Wind turbine and wind farm sound levels - a prototype journey

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ABSTRACT

The wind energy industry is booming in New Zealand and presents a new set of challenges to the acoustics industry. The NZ Standard NZS 6808:1998 'Acoustics – The Assessment and Measurement of Noise from Wind Turbines' provides guidelines on the way in which sound from wind turbines should be measured and assessed and the levels of sounds that are acceptable. As part of its resource consent conditions for a New Zealand designed and manufactured 500 kW wind turbine at Gebbies Pass near Christchurch in July 2003, Windflow Technology offered to do better than the guidelines and be no louder than 30 dBA at the boundary (including tonal penalty). This paper outlines the process of estimating the sound levels in the area, determining the sound source when the turbine was found to be over the limit, and the results from implementing the solution. The paper also raises other unique issues in measuring sound levels in different wind conditions and topography, and low frequency/infrasound concerns that are arising.

INTRODUCTION

New Zealand is situated in the southern part of the Pacific Ocean and lies directly across the path of the well-named "Roaring Forties" winds. These strong winds provide a consistent resource for wind power.

However the abundance of relatively inexpensive water, geothermal steam, natural gas and coal resources has allowed them to meet the steadily increasing demand for electricity until recently. This and the country's unsubsidised manufacturing economy has inhibited the development of a wind power industry.

Recent events have encouraged the country's major electricity generators to look to wind power as a means of increasing supply: the country's Resource Management Act (1991) has made it increasingly difficult for generating companies to further exploit major water sources; geothermal and gas supplies are dwindling; and there is strong public debate over the acceptability of coal-fired generation.

Since 1990 the author has had a vision of designing and manufacturing wind turbines in New Zealand for the country's high wind and unsubsidised conditions. This has required the development of technologies to provide a turbine light enough to be manufactured commercially in New Zealand and yet resilient enough to withstand the high and often turbulent wind conditions experienced in the better wind resource areas of the country.

Certain that the combination of a two bladed teetering system and a torque-limiting gearbox would provide a resilient and commercially viable wind turbine, the author established the company Windflow Technology in 2000. The company raised funds in 2001 for the design and manufacture of the full-scale prototype of the "Windflow 500", a 500 kW wind turbine.

The Company

The company's mission statement is "to be a global leader in wind turbine technology innovation". We now have over 700 shareholders and are listed on the NZAX share market. We specialise in the design, development and manufacture of utility size wind turbines, which are manufactured and installed with over 90% New Zealand content.

We have 14 staff, mainly professional engineers, and a subsidiary company, Wind Blades Ltd, which manufactures our wind turbine blades.

The Wind Turbine

Our turbine has a 33 m rotor and a rating of 500 kW. It combines two proven technologies based on the author's experience in Britain in the 1980's:

- two bladed teetering with pitch-teeter coupling
- a patented torque-limiting gearbox driving a standard synchronous generator, which runs synchronised with the grid, ie at constant 1500 rpm.

The 16 m blades are made of laminated wood-epoxy and fibreglass. The wood species is pinus radiata (New Zealand's main commercial species). The structure is based on a stressed shell concept, similar to that used by Vestas' 40 m blades from their factory on the Isle of Wight, England. The blades are made in Auckland by Wind Blades Ltd.

Figure 1. Windflow 500 Turbine

The gearbox (Figure 2) is made in Auckland by AH Gears Ltd. It is a 4 stage design with an overall ratio 30.94 and rated power 548 kW (mechanical). Starting from the low speed end the stages are planetary – planetary - parallel - epicyclic with patented torque limiting on the fourth stage. The torque limiting gearbox (TLG) system was developed by the author in the late 1980's to solving the wind turbine gearbox torque control problem, which it does by including a differential stage and a simple hydrostatic torque control circuit. The TLG system is patented in several countries including the USA.

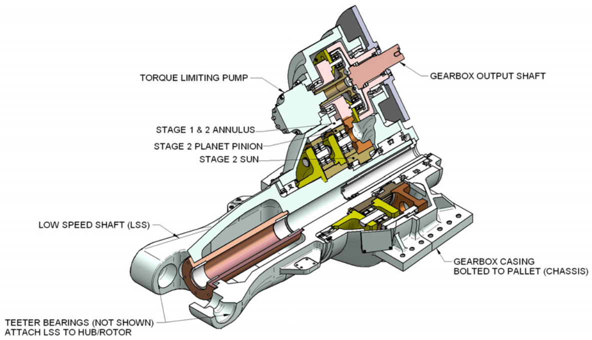


Figure 2. Internal Cutaway of Gearbox

Input speed varies from 48.5 to 51 rpm while the output speed is constant at 1500 rpm.

The parallel stage is helical and the three epicyclic stages use straight cut spur gears. Flexible spindles (as patented by Ray Hicks in 1964) carry all planets, enabling a multitude of planets and a compact design. The first planetary stage has eight planets, the second has four and the fourth stage has six planets. The gearbox has an integral low speed shaft (LSS) so that the main bearings in the gearbox carry the loads from the wind turbine rotor.

Lubrication is based on a dry sump draining to a de-aeration tank and being injected via an external filtered cooling circuit. The casing is SG iron and total weight of the gearbox is 2.6 tonnes including the LSS extension.

The turbine operates in wind speeds from 5.5 to 30 m/s and uses a synchronised, synchronous generator. The design is light-weight throughout, using approximately 50% less steel and concrete than comparable 3-bladed turbines.

Background to the Noise Problem

Prior to installing the prototype, we consulted with the local neighbours from a standpoint that measurable sound levels should conform to community-set standards (40 dBA being the local council's requirement) and if possible go even better. Normally wind farms do much better, and we agreed to a particularly low sound level (30 dBA including any tonal penalty at the house of the nearest objecting neighbour) as part of our resource consent. Why did we do this when we did not have to? There were three main reasons:

a) the nearest objector lived 1.4 km away and we believed we would easily meet that standard
b) the neighbour is question experienced very low background sound levels in a sheltered valley (sometimes as low as 20 dBA or lower) and expressed the strong value that she placed on that sound quality
c) the turbine was a prototype. Therefore we accepted the need to "go the extra mile" for the local community. We also knew that if the sound levels exceeded 30 dBA at that distance, we would have a serious marketing problem with the turbine.

The topography of the Gebbies Pass area including the turbine location and neighbouring valley is shown in Figure 3.

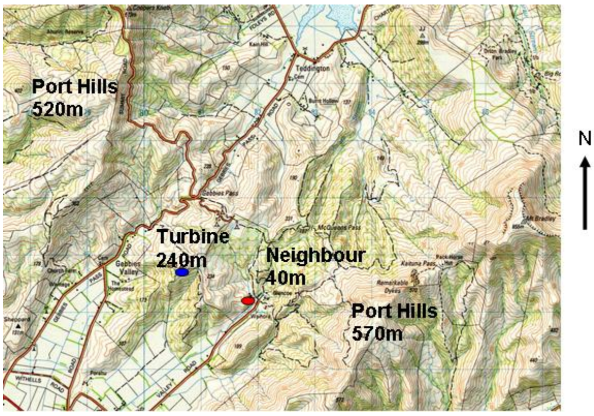
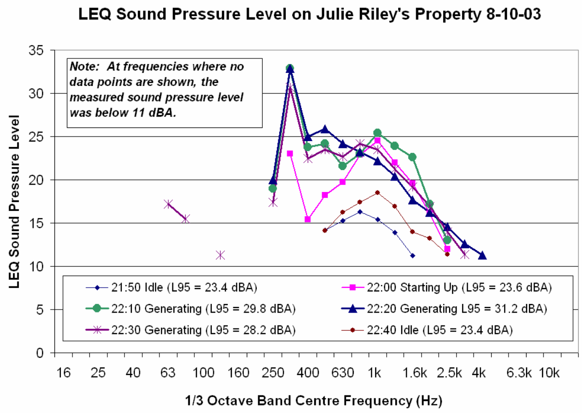


Figure 3. Gebbies Pass Topography

THE NOISE PROBLEM

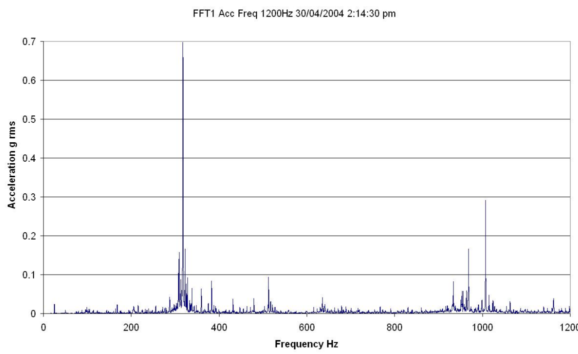
After commissioning, the prototype generated noise complaints from the neighbour in question. Working with the University of Canterbury and a number of different acoustic specialists, we made a range of measurements of the offending sound levels. It was difficult to obtain the right conditions to determine the exact sound level but eventually we obtained an evening measurement showing a level of 31.2 dBA at that residence against a background of about 23.4 dBA (see Figure 4).



Source: (Author 2006)

Figure 4. Evening measurements at affected residence, generating and idle

From the outset there was a clear tonal component at around 315 Hz (as shown in Figure 4 and even more clearly in narrow-band vibrations measurements like Figure 5).



Source: (Author 2006)

Figure 5. Gear case vibration measurements showing 311 Hz peak

This added another 5 dBA to make the assessed level 36 dBA. Therefore we voluntarily restricted operation to daylight hours, five days a week. After three months of trying various remedial measures, in November 2003 we shut down the turbine completely in accordance with our resource consent and took the time to get it right.

IDENTIFYING THE ROOT CAUSE

Tower

While the tone was clearly coming from the gearbox, and closely coinciding with the Stage 2 gearmesh frequency, initially our attention focussed on the tower. Why? Because there was obviously some resonance occurring in the tower. Sound levels in the nacelle right beside the gearbox did not seem excessive, whereas sound levels at the base of the tower were unusually high. Not only were measured sound levels there close to 100 dBA, the experience was like being inside a bell, with the vibration being able to be felt in one's body.

Therefore we examined the prospect of stiffening the tower panels with steel ribs. However finite analysis of a range of different rib configurations showed that there were simply too many modes in the range 300-320 Hz. Addition of stiffening ribs would simply shift the modes, not eliminate them.

Damping therefore seemed an attractive option. After considering various options, we decided to pursue rubber matting, glued to the interior of the tower. Laboratory testing showed that two layers of 25 mm rubber were considerably more effective than one at damping vibration in the range 200-500 Hz. Some tuned absorption was taking place with two layers, so we decided to proceed. The product was a type of matting made from recycled rubber, commonly used as a playground surface.

Lining about 20% of the tower interior with 50 mm of this product produced a major reduction in tower base sound levels, which came down about 8 dBA. A success of sorts!

However sound levels at a distance were unaffected. The tower was not the main problem after all.

Nacelle Cladding

Similar efforts were made to improve the sound reduction properties of the nacelle cladding. However nothing made any measurable difference to sound levels inside or outside the cladding.

Sound intensity measurements were made at this stage. Even with the measurement problems of using a stationary sound intensity meter on top of the nacelle aimed at the rotating blades, it became clear that 92% of the sound power was coming from the blades, with the balance coming from the tower and nacelle cladding.

Therefore attention shifted to the blades and the mechanism by which Stage 2 gearmesh vibration was being amplified.

In Search of the Hidden Resonance

By this time we had a large team of advisers working on the problem, drawing on the best acoustic advice available to us in Christchurch. The strength of the peak in the sound and vibration measurements indicated a structural resonance somewhere in the system. Based on the experience with the tower we realised that the blades themselves were probably not the root cause, but simply providing panel vibration or broad-spectrum resonance to propagate the vibration.

We tried to identify a component that would be more clearly resonant at about 311 Hz. In retrospect the answer was obvious, but we came to it in a roundabout way. We examined the gearbox/pallet sub-system, using both FE analysis and bump tests. A local company, Commtest Instruments Ltd, provided their "VB" vibration analyser initially on a loan basis. (We have since bought two VB units from them.)

However none of the initial bump tests on the gear case and pallet showed a clear natural frequency in the suspect range. Similarly the FE analysis showed a range of minor modes rather than a strong single mode at those frequencies.

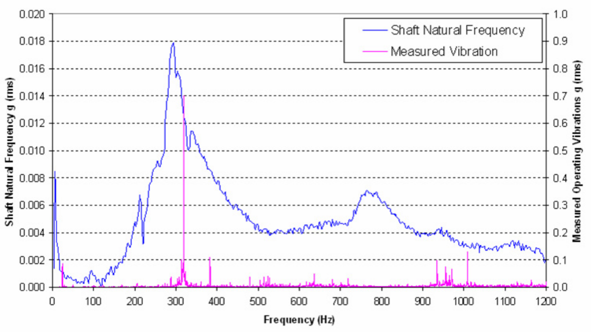
But we felt we were on the right track so we commissioned the government-owned research company, Industrial Research Ltd (IRL), to investigate natural frequencies by doing bump tests on the gearbox/pallet system.

Low Speed Shaft

Finally it became apparent why our early bump tests had not uncovered the real "culprit". All those tests had been on the external components which were easily accessible.

In situ access to the low speed shaft (LSS) was difficult, and made somewhat more difficult by the teetering hub (see Figure 2 for internal cutaway of gearbox). On a fixed-hub wind turbine, bump tests on the hub would provide good information about any LSS modes. However in our case the teeter bearings were isolating the hub from the LSS. Bump tests on the hub showed only a hint of a resonance but we were able to get direct access by removing hub inspection covers and positioning the accelerometer directly on the LSS.

Figure 6 shows the result. A significant bending mode of the LSS was apparent at about 290 Hz.



Source: (Author 2006)

Figure 6. LSS bump test result compared to measured gearbox vibration

Finally the noise problem made sense. The Stage 2 sun gear rides directly on bearings on the LSS, which in turn is mounted on the main gearbox bearings. The turbine rotor is mounted on the cantilevered section of the LSS. The front LSS bearing is a spherical roller bearing, which is self-aligning. Any forcing from the Stage 2 gearmesh was thus able to bend the LSS between its main bearings, giving rise to deflections of the cantilevered section out the front. The turbine rotor was being shaken at 311 Hz as it rotated!

This became our "tuned music system" model to explain the problem:

- the Stage 2 gearmesh was the "CD player"
- the LSS resonance was the "amplifier"
- the blades (being large hollow wooden items) acted as "speakers".

The “music” being propagated into the neighbourhood was a very boring single note, about E flat above middle C (311 Hz).

DEVELOPING AND IMPLEMENTING THE SOLUTION

Having identified that the root cause was deep inside the gearbox, we decided in March 2004 to remove the gearbox and return it to the AH Gears factory in Auckland. Using a full-load test rig, we carried out baseline sound and vibration tests at full and part load while working on a programme of retrofits to modify the gearing.

Our approach was to (as far as possible) “turn off the CD player”. In addition we changed the Stage 2 gearmesh frequency away from 290 or 311 Hz, increasing it to 375 Hz by changing the gear module. However by this time our researches had produced an innovative approach to the problem, so that the shift in frequency became something of a precaution, rather than a key part of the solution. Indeed we did not want to rely on simply shifting frequencies (forcing or natural) because we had already experienced how difficult it can be to eliminate resonances altogether.

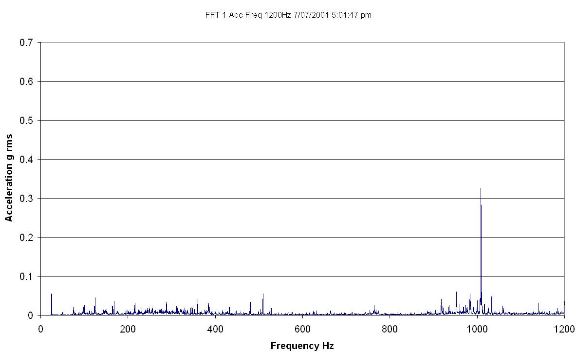
Rebuilding a gearbox is an expensive business and we did not want to start a process of trial and error looking for the quietest part of the spectrum in a complex system response!

Our key innovation in gearbox design is the subject of a current patent application so we are unable to reveal the full details. It is a unique combination of technologies which came together for the first time in our gearbox. Our researches showed that there was a theoretical possibility of substantially eliminating the gearmesh forced vibration in any planetary stage, and we decided to pursue this.

However it was not at all certain that theory would translate into practice. The gearbox manufacturer in particular did not want to rely on it and advised us to try other approaches as well.

Accordingly we planned a series of three main retrofits to the gearing, which would progressively establish whether the new theory, or more traditional approaches such as tip relief modification, would be more effective. These three retrofits were carried out and tested between April and June, 2004.

Retrofit 3 involved full implementation of the new theory throughout the gearbox, not just for Stage 2 but Stages 1 and 4 also. Testing confirmed this gave the best results, as shown in figure 7.



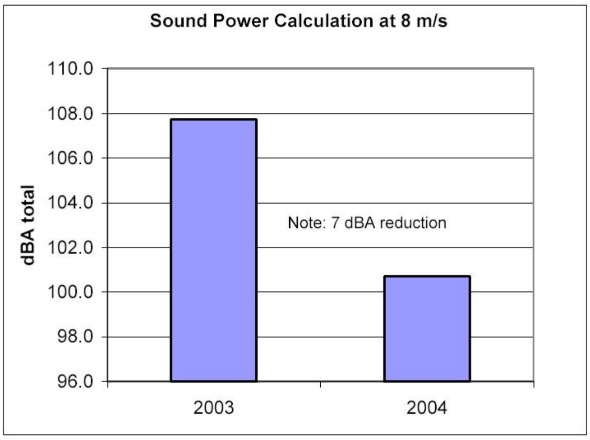
Source: (Author 2006)
Figure 7. Gear case vibration measurements after Retrofit 3

Comparison of Figures 5 and 7 shows that we have substantially eliminated the 311 Hz vibration at the heart of our noise problem. The only significant vibration peak is at Stage 3 gearmesh frequency (1008 Hz) which is the only parallel stage in the gearbox. All the vibrations at planetary gearmesh frequencies have been substantially eliminated, as predicted by the new theory. Retrofits 1 and 2 showed only partial elimination. Again this served to confirm the new theory and its superiority over conventional methods of gear vibration reduction.

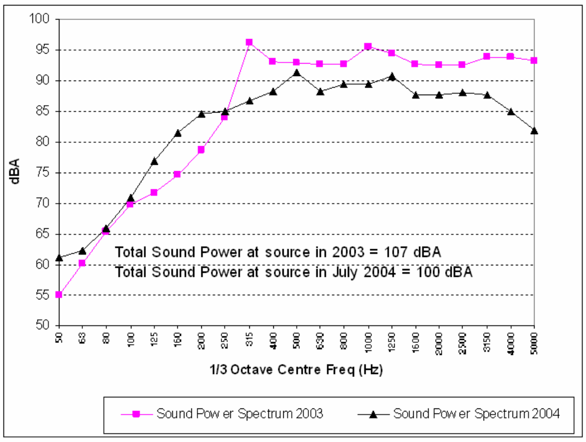
Following retrofit 3, the gearbox was returned to Christchurch and refitted to the prototype windmill in July 2004.

THE RESULT

Compliance testing carried out for the Banks Peninsula District Council indicates that sound levels at the affected residence have reduced to well below the consent requirement of 30 dBA. Again the very low levels made it difficult to be precise but we estimated the new level to be about 24-27 dBA. This compared to our original assessment of 36 dBA (31 dBA measured plus 5 dBA tonal penalty). Near field measurements to determine sound power (Figures 8 and 9) also showed about a 4-7 dB reduction plus elimination of the tonal component.



Source: (Author 2006)
Figure 8. Overall sound power levels before and after



Source: (Author 2006)
Figure 9. Sound power spectra before and after

THE FUTURE

Windflow Technology Ltd is undertaking testing of the Gebbies Pass prototype turbine towards International Electro-Technical Commission (IEC) WT01-1A. The sound power level of the turbine will be certified as part of this process.

As at September 2006, Windflow Technology Ltd has completed its first batch of five production machines which have been installed on a wind farm near Palmerston North. NZ Windfarms Ltd obtained resource consent for the Te Rere Hau wind farm from the local council for this 97 turbine project of 48.5 MW, which will be built in stages over the next 2-3 years. As part of the resource consent, the wind farm has a 40 dBA sound limit at the notional boundary. At time of writing, background sound level monitoring had been completed, and sound level compliance testing will be carried out after each stage of the wind farm is commissioned.

The wind energy industry and acoustics consultancy in New Zealand has learned a lot about wind turbine noise during the last decade’s growth period. The resource consent process requires an assessment of the likely noise effects of the wind farm before installation and also requires compliance monitoring after commissioning.

Concerns about low frequency and infrasound from wind farms have started to be raised, however two recent reports carried out by independent New Zealand professionals (Bel Acoustic Consulting 2004, Hegley Acoustic Consultants 2004) concluded that there is no evidence to indicate that low frequency sound or infrasound from current models of wind turbine generators should cause concern.

CONCLUSIONS

Windflow Technology has encountered and overcome a classic wind turbine noise problem. Like many such problems:

1. gear noise has been central to the problem and thus difficult to rectify
2. residents in a sheltered valley nearby have been affected, and focussed attention on it
3. a resonance was involved, though this was not easy to pinpoint
4. the blades and tower were providing panel vibration to propagate the sound.

We have achieved a dramatic reduction in assessed sound level, due to the combination of:

- the LSS resonance being a big part of the problem
- the theoretical breakthrough in planetary gear vibration which we invented and validated in the course of our researches.

ACKNOWLEDGEMENTS

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