Towards a better understanding of helical gears vibrations – dynamic model validated experimentally

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In order to simulate the vibration signature of gears, an accurate calculation of the gear mesh stiffness (GMS) is required. The time varying GMS, which is the main excitation that determines the dynamic response of transmissions' vibrations, is well understood for spur gears, but that of helical gears was less investigated. Although there is work dedicated to helical gears vibrations, a comprehensive analysis of their GMS compared to spur gears and their time and spectral domains have yet to be made. This paper deals with the dissimilarities and provides a better understanding of helical gears behavior, as they are a key component in many complicated and costly machines. With this new knowledge a more educated approach to diagnostics might be achieved.

The main difference between spur and helical gears is in the contact line pattern. In spur gears the contact line is parallel to the tooth's base and so calculating the GMS in any given moment is rather easy. Helical gears on the other hand have a diagonal line of contact which makes the moment applied by the meshing gear in respect to the tooth's base change along the tooth's width. To overcome this challenge a 'multi slice' method is utilized [1-4], in which the helical tooth is divided into many infinitesimally narrow slices which are treated each as a spur tooth. The total helical tooth stiffness is the sum of all those spur slices.

For the purpose of simulating the vibrations of helical gears a fourteen degree of freedom spur-teeth dynamic model [5] was upgraded to include helical gears a well. The dynamic equations and stiffness calculation were not changed and thus are discussed only briefly. The focus is dedicated to the modeling of the contact line using the multi slice method and other adjustments made to the model.

The challenge with the slice method is determining how many slices are in mesh at every given time, along with determining their mesh "height" (distance from the tooth's base). The solutions found in the literature are rather complicated and require knowing niche data about the gears, such as the transverse operating pressure angle, which are often not provided by the manufacturer. In contrast, the method suggested in this work is based on only a few common parameters such as the gears module, number of teeth and the involute profile.

The model was validated by an experiment conducted on a helical gearbox and recorded with a tri-axial accelerometer. The signals were compared in terms of their load and RPM dependency and exhibited a similar behavior, as can be seen in Figure 1. After obtaining a healthy baseline, a broken tooth case with three severity levels was studied. The fault severities were removal of 25% of the tooth's width in a diagonal line, removal of 50% and a missing tooth (Figure 1). This kind of diagonal material removal was chosen because when helical teeth breaks it happens in a pattern parallel to the contact line. The light and medium fault were challenging, but the missing tooth was seen clearly, mainly in the Kurtosis and Crest Factor of the difference signal. The statistical distances of the SA spectrum around the first and fifth GM harmony proved to provide better sensitivity, and showed clear detection of various fault severities, mainly in the tangential direction. A calculation of the Z-score index around the first GM even showed capability of ranking by fault severity (Figure 2).

Figure 1: Three level of fault severity. Left to right: removal of 25% of the tooth's width, removal of 50%, and a missing tooth.

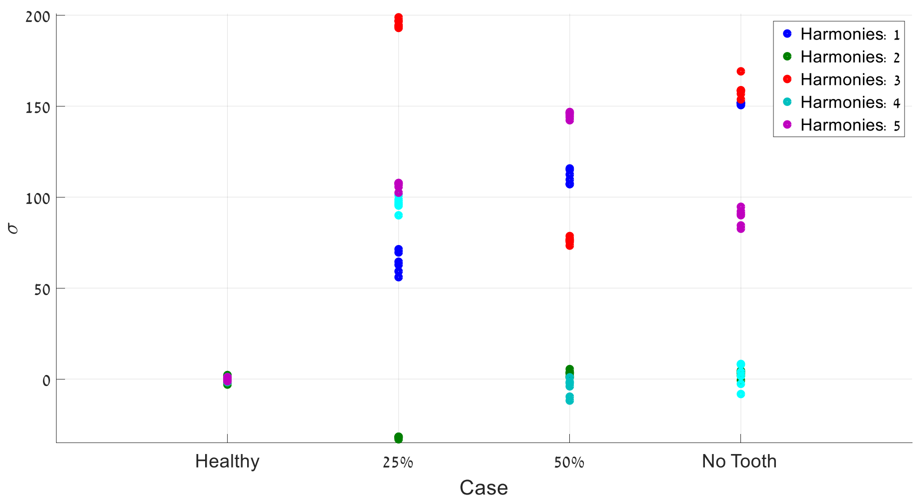

Figure 2: The Z-Score index for each GM harmony. Notice the first harmony which shows fault ranking and the fifth, which shows detection even at the smallest fault.

Keywords: Helical gears, Dynamic model, Multi-slice method, Broken tooth.

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