Spiral-Bevel Gear Noise: An Approach to Solving In-Field Issues

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Introduction

Gear noise is a common evil any gear manufacturer must live with. It is often low enough not to be a major problem but, at times, gear whining may appear and then, tracking the source and, especially, curing the ill can be tricky at best.

A large number of publications, too numerous to be listed here, have been published on gear noise; the work of Endo and Sawalhi is a typical example (Ref. 1). However, once the gearbox is in production, such models are of little use.

The Gleason Works also markets advanced software and machinery systems (Ref. 2) to run gear sets and determine at which positions the best behavior is found. Such systems are well-adapted to large-batch manufacturing.

This paper presents an approach that allows identifying the source of the noise and offers avenues in correcting the issue using more limited means, which is typical of small-to-medium- size gear manufacturers.

Problem Statement

A reversing gearbox comprising a spiral-bevel gear set has been in production for several years. Noise had not been mentioned by the customer until recently, when different testing procedures were introduced. Gear stall torque is ~80 Nm, but testing rather replicates nominal conditions.

The pinion convex flank can be operated at different torques and rpms without undue noise, but the pinion concave flank exhibits annoying noise levels around 1,400 and 3,200 rpm.

Table I lists the main specifications of the gear set. Both members are hard finished using CBN grinding wheels of different point widths.

Table I Spiral bevel gearset specifications

	Pinion	Gear
Z	9	59
Module [mm]	2.5	
Spiral angle [0]	35	
Cutting Process	Duplex Helicall	Non generated
Finishing	Grind — CBN GW	Grind — CBN GW
Max RPM	3,500	
Nominal Torque [Nm]	3	19.67

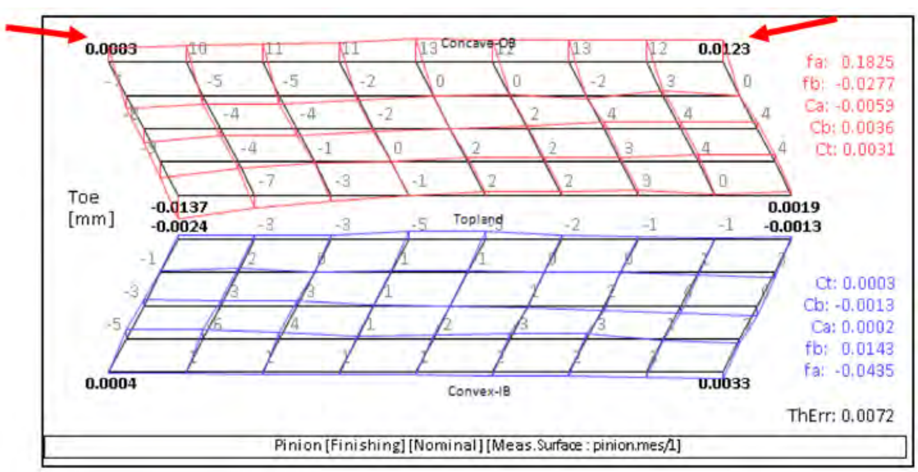

Figure 1 Pinion CMM results.

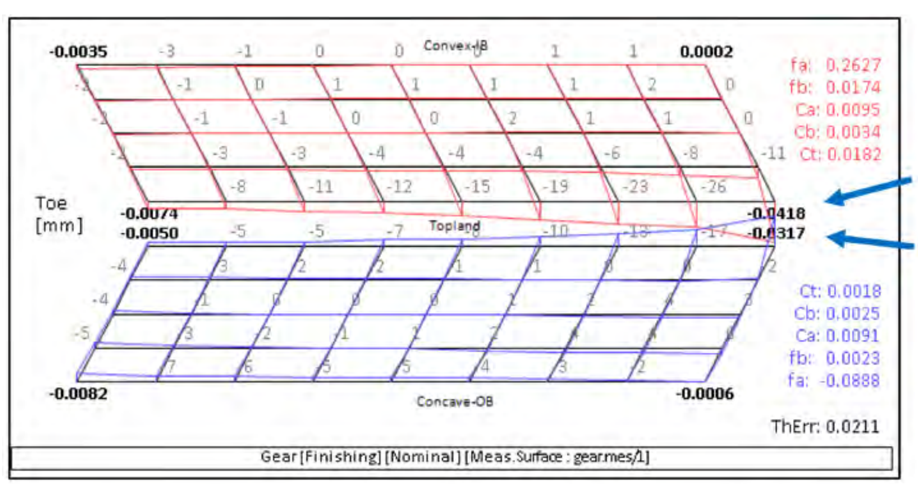

Figure 2 Gear CMM results.

Preliminary Assessment

The pinion and gear were measured on a Wenzel CMM, giving the results shown in Figures 1 and 2. Both the pinion and gear show overall minor deviations relative to the nominal surface, but we note that on the concave flank, (Fig. 1), the pinion exhibits some positive profile curvature near the root (red arrows), which is expected to cause premature contact entry. The gear, Figure 2, exhibits tip relief towards the heel (blue arrows), which is usually beneficial.

Because of manufacturing tolerances on both the gearbox housing and the gear teeth, the pinion and gear were shimmed in order to provide the visually, and therefore subjectively, best contact patterns.

Actual contact positions in the gearbox are (definitions shown in Fig. 3):

- E = -0.007 mm, caused by gearbox manufacturing tolerances
- P = -0.019 mm, to obtain an acceptable contact pattern position
- G = +0.005 mm, to obtain the correct backlash

The contact patterns measured in the above positions are shown (Fig. 4). Contact patterns on both gear flanks are fairly well centered lengthwise on the tooth, and appear to cover most of tooth depth.

Using the *HyGEARS* (Ref. 3) software and accounting for the measured errors shown in Figures1 and 2, the contact patterns shown in Figure 5 are obtained.

Although the calculated contact pattern on the gear concave flank is quite similar to that of Figure 4, the contact pattern calculated on the gear convex flank differs substantially from that shown (Fig. 4).

The contact patterns were therefore re-measured — this time using a much thinner marking compound to better reveal the boundaries. The results are shown in Figure 6 where, clearly, the calculated contact patterns now correlate very well with the measured ones shown in Figure 6.

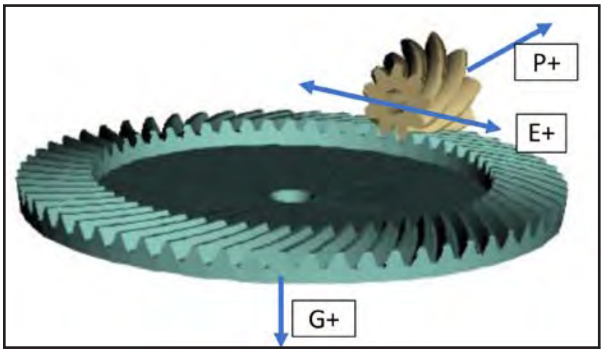

Figure 3 E-P-G definitions.

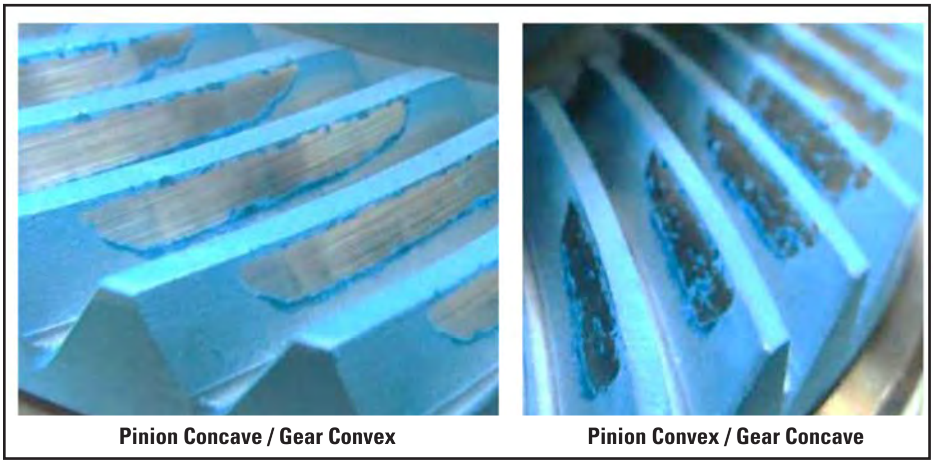

Figure 4 Measured contact patterns: thick marking compound.

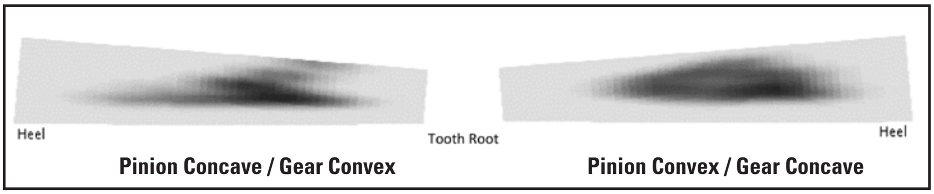

Figure 5 Calculated contact patterns.

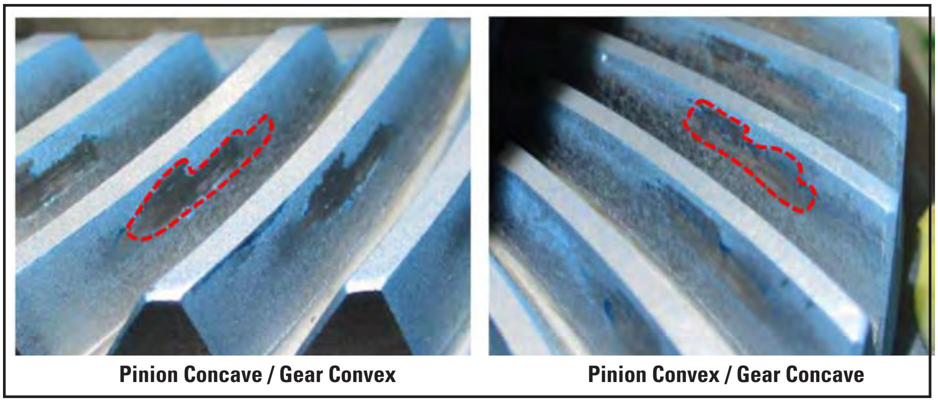

Figure 6 Measured contact patterns: thin marking compound.

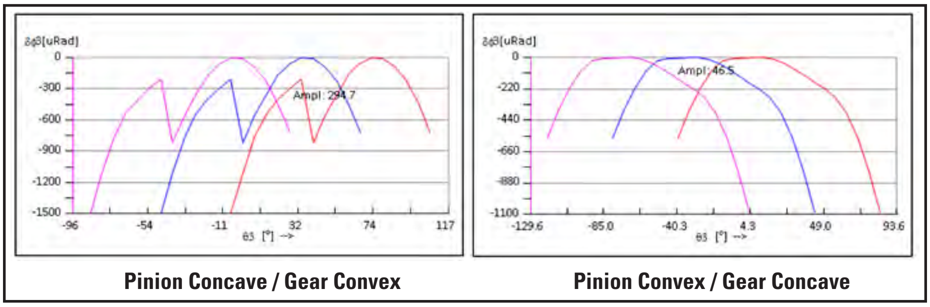

Figure 7 No load transmission error — calculated from the measured surfaces.

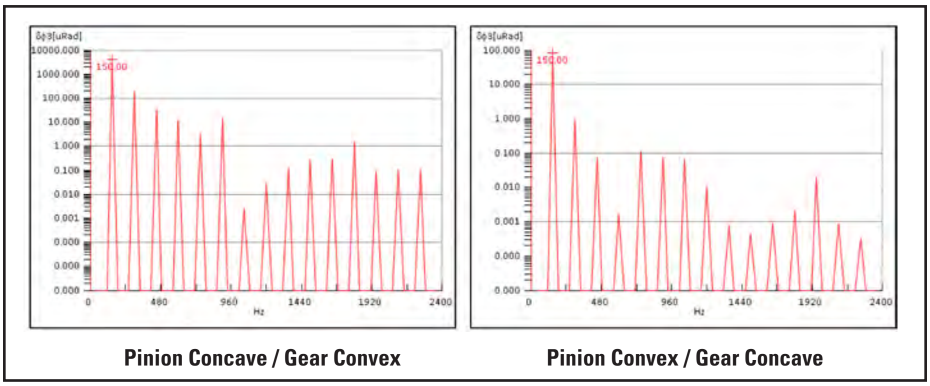

Figure 8 No-load TE FFT — calculated from the measured surfaces.

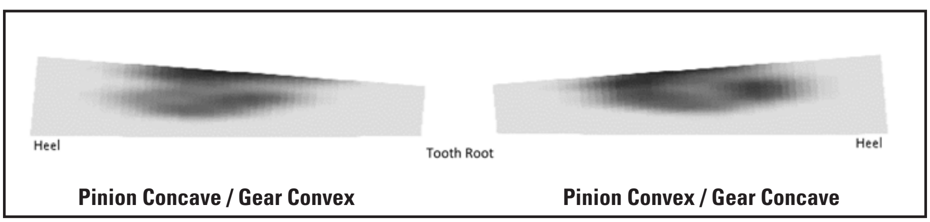

Figure 9 Improved contact patterns — calculated using the measured surfaces.

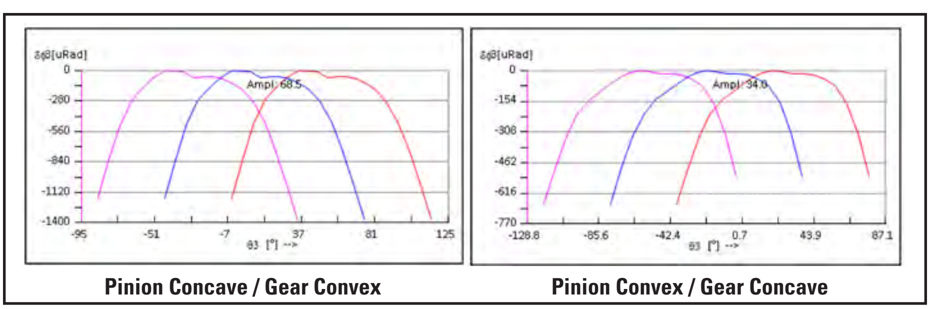

Figure 10 Improved TE — no-load — calculated using the measured surfaces.

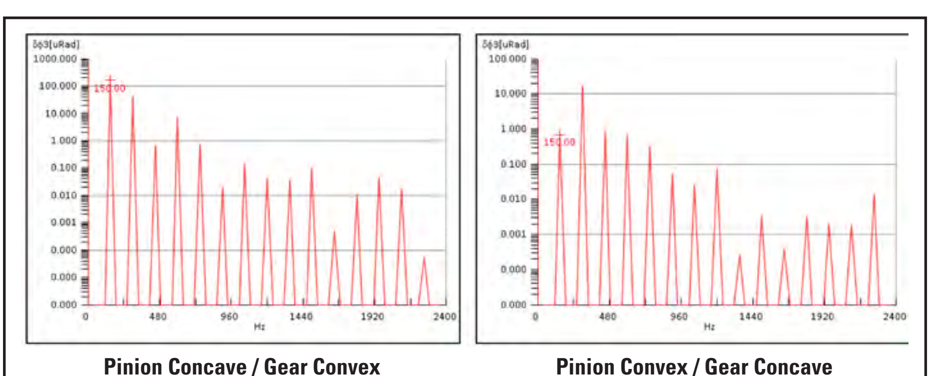

Figure 11 Improved no-load TE FFT — calculated using the measured surfaces.

Transmission Error and Gear Noise

The good correlation between the measured and calculated contact patterns shown in Figures 5 and 6 indicates that the tooth contact analysis (TCA) model evaluates correctly the no-load kinematics (Ref. 4). The resulting transmission error curves (TE) for 3 consecutive meshing tooth pairs — pink, blue, orange — are shown (Fig. 7).

The TE curves on the pinion convex flank (right figure, below) are of convex shape, continuous, and with transfer points (TP) at a depth of ~46 µ*Rad* — a value frequently found in gearsets of this module; rotation proceeds from left to right in the graph.

By contrast, the TE curves on the pinion concave flank, although also of generally convex shape, show a much deeper TP at 295 µRad, which is likely to cause a sharp acceleration when motion is transferred from one tooth pair to the next. Rotation proceeds from left to right in the graph.

Figure 8 shows the FFT of the TE curves displayed in Figure7. Clearly, based on the amplitude of the 3–4 first harmonics, the pinion concave flank (left below) is expected to be noisier than the convex flank (right below) which, again, correlates with what was noted.

Of course, the graphs shown in Figure 8 do not indicate at which rpm, or torque, noise is to be more prevalent since the actual gearbox, bearings, shafts, etc. are not modeled. However, they indicate that there is a potential noise issue — which is validated in practice.

Improving Contact Pattern and Gear Noise

There are basically three solutions to the situation depicted above, which derives from the measured manufacturing errors:

a. Apply closed loop to eliminate the profile curvature at root noted on the pinion concave flank — Figure 1.; of course, this means new pinions and that the gearboxes already in use are to be recalled — a very expensive operation at best.

b. Measure and correct the CBN grinding wheel used for pinion finishing, concave flank; the same gearbox recalls and cost issues as in a) are expected.

c. Modify the pinion and gear installation in the gearbox to try and improve the contact patterns and TE.

Of course, solution c), if a practical combination of pinion and gear shimming can be obtained, is by far the easiest and least expensive as it involves a simple operation that can easily be performed in the field.

Again, using the gear simulation model, the following combination of EPG was found; note that only P is changed.

- E = –0.007 mm (imposed by the manufacturing tolerances on the gearbox housing)
- P = +0.173 mm
- G = +0.005 mm (no change)

The resulting no-load contact patterns, TE and FFT, are shown in Figures 9–11. In particular, for the pinion concave flank, Figures 10–11 show a dramatic change when compared to Figures 7–8.

Of course, other P and G combinations could be used, but the above-selected combination involving pinion shimming only results in lower efforts and costs.

Assessment of Improved Contact Pattern and Vibrations

Figure 12 below shows the contact patterns measured in the modified EPG positions while Figure 13 shows the calculated contact patterns in the same gear tooth orientation to ease comparison. Clearly, the predicted behavior is obtained.

The gearset was run at different torque levels and rpms in the original, and improved EPG positions and vibrations levels were recorded. The results, appearing in Figures 14–15, show a significant improvement in vibration levels when shimming the pinion position from P = –0.019 mm to P = +0.173 mm; not only on the pinion concave flank, which was originally the problem tooth flank, but also on the pinion convex flank, which was originally found as acceptable.

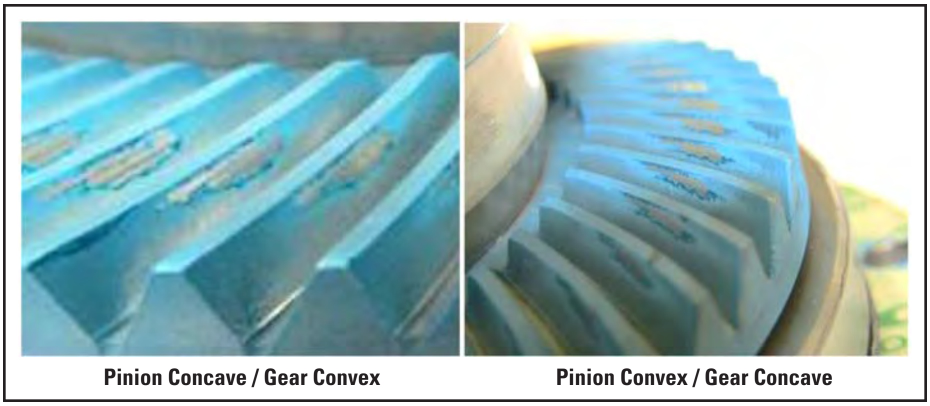

Figure 12 Measured contact patterns P = +0.173 mm.

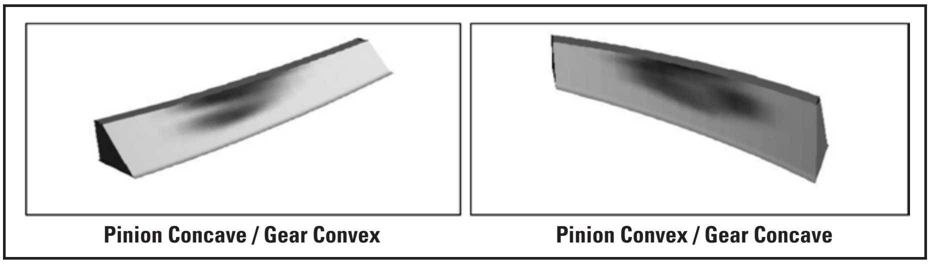

Figure 13 Calculated contact patterns P = +0.173 mm.

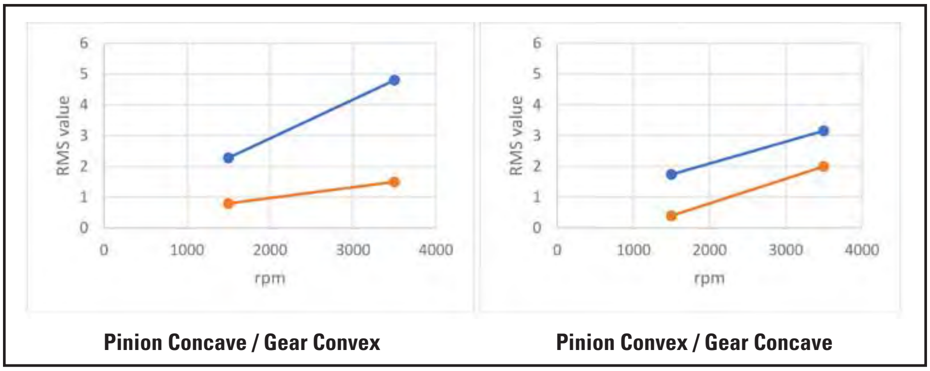

Figure 14 Mechanical vibrations: 0 Nm pinion torque P = –0.019 mm P = +0.173 mm.

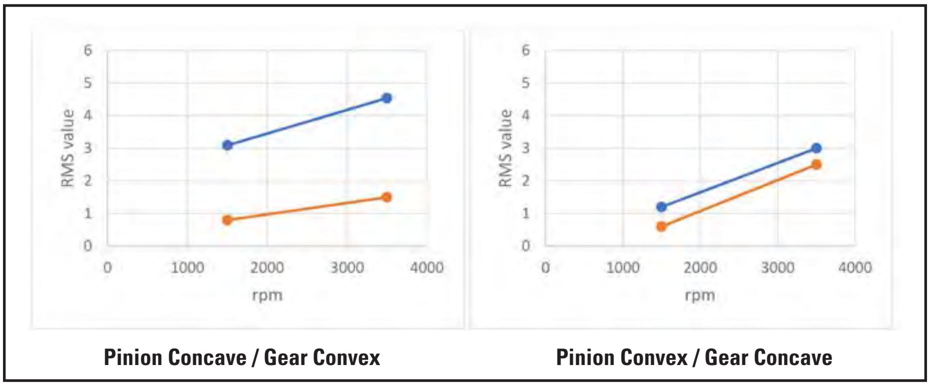

Figure 15 Mechanical vibrations: 3 Nm pinion torque P = –0,019 mm P = +0,173 mm.

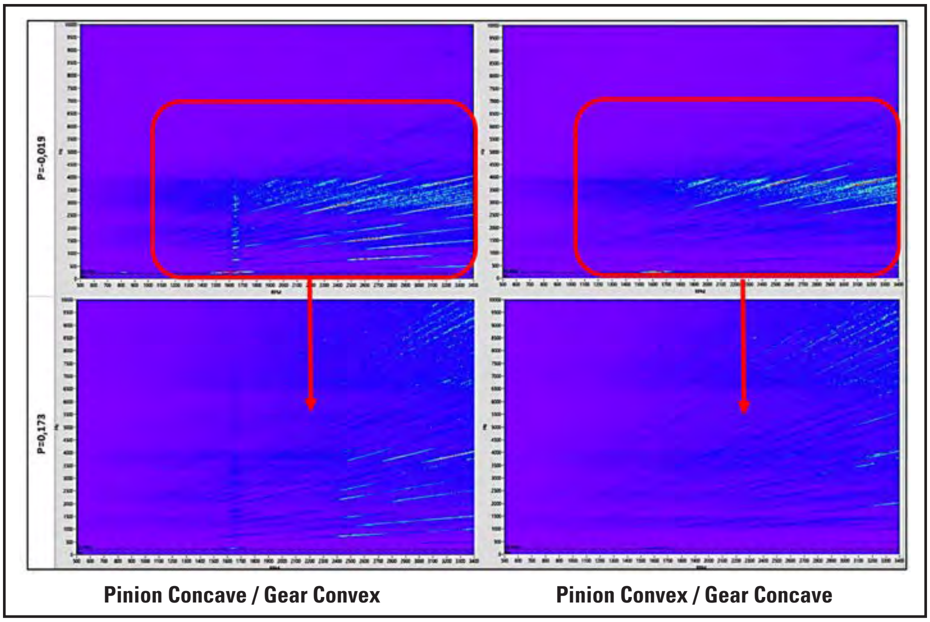

Figure 16 Waterfall plots: 0 Nm pinion torque.

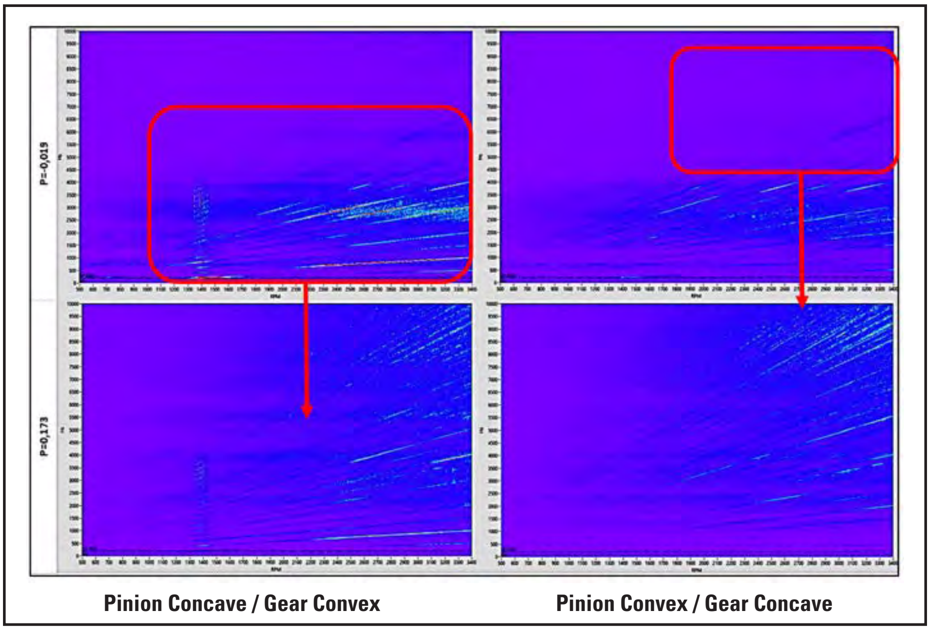

Figure 17 Waterfall plots: 3 Nm pinion torque.

Figures 16–17 show the waterfall plots for the original and modified operating positions for, respectively, 0 Nm and 3 Nm pinion torque. Of course, 0 Nm pinion torque means that no braking other than friction is applied at the output. rpm on the horizontal axis ranges from 500 to 3,400, whereas frequency in Hz, on the vertical axis, ranges from 0 to 10,000.

Both graphs show the same dramatic reduction in noise for the pinion concave flank operating in the modified position. On the pinion convex flank, Figure 17 shows a slight increment at frequencies above 5,500 Hz for 3 Nm pinion torque.

Overall, noise improvement on the gearbox renders it *more than acceptable* to the customer, which was the aim with this analysis.

Conclusion

Gear noise, which results from mechanical vibrations transmitted to the gearbox housing and environment, is a problem that is often difficult to cure. While highly evolved mathematical models of gear trains, including shafts, bearings and gearboxes, are available, they are of little use to the manufacturer once the gearbox and components are in production and noise is experienced at certain rpm and torque levels.

This paper presents an approach focusing on the use of tooth flank topography measurement (CMM) data coupled to spiral bevel gear modeling software to analyze how a problem gearset operates in a given installation. It is shown that the software calculated contact patterns closely match those actually measured on the gearset, which leads to the conclusion that the calculated transmission errors are also close to what is actually occurring.

This leads to the use of the software to try and improve the contact patterns by shifting the pinion and gear positions, an operation easily achieved in practice through shimming, thereby improving transmission error and allowing in-field correction of the noise problem.

A sample 9x59 duplex helical pinion / non-generated gear spiral-bevel gearset is used to validate the approach. While the pinion convex tooth flank could mesh at different torque levels without undue noise up to 3,400 rpm, the pinion concave flank exhibited significant noise at around 1,400 and 3,200 rpm.

The spiral bevel gear modeling software was used to calculate new pinion and gear installation positions, based on the CMM data, in order to improve on both the contact patterns and transmission error.

Results confirm that the improved contact patterns actually resulted in significantly lower noise levels on *both* tooth flanks, thereby allowing in-field correction of the noise problem at a very low cost.

While the demonstrated methodology is applied to a spiral bevel gearset, it is applicable to any type of gear to allow the identification of the noise source, and is easily used with all types of bevel gears since their respective mounting distances directly affect contact pattern location and transmission error.

References

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