

Methods of Evaluating and Mitigating NVH when Operating an Engine in Dynamic Skip Fire

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ABSTRACT

Cylinder deactivation is a technology seeing increased automotive deployment in light of more demanding fuel economy and emissions requirements. Examples of current production systems include GM's Active Fuel Management and Chrysler's Multi-Displacement System, both of which provide one fixed level of deactivation. Dynamic Skip Fire (DSF), in which the number of fired cylinders is continuously varied to match the torque demand, offers significantly increased fuel savings over a wider operating range than the current production systems. One of the biggest challenges in implementing cylinder deactivation is developing strategies to provide acceptable Noise, Vibration and Harshness (NVH); this paper discusses those challenges and the methodologies developed. This work covers theoretical root causes; proposed metrics to quantify the NVH level; algorithmic and physical mitigation methods; and both subjective and objective evaluation results.

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INTRODUCTION

Cylinder deactivation technology has been demonstrated as a durable and reliable means to achieve improved fuel economy in spark ignited gasoline engines. Notable current production systems include GM's Active Fuel Management (AFM) [2] and Chrysler's Multi-Displacement System (MDS) [3], both deployed on production V8 engines. Recently increased Corporate Average Fuel Economy standards in the US and CO₂ targets in Europe have driven application of cylinder deactivation to smaller displacement and lower cylinder count engines. This can be seen with Volkswagen's application of Active Cylinder Technology (ACT) in its new 1.4L turbo GDi engine in 2013.

Cylinder deactivation achieves fuel economy improvement by operating a reduced number of cylinders at a higher operating load per cylinder to produce the same engine torque output. These systems are examples of two-mode deactivation where either full cylinder count or half cylinder count is provided, V8 or V4 mode and L4 or L2 mode as shown in Figure 1. Such systems are even firing, which means a skipped cylinder event follows a firing cylinder and the firing sequence or pattern is completed with each engine cycle.

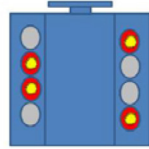
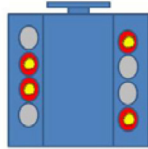
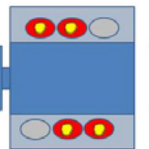
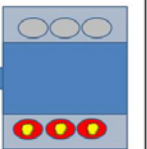
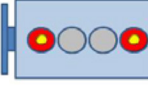
GM Small Block V8	CHRYSLER HEMI V8
AFM Active Fuel Management	MDS Multi-Displacement System
V8 1-8-7-2-6-5-4-3 V4 8-2-5-3	V8 1-8-4-3-6-5-7-2 V4 8-3-5-2
	
HONDA V6	VW 1.4L TDI
VCM Variable Cylinder Management	ACT Active Cylinder Technology
V6 1-4-2-5-3-6 V4 1-2-5-6 L3 4-5-6	L4 1-3-4-2 L2 1-4
 	

Figure 1. Cylinder deactivation systems as deployed by GM, Chrysler, Honda and VW

Automakers have also implemented cylinder deactivation systems that result in uneven firing. Honda introduced a V4 mode to its V6/L3 deactivation engine [4, 5]. GM recently introduced a V4 mode to its fifth generation small block based V6 engine family for 2014 model year. These systems are also shown in Figure 1. They provide two cylinder firing events followed by a skipped cylinder event. Similar to the 2-mode applications these systems complete the firing sequence or pattern with each engine cycle.

An alternative approach is to consider firing sequences that require more than one engine cycle to complete their sequence or pattern. This idea is not new; the first example is described US patent 2771867 [6], which describes a method for running a four cylinder engine at multiples of 1/9 of full power. US patent 4509488 [7] extends this idea to many fractions of full power. If we extend this paradigm to any possible firing sequence with even or uneven firing we have DSF operation where the engine is continuously operated at high thermal efficiency. DSF incorporates any-time, any-cylinder deactivation and can continuously vary the number of cylinders firing, along with cylinder load to provide for flexible control of acoustic and vibrational excitations from the engine. The fuel economy benefits of DSF are shown to be 14-17% composite fuel economy improvement over standard V8 operation [1].

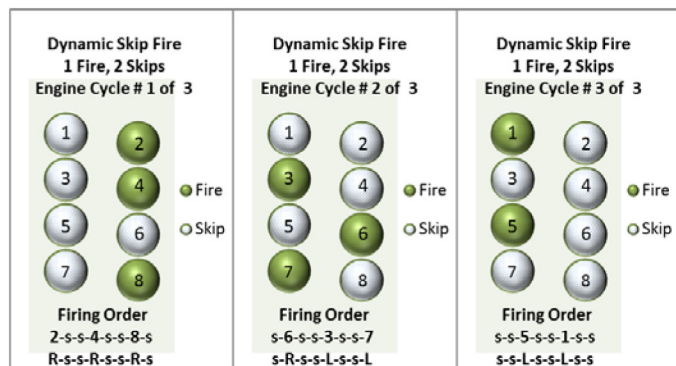


Figure 2. An example of a firing sequence that fires one cylinder followed by two skipped cylinders. The pattern will repeat after three engine cycles for an 8 cylinder engine.

In this paper we will investigate one particular sequence of interest, a cylinder firing followed by two cylinder skips. If applied to an engine of eight cylinders the patterns will repeat after three engine cycles. If we start the sequence on cylinder #2, then using the GM small block firing order from Figure 1 we have three firing cylinders in the first engine cycle, followed by three firing cylinders in the second engine cycle and two firing cylinders in the third engine cycle as shown in Figure 2. Although the number of fired cylinders per engine cycle varies for our eight cylinder engine, the sequence itself is defined as even firing with each firing followed by two skips. Another characteristic of this sequence is that all the cylinders will be fired in the three engine cycles providing a rotating pattern to distribute thermal load and avoid any prolonged cylinder deactivation that can result in spark plug fouling. It should be noted that the sequence is not limited to start on cylinder #2 but can be entered on any cylinder number and followed for

three engine cycles where it will repeat. Of course this is not the only sequence to fire eight cylinders in three engine cycles. In fact there are 735,471 sequences that fire eight times and skip 16 times in three engine cycles. The challenge, then, is to select the patterns with acceptable NVH.

1. VIBRATION

One of the primary challenges of using DSF is the resulting increase in the level of vibration that running the engine in a skip fire mode induces in the vehicle. One straightforward source is the fore-aft vibration on the entire vehicle induced by the changing levels of vehicle acceleration.

Each time a cylinder fires, the torque generated accelerates the vehicle, causing an excitation that may be felt by the vehicle occupants. At 1500 engine RPM a four-stroke V8 will fire 100 times per second. If we were to assume that all of the torque pulses generated by the cylinders were identical, then the lowest fore-aft acceleration frequency would be 100 Hz. However, when running in DSF, the frequencies of vibration will be quite different. For a given pattern, though, the spectrum of the fore-aft acceleration can be approximately determined. Continuing the example from before, if the skip/fire pattern is one fire followed by two skips (*fire-skip-skip*), then the lowest fore-aft acceleration frequency will be 33.33 Hz, since the pattern repeats every 30 ms. The time domain waveform of the flywheel torsional speed that arises from such a pattern is shown in Figure 3. Its frequency spectrum is shown in Figure 4. (The constant portion of the waveform has been removed.)

This very simple model is informative and tells us that the fundamental frequency of this pattern is high enough that it should be quite acceptable (see Section 3). However, two things can occur that make it less ideal. First, if a resonance at about 33 Hz exists in the powertrain or between the powertrain and driver, the magnitude of the vibration can be significantly increased. In this case this pattern may be unsuitable at this RPM. Second, if the resonance occurs instead at, say, 25 Hz, then the pattern may be unacceptable when the RPM is 1100-1150. Thus the quality of the various patterns can change markedly as the engine speed changes.

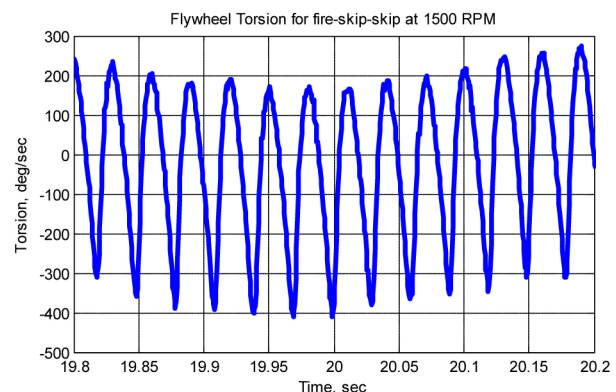


Figure 3. Flywheel torsion versus time for a fire-skip-skip pattern. DC term removed

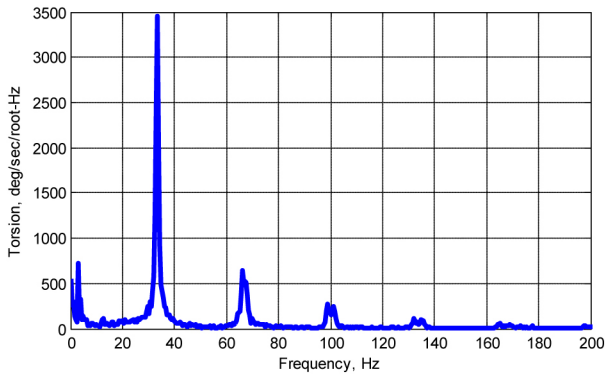


Figure 4. Flywheel Torsion versus frequency for a fire-skip-skip pattern

The way in which the powertrain excitation enters the cabin as low frequency vibration depends upon the many transfer path sensitivities as well as the primary rigid body modes of the body and the powertrain. For an SUV with a body-on-frame there can be significant low frequency modes (as shown on Figure 5) that have to be managed to minimize excitation. Human vibration sensitivity tends to occur between 1 to 18 Hz with peak sensitivity being around 4 Hz

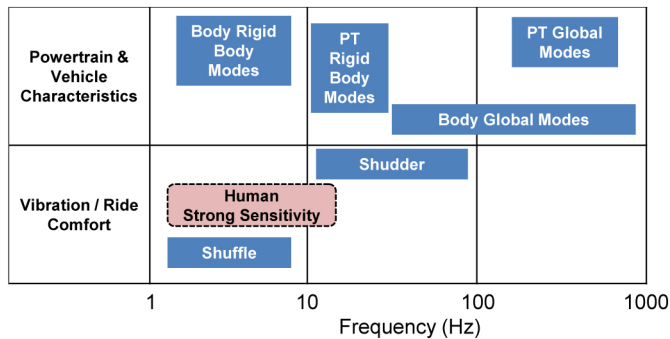


Figure 5. Comparison of vehicle modal characteristics and highest vibration sensitivity for humans

With DSF it is therefore important to optimize the excitation and the transfer paths in order to minimize vibration into the vehicle.

While the frequency of the torsional vibration depends on the pattern being used, the peak-to-peak variation depends on the firing density. Figure 6 shows the change in velocity of engine speed for different firing densities. As can be observed, the velocity amplitude decreases as the firing density increases.

In terms of human sensitivity to these speed fluctuations it was found that only the 20% firing density was considered unacceptable at this speed & load. So while care has to be taken in the calibration of the DSF system to ensure that N&V and comfort targets are met, the task is not insurmountable.

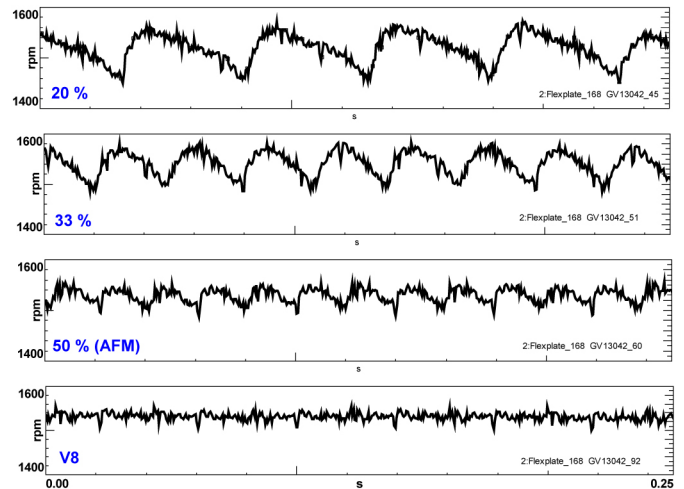


Figure 6. Comparison of engine speed velocity during constant speed (1500rpm, 3rd gear, same TCC slip) for various firing densities

2. ACOUSTICS

Each skip/fire pattern creates not only a signature vibration, but also a signature acoustic pattern. However, the basic model to describe the acoustic effect is more complicated than was the case for fore-aft vibration. With vibration, a simplifying assumption is that each torque pulse from firing a cylinder is identical. For acoustics, we assume that the sound of each firing is the same, except that each cylinder has a different length path from the exhaust valve to the tailpipe outlet. As a result, the position of each pressure pulse is modulated over time, potentially introducing additional frequencies. This can be illustrated with a simple model that assumes that 1) the *fire-skip-skip* pattern discussed is used, 2) the RPM is 1500, and 3) the path length for odd cylinders is longer than for even cylinders because of an uneven Y-pipe length. Such a Y-pipe is often found in production applications.

Figure 7 shows a time plot of a simulation of a series of sound pressure pulses from a cylinder firing sequence. The green line shows the case that the Y-pipe is equal length, while the blue line shows the case when the Y-pipe has a difference of about 1 meter between legs. This path length difference shows up when four consecutive firings are delayed relative to the other four. If, as occurs in a V8 engine like the GM small block from Figure 1 with firing order 1-8-7-2-6-5-4-3, then with this pattern for *fire-skip-skip*, the firing sequence starting on cylinder 2 is 2-s-s-4-s-s-8-s-s-6-s-s-3-s-s-7-s-s-5-s-s-1. (Each fire is given by the cylinder number and is separated by two skips.) So this pattern fires four times on one bank (even cylinder numbers) and then four times on the other bank (the odd cylinder numbers.) See Figure 2 for illustration where the two lines overlap for the firings on one bank but are offset for the firings on the other bank.

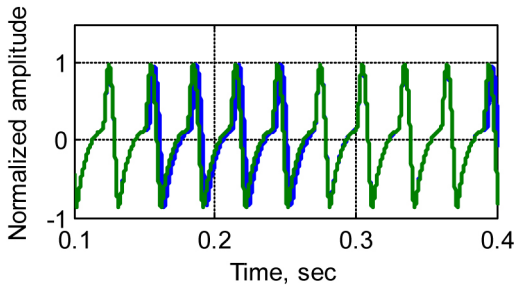


Figure 7. Sound pressure versus time for equal-length Y-pipe versus one with unequal length at tailpipe outlet

The following figure (Figure 8) shows the two frequency responses for the two time waveforms in Figure 7. The green line corresponds to the equal length Y-pipe, and has frequency content only at 33 1/3 Hz and its harmonics. The blue line, on the other hand, corresponds to the unequal Y-pipe. It has additional spurs from the equal length case, displaced from the main spurs by 4 1/6 Hz, the rate of modulation of the pulses' position.

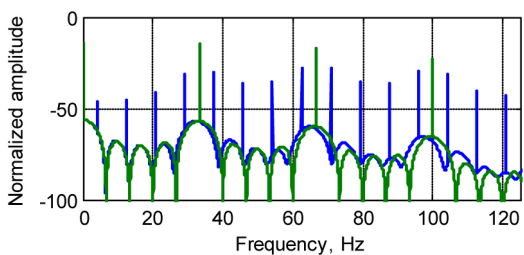


Figure 8. Sound spectra for the waveforms of Figure 7

These additional spurs act to cause an audible change at a frequency of 4 1/6 Hz, or 1/8 of 33 1/3 Hz. Psycho-acoustic effects of this firing pattern will be discussed more in section 3.3 where the concept of fluctuation is discussed. But one additional thing to notice here is that the extra spurs increase in magnitude relative to the center spurs (the 33 1/3 Hz harmonics) with increasing frequency. This is a mathematical consequence of the modulation of the pulse position, and manifests itself as more noticeable modulation with increasing frequency. Thus the amount of modulation grows precisely in a frequency band to which the ear is more sensitive.

3. METRICS FOR VIBRATION AND ACOUSTICS

3.1. Measuring Comfort

This section describes two techniques for evaluating ride comfort: ISO-2631 [8] and the Sperling Index [9,10,11,12]. ISO-2631 provides a standard model for evaluating human exposure to whole-body vibration. The basic measure is the weighted root-mean-square (RMS) acceleration as given by

$$a_w = \sqrt{\frac{1}{N} \cdot \sum_{n=0}^{N-1} a_w(n)^2}$$

where $a_w(n)$ is the weighted acceleration as a function of samples and N is the number of samples in the measurement. The weighting of the acceleration is given by one of a set of comfort weighting functions (W_c , W_d , W_e , W_f , W_k , and W_j) specified in the frequency domain for different applications and can be decomposed into component transfer functions. Refer to [8] for the transfer functions and parameter definitions. For evaluating comfort, W_d is used for weighting the horizontal directions and W_k is used for weighting the vertical direction.

Sperling's method, also known as the W_z (Wertzungzahl) method, was initially proposed to measure ride comfort in locomotives [10, 12]. The Sperling index can be implemented using the discrete Fourier transform (DFT) with the following calculation [9]

$$W_z = \sqrt[10]{\sum_{i=0}^{N_f} [2 \cdot a_i \cdot B(f_i)]^3}$$

where a_i is the magnitude of the i^{th} frequency index (from the acceleration DFT) in m/s^2 , N_f is the spectrum of acceleration response, and $B(f)$ is the Sperling weighting curve given by

$$B(f) = k \cdot \sqrt{\frac{1.911f^2 + (0.25f^2)^2}{(1 - 0.277f^2)^2 + (1.563f - 0.0368f^3)^2}}$$

where $k = 0.737$ for horizontal and 0.588 for vertical direction.

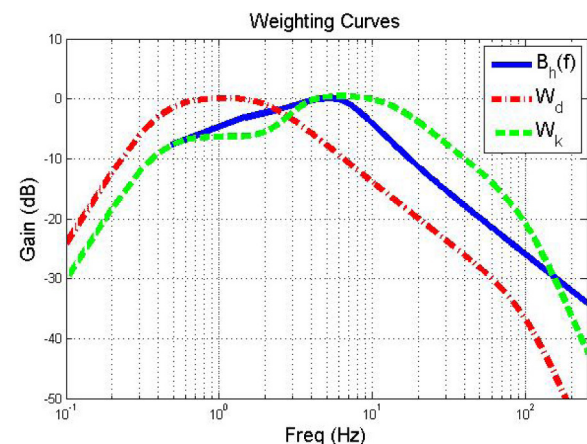


Figure 9. Comfort weighting curves: W_d , W_k , and B (Sperling)

The frequency weighting curves for the ISO-2631 and Sperling comfort metrics are shown in Figure 9. The W_d curve is used to scale fore-aft and lateral vibration, and has a lower frequency response (bandwidth from ~0.4 - 2.8 Hz) while W_k is used for scaling vertical vibration (bandwidth from ~2.8-18 Hz). The bandwidth of the Sperling weighting curve is between W_d and W_k (from 1.5-9 Hz). For both metrics, human sensitivity to comfort is weighted more heavily at low frequencies.

The Sperling index maps to a subjective evaluation of ride comfort in the range of 1 to 4 given in Table 1.