

# Input Torque Shaping for Driveline NVH Improvement and Torque Profile Approximation Problem with Combustion Pressure

Kazuhide Togai and Michael Platten

**Abstract** The powertrain is the largest excitation source of automotive noise and vibration. Torque variations from the engine cause noise and vibration in two frequency regimes: Engine firing fluctuations at the engine rotation frequency and its harmonics and driver-demanded propulsion torque variations at lower frequencies. This paper examines the mechanisms of gear rattle noise caused by engine firing fluctuations and shock and jerk caused by propulsion torque variations. Gear rattle behaviour is simulated and various approaches for reducing gear rattle are investigated. It is found that isolation of engine firing fluctuations from the transmission is the most effective practical way to reduce rattle. Torque profile shaping by robust control is a method that can be used to reduce shock and jerk. A pre-compensator approach based on knowledge of the driveline resonant properties is proposed and the sensitivity of the controller to errors in the assumed resonant frequencies is investigated. It is shown that one of the constraints of this strategy is that the resolution of control system depends on the firing frequency.

**Keywords** Transmission • NVH • Gear rattle • Modelling • Active control

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## 1 Introduction

A passenger car can be thought of as a flexible structure of masses and springs distributed over the vehicle to balance handling, fuel efficiency and comfort. Resonances in this structure can lead to noticeable vibration and audible noise. As part of this flexible system the transmission and driveline transfers the power from the engine to propel the vehicle. But the dynamic behaviour of the driveline can also generate or amplify unwanted noise and vibration.

There are some noise phenomena that are generated by the driveline itself such as unbalance excitation or gear whine. In others, the driveline is a passive system that is excited externally, often by forces from the engine. In the case of gear rattle and shock and jerk it is the interaction of the engine acting as a source and the driveline acting as a transfer path that cause the noise and vibration problems. Generally with noise problems it is more effective to reduce noise at source, but sometimes there are limits to what can be achieved and changes to the transfer path may also be required to achieve an economically satisfactory result. In this paper mathematical models are used to simulate both the excitation and transfer path. Results of these simulations for gear rattle and driveline shock response are used to investigate some proposed methods of reducing annoyance caused by these phenomena.

Gear rattle is common in manual transmissions and is caused by engine torque fluctuations associated with engine firing which lead to torsional vibrations inside the transmission. Idling gears can lose contact in these conditions and the resulting noise when they come back into contact causes an unpleasant rattling sound. Rattle occurs when the fluctuating dynamic torque at the gear mesh exceeds the static driving torque of the gear—this is the rattle. In this study, the gear meshing behaviour with backlash is investigated and the effects of the reduction in lubricant viscosity, backlash and engine firing torque fluctuations on gear mesh impact force are calculated.

Driveline vibrations can be excited by sudden changes in supplied torque caused by changes in acceleration demanded by the driver such as sudden acceleration for overtaking or tip-in/tip-out. The driveline torsional vibrations that are generated by this process can cause noticeable longitudinal vibrations (“shunt” or “shuffle”) of the chassis and sometimes also noise (e.g., “clunk”).

One way to mitigate this problem is by input torque shaping where a control system adjusts the throttle input supplied by the driver to change the combustion of the engine. When combined with knowledge of the resonant behaviour of the driveline, this can reduce the longitudinal vehicle vibrations. There is a fundamental upper limit to the frequencies that can be suppressed in this way due to the fact that engine torque can only be controlled during firing. This means that the method is less effective as engine rpm decreases. Simulations are used to illustrate this method.

## 2 The Engine as a Vibration Source

An ideal transmission and driveline transfers power efficiently from the engine to a vehicle. Along with the power needed for propulsion there is also a dynamic aspect to the transferred power which can lead to noise and vibration (Fig. 1). Sometimes this dynamic behaviour is caused by self-excitation within the driveline (unbalance and gear whine are examples of this). However, there is class of vibration problems where the source of the vibration is the engine and driveline plays an important role in the amplification and transfer path for this engine excitation.

This engine excitation is in the form for torque fluctuations and occurs in two frequency regimes: Driver torque demand and combustion pulses.

### 2.1 Driver Torque Demand

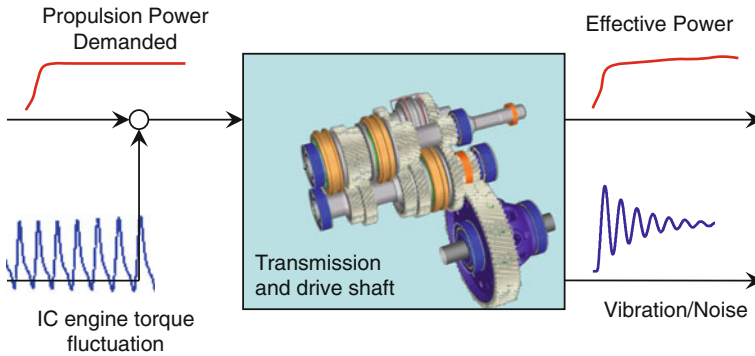
This is the changing mean engine output torque as demanded by the driver to achieve the necessary speed changes required to control the vehicle. When these changes occur fast enough, (e.g., sudden acceleration, sudden change from driving to coasting—tip-in/tip-out, clutch re-engagement after shifting) then they can excite the resonant behaviour of the vehicle and driveline system. These torque variations are non-periodic.

### 2.2 Combustion Pulses

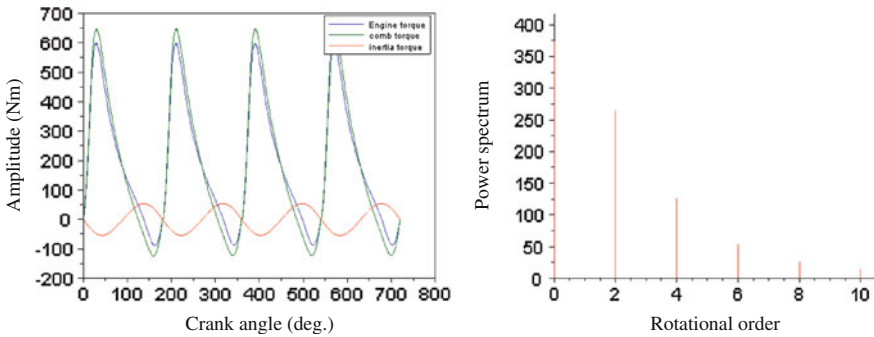
Superimposed on the mean torque signal is a large fluctuating torque which comprises two components: the torque that results from the change in cylinder pressure during combustion and the torque that results from the changing inertia seen at the crankshaft due to motion of the piston and conrod. The torque variations are periodic for a steady speed and load.

Figure 2 shows the individual contributions from piston inertia and combustion pressure to the total crankshaft force on a 4-cylinder 4-stroke engine. Also shown is the corresponding frequency content of the combined signal. It can be seen that as well as the main contribution at 2nd order (firing frequency) there is also contributions at higher orders. Typically in powertrain management, frequencies up to 8th order are considered significant for a 4 cylinder engine.

Flywheel design is the usual method for reducing the transfer of both kinds of torque fluctuation to the rest of the driveline. A single mass flywheel clutch attenuates engine vibrations with a roll-off of 40 dB per decade. For a dual mass flywheel the roll-off is 80 dB per decade (Fig. 3). This means that the drive torque and very low frequency vibrations can pass through the flywheel to the trans-



**Fig. 1** Power transfer concept via a transmission



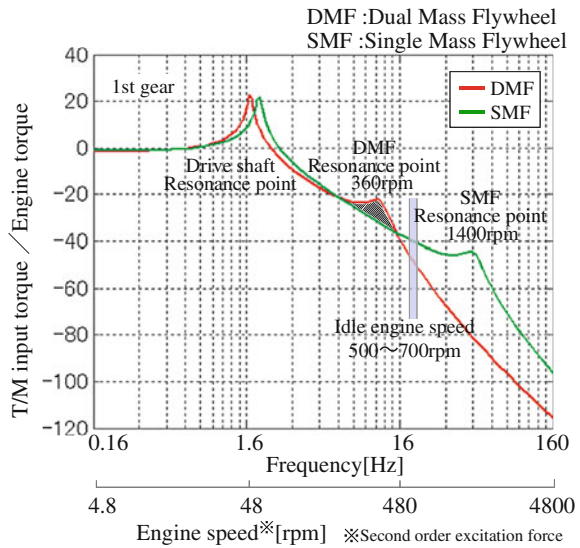
**Fig. 2** Excitation torque with an IC engine model

mission, but higher frequencies vibrations are reduced. The flywheel subsystem has its own internal dynamic resonance behaviour which interacts with the resonant behaviour of the rest of the driveline (transmission, driveshafts etc.)

### 3 Gear Rattle with Internal Combustion Engine

Gear rattle occurs inherently in a powertrain consisting of an internal combustion engine and a manual type transmission. A meshing gear pair with backlash is excited by fluctuation torque in the powertrain which causes idling gears (usually) to lose contact. There is an impact when the gear teeth come back into contact again and when this happens repeatedly an unpleasant rattling sound is perceived.

**Fig. 3** Input torque fluctuation attenuation with a single and dual mass flywheel clutch



### 3.1 Gear Rattle Mechanism and Simulation

If driven gears are sufficiently loaded, meshing gear pairs don't lose contact under torque fluctuations from the engine. When driven gear load is small, contact between two meshing gear pair can be lost temporarily as the drive gear decelerates due to the fluctuating combustion torque. Strong impulsive contact then occurs when the drive gear accelerates under the influence of the next combustion pulse. Even when an idling gear is not supplied by a driving torque from the engine there is still some torque applied by the drag of the gear through the lubricant sump and by the drag of the lubricant in the bearing under the gear. The most severe forms of gear rattle involve the gear mesh repeatedly bouncing across the backlash between the drive and coast flanks of the gear pair. The most basic form of a system which exhibits gear rattle is shown in Fig. 4. It comprises a single degree of freedom system with backlash and represents a single gear pair.

In a real driveline, the model is not as simple as this as there may be many degrees of freedom with many backlashes and there are many factors that influence the response such as temperature, lubricant viscosity, gear inertia and amount of backlash. The general resonant dynamic behaviour of the entire driveline also has an effect on the occurrence of rattle. However, in all cases, the fundamental noise-generating mechanism is the same with tooth impacts causing vibrations in the gears which travel through the shafts to the transmission housing where they are either radiated directly as noise or the vibrations are transferred to the chassis and are radiated elsewhere in the structure.

The resonant behaviour of the transfer path is usually less important as it is the impulsive nature of the force that causes annoyance rather than its tonal qualities.

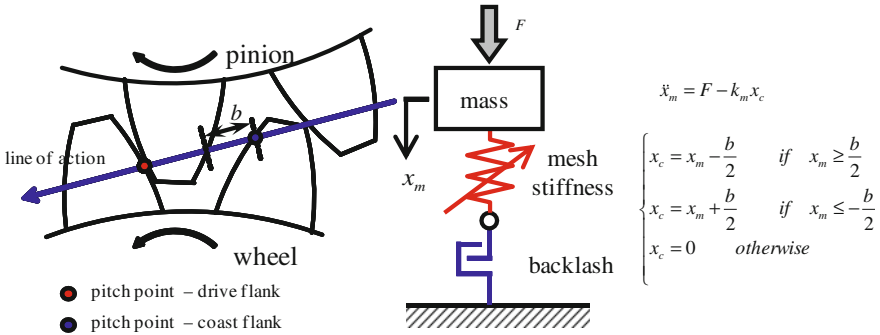


Fig. 4 Gear rattle concept

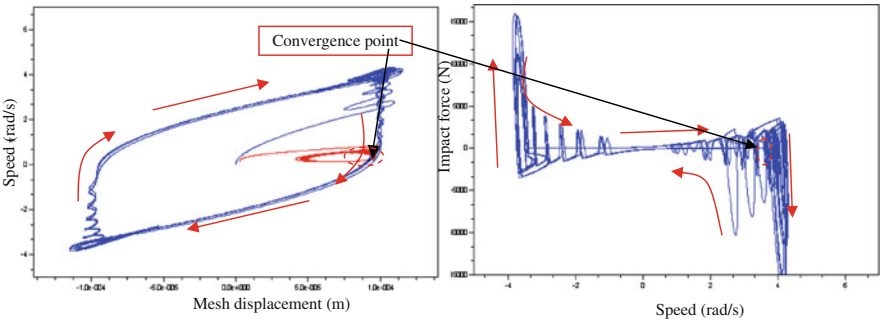


Fig. 5 Mesh displacement—relative speed—impact force

Many different metrics are used to quantify annoyance from gear rattle and most of them are related to the peak level of gear mesh force at the point of impact.

The gear rattle behaviour in a driveline can be simulated using low-level simulation tools, multi-body dynamic software or by dedicated software for gear rattle analysis [1].

### 3.2 Gear Rattle Vibration Response and Rattle On-Set Threshold

Gear rattle vibrations are often of the limit cycle type. Mesh displacement vs relative speed for a rattling gear pair is shown in Fig. 4. Both double and single sided impact events can be observed. As the speed fluctuations caused by the input torque fluctuations decreases the double-sided impacts become single-sided impacts and below a certain level, they converge on a point where no mesh displacement occurs. The magnitude of the impact force also decreases as the speed fluctuations decrease and eventually become zero (Fig. 5).

### 3.3 Gear Rattle On-Set Threshold

The magnitude of torque fluctuation below which no rattle occurs is called the rattle on-set threshold. This is a function of both the torque fluctuation at the gear mesh and the drag torque on the driven gear. Provided the instantaneous dynamic torque at the gear mesh does not exceed the static drag torque then no rattle will occur and the teeth will remain in contact [2].

Mathematically the rattle on-set threshold can be derived as follows: A driven gear is accelerated by the drive gear and will rotate freely. If deceleration of the wheel is larger than that of the pinion during deceleration, contact between two teeth is kept. The pinion and wheel behaviour is expressed mathematically as.

$$\begin{aligned}v_w(t) &= v_{p2} - \int_0^t \left(\frac{T_{dr}}{I_w}\right) d\tau \\v_p(t) &= v_{p2} + \int_0^t \left(\frac{T_{in}(t)}{I_p}\right) d\tau \\v_w(0) &= v_p(0)\end{aligned}$$

It is assumed that the drag torque is constant and the fluctuation torque is monotonously decreasing during a half cycle of combustion pulse. When deceleration of the wheel is smaller than that of the pinion, then, contact between teeth is lost. And the gear rattle on-set threshold is written as

$$\frac{T_{dr}}{I_w} < \frac{T_{fl1}}{I_p} \rightarrow T_{dr} < \frac{I_w}{I_p} T_{fl1}.$$

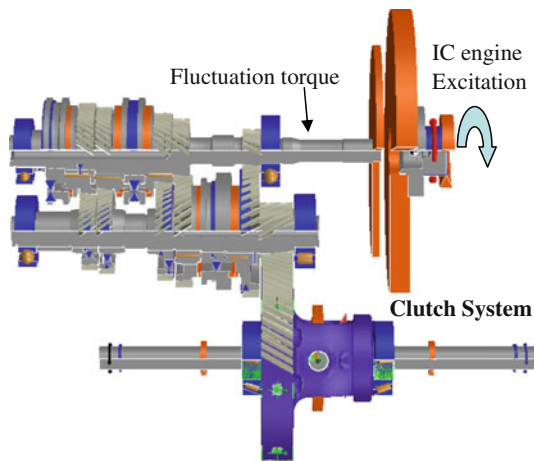
### 3.4 Drag Torque Evaluation

It is clear from the above discussion that for accurate simulation of gear rattle on-set, the drag torque must be known. Drag torque between an input shaft and an output shaft of a transmission can be measured in a carefully designed test rig. However, drag torque measurement for individual gears is more technically challenging. Using calculated drag torque with a transmission model is a practical and rather accurate method. Drag torque correlation between measurement and a transmission model calculation has been published previously [2].

### 3.5 Calculation of Torque Fluctuations at the Transmission Input

In the complete driveline, the amplitude of the excitation torque from an IC engine (Fig. 2) is attenuated mechanically with a clutch/flywheel arrangement. Typical attenuation properties of such a clutch system is shown in Fig. 3. In some driveline

**Fig. 6** Transmission system resonance calculation model



arrangements, the resonant frequencies of the driveline may be close to firing frequencies of the engine and the input fluctuation torque must be calculated using dedicated software which can accurately model the dynamics of the entire driveline (Fig. 6).

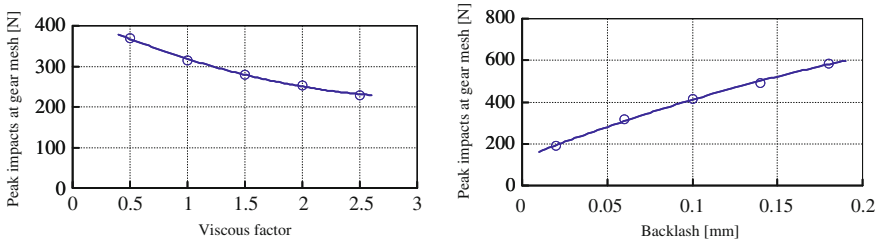
### ***3.6 Measures for Reducing Gear Rattle***

There three main ways in which gear rattle can be reduced are backlash reduction, drag torque increase and engine fluctuation torque amplitude reduction. The effects of these can be simulated using the models described previously.

The effect of lubricant viscosity (which directly affects gear drag) on the peak impact force is shown in Fig. 7. Increasing the lubricant viscosity reduces the peak force by reducing the speed of the driven gear during a rattle event. In the extreme case, increasing the lubricant viscosity may cause the drag torque to be great enough to bring the system below the rattle on-set threshold and eliminate gear rattle entirely. Increasing the amount of lubricant in the gearbox may have the same effect. However, increasing the drag torque of course reduces the efficiency of the transmission and increases fuel consumption which is not generally acceptable. Also lubricant viscosity is chosen for particular operational and durability reasons and should not be changed without consideration for these effects.

It can be seen that reducing backlash has an effect on the reduction of the peak mesh force however gear rattle can never be eliminated by backlash reduction (unless the backlash is zero, which is undesirable for other reasons). Again, backlash reduction is not an acceptable solution in a production environment as the increased quality of manufacturing required to control backlash to this degree would be prohibitively expensive for a mass production transmission.





**Fig. 7** Gear rattle sensitivity to lubricant viscosity and backlash

The conclusion is that the engine torque fluctuations must be reduced if rattle is to be avoided. Engine calibration is not an option in this case as changing the engine management system to reduce torque fluctuations has a negative impact on fuel economy. The only practical way to achieve gear rattle reductions is by careful design of flywheel and clutch assemblies to reduce the transfer of the engine torque fluctuations to the transmission input.

Fortunately, as has been demonstrated here, the kinds of specialist tool required to simulate the driveline dynamic behaviour are now available and much of this work can now be carried out by CAE methods with rattle noise can be reduced or eliminated before a design is finalised. Gear rattle is still a major challenge and this problem will only get worse with the current trends for downsizing engines and increasing combustion pressures and the increased engine torque fluctuations they generate.

## 4 Acceleration/Deceleration Vibration

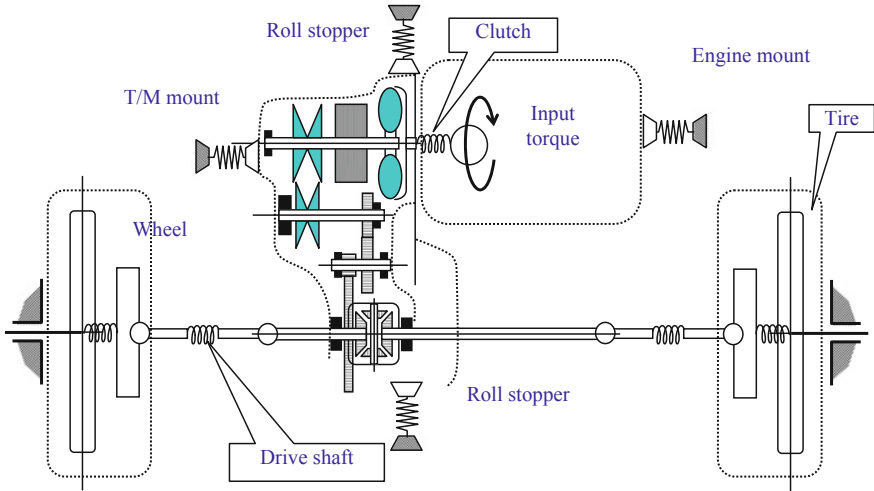
The gear rattle phenomena described above were caused by the periodic engine torque fluctuation pulses associated with combustion. The phenomena such as shunt and tip-in/tip-out which cause longitudinal vibration of the whole vehicle are caused by longer, lower frequency, transient torque fluctuations caused by driver demand through the throttle pedal.

The entire driveline from engine to tires is involved in these acceleration/deceleration drivability events (Fig. 8). The torque variations are supplied by the engine and the driveline acts as a passive resonator with many natural frequencies. Vibration is transferred to the vehicle either through acceleration and deceleration of the entire vehicle by the tires or as structure-borne sound through the various drive train mounts and suspension.

The driveline can be represented as a system of equations of motion of the form

$$M\ddot{x} + C\dot{x} + Kx = f$$

Where  $M$ ,  $C$  and  $K$  are respectively the mass, damping and stiffness matrices,  $x$  is the component displacement vector and  $f$  is the equivalent vector of external



**Fig. 8** Driving force transmission path

forces which may vary with time. For the types of problem studied here, it has been shown in other studies [3] that the number of degrees of freedom may be reduced to just the effective referred inertias upstream and downstream of the driveshaft with the driveshaft as a flexible component. The transfer function of the reduced order model is described as

$$G_p(s) = \frac{K\omega_p^2}{(s^2 + 2\zeta_p\omega_p s + \omega_p^2)}$$

Where  $\omega_p$  is the resonance frequency of the driveline and  $\zeta_p$  is the damping ratio of the driveline

#### **4.1 Methods of Vibration Reduction**

Measures to reduce the sensitivity of the transfer path are increased damping and amplitude attenuation (gain reduction). As the transfer path is a torsional rotating one, any attempt to remove energy by damping may also lead to reduced driveline efficiency and hence fuel economy. Mechanical amplitude reduction measures will not be effective without major vehicle structure modification, sometimes tuned mass dampers are employed as a method of last resort but the add additional weight and cost.

As with gear rattle, the best solutions come from reducing the source of the vibration, the engine torque fluctuation. One method which is described in detail here is “input torque profile shaping”. This is a combustion control technique and

as such is relatively low cost and does not add any heavy or unreliable mechanical components to the system.

With electronic throttle or injector control, the inputs from the accelerator pedal can be mitigated by a controller. In its most simple form a passive low pass filter can be used to prevent sudden acceleration demands from reaching the throttle/injector system. This type of control is not suitable because drivers complain of the accelerator response being too low.

Input torque shaping is a more sophisticated system that replaces the low pass filter with a notch filter tuned to the dominant resonant frequency of the driveline. This method requires that the driveline resonance frequencies are known a priori for all ratio conditions. The method must also be robust enough to cope with changes in vehicle mass which will also change the driveline resonance frequency.

## ***4.2 Increasing Damping by Speed Feedback***

One way to add damping is to measure speed changes and use this as feedback to create an additional input force proportional to the output speed. This form of motion feedback active vibration control is very robust and can provide high levels of damping. However, this only works well for very low frequency vibrations.

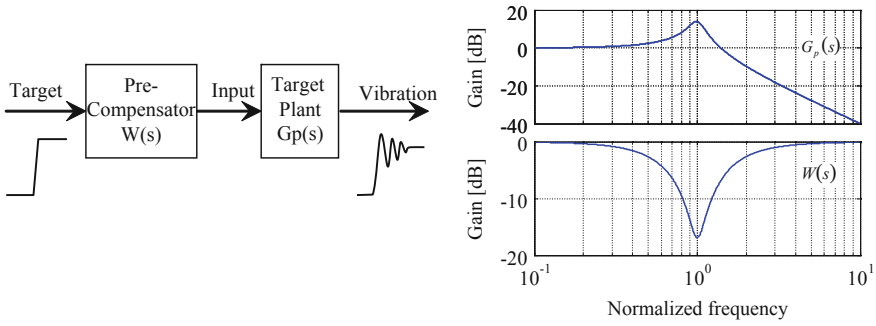
There is a delay from increasing throttle/fuel injection to increased output torque. This is due for the time taken for the increased fuel/air mixture to be delivered to the cylinders and also because the system must wait for the next firing before any torque control can take place.

Model prediction can be used to minimise such delay and increase the useful frequency range. This works by replacing the measured speed feedback with a simulated speed feedback calculated from a simplified dynamic model of the driveline and torque response delay of the engine is avoided. Two model prediction approaches are described below.

## ***4.3 Pole Cancellation of Transfer Function with Pre-compensator***

This model prediction control method is implemented by using a pre-compensator. The compensator has polynomials corresponding to denominator of the driveline transfer function as its numerator and transfer function with desirable damping in its denominator. The compensator shapes input torque and functions as a notch filter. The method was proposed by authors [3–5] and applied to production cars [6]. The concept is shown graphically in Fig. 9.

The compensator transfer function to counteract the dynamic behaviour of the driveline is



**Fig. 9** Pre-compensator strategy and frequency property

$$W(s) = \frac{\omega_m^2 (s^2 + 2\zeta_p \omega_p s + \omega_p^2)}{\omega_p^2 (s^2 + 2\zeta_m \omega_m s + \omega_m^2)}$$

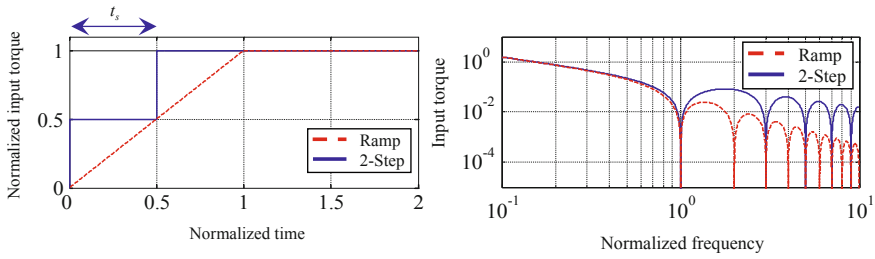
where  $\omega_m$  is the resonance frequency of the model and  $\zeta_m$  is the damping ratio of the model.

#### 4.4 Nonlinear Input Torque Waveform Conversion

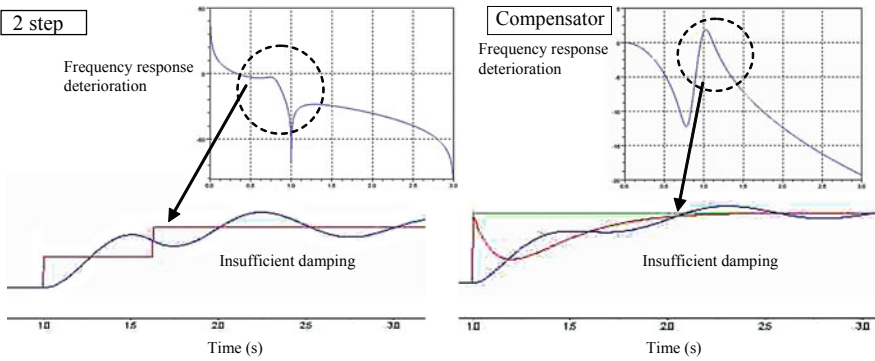
An alternative method is to replace the driver demanded torque with a torque that aims to avoid exciting resonance. When a sudden step change in torque demand is requested by the driver the torque should instead be delivered in a way which suppresses the resonance of the driveline and thus minimises vibration. Two such waveforms are shown below (Fig. 10). One is a ramp input which lasts for a full cycle of the resonance frequency. The other is a 2 stages step input where each step lasts a half cycle of the resonant frequency. The Fourier transform of these functions show the required notch filter behaviour at the model resonant frequency.

#### 4.5 Robustness to Model Errors and Driver Operation

Both the pre-compensator and nonlinear input conversion methods rely on the model used to predict the driveline resonances being accurate to achieve maximum calculation. This cannot be achieved in practice. For example an increase in mass of the vehicle due to more passengers or more load will change the driveline resonant frequencies. It is important to evaluate performance deterioration of the proposed methods which results from such model errors.



**Fig. 10** Nonlinear input profile to avoid vibration



**Fig. 11** Influence of model error in time domain and frequency domain

Resonance frequency is the most important property in pre-compensator or nonlinear input shaping applications. The frequency depends on total inertia of powertrain and stiffness of drive shafts. Relationship between the inertia and the stiffness changes as the gear ratio changes. This means that the target frequency to be compensated changes during gear shift. For the CVT example shown in Fig. 8 for example, the controller needs to take account of a continuous range of ratios and associated resonant frequencies.

The effect of an error in the expected resonant frequency is shown below (Fig. 11) for both methods. The effect is more severe for the 2-step non-linear input conversion method. This is because the notch is very narrow and a small frequency error can lead to a large deterioration in performance. In the case of the pre-compensator, the damping can be altered by changing the damping in the transfer function. There is still some fluctuation in the frequency response as the effects of damping quickly become reduced away from resonance.

An additional drawback of the nonlinear input conversion method is that there is an inherent delay in tracking the driver’s requested torque demand. This is because the step or ramp input can only be calculated once the accelerator pedal

action is completed whereas the pre-compensator method can start to act from the moment the pedal starts to move. As a result, drivers may complain of a response delay of acceleration or strange feel in engine response.

#### ***4.6 Realisation of Target Torque Profiles***

So far it has been assumed that any target torque profile requested by the controller can actually be generated by the engine. Reasons why there may be errors in the generated torque profile include:

- Negative torque is uncontrolled.
- Maximum torque is limited. Any torque up to the maximum is feasible at each stroke with a common rail diesel engine.
- Torque generated from the IC engine is not continuous but in discrete pulses as shown in Fig. 2.
- Suppression of the resonance requires a high enough engine speed so that the “sampling frequency” of the engine pulses is high enough to generate the signal requested by the controller as per the Shannon-Nyquist sampling theorem.

A gasoline engine has inherent delay in air intake and fuel transportation process to the combustion cylinder. There also is nonlinearity in throttle-torque relationship. Despite these limitations, the control strategies described here were applied to a vehicle with a gasoline port fuel injection engine. The tip-in shock was found to be well suppressed for comfort, even though some measurable vibration remained [7].

### **5 Conclusion**

- Two examples of NVH phenomena caused by engine torque fluctuations have been presented.
- Methods for simulating gear rattle have been presented for a complete transmission with backlash.
- Based on these simulation models the effects of backlash and lubrication on gear rattle performance have been investigated.
- It was concluded that neither backlash reduction nor lubrication viscosity increase were suitable approaches for rattle reduction.
- The best approach to reduce rattle is to reduce the engine torque fluctuations transferred to the transmission by careful design of the clutch/flywheel assembly.
- Methods for controlling drivability NVH issues caused by changes in driver demanded torque were investigated.

- Three active control methods were discussed—Speed feedback control, compensator model feedback control and model-based nonlinear torque waveform conversion.
- Measured speed feedback control is not suitable for this problem due to the delay between throttle command and engine response.
- The effect of errors in the model resonance frequency was investigated for the other methods and other limiting factors were discussed.
- The resolution of the torque control is shown to be proportional to engine speed and therefore there is an inherent lower speed limit for any torque shaping approach.

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