Does Turbine Noise Affect Human Health?

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Today’s utility-scale wind turbines are uniquely challenged in the variety and severity of transient torsional events (or TTEs) that can cause potentially damaging torque reversals and high-amplitude torsional oscillations [4].

An RTD (Reverse Torsional Damping) device, a special type of torsional damper, limits torsional oscillations in turbine drive components during transient events. A typical turbine drivetrain shows that the device would mount on the generator. During normal turbine operation, there is no damping or other impact on power production.

An RTD device can be installed on the generator shaft and adapted to the existing high-speed coupling, providing an economical and easily “retrofittable” mechanism to mitigate the damaging effects of TTEs.

Real-world recordings of torsional load in drive systems of many different turbine models show that the worst torsional vibrations and torque reversals generally occur during transient events, such as emergency stops, grid faults, and many other hard stops. These are

Using an FMEA approach to quantify value of an RTD

This article is the first of a two-part series in which a Failure Mode and Effects Analysis (FMEA) is used to evaluate how torsional oscillations and reversals can damage many expensive turbine components. It also compares the effects of adding a Reverse Torsional Damping device to mitigate the damage. The FMEA calculates a projected range of cost reductions based on the credibility of evidence, contribution to overall failure mode, and the estimated life extension from the damping device.

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The illustration provides the general layout of a typical wind-turbine drivetrain.
Illustration: NREL

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detrimental to bearing and gearbox life. The Failure Mode Effects Analysis or FMEA covers this in detail. Less well understood are the ways these loads may impact other turbine components.

Many types of machinery see transient loads during startups, shutdowns, and events of unusual severity. However, they rarely result in a significant reversal of drive-system torque unless it is intended to operate in reverse, in which case reversals are smooth and controlled.

In modern wind turbine designs, blade pitching protocols provide the primary means of braking. Many events can trigger a stop command that results in the blades pitching rapidly to decelerate the turbine. Most of this aero-braking effort goes into decelerating the mass of the rotor, but a portion of the braking goes throughout the drive system to decelerate the rotating mass of the generator. For example, the left plot in Aero-Braking only during a hard stop shows the recorded main-shaft torque during aero braking. It causes torque reversals equal to 75% of turbine rated torque and excites significant torsional oscillations at the natural frequency of the blades in the drive system.

The blue line of the overlaid torque plot (right) shows the same aero-braking stop recorded on a nearby turbine equipped with an RTD device. Negative torque excursions were limited to 40% of rated turbine torque throughout the drive system and the amplitude of the oscillations has been reduced by almost 50%. The damping action has effectively limited the torsional elastic energy that is stored in the drivetrain during the aero-braking.

A hard stop is defined in this report and characterized as a rapid shutdown procedure initiated by the control system or by operator intervention, triggered by a stopping protocol using the emergency-stop function. In more critical hard stops, including many emergency stop protocols, caliper-braking is triggered simultaneously with maximum aero braking. The caliper-braking effort is divided, with most of the braking effort going toward slowing of the blade mass.

Caliper braking causes a torque in the gearbox and main shaft that is large enough that the combined aero and caliper braking is seen as generally positive. A highly oscillating torque, is shown in Torque from a hard stop (on next page). It was recorded on the main shaft of a 1.65-MW turbine without an RTD device. The amplitude of these torsional oscillations is far greater than any recorded during normal turbine operation, and is reasonable expected to add to the cumulative fatigue damage of many turbine components.

Combined aero and caliper braking also shows a large torque reversal that begins at 14 seconds, when the shaft rotation stops due to caliper braking. While torque reversals in general damage the drive-system bearings, a torque reversal of this magnitude can be more damaging when bearings are not rotating, such as during a shut down. The potential for surface damage is further exacerbated when the stationary bearings are subject to simultaneous axial forces and movement. This is true for gearbox bearings that support helical gears and for the main-shaft bearing, which is restraining the high fore and aft oscillations of the tower during the stop. For instance, gearbox borescope inspections of many bearings, high speed to low speed, have produced convincing evidence linking TTEs to scuffing and secondary abrasive cutting wear [6]. Torque reversals also promote bending fatigue damage in gear teeth.

Hard stops on turbines with and without an RTD (next page) shows the torsional behavior of the turbine with an RTD device in solid blue line overlaid onto the turbine without it, during side-by-side monitoring of the same hard stop. The damping action reduced the magnitude of the oscillation by more than 70%, effectively limiting the torsional energy that was stored in the drive system and protecting the gearbox and other components.
drive components from the worst torsional vibrations. More importantly, the overlaid plots show that the reverse torsional-damping action almost eliminates the large torque reversal when the shaft rotation stops.

Research shows that such reductions in high-torsional oscillation amplitudes and torque reversals will provide significant life improvement to the gearbox and its bearings, and could reduce O&M costs to most of the drive system and many other turbine components. The Life-Cost-Based FMEA detailed in the second article of this series, is intended to help quantify the resulting cost benefits.

**Explanation of the FMEA structure**

To best understand the FMEA structure, the discussion is split into life cycle costs, and the potential life-extending effects of RTD devices on failure modes. To populate the FMEA spreadsheet, the authors used this hierarchy of preferred references:

- Wind turbine, bearing, and gear standards,
- Commonly available public documents and sources,
- Turbine operation first-hand experiences, and
- Best estimates based on experience.

To identify components to focus on, a traditional FMEA uses a Risk Priority Number (RPN), which is the product of severity, occurrence rates, and detectability. This FMEA model was based on a modified RPN number created to quantify the benefit of the overall wind-turbine system from an RTD device in the drivetrain. The FMEA calculates a projected range of cost reductions based on the product of the credibility of evidence, contribution to overall failure mode, and the estimated life extension from an RTD device.

The FMEA model is based on a generic list of known issues affecting turbines in the 1.5 to 2.0-MW range. It is important to note that not every individual turbine will experience failure on all listed components. That is, a turbine with a component designed and or manufactured in a certain way may have less (or more) incidence of a particular failure. It is believed that the value of this particular FMEA can be enhanced when populated using site-specific values taken from the statistics from one particular turbine type, in one location, by one owner.

Such an approach would provide the necessary backup for a strong site-specific business case, and should support the installation of RTD devices. The more generic case presented in this article is a conservative estimate of what an operator may expect from such an exercise. It provides a useful benchmark for any site-specific version of an FMEA and draw attention to several chronic and acute modes.

Estimates of gearbox repair costs have a wide range in the FMEA model. Lacking sufficient public domain cost data, it was necessary to make assumptions, such as:

- Repair parts and work are of high quality. For example, that proper bearing heating is used.
- All bearings are replaced when a shaft assembly is repaired.
- For a minimum cost, damage is confined to a single part and the repair is done under ideal conditions.
- For a maximum cost, everything that could go wrong would go wrong, and there is secondary damage from the failure. For example, housing bore damage, metal fragment contamination of the entire filtration and cooling systems, planetary failure with housing rupture, and the associated clean up. The ratio between bearing and gear failure rates for each of the four shaft assemblies was estimated using 2013 Gearbox Reliability Collaborative data [15].

When performing a system FMEA, it is normal to break the system into its component parts or key assemblies in *Life and cost headers from the FMEA*. The first
Column heading is “Component,” which consists of assemblies, sub-assemblies, and components. The “Component” group related to the Gearbox is grouped into four shaft assembly categories. Also included in these shaft assemblies are the potential effects of gearbox housing deflection and deformation.

Some failure modes show a large value from the mitigation of TTEs, while other failure modes show a lesser value in Modes of failure and potential improvements from an RTD. Important items to extract from this data, presented in the next issue, will come the modes of high value are the failure modes that would benefit most from mitigation, and the total amount of incremental savings from all individual improvements shows the total value to the system from mitigation due to an RTD device installation.

**FMEA Inputs**

This section on inputs to the FMEA is divided into gearbox and non-gearbox components. The ways in which TTEs may impact a gearbox are relatively well documented and there is high confidence in these links. The impact of TTEs on other components in the wind turbine is less well documented and is an area that would benefit from more research, along with review of available data.

**Evaluating the Impact of TTEs on the Gearbox**

Providing quantitative input to the gearbox-FMEA model called for an understanding of the relationships between loading on the wind-turbine gearbox and the failure modes of its bearing and gears. In particular, the necessary understanding is of how TTEs affect bearing and gear failure modes and their longevity.

To perform the FMEA, the failure-mode information was organized around its causes for each mode. This differs from textbooks and standards that discuss bearing and gear-failure analysis, which are organized around the failure modes alone. This situation necessitated systematically reviewing and analyzing every bearing and gear failure mode described in the established standards and failure analysis books, looking for potential links between causal factors, such as TTEs, and the many failure modes. Where there were clear references to causes attributable to TTEs, confidence in the link is considered high. Failure modes that were not in any way attributable to TTEs were grouped together under the Other Failure Modes and given a zero for creditability of evidence. The results of this research and analysis were summarized and populated in the FMEA model.

Two quite different but related sources of bearing and gear failure mode data were reviewed and analyzed for causal relationships between TTEs and failure modes:

1. Bearing and gear-failure standards and failure analysis books [7, 8, 9], including ISO 15243 [12] and AGMA 1010 [5] and 6006 [4].
2. Wind Turbine gearbox technical articles, research papers, presentations, and dissertations pertaining to the causes and effects of failure [2,3,15].

Axial cracking is not included in the established standards or failure-analysis references, and there are numerous competing explanations for the failure mode and its solutions. A simple criteria was used to narrow the field of explanations, based on the fact that axial cracking in wind-turbine gearbox bearings did not surface until turbines reached a threshold of about 1.5 MW. Therefore, hypotheses unable to explain this threshold were eliminated. This left only two plausible root-cause hypotheses:

1. The through-hardened steel used throughout wind-turbine gearboxes has poor resistance to the formation and propagation of cracks. That is, it lacks sufficient toughness (crack resistance) for the specific application [7, 18].
2. When bearing surface speed reaches a threshold under high-load events, the allowable shear rate of the bearing material is exceeded, leaving behind an initiation point for axial cracking, which then propagates under normal operation and accelerates by high-amplitude transient loading [11].

Some gearbox-failure modes were not included in the FMEA model for one of three reasons:

- The mode does not affect wind turbine gearboxes. The mode could be electric current erosion (fluting) and lightning damage, contact corrosion, or cavitation,
- The modes were not initiated or propagated by TTEs, and
- The modes were low-cycle fatigue failures which occurred within this first 10k cycles.

Although gearbox and generator failure is normal and expected – and are budgeted items in the wind business – in this FMEA they are considered chronic problems. Typically, there is much greater value in reducing chronic failures than unexpected, costly, sporadic events [13, 14, 15]. The FMEA model includes chronic and acute failures.

The next challenge was to find a way to combine data from the different failure classification systems used in the failure mode standards and failure analysis references. For simplicity, the numerous individual failure modes were sorted into three broad classes: fatigue, wear, and overload.

Fatigue is defined by ISO 15243, the international standard for bearing failure modes, as: “The change in the structure, which is caused by the repeated stresses developed in the contacts between the rolling elements and the raceways... Fatigue is manifested visibly as a flaking of particles from the surface.” AGMA 1010-F14, the recently updated version of the gear failure mode standard, notes that fatigue involves the initiation and growth of cracks and defines high-cycle fatigue as “…fatigue where the cyclic stress is below the yield strength of the material and the number of cycles to failure is high.” Wind-turbine gearboxes are not prone to low-cycle fatigue, which occurs at 10k cycles or less, and requires that each cycle result in macroscopic plastic strain. Fatigue damage is permanent and cumulative and follows a logarithmic curve, so small increases in cyclical stress levels can lead to rapid decrease in fatigue life [17].

Wear is broadly defined as the progressive removal of surface material due to mechanical, chemical, or electrical action. Wear modes relevant to this FMEA are adhesion, abrasion, fretting, and false brinelling. Wear modes excluded from this FMEA include chemical corrosion, electrical erosion, and polishing. Although micropitting and macropitting also remove surface material progressively, fatigue and its effects are not included as forms of wear, and will be referred to here as “fatigue wear.”

Overload occurs when applied loads exceed the yield or ultimate strength of the material in the stressed area. It can range in size from small, localized debris dents to covering larger areas by true brinelling, caused by roller indentation into a raceway. Overloading often ends the useful life of a part by fracture. Fatigue cracking often precedes fracture by cracking through a significant portion of the material.

This FMEA divides the gear and bearing failure sequence into three distinct stages, and examines the contribution of TTEs to each failure stage: initiation, propagation, and failure.

**Initiation**

Hertzian fatigue modes such as macropitting and micropitting often take many years to initiate failure [8]. Rapid failure initiation is generally due to stress concentrations, not Hertzian fatigue modes. For example, geometric stress concentrations increase effective loads by 150 to 200% [17]. There are long lists of the causes of stress concentrations but for this
FMEA, they were limited to two broad categories: damage and flaws.

**Damage**, which occurs during turbine operation, includes adiabatic shear bands, debris dents, shaft misalignment damage resulting from housing deflection or deformation [10], rapid changes in bearing load-zone locations, or a single extreme event causing scuffing [6], true brinelling, or root fillet cracking. Debris dents in bearing raceways often initiate point-surface-origin (PSO) macropitting bearing failure [9].

**Flaws**, which occur prior to operation, are caused by everything from design to flaws in the steel and errors during hardening, grinding, or tempering. Common examples are subsurface non-metallic inclusions, hardening and grinding cracks, or grind temper flaws. Bearing failure often initiates at an inclusion [5, 7, 12]. Steel cleanliness is therefore a primary key to preventing premature failure, while globally steel quality has become increasingly questionable. Initiation cracks begin very small, on the order of the steel's grain size. Over time, these tiny initial cracks join together and extend across several grain boundaries. Crack growth may begin at this point. It is important to note that without initiation, fatigue failure will not occur, so delaying initiation will prolong gear and bearing life.

Part II of this article, scheduled for the April issue, will include a portion of the FMEA in a spreadsheet (it’s quite large and detailed) and a conclusion.

**For further reading:**

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