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## Comparison of vibration and transmission error in gear crack diagnostics

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### Abstract

Vibration has long been established for gear diagnostics. In the literature, the study of gear-crack induced vibration can be interpreted and divided into two main groups. The first group of studies treat crack-induced gearmesh stiffness and tooth-specific profile deviations due to plastic deformation (e.g., crack being permanently open) as excitations of gearbox dynamics. The two excitations can be combined as a single component, leading to static-transmission-error (STE). STE then passes through a certain transfer function, leading to outputs such as measurements of transmission error (TE) and vibration. Hence, signal processing based on this traditional interpretation of crack symptoms focuses on the first several gearmesh harmonics and their sidebands in the low frequency range, and is valid when the system can be suitably considered as linear-time-invariant (LTI), supported both experimentally and theoretically (i.e., modelling) by recent studies. The second group of studies instead look into the free-vibration-like system impulse responses caused by the impulsive events generated by the crack at high frequencies as abrupt changes in gearmesh stiffness and profile occur. This paper further explores this area and compares it to TE-based diagnostics. It is shown that TE is the best measurement to capture crack symptoms directly related to compliance and profile changes in the low-frequency region, whereas vibration is sensitive to high-energy impulsive events in the high-frequency range. As future work, the combination of these two measurements can be further exploited to develop more effective strategies for gear condition monitoring.

**Keywords:** gear cracks, transmission error, vibration, gear diagnostics.

### Introduction

Vibration has been the traditional gear diagnostic tool for decades. Among the multitude of studies on knowledge-based signal-processing for this application, it is possible to distinguish two main categories, based on their interpretation of the key physical phenomenon to target for the identification of tooth crack symptoms.

A first category follows a model in which crack-induced gearmesh stiffness and profile deviations due to crack-related plastic deformation are treated as standard “excitations” of the gearbox dynamics. This approach has been recently linked to the concept of static-transmission-error (STE). Under this modelling interpretation, the gearmesh compliance (inverse of stiffness)  $s(t)$  and geometric error  $\varepsilon(t)$  combine additively to give STE, with the average load  $F$ , to act as a weight:

$$STE(t) = F \cdot s(t) + \varepsilon(t) \quad (1)$$

STE is seen as an excitation source, including not only the gearmesh-periodic characteristics of a healthy gear-pair, i.e., time-varying gearmesh stiffness (TVMS) and design profile modification (e.g., tip-relief), but also crack-symptomatic shaft-synchronous components, i.e., a tooth-specific increase in compliance and tooth-specific deviations of the profile due to the crack being permanently open as a consequence of plastic deformations. Vibration then responds to STE based on the system's transfer function from the gear-contact point to the sensor location. Recent experimental studies have confirmed that such transfer functions exist, and that they can be identified and "removed" from diagnostic measurements such as encoder-based transmission-error (TE) [1] and vibration [2].

Signal processing techniques following this interpretation reasonably focus on the first several harmonics of gearmesh and their sidebands. It is important to clarify that both approaches targeting shaft harmonics and ones targeting shaft sidebands around the first few gearmesh harmonics can be justified by this STE-based model. The second [3] and perhaps more widespread interpretation, targeting sidebands, interprets crack-induced vibration as a variation from uniformity between the teeth that 'modulates' the regular meshing waveform, giving rise to sidebands spaced at shaft speed around the gearmesh harmonic(s). The first [4] sees instead each tooth's meshing as a separate event, but one repeated once per shaft revolution, with each tooth then 'responsible' for a certain distribution of shaft harmonics in the spectrum. A cracked tooth for example would be seen by the latter to give the most marked increase in shaft harmonics mid-way between the gearmesh harmonics, while the former would interpret this as a very sharp modulation of gearmesh, generating a wide distribution of sidebands (explained by the convolution theorem).

Another group of studies has instead focussed on the impulsive events happening at the meshing of cracked teeth, triggering impulse responses of the system at high-frequency. A study on the regularity of gearmesh sidebands across multiple gearmesh harmonics [5] has suggested that the two phenomena might interact, producing different effects in different frequency ranges, and thus limiting the valid frequency range when analysing the sidebands due to the presence of cracks. An intuitive interpretation of the two phenomena is a combination of a direct forced-response-like vibration and a series of free-vibration-like impulse responses happening due to abrupt changes in gearmesh stiffness and profile. The traditional forced-response model is a good approximation of what happens at low frequency, where the system can be considered almost linear-time-invariant (LTI), while the free-response approximation is a more suitable candidate for the high-frequency range, where resonances correspond to short-lived impulse responses, which can be considered separately between subsequent impulsive events. Separating between low- and high-frequency components is a first approach to distinguish between these two, which will be adopted in this paper. In future works, the analysis of the two across multiple domains, and especially in variable speed conditions, could provide further insights into the validity of this interpretation, given that impulse responses are defined in the time domain while variation in stiffness and profile are synchronous with the shafts.

Supporting the LTI interpretation of the low-frequency components of vibration are both an experimental work [2] and a recently submitted MSSP paper [6], where a model was used to demonstrate the possibility to extract the approximate linear-time-invariant frequency response functions (FRF) linking STE inputs (combining gear compliance and profile errors) with TE/vibration. A key finding was that these FRFs are indeed only valid at low frequency, i.e., in the range where the natural frequencies of the system are almost invariant to changes in the gearmesh stiffness. Intuitively, this is equivalent to saying that the system is only LTI as long as the time-varying gearmesh stiffness can be approximated as constant for the modal analysis

of the gearbox. The aforementioned study then mostly focussed on the low-frequency area where FRFs are valid, and proposed a model-based removal of the FRF to estimate the original STE, a much clearer and unbiased measure of crack severity. However, the higher frequency range beyond the validity of the LTI assumption remains an area of study for the diagnostics of cracks.

This paper further elaborates on this finding by focussing on the area where the FRF is no more valid and the assumption of an LTI system fails. In this high-frequency range, the modes themselves have natural frequencies which are time-varying, strongly affected by the change in instantaneous gearmesh stiffness, mostly due to the number of teeth in contact and the crack itself. A rigorous analytical discussion of such a topic is very complex due to the non-linearity of the model in question, where compliance, geometric errors and rotation/displacements of the gear interact in multiplicative ways. As such, this is beyond the scope of this paper, which aims more modestly at analysing the possibility to exploit this area for condition monitoring, and quantifying its value relative to the low-frequency zone.

### **Data description**

The data used in this paper was collected from the same experimental campaign described in [1]. In that experiment, other than the encoder measurements which were already previously mentioned, vibration data was also acquired using B&K4396 accelerometers mounted on the gearbox casing. Results in the subsequent section of the paper were calculated using 48 measurements, which were acquired at different combinations of 3 nominal speeds (10, 15 and 20 Hz), 4 nominal loads (0, 10, 15 and 20 Nm), and 4 crack severities (healthy, small crack, medium crack and large crack). Images of the cracks are also available in [1].

### **Signal processing of TE and vibration**

Conventional (zero-mean) TE signals were calculated using phase-time maps of the input and output shafts obtained from the 3600-pulses-per-rev encoder measurement of the respective shaft, scaled along the line of action to obtain the signals in  $\mu m$ . The phase-time maps were zeroed at an instant where a once-per-rev tacho pulse of the corresponding shaft rises. The TE signals were then order-tracked and synchronous-averaged with respect to the input shaft, since the crack was always on the driving gear. While the driving gear was installed such that the crack was always opening in mesh, it should be noted that since the cracks were actually slots artificially generated using electrical discharge machining (EDM), the crack effect was most visible in the zero-load TE signals, due to the release of residual stresses when the slot was machined, leading to a plastic deformation that appears as a “closing” crack, as opposed to a naturally-developed crack which is expected to have a plastic deformation appearing as an “opening” crack. To remove these profile effects and focus on compliance, nominally zero-load TE signals were subtracted from each corresponding non-zero-load TE signal, with the same speed and crack severity, resulting in “differential” TE signals that describe the load-dependent deflection components, plus the transfer-function effects. The gearmesh (GM) harmonics were then removed from these differential TE signals, and a band-pass filter was applied to retain the 3<sup>rd</sup> – 135<sup>th</sup> input shaft orders (inclusive). This passband would essentially remove any shaft-dominant effects (e.g., eccentricity) and high-frequency components beyond the 5th GM harmonic, so as to emphasise the effects of the crack-induced components. Finally, transfer-function effects were removed using an STE-to-DTE transfer function with a pole at ~580 Hz and damping ratio of ~1%, as defined and identified in Ref. [1].

A similar procedure of order-tracking and synchronous averaging (with respect to the input shaft) was also applied to the vibration signals. For vibration, in addition to the removal of GM harmonics, the first pair of sidebands around each GM harmonic was also removed, and a band-

pass filter was applied to retain the components at input shaft orders equivalent to the frequency range of 5-10 kHz, to focus on the crack-induced impulsive events in the high-frequency region.

## Results and Discussion

The vibration and differential TE results obtained after the aforementioned processing procedure are shown in Figure 1, Figure 2 and Figure 3, respectively for nominal loads of 20, 15 and 10 Nm. In each figure, the columns from left to right represent different nominal speeds 10, 15 and 20 Hz, whereas the rows from top to bottom represent increasing crack sizes. Observing all the differential TE results, it seems that the crack is always visible for medium and large cracks, while there is no obvious distinct indication in the small-crack cases. More interestingly, speed does not compromise the visibility of the crack effect, but this is not surprising since the crack manifests itself as an additive component, affecting only the low-order region, which has been sufficiently included in the passband.

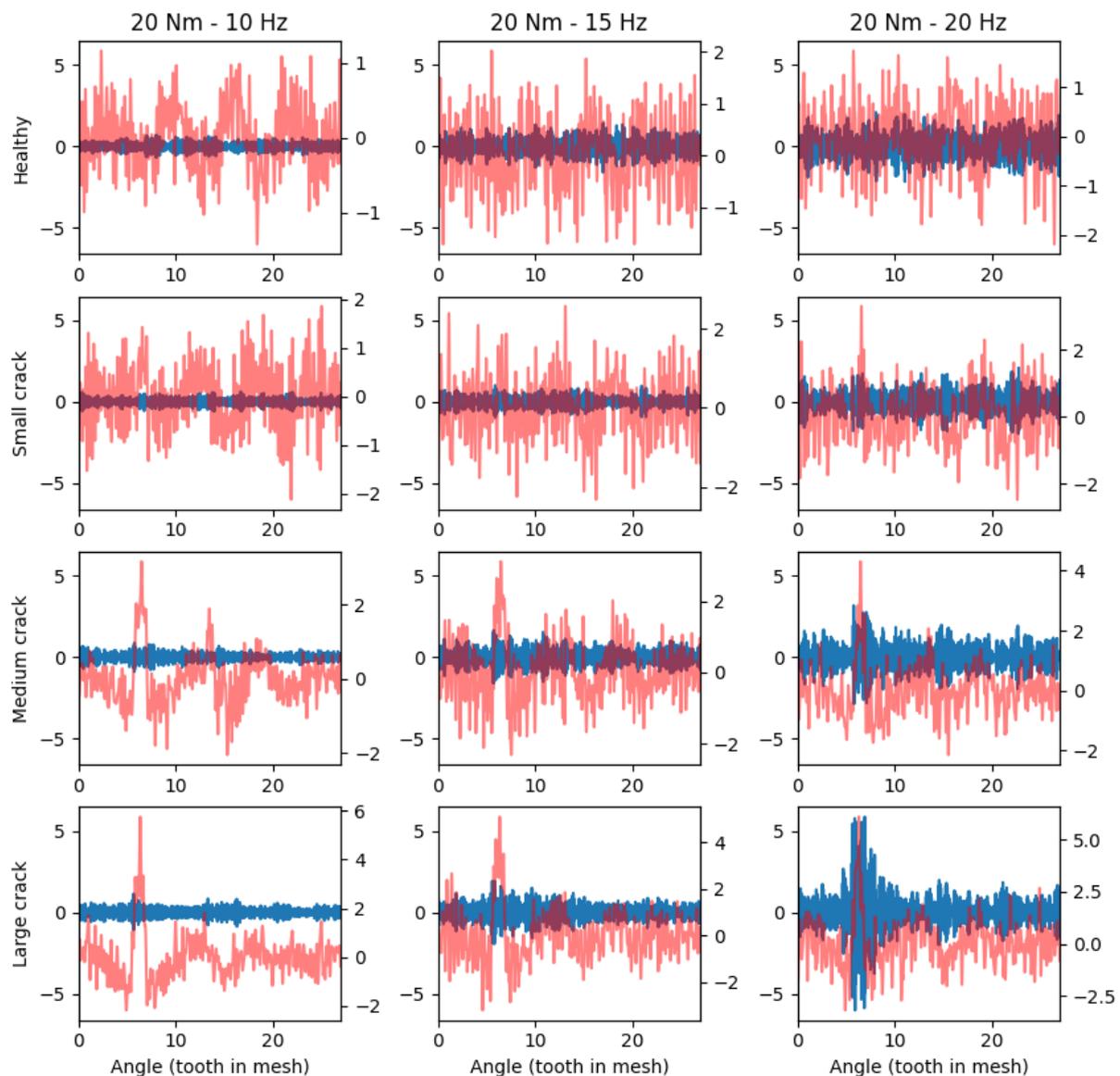


Figure 1. Vibration (blue, in  $m/s^2$ , left vertical axis) and TE (red, in  $\mu m$ , right vertical axis) after pre-processing. 20 Nm tests.

On the other hand, high-frequency vibration appears to only show the crack symptom at higher speed and load, in the form of impulse responses excited by the impulsive events due to the

crack in the higher frequency range. This phenomenon is most evident in the case of highest speed (20 Hz) and highest load (20 Nm), suggesting that these strong crack symptoms due to high-frequency non-linear effects are only visible if the sudden changes in stiffness and profile are fast and strong enough to trigger this high frequency resonant vibration. The results from the modelling paper [6] further support this interpretation, by showing that the range of the vibration pass-band (around 6 kHz) is the only one containing a resonance which strongly depends on the instantaneous value of the TVMS.

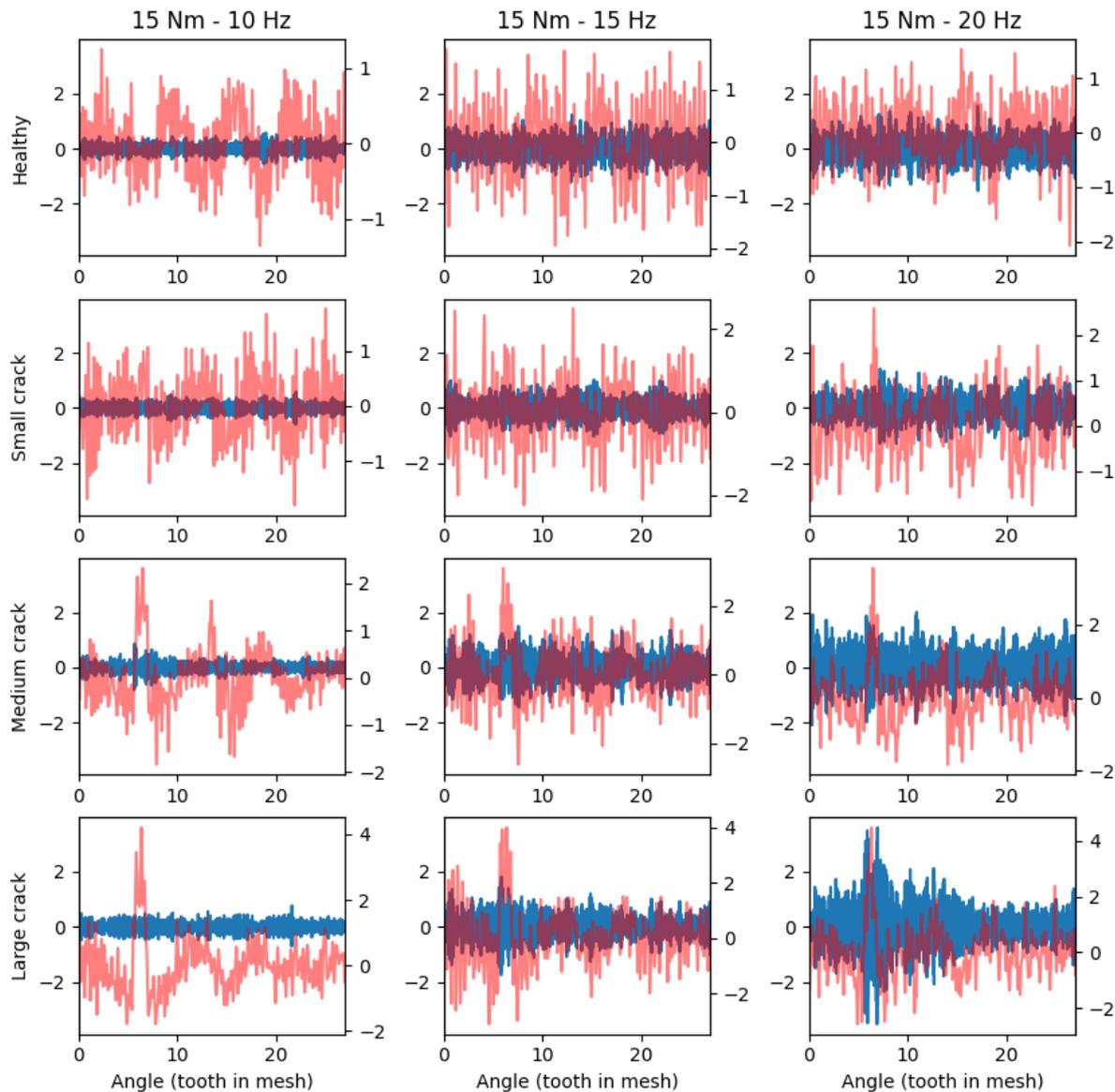


Figure 2. Vibration (blue, in  $\text{m/s}^2$ , left vertical axis) and TE (red, in  $\mu\text{m}$ , right vertical axis) after pre-processing. 15 Nm tests.

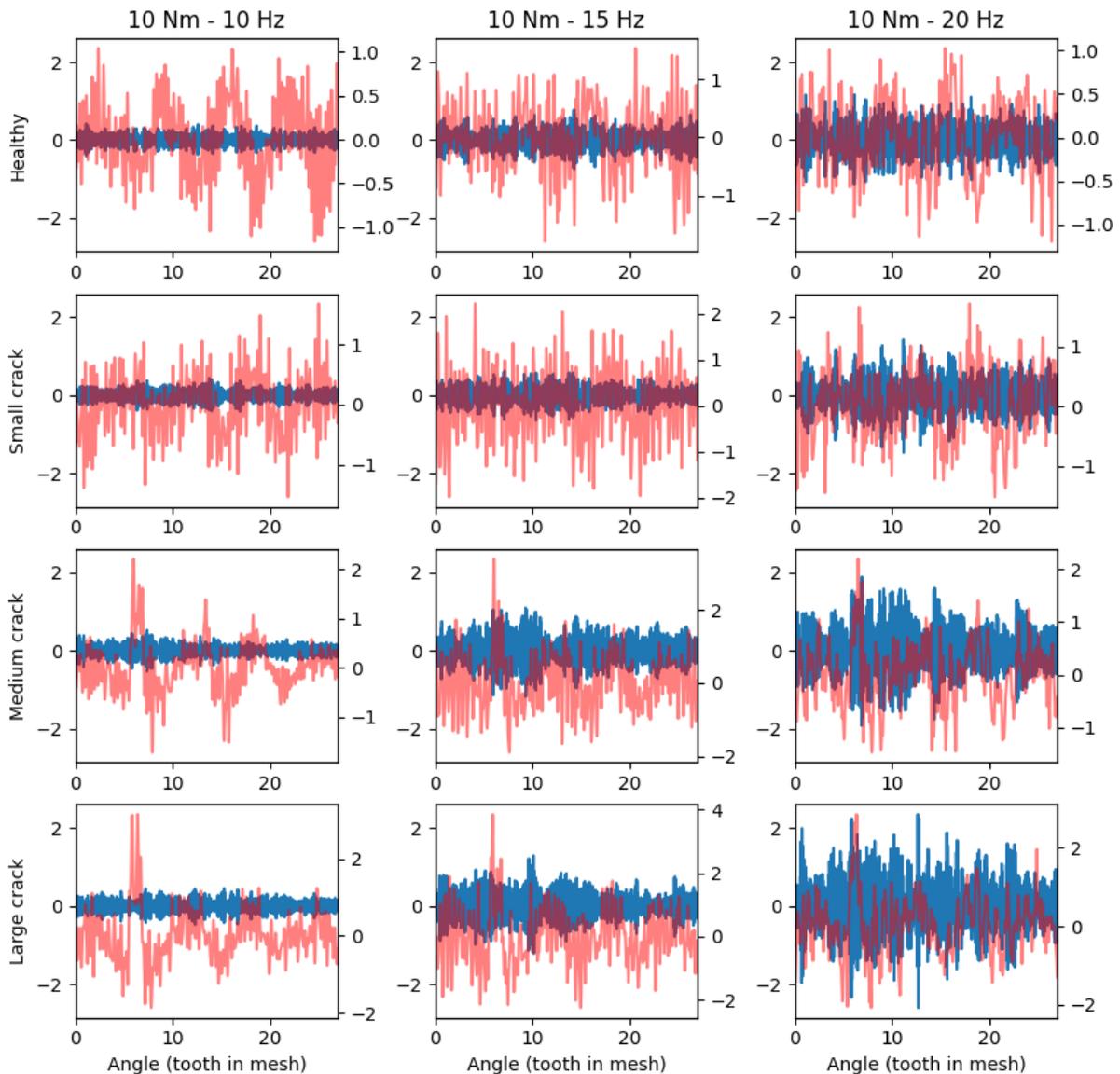


Figure 3. Vibration (blue, in  $\text{m/s}^2$ , left vertical axis) and TE (red, in  $\mu\text{m}$ , right vertical axis) after pre-processing. 10 Nm tests.

### Conclusions

This study has shown that TE and vibration are both sensitive to cracks and their severity, and that two different symptoms are captured by the two measuring techniques. TE is the most suitable technique to capture the low-frequency component of crack-induced symptoms, since it is directly linked to the corresponding compliance and profile changes. Vibration is instead sensitive to impulsive events which have significant energy in the high-frequency range. Both are deemed useful, and their combination for a more effective condition monitoring strategy remains a key avenue for future work.

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