Effects of Compliant Geartrains on Engine Noise and Performance

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Geartrain rattle has been an issue in heavy-duty diesel engines for a number of years, affecting not only noise but also engine performance and durability. A project at Caterpillar that completely removed the front geartrain from the engine (all ancillary devices and drove the camshaft from the rear) demonstrated a large noise reduction potential in heavy-duty diesel engines. This work prompted an effort to identify production viable devices that would reduce engine noise through geartrain compliance without affecting performance or the engine envelope. This article shows that compliant geartrains effectively reduce gear rattle (and hence noise), increase the life of geartrain components, and may even improve the performance of the engine.

Within the heavy-duty diesel engine community, gear rattle has been an issue for many years. Gear rattle is the phenomenon during which gear teeth come out of mesh and are forced back into mesh by a backside tooth impact. More complete descriptions of gear rattle, especially the modeling of the phenomenon, are described in the literature.[1,2] Note, though, that while the automotive industry deals with gear rattle at certain conditions (normally low idle), mechanically injected heavy-duty diesel engines usually exhibit gear rattle at most operating conditions if not properly dealt with. This has been further exasperated by the need to reduce gaseous emissions in diesel engines, which has been partially accomplished with elevated cylinder peak pressures and injection pressures.

A previous unpublished Caterpillar project demonstrated a large noise reduction by completely removing the front geartrain from a heavy-duty diesel engine (all ancillary devices and drove the camshaft from the rear). Thus, Caterpillar has focused its efforts on geartrain noise in the past few years. Traditional gear rattle literature suggests the following for gear rattle reduction: modified gear positioning, modified phasing, mass variation of gears, mechanical pre-load system, viscous dampers, changed number of teeth, modified engagement factor/helix angle, or modified backlashes.[1,2] While these methods can be useful for fine-tuning geartrains, larger changes are necessary for significant noise reductions.

Several options were considered that parallel the investigations of Zhao.[3] Specifically, the focus was on introducing additional compliance in the geartrain while maintaining cam to crank timing. Croker's relationship for rattle in heavy-duty diesel engine geartrains[4] (Figure 1) shows a nonlinear jump where increased backlash quickly becomes beneficial. Tolerance restrictions in heavy-duty diesel engines can make it very costly to take advantage of tight backlash; therefore large backlash is the desirable region of operation. This concept of increased backlash was combined with nonlinear, frequency-dependant isolation principles to reduce rattle. While the specifics of the actual hardware remain undisclosed, this article will discuss the effects of this compliance on engine noise, gear loads and engine performance.

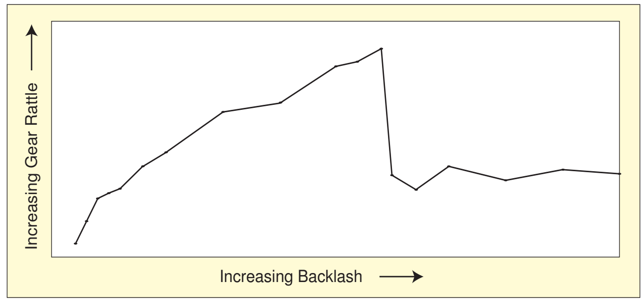

Figure 1. A replication of Croker's graph.[4]

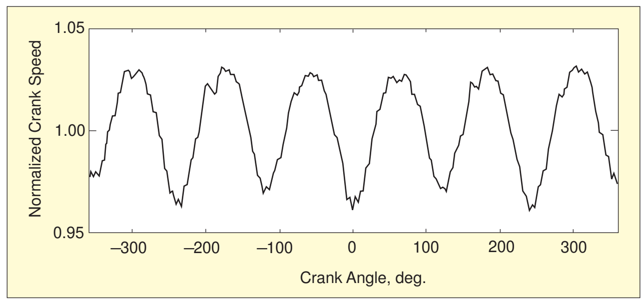

Figure 2. Typical crankshaft speed variation at a lug condition, normalized.

Geartrain Isolation

Due to the impulsive nature of reciprocating engines, crankshafts do not rotate at constant speeds, but instead often vary from 3-5% within one cycle of rotation. Additionally, the mass-elastic system of the flywheel-crankshaft-damper has torsional modes which, when excited, add to this speed variation as well. See Figure 2 for a typical time history of crank speed at a lug condition. Thus, for the first engine to be quieted, devices were devised to isolate the geartrain from this rattle-producing event. One such device effectively places a nonlinear, tuned torsional spring between the crankshaft and crank gear. The geartrain in Figure 3 for Engine A shows the location of the crankshaft gear, which was where this device was located.

Even though the geartrain was now isolated from the crankshaft torsionals, rattle still existed in the geartrain. The impulsive nature of mechanical fuel injection systems also causes the camshaft speed to vary 3-6%. This speed variation was causing the rest of the geartrain to rattle. To address the issue of camshaft induced geartrain rattle, another device was needed to isolate the geartrain from cam-induced excitation. This device isolated the rest of the geartrain from these cam torsionals. Note that further torsional activity comes from the pumps, but this study found that these pump torsionals were negligible in this geartrain.

The layout of the geartrain for the second engine on which a compliant device was pursued is shown in Figure 4 (Engine B). Note the significant differences between the two geartrains. Instead of separating the cam and crank by four meshes, this engine is only separated by two. Additionally, due to some size restrictions at the crankshaft, similar devices from the previous engine could not be applied. Instead, new devices were constructed that replaced both of the idler gears in mesh with the crank gear. The cam-to-crank idler gear isolated the camshaft and crankshaft from each other, as well as isolating the pumps from the crank and cam torsionals. The other idler gear isolated the pumps from the crankshaft torsionals. Although the method by which compliance was achieved for these devices differed from previous geartrain treatments, the general trends of the effects of isolation remained the same.

Based on a paper presented at IMAC-XXIII, the 23rd International Modal Analysis Conference, organized by the Society for Experimental Mechanics, Inc., Orlando, FL, January 31-February 3, 2005.

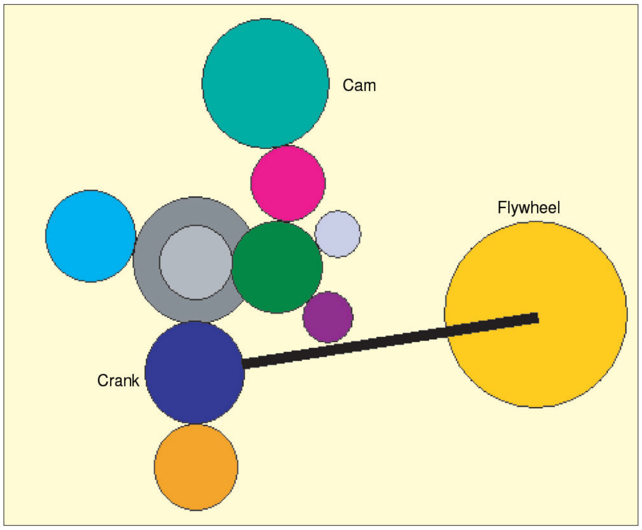

Figure 3. "Engine A" geartrain schematic.

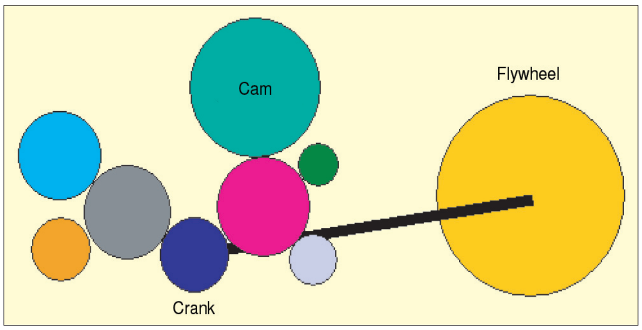

Figure 4. "Engine B" geartrain schematic.

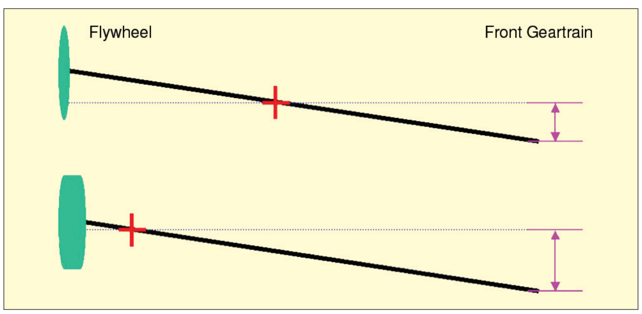

Figure 5. Effective driveline inertia effects on torsional modes.

A second, interdependent issue related to geartrain rattle is driveline inertia. Caterpillar's engines are used in numerous machines including, but not limited to, on-highway trucks, generator sets, industrial tools, marine applications, and construction equipment. Each of these applications can utilize different engine and driveline components, which will affect the crankshaft dynamics as the mass-elastic system is modified. When rigid body motion of the crank dominates the front crank torsional activity, such as when the engine idles, additional inertia reduces speed variation. This is validated by the basic dynamics equation, showing inertia and angular acceleration to be inversely proportional.

$$\Sigma T = I\alpha \quad (1)$$

where
T = Torque
I = Inertia (Polar Moment)
α = Angular Acceleration

At most other engine operating conditions, the dynamic torsional response is dominant and an increase in flywheel inertia will increase speed variation (motion) at the front of the crankshaft where the geartrains are located (for engines with front geartrains, as are both Engines A and B). In this case, the additional effective flywheel inertia moves the node of the first torsional mode toward the rear of the engine, typically increasing the overall front motion. Any simple lumped-mass calculation validates these results. Figure 5 demonstrates this phenomenon.

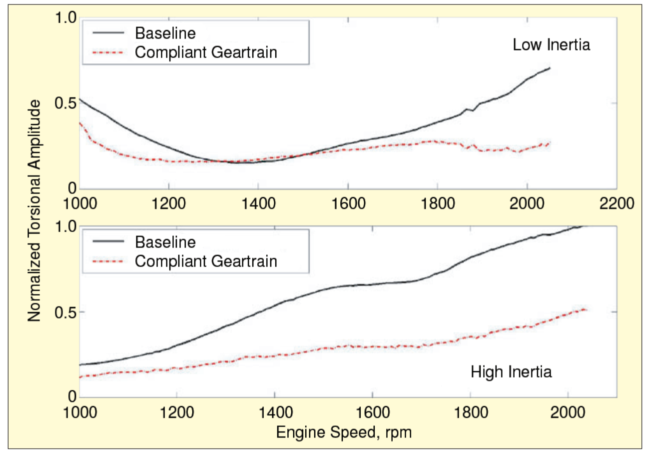

Figure 6. Torsional isolation by a compliant geartrain at two inertia levels.

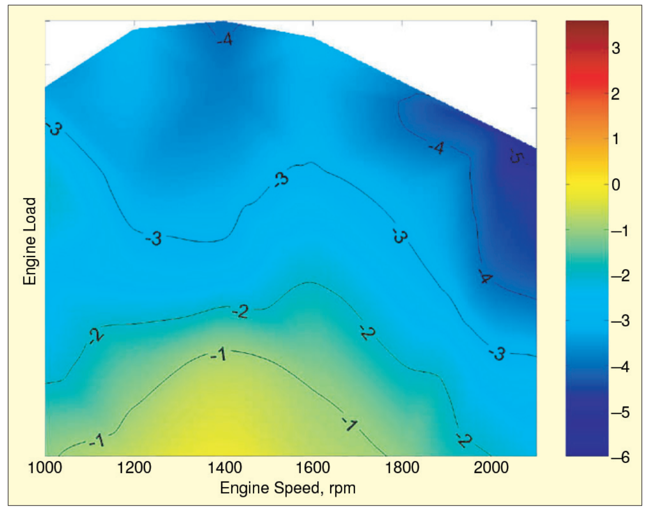

Figure 7. Overall engine sound power reduction – "Engine A," low driveline inertia.

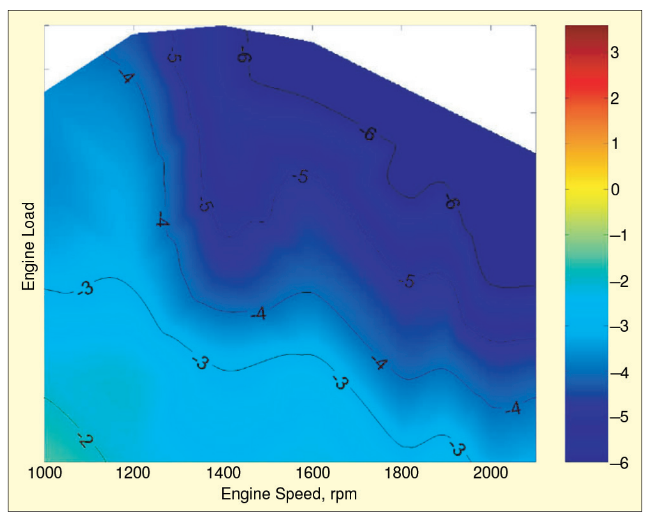

Figure 8. Overall engine sound power reduction – "Engine A," high driveline inertia.

Typical results of the isolation can be seen in Figure 6. Note that the upper plot is for a lower driveline inertia while the lower plot is for a higher driveline inertia. The plots show the normalized torsional response of the summation of the domi-

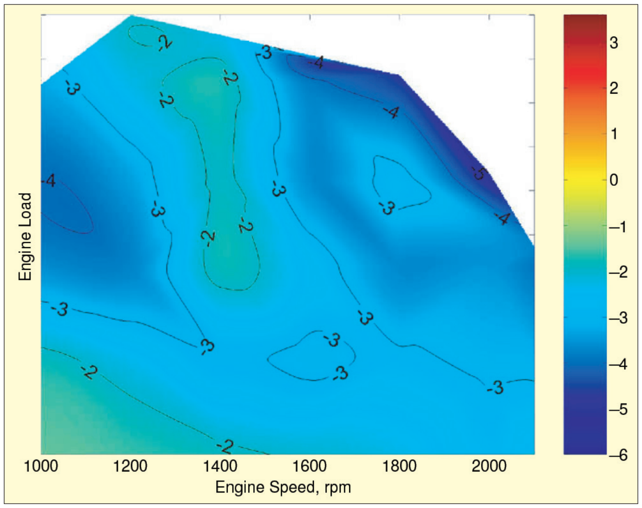

Figure 9. Overall engine sound power reduction – "Engine B," low driveline inertia.

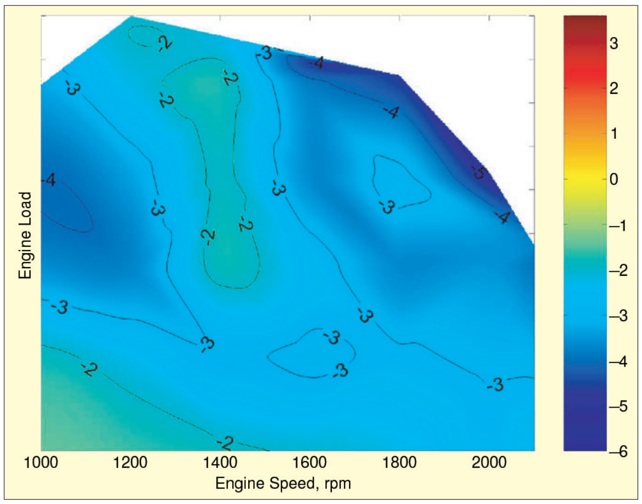

Figure 10. Overall engine sound power reduction – "Engine B," high driveline inertia.

nant engine torsionals (1.5, 3.0, 4.5 and 6.0 for an inline 6-cylinder engine). The fact that the benefits of the compliant geartrain were amplified by increased driveline inertia at higher speeds plays a large role in quantifying both the noise benefits and the gear load reductions.

Effects on Overall Engine Sound Power Levels

The effects of compliant geartrains on engine noise were studied on the two engines in Figures 3 and 4. Zhao[3] indicates that engines with more gear meshes tend to have more gear noise. This means that Engine A contains more gear rattle noise reduction potential. Figures 7 and 8 show these reductions at both high and low driveline inertias. Note that just as torsional activity of the front geartrain increased with speed above 1600 rpm, the measured noise reductions became much larger there as well. Additionally, it can be seen that the driveline inertia has a large effect on the sound power levels as well as the front geartrain activity.

Engine B also showed significant reductions, although not as large as Engine A. Figures 9 and 10 show these noise reductions. When 1/3 octave and narrow band data from both engines were analyzed, large reductions occurred in the broadband levels while the prominent tonals at each 1/2 order remained relatively unchanged. This proved consistent with the knowledge that gear rattle produces broadband noise.

Effect on Gear Loads

A common dynamic measurement at Caterpillar is to calculate the mesh torque, which is defined as the instantaneous

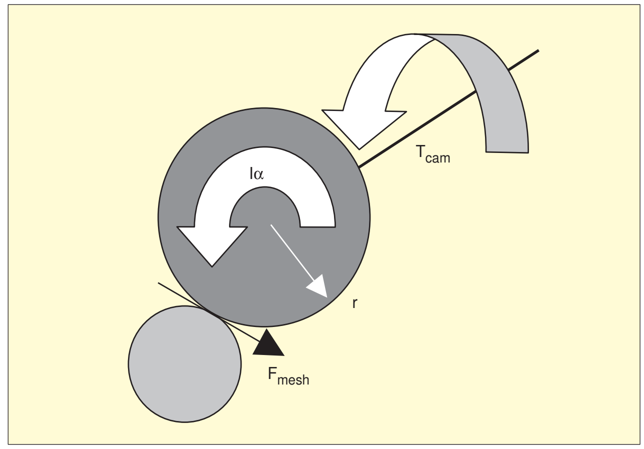

Figure 11. Free body diagram of a cam gear for calculating mesh torque.

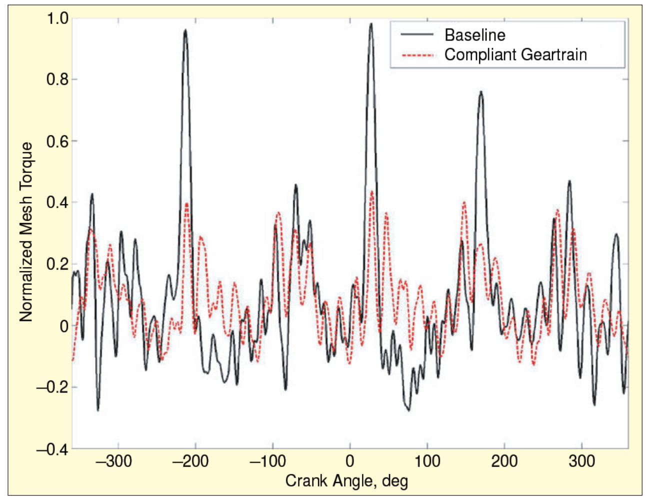

Figure 12. "Engine A" cam mesh torque comparison.

torque on a given gear independent of angular position. Figure 11 shows a free-body diagram of the gear. At the camshaft, for example, the shaft torque T_{cam} and gear acceleration α would be measured, and the inertia I calculated from geometric definitions. By simply applying Equation 1, the mesh torque can be calculated.

The cam mesh torques on Engine A were dominated by impacts to the driving side of the gear teeth (one-sided contact), but still were composed of large, impulsive spikes. Some of the spikes were up to 8× the mean torque. Figure 12 shows that the magnitude of these spikes was substantially reduced by the compliant geartrain over one engine cycle. In this case, the maximum value was reduced by almost 60%. Despite the reduction, the injectors and valves are still getting the necessary torque for operation, as the mean value of the two traces is the same.

Engine B, with its different geartrain, had a very different baseline mesh torque signature. The mesh torque shows definite activity against both the front and backsides of the gear teeth (two-sided contact), resulting in peaks as much as 12 times the mean level. The compliant devices again reduced these peaks significantly, as can be seen in Figure 13. As with Engine A, the positive peak reductions were as much as 60%. Additionally, the backside collision magnitude was reduced by almost 90%. Again, the mean value of the two traces is the same.

Taking the concept of a less impulsive operation for a gear further, the ideal geartrain can be extrapolated to operate at a constant value – exactly the average value needed for the camshaft to operate the valves and injectors. This would be the ultimate goal of a compliant geartrain, although factors such as cost, reliability and durability require that reasonable compromises be made.

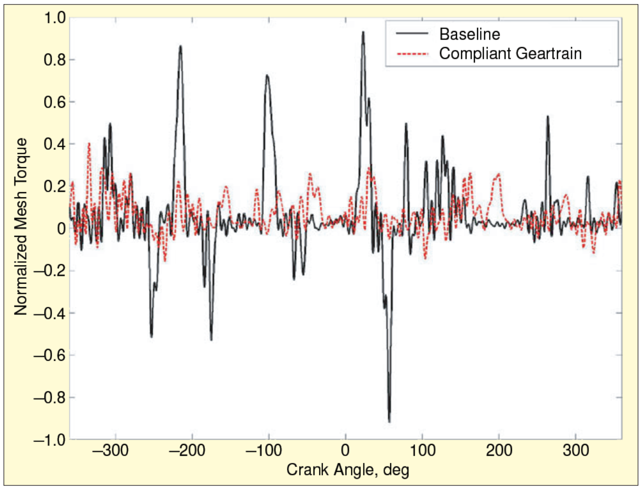

Figure 13. "Engine B" cam mesh torque comparison.

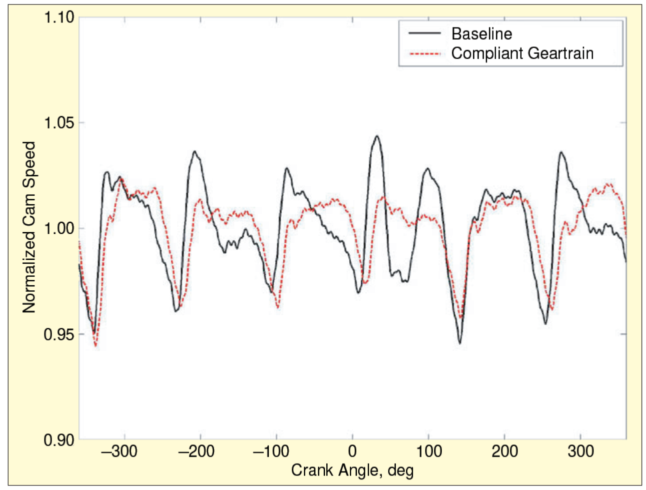

Figure 14. "Engine B" average camshaft speed.

In either case, the effects on gear life are obvious. The significant reductions in gear loads can lead to two options (or some mixture of both). First, the size and/or quality of the gears can be reduced, thus reducing manufacturing costs. Additionally, failures will be reduced (or eliminated depending on the nature of the failure) with such substantial reductions. Such financial gains become important in creating a business case for compliant geartrains, as they themselves are not without substantial cost.

Effect on Engine Performance

In validating the effect of compliant geartrains, the issue of gaseous emissions was raised. In studying the effects, two significant results were found. First, as expected, a slight timing loss was measured. This loss is easily calculated knowing the mean loads of the various parasitic loads (injectors, valves, pumps, etc.). This was an important factor that influenced the original design, as there is not much room for loss of timing between the cam and crank before pistons begin to hit valves. The results showed that the timing losses were as expected. Such timing losses, as long as they are able to be calculated and repeated, can be corrected in the fuel system by remapping the engine's software.

The second result was discovered when evaluating the effects on camshaft speeds. Engines A and B both drive mechanical injectors with the camshaft, whose speed variation has direct implications on the injection and valve events. The camshaft, now isolated from the crankshaft torsionals, was allowed to operate at a more constant speed. Instead of the injection events being driven by random, high-energy, short-duration impacts between gear teeth, they are being driven by a combination of more controlled forces from the crank and gear inertia. Figure 14 shows this change in the average cam speed for Engine B.

Not only did the average cam speed appear to be more desirable, but also the repeatability of cam speed between cycles showed a large improvement. Figure 15 shows the average cam speeds plus and minus two instantaneous standard deviations $\pm 2\sigma$. Consequently, the area between the curves represents 95% of the possible cam speed traces assuming a normal distribution. On average, for this load/speed condition, the compliant geartrain reduced cam speed variability by 60%. This could be further reduced with additional inertia at the camshaft with a compliant geartrain. For a baseline geartrain, though, this added inertia may only worsen the mesh torque impacts, as the gears will have more rotational momentum to dissipate during each impact.

The effects on gaseous emissions are negligible once the timing loss is accounted for in the software. Additionally, the more consistent cam speeds may actually allow for a better fine-tuning of the injection system. Further work is required on this topic before any firm conclusions can be made.

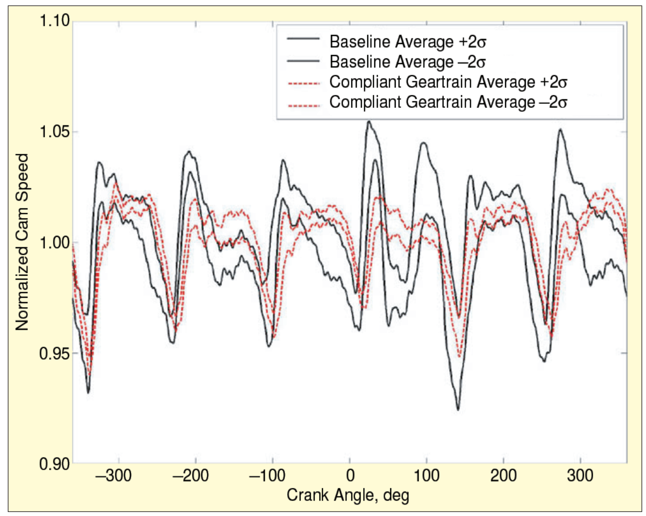

Figure 15. "Engine B" cam speed envelope $\pm 2\sigma$.

Conclusions

The application of geartrain compliance in two Caterpillar heavy-duty diesel engines has been successful on several fronts. The revolutionary devices invented by Caterpillar reduced the torsional activity of the front geartrain by up to 50%. The overall sound power levels of the engines were reduced by 3-7 dB at higher speeds. Additionally, the cam gear mesh loads were reduced by 60%. Furthermore, the cam speed variation was reduced by 60%. Large reductions in gear rattle are possible with the caveat that these results are all speed, load, and driveline inertia dependant. Compliant geartrains should be considered as a viable, effective treatment for geartrain rattle, excessive loads, and to improve the repeatability of camshaft speed.

References

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