gearbox arrangements is the Gear Mesh Frequency (GMF). In a simple pinion / wheel gear arrangement, the GMF is the product of the gear rotational speed (in Hz), and the number of teeth on the gear.

Calculation of the GMF of the planetary stage takes into account the planets meshing with both the sun gear and the ring gear, with the planetary fundamental gear mesh frequency calculated in accordance with equation (1):

$$F_{mesh} = (\omega_s - \omega_c)Z_s = (\omega_p - \omega_c)Z_p = \omega_c Z_r \tag{1}$$

For n_p planets orbiting the sun gear, the prominent gear mesh frequency is then given by equation (2):

$$GMF = n_p F_{mesh} = n_p \omega_c Z_r \tag{2}$$

Table 1 presents the calculated gear mesh frequencies for a multi-stage wind turbine gearbox.

Table 1 – Gear rotation and gear mesh frequencies of a wind turbine drive train

Component	Number of teeth	Speed (Hz)	GMF (Hz)
Ring gear	87	-	-
Planet gears	33	0.7	22.1
Planet carrier (main shaft)	-	0.3	-
LSS sun gear	18	1.5	66.2*
LSS wheel	72	1.5	106.5
IMS pinion	18	5.9	106.5
IMS wheel	93	5.9	550.0
HSS pinion	22	25.0	550.0

** prominent gear mesh frequency for 3 planet gears*

Also important in investigation of the measured spectrum is the sidebands of the gear mesh frequency (example shown in Figure 4), which can indicate (15):

- irregularities in teeth spacing, causing impacts between teeth to not be equally spaced, and resulting in the acceleration and deceleration of the mating gear
- eccentric gears or shaft deflection / misalignment which causes gears to modulate the speed of the meshing gear (evident as mesh frequency sidebands at multiples of the pinion or wheel speed)

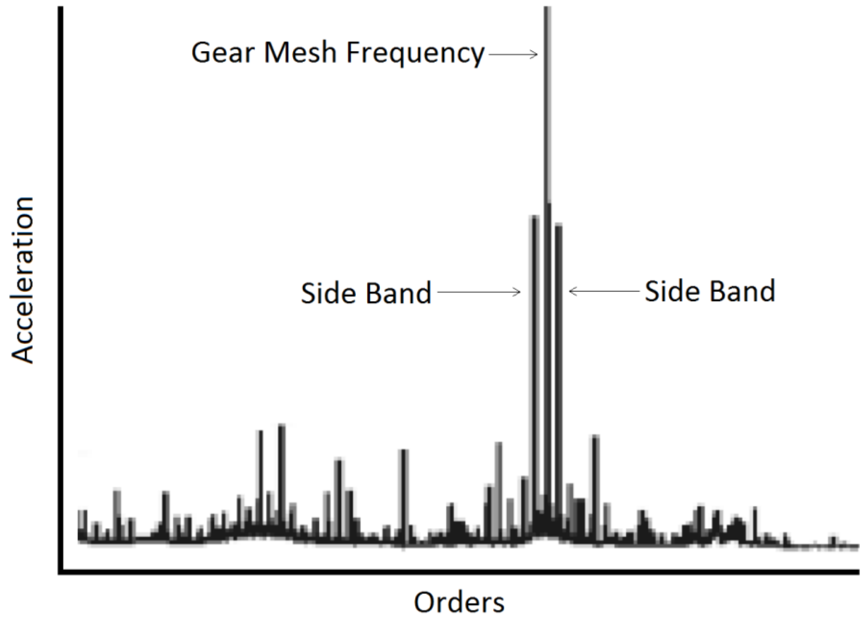

Figure 4 – Order analysis (zoomed) showing the gear mesh frequency and side bands

Where an issue exists with one of the gears (e.g. eccentricity), the sidebands around the GMF are typically 1x rpm of the subject gear (due to eccentricity / imbalance of the gear rotation) or 2x rpm (due to gear coupling / shaft misalignment issues) (12).

Mesh defects which occur due to damage of the gears or wear with age will also cause adverse vibration as the teeth engage such as defects in the gear teeth (i.e. wear, chips, dents, cracks and pitting) and the condition of the contact between teeth. These defects result in discrete spectral lines in narrow band spectra which are investigated to identify drive train faults and damage.

5. EIGENMODES OF THE GEAR BOX

Each of the excitations discussed in the previous section is useful in determining drive train faults, damage and degradation, however if these excitations occur at or near the natural frequency of another element within the wind turbine (including gearbox, turbine blades, nacelle casing, etc), the effect of the vibration will be magnified and the associated radiated noise can increase.

5.1 Rigid Body Analysis (One DOF)

Peeters et al (9) notes "in order to avoid resonance, it is important to know which eigenmodes can considerably influence the loads in the drive train. This means that in one way or another there should be a coupling between these modes and a source of excitation". The gearbox housing, shafts and gears are not a rigid structure, and stiffness of each element (including stiffness between meshed gear teeth) play an important role in the response of the drive train. Conventionally this has been modelled as a one degree of freedom system, with movement restricted to rotation of the drive train components.

Detailed investigation of the mode shapes was undertaken based on the work by Todorov and Vukov (10) which provides a frame-work for dynamic modelling of a three-stage wind turbine gear box (10 bodies and 11 degrees of freedom), with the tooth meshing modelled as a linear spring with stiffness as a time function. The differential equations describing the torsional vibrations of the wind turbine as outlined by Todorov and Vukov are given by equation (3).

$$[M]\{\ddot{q}\} + \left[C(t) - \omega^2 C_\omega\right]\{q\} = \{T\} \tag{3}$$

where M is the inertia matrix, C is the stiffness matrix and C_ω matrix results from the carrier rotation.

Based on the method described in Todorov and Vukov (11), the natural frequencies and mode shapes are given by equation (4), with the simplification of mesh stiffness being constant and equal to the average stiffness over a cycle, and all external forces assumed to be zero.

$$\left([M]^{-1}[C] - \lambda[E]\right)\{q\} = 0 \tag{4}$$

Using the non-zero elements of the M and C matrices as outlined in the Todorov and Vukov paper, the following natural frequencies of the drivetrain were calculated based on a rotor operating speed of 15 rpm (frequencies given in Hz):

753.8, 456.8, 381.7, 245.9, 125.3, 15.9, 75.3, 59.9, 59.2, 58.7, 20.2

Coupling of the calculated natural frequencies of the drive train with the excitation frequencies calculated for gear meshing was then analysed using a Campbell diagram, where intersection of the plotted natural frequencies and the gear mesh frequencies for the rotor speed of interest show sources of tonal excitation sufficient to transfer to other radiating elements of the wind turbine (e.g. nacelle shell, rotors, generator casing, etc).

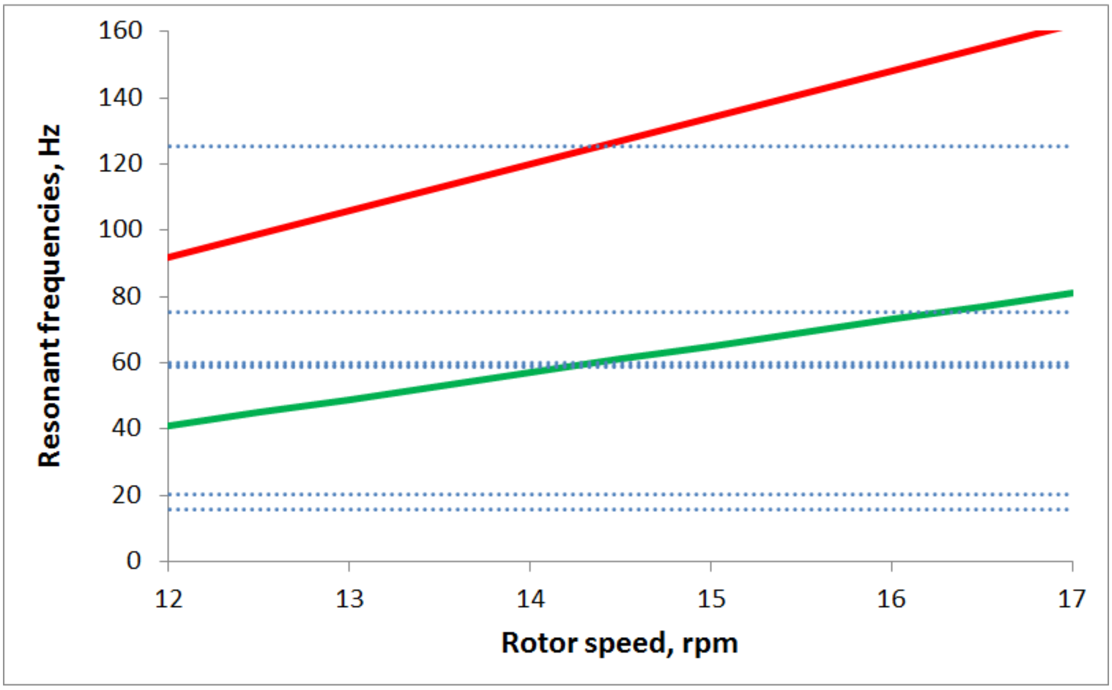

Figure 5 – Example Campbell diagram which indicates GMF of the LSS and ISS (shown in green and red) in comparison with the dominant resonant frequencies (blue dotted lines). Intersections within the turbine operating rotor speed range indicate important radiating elements.

5.2 Multi-Body Analysis

Conventional modelling methods using one degree of freedom (DOF) have been sufficient for modelling the dynamic behavior of the overall wind turbine, however analysis using these methods inadequately models the dynamic behavior of internal drive train components (8) at higher frequencies. Peeters et al (8) notes insight from additional analysis is useful for noise radiation calculations (and vibration monitoring), given internal drive train dynamics are in a frequency range well above overall wind turbine dynamics.

Peeters et al (8) describes the multi-body simulation technique, with differing levels of complexity, from the simplest one DOF torsional model, to rigid multi-body models that allow six DOF (translation/rotation) of individual drive train components, to flexible multi-body models. The modelling techniques (using LMS International's DADS – "Dynamic Analysis and Design System' software) are beyond the scope of this paper, however the techniques highlight the inaccuracies of assessing vibration using one DOF models.

Peeters provides a general overview of the flexibilities in the different models:

- Tooth Flexibility: Represented as tooth stiffness between components;
- Component Flexibility: Represented as a stiffness between components or integrated stiffness of a component;
- Bearing Flexibility: Represented as the stiffness between the component and the bearing housing.

Peeters presents a comparison of each modelling method at the high-speed parallel stage of a wind turbine gearbox, being a helical gear pair, as shown below in Table 2. Peeters also presents results for a complete wind turbine drive train comparing only a torsional model with a rigid multi-body model. The significant reduction in frequency of the fourth mode ("y-z rotation of the pinion") between the torsional to rigid model and further reduction when flexibility of components is taken into account highlights the inaccuracy of one DOF modelling methods. The significant additional modes, all within the audible frequency range, further highlights the need to use more complex models to understand the potential for vibration and noise radiation from drive train components.

Table 2 – Comparison of modes of a helical gear pair using a torsional rigid body model, a rigid multi-body model, and a flexible multi-body model (8)

No.	Torsional model	Rigid MBM	Flexible MBM	Mode Shape
1	0	0	0	Rigid body mode
2		400	395	x translation pinion
3		510	497	x translation gear
4	1479	702	518	y-z rotation pinion
5		755	519	y-z rotation pinion
6		761	635	y-z rotation gear
7		775	696	y-z rotation gear
8		801	702	y-z translation gear
9		821	718	y-z translation gear
10		1003	848	y-z translation pinion
11		1197	1012	y-z translation pinion
12		1947	1634	x rotation pinion and gear

6. TONAL NOISE EMISSION AND CONTROL

6.1 Transmission Paths

Tonal audibility issues can arise where the GMF excitation within the drive train is transmitted to other parts of the wind turbine, resulting in re-radiated structure borne noise. This includes noise radiation from the generator casing and other drive-train elements (e.g. brake, gearbox casing), transmission through the fixed torque arms with re-radiated noise from the drive train bed plate, the nacelle shell, or the wind turbine mast, or vibration transmission along the main shaft into the rotor, resulting in re-radiation from the hub or rotor blades. Figure 6 presents the key vibration transmission paths which require suitable vibration isolation treatment.

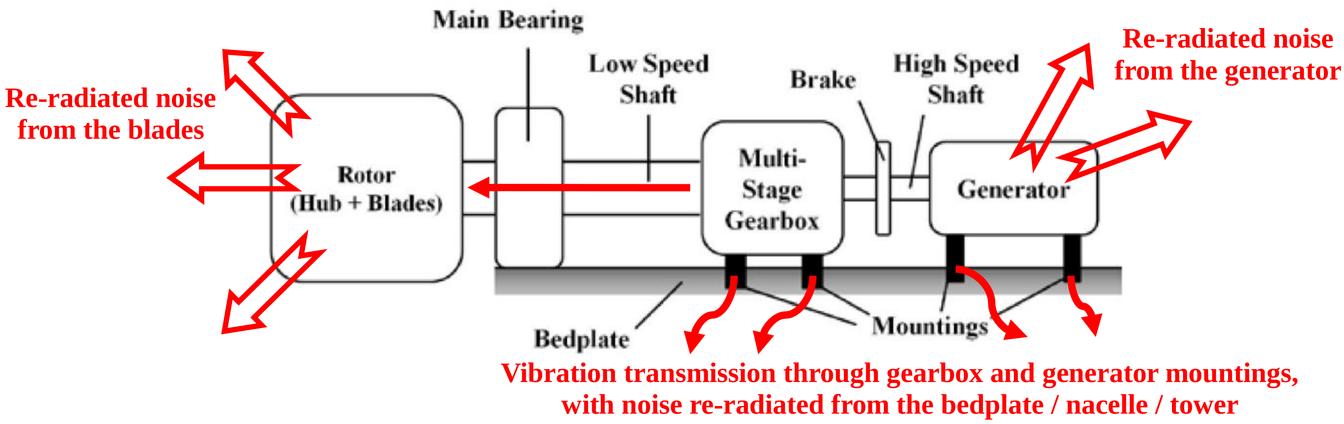

Figure 6 – Drivetrain configuration of a wind turbine (5)

6.2 Acoustic Treatment

There are two possible solutions for keeping a transmission unit quiet as described by Tuma (7):

- Introducing an enclosure to prevent noise radiation, which provides the easiest solution but has adverse impacts of difficult maintenance and inefficiency, along with issues associated with practical installation for a wind turbine
- Solving the noise problem at the source by improving the gear design, manufacturing or operation which is much more efficient but is technically challenging

In terms of solving noise, and in particular tonal characteristic problems at the source, isolating particular elements of the drive train is often effective in controlling structure-borne noise being re-radiated from other elements of the turbine. Treatment at the source also has the advantage of avoiding bulky enclosures, which would impact on both weight and accessibility within the confines of the nacelle.

Vibration transfer through the torque arms is typically managed by implementing rubber bearings at the anchor points on both sides of the drive train, and is one of the main transfer paths of noise from the drive train into the wind turbine structure (6). However, degradation of the rubber bushings over-time or improper design for the required torque loads may result in vibration flanking through the torque arms, and noise re-radiation from the wind turbine.

The generator and associated mounting locations on the bed plate are another potential path of vibration transmission, and elastomeric bearings are used for the suspension of the generator to control this flanking path. This flexible mounting is used to control structure-borne noise from the generator, while also ensuring that no stresses are produced in the mechanical drive train in the case of structural deformations.

Mechanical couplings are used to connect the HSS (output) shaft of the gearbox to the generator. Flexible shaft couplings are implemented to tolerate slight parallel and axial misalignment between the gearbox and the generator which would otherwise result in premature wear of bearings and equipment on both sides of the coupling (however tolerance of misalignments works only to a degree). Different types of flexible coupling can be implemented to the drive train including composite-disc couplings (shown in Figure 7) and link type couplings (where links located on each end of a tube are connected with rubber elements to tolerate misalignment and control structure-borne noise).

Figure 7 – Composite disc coupling between the HSS and generator (18)

Depending on the source of the vibration within the drive train, tuned mass dampers (TMDs) may also be applied to various elements of the drive train / wind turbine to control structure-borne re-radiated noise based on an understanding of the natural frequencies of the wind turbine (e.g. the blades, tower, etc). Zhang and Nielsen (19) discuss passive control of edgewise vibrations using roller dampers and tuned liquid column dampers, where dampers were incorporated into a 13-DOF wind turbine model to reduce fluctuations of the generator torque (increase the quality of the generated power), and the coupled blade-tower-drivetrain vibrations. Other methods include rotational tuned mass dampers (RTMDs) attached to the main shaft to control transmission of gear mesh frequencies into the turbine blades, or tuned mass dampers fixed to the torque arms of the gear box (shown in Figure 8).

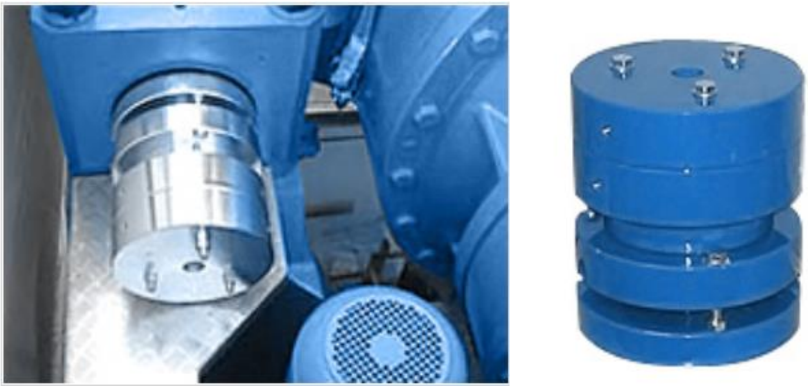

Figure 8 – Structure-borne noise damper (3)

7. CONCLUSIONS

Assessment of tonal noise emissions from the wind turbine into the environment, and in particular vibration transmission from the drive train into other elements of the wind turbine (e.g. blades, nacelle casing) will typically require a detailed understanding of the tonal characteristics at environmental receivers (i.e. narrow-band measurements undertaken for various operating scenarios, wind speeds, wind directions, meteorological conditions) to confirm the frequencies of interest. Subsequent investigation of the excitation frequencies within the wind turbine can then be undertaken to confirm the source of the tonal characteristic, and investigation of the natural frequencies of the drive train and other elements of the wind turbine used to determine the path of excitation and any re-radiated structure-borne noise to the environment.

Understanding the eigenmodes of the gearbox can be a complex process. Multi-body modelling techniques and analysis incorporating the dynamic behavior of internal drive train components will provide superior results compared to the rigid body (1-DOF) analysis commonly used for overall wind turbine dynamics. Vibration measurements of the wind turbine at the drive train / gearbox casing, above and below the vibration isolation bushings on the torque arm, the bed plate, the turbine rotors, the nacelle shell / casing, and the mast will assist with identifying the vibration sources / transmission paths, and calibration of the model.

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