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APPLIED NUMERICAL MODELLING OF GEAR IMPACTS IN POWERTRAINS - DRIVELINE SHUFFLE AND CLUNK

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ABSTRACT

This paper presents a study on the transient response of powertrain systems at the lowest mode of torsional vibration (driveline shuffle). The response can result in gear impacts in the transmission and driveline (driveline clunk). A powertrain system dynamic model is developed for an automatic transmission fitted passenger vehicle. The model used in numerical simulations is a stiff system with strong non-linear elements. In these simulations; damping of the lowest mode was found to reduce multiple impacts of driveline clunk, reducing the transmitted engine harmonics such as is the effect of a torque converter had little impact on clunk, increasing the lash was found to reduce the occurrence of multiple clunks.

1. INTRODUCTION

Powertrains with automatic transmissions do not usually exhibit transmission gear rattle due to high damping from the torque converter. However, in these systems gear impacts may occur in one instance due to transient vibrations at the lowest mode of the torsional system. This low frequency torsional vibration is at 2-10Hz depending on gear ratio, inertias and stiffness and can be a result of engine tip-in, tip-out, automatic or CVT gear ratio change, clutch engagements and braking. A large enough transient can cause a torque reversal and gear separation. Their impacts cause high frequency vibrations which transmit through the bearings and casing then to the ear as noise. This impact is likely to be cause of driveline 'clunk' - a term for the noise when heard in the final drive. Passenger vehicle users can be aware of this noise and manufacturers want to reduce its occurrence to improve product quality. Research into transient vibration in vehicle powertrains for problems such as this 'clunk' is ongoing. Tooth separation can occur as geared systems require clearance between mating gears for smooth operation. The clearance is termed lash. The mating gears can be modelled with a mesh stiffness which is non-linear across the lash. It can be set to zero across lash [1] or approximated to this with some sort of smoothening function [2]. The mesh stiffness can also be treated as time varying including interaction with clearance [3]. In [4] design factors which determine the severity of clunk and shuffle were stated as engine torque rate rise – the faster the more severe the response and also the amount of driveline lash - clunk severity was found to increase with lash but not necessarily linearly. Many papers have been published for investigations into gear impacts in manual transmission gear sets. For driveline clunk, papers concentrate on powertrains with manual transmissions [5, 6]. Tip-in tests have confirmed the response frequency of driveline shuffle corresponds well with lumped mass models [5]. Free vibration tests were conducted by exciting the unloaded powertrain test rig with the impulse on the flywheel [6]. High frequency transient response was found to be highest at the accelerometers located on the transmission and final drive. This corresponded to the locations with the largest lash. Spectral results showed high frequency content from 1.5 to 5 kHz for the propeller shaft. They developed of a finite element model for modal analysis of this shaft. Significant response was found at various levels in the frequency range of 1 to 4 kHz. Some of the levels concurred with the experimental results. Their findings were that different damping materials for the shaft can affect the response of different shaft modes and that this could be used in design to reduce clunk radiated noise. In [5] it was noted that 'the noise (clunk) is either attenuated through hollow driveline pieces or radiated in the acoustic cavity formed by the clutch bell housing'. They included in a powertrain model a propeller shaft with a distributed element to include its high frequency contribution to the natural modes. Research [7] on computational issues with gear rattle problems has found the strongly non-linear clearance algorithms used in gear rattle problems cause the system matrix to be ill-conditioned. This ill-conditioning is defined as the ratio of the smallest to largest eigenvalues being very large. This ratio is one measure of numerical 'stiffness'. They presented procedure and evaluation criteria useful for gear rattle problems, they provide ways to check solutions and to reduce solution difficulty and times by reducing the numerical stiffness. They helped in choosing an appropriate solver, low order fixed step solvers are found to be efficient for highly non-linear and strongly stiff systems. In this paper a simplified model for a powertrain system with automatic transmission is presented. It is a torsional lumped-mass model for fixed gear transient analysis. Numerical simulations were performed for a second gear throttle tip in-out that causes shuffle and clunk. Results are presented for an investigation of the effects on clunk from changing the damping of the lowest mode, the amplitude of transmitted engine harmonics and the gear lash displacement.

2. POWERTRAIN MODEL FOR TRANSMISSION AND DRIVELINE CLUNK

Figure 1 presents the dynamic model for a powertrain fitted with an automatic transmission. The system is modelled with 10 degrees of freedom as a torsional lumped mass model. This model is reduced from a larger degree of freedom model for the same powertrain system – a model used for gear shifting [8]. The transmission and final drive are both modelled with a driving and a driven gear so as to include a non-linear stiffness describing gear backlash in the numerical analysis. The mesh stiffness used is $5\text{x}10^8$ N/m. In second gear the automatic transmission is unlocked so the powertrain is modelled from the turbine and an effective engine inertia is added at to this coordinate. The wheels are modelled with one coordinate – reducing model size and thus computation time. It should be noted that the wheels are often modelled with a coordinate for each wheel in which case there is an additional mode of vibration for the system. For this same system including both wheel coordinates the mode is at 1.25 Hz, it is local to the driveshafts and tires. For this mode there is little relative displacement between gear teeth so whether the model is with two wheels or one has little effect on the investigation of vibration in the gear meshes. The mode is excitable by asymmetric brake torque and tire loading, which are not applied in these numerical simulations. The system of equations of motion is taken as standard form:

$$\mathbf{I}\ddot{\boldsymbol{\theta}} + \mathbf{C}\dot{\boldsymbol{\theta}} + \mathbf{K}\boldsymbol{\theta} = \mathbf{T} \tag{1}$$

Where

$$\boldsymbol{\theta} = [\theta_1, \theta_2, ..., \theta_{10}]^T, \quad \mathbf{T} = [T_1, T_2, ..., T_{10}]^T. \tag{2}$$

Torsional finite elements are used to construct the inertia matrix, **I**, and the stiffness matrix **K**, they are straight, elastically geared and branched types. There are ten elements which are assembled global **I** and **K** matrices with global coordinate and torque vectors (2). In [9] detail is provided on modelling and assembly with torsional finite elements. The global inertia matrix is diagonal. The vehicle mass is lumped on one tire coordinate θ_{10}. The global stiffness matrix is symmetric.

It is difficult to model damping in these types of systems. In this study modal damping has been used for simplification. The natural frequencies and the corresponding modal shapes of the system are firstly found from the eigenvalues and eigenvectors of the system matrix:

$$\mathbf{A} = \mathbf{I}^{-1}\mathbf{K} \tag{3}$$

The damping ratios can then be used to construct a damping matrix for the uncoupled system. This uncoupled damping matrix is then transformed for use with the original coordinate system. The damping matrix is given by:

$$\mathbf{C} = \mathbf{I}\psi[diag(2\xi\omega_n)]\psi^{-1} \quad (4)$$

Where **I** is the diagonal inertia matrix, ψ is the eigenvector matrix for (3) and $diag(2\xi\omega_n)$ is a diagonal matrix for $\xi_{i=1\to10}$ damping ratios of $\omega_{ni=1\to10}$ natural frequencies. This system can be formulated in state space representation to solve the eigenvalue problem for damped frequencies, mode shapes and phase. The damped frequencies and damping ratio are given in table 1.

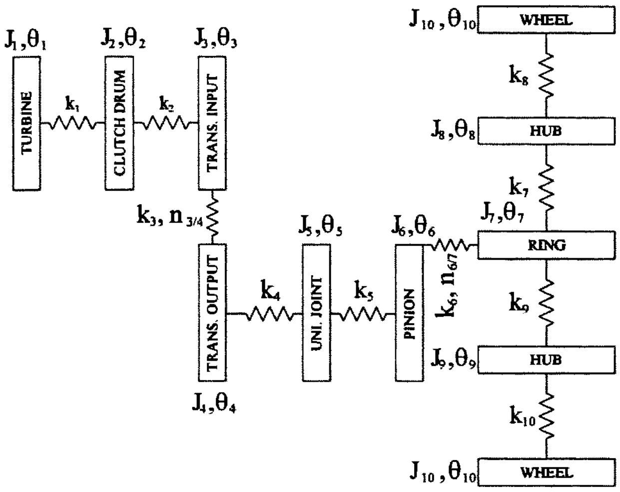

Figure 1: Powertrain model

Damped natural frequency (Hz)	Damping ratio
5	5%
30.5	2%
32.6	2%
91.6	2%
119	2%
401	2%
1591	2%
3343	2%
3567	2%

Table 1: Damped Frequencies

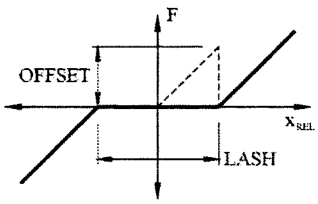

Figure 2: Gear mesh function

3. NUMERICAL SIMULATIONS FOR TRANSMISSION AND DRIVELINE CLUNK

Numerical simulations are carried out to investigate gear impacts from transient vibration in powertrain systems. This simulation is for engine torque tip-in and then tip-out to observe clunk. We investigate the influence on clunk from damping, engine excitation and lash. We solve equation (1) for a non-linear torque vector with a non-linear stiffness matrix and a modal damping matrix. The simulation has been programmed in Matlab using the ODE15S solver. This solver is appropriate for this system as it is low order and can be set to use Gear's methods. Backlash is modelled with a non-linear stiffness algorithm with the stiffness as a function of the relative displacement of meshing teeth (Figure 2). For the transmission sun and ring gear the mesh stiffness is given by (same algorithm for final drive mesh):

$$k_3 = \begin{cases} k_3 & if \ \left|(x_{rel})\right| \geq lash/2 \\ 0 & if \ \left|(x_{rel})\right| < lash/2 \end{cases} \quad (5) \qquad \text{where,} \quad x_{rel} = r_3\theta_3 - r_4\theta_4 \quad (6)$$

Using the non-linear spring force a torque vector is needed to account for the offset of force from the zero position. The global torque vectors are modified to include the torque offset:

$$T_3 = -sign(x_{rel})r_3k_3\, lash/2 \quad (7) \qquad T_4 = sign(x_{rel})r_4k_4\, lash/2 \quad (8)$$

In the simulation engine harmonics are added at 1st, 3rd and 6th order of engine speed. The vehicle resistance torque was taken as a constant. Drag torques were not included in this model. The torque vector for the system includes the engine torque with harmonics (Figure 3), torque offsets for the gear mesh and the vehicle resistance torque. The time step for the simulations was $1x10^{-5}$s. This gives 20 points in a 5000 Hz wave. Solutions converged as the timestep was reduced towards this value. Gear lash at both meshes is 0.001m. The simulation uses the damping ratios given in table 1. The component speeds, input and propeller shaft relative angular displacements and mesh relative displacements are given in Figures 4 to 8. The tip in and out causes a sharp rise and fall in angular displacement twist and transient response mainly in the lowest torsional mode for the system, 5Hz. The transient response is large enough for a torque reversal (relative displacement reversal) in the gear meshes – after separation the gears have to impact to reengage contact. The horizontal lines in Figures 7 and 8 show the gear backlash, in between the lines is the zero stiffness region. Double and single sided impacts can be seen in this solution. Due to the impacts there is high frequency response that is transmitted through the bearings and case as sound – the clunk. Multiple clunks are shown in the simulations to occur closely to the frequency of the lowest mode.

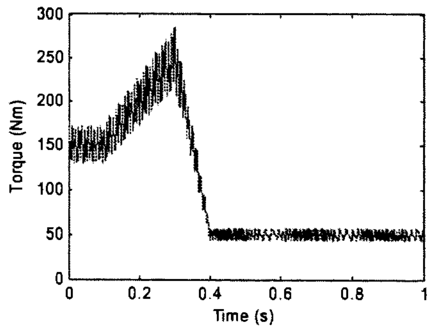

Figure 3: Engine Tip-In Torque and Harmonics

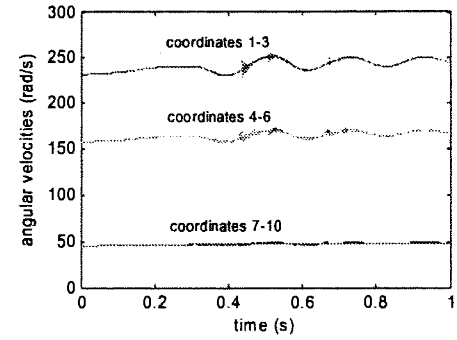

Figure 4: Component Speeds

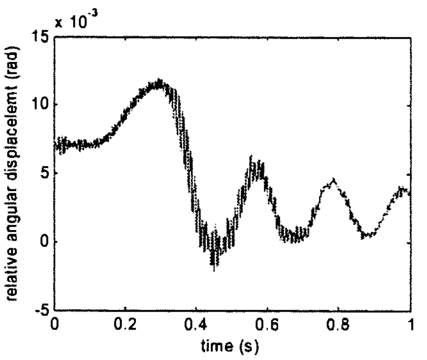

Figure 5: Input shaft

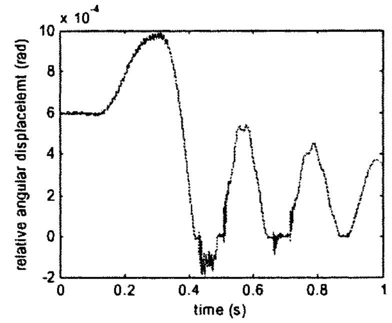

Figure 6: Propeller shaft

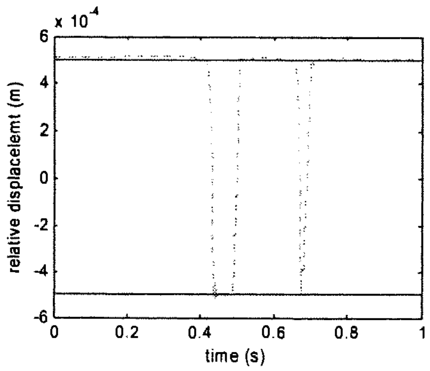

Figure 7: Transmission mesh

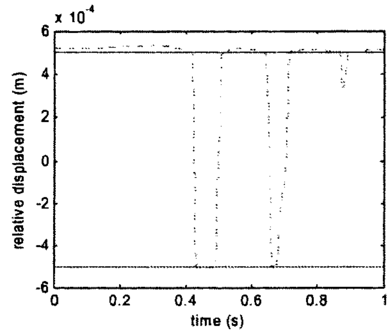

Figure 8: Final drive mesh

The simulation was rerun with the damping of the lowest mode increased to 10%. There is now only one double side impact for each gear pair (Figures 9 and 10). The increased viscous damping has damped out the shuffle response enough to reduce multiple impacts. Damping for the lowest mode was increased further to 15%. The result for clunk was basically the same as with 10% damping. Increasing the damping further is not greatly reducing the amplitude of the first cycle of transient response. The increased viscous damping has damped out the shuffle response only enough to reduce multiple impacts. The engine tip-in function is simplified and designed to be jerky for the powertrain. The lag of torque transmission through the torque converter is not considered. However transient vibration at the shuffle mode could be caused by throttle change, gear shifting and braking and combinations of these will be jerky. The tip in-out function allows us to simulate powertrain dynamics to investigate some influences on gear impacts. We doubled the period of the tip-in and out and reran the original simulation. Naturally the shuffle mode was excited with less response and there was only one single sided impact in each mesh. The original engine harmonics had components at 1st, 3rd and 6th order and amplitudes of 2%, 10% and 5% of the mean engine torque. The amplitudes of the harmonics set to zero and the original simulation rerun. For this simulation, Figures 11 and 12 show the transmission clunk is hardly changed but that there are no engine harmonics in the input shaft. The original harmonics were doubled and the simulation was rerun. There was increased transmission of engine harmonics through the flywheel shaft but the occurrence of clunk was much the same in the transmission mesh. For the final drive it was basically identical. The original simulation was rerun with each lash doubled to 0.002m and also halved to 0.0005m. Figures 13 and 14 show the relative displacement results for the final drive mesh. The number of impacts can be seen to be decreasing with increased lash. In [4] it was mentioned that driveline clunk severity increased with lash but not necessarily linearly. In this case we believe the number of impacts is decreasing as the period of the lowest mode is slightly increased by larger lash and this allows more time for the mode to be damped in between lash cycles. In fact increasing each lash to a large 0.005m gave a result with even less impacts.

4. CONCLUSIONS

This paper provides some numerical results for our research into clunk. Clunk in vehicle powertrains is caused by of transient torques applied to the system. The rate of rise and fall of accelerating engine torque is an important aspect of vehicle performance, reducing it reduces shuffle and thereby clunk - but this is detrimental to performance. Damping of the lowest mode reduces multiple clunks and partially inhibits the first clunk from occurring. The filtering of engine harmonics from the torque converter was found to have little effect on clunk. We expect drag torques from the torque converter to have a significant effect. Our ongoing research includes further investigation into the effect of vehicle assent and descent loading, braking, damping, drag torques and mesh excitation. Also we would like to run simulations and experimental verification for driveline clunk with our vehicle powertrain test rig.

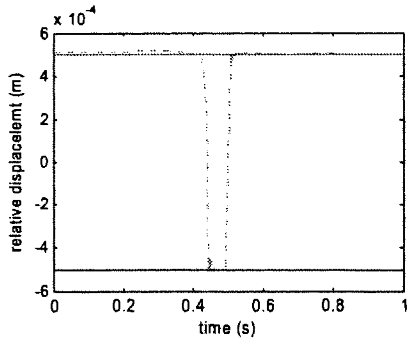

Figure 9: Transmission mesh

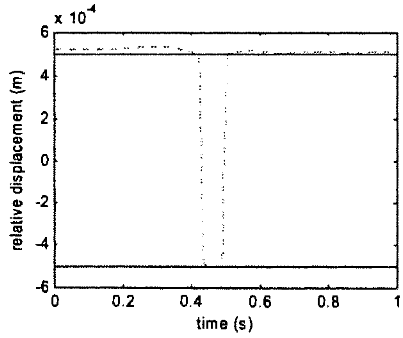

Figure 10: Final drive mesh

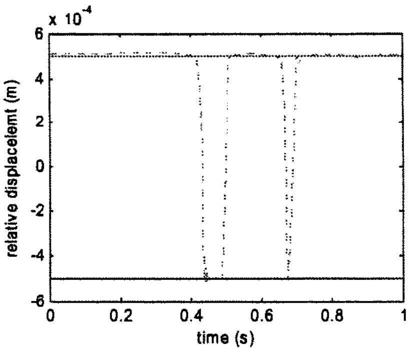

Figure 11: Transmission mesh

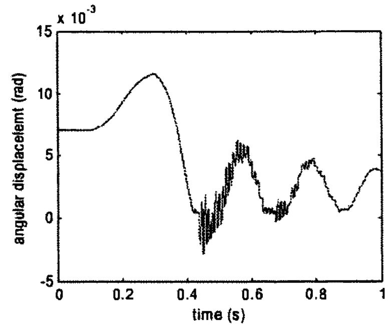

Figure 12: Input shaft

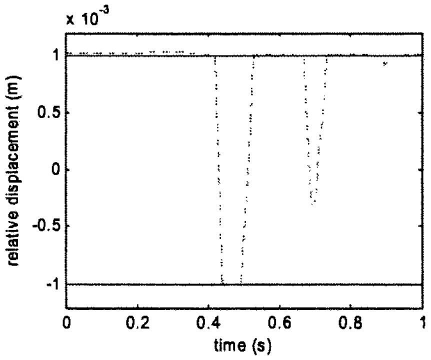

Figure 13: Final drive mesh (lash doubled)

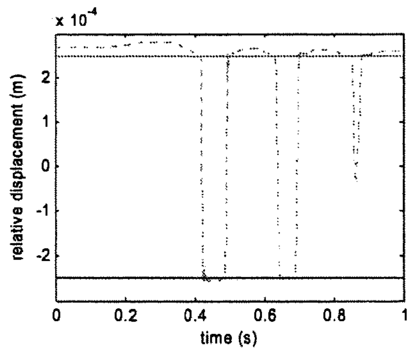

Figure 14: Final drive mesh (lash halved)

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