Clonk: an onomatopoeic response in torsional impact of automotive drivelines

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Abstract: Vehicular driveline is a lightly damped non-linear dynamic system that is prone to noise and vibration response when subjected to excitation. There are many sources of excitation such as torsional impact caused by the take-up of backlash in the powertrain system. Such sources of excitation exist in transmission backlash, in driveline splines and in pinion–ring gear contact in the differential. Abrupt application or release of the throttle in slow moving traffic or rapid engagement of the clutch can be followed by an onomatopoeic response of the driveshafts, referred to in the industry as clonk. This is a disagreeable, audible and tactile response in some vehicles and can also coincide with every cycle of low-frequency longitudinal vehicle response, commonly referred to as shuffle or shunt. This paper describes the phenomenon of clonk and investigates its occurrence both by an experimental technique and by detailed modal analysis of driveshaft pieces. It is shown that finite element predictions agree well with the experimental findings and that the high-frequency structural modes can lead to discernible radiated noise. The preliminary findings reported here point to a need for a more detailed elasto-acoustic analysis.

Keywords: automotive driveline, driveline clonk, torsional impact, hollow cylinder, modal response

1 INTRODUCTION

The vehicle powertrain system comprises the engine, clutch, transmission system, driveshafts and rear axle and the connections to the vehicle body. This is a complex assembly of active and reactive dynamic elements. The system is highly non-linear and lightly damped and is readily excited by engine and road inputs. As such, the driveline system is a source of noise, vibration and harshness (NVH) concern that spans a large spectrum of response frequencies. Of particular interest in this paper is driveline clonk (an onomatopoeic response), a short-duration, audible, high-frequency transient response. This occurs as a result of a load reversal in the presence of lash in the system. The load reversal is usually caused by torsional impact of contacting elements by take-up of backlash, for example in gearing in transmission, spline joints or at the differential. Load reversal can also manifest itself in low-frequency longitudinal driveline shuffle. The initiating cause of load reversals is often the rapid application or release of the throttle from coast or from drive condition while in low gear at low road speeds. It can also occur after gear selection and when the clutch is rapidly engaged.

The rapid throttle demands are converted to system torsional impulses. The driver requires a responsive vehicle. The ill-configured driveline, however, can excite an unwanted low-frequency longitudinal mode of vibration known as *shuffle*. Clonk may be heard on the first cycle of the shuffle response. Krenz [**1**], Hrovat [**2**] and Tobler and Tsangerides [**3**] have studied the relationship between audible clonk and shuffle. They have all shown that the first swing on the transient torque, which can be 2–3 times the steady state response, is perceived as a short-duration jerk pertaining to clonk. The initial clonk response is then followed by shuffle cycles. In severe cases, multiple clonks may occur with each cycle of the shuffle response. The shuffle frequency is usually in the range 2–7 Hz, with the clonk duration varying from 0.25 to 5 ms, both depending on driveline configuration and structural materials.

Tsangerides *et al.* [**4**] have also made an important contribution in the torsional analysis of drivelines, using a lumped parameter model, including the effect of backlash using zero-rated springs. They defined tip-in

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jerk as the rate of change in vehicle acceleration that may be accompanied with an audible clonk. Petri and Heldingsfeld [**5**] have also observed that sudden throttle change transforms into vehicle acceleration/deceleration which may excite system shuffle accompanied with a hard metallic clonk. They observed such acceleration peaks at the input shaft and the differential.

It is therefore envisaged that the NVH spectrum will contain contributions from low frequencies, mainly owing to rigid response of the entire system to high-frequency responses due to flexure and combined torsion–deflection modes of system components such as driveshafts. The low-frequency content is accounted for by shuffle and driveline angulation and these can be obtained by rigid multibody analysis [**6**, **7**]. The high-frequency content of the spectrum is expected to be due to the response of the elastic members which can be obtained by modal analysis.

Arnold and Warburton [**8**] have shown that for thin cylindrical shells an infinite number of axial vibration modes, m, are theoretically possible, each with a corresponding number of circumferential forms, n. Therefore, to define a mode, the m and n values must be specified. Forsberg [**9**] used a method outlined originally by Flugge in 1934 and concluded, as did Arnold and Warburton, that for any given set of values of (m, n) there are three natural frequencies corresponding to three mode shapes that are obtained by the solution of the cubic energy equation.

Recent investigations have employed the use of finite element analysis, e.g. to compute the natural frequencies of a clamped cylinder [**10**]. The driveline characteristics can be regarded as piecewise linear, where the transition from a very low torsional stiffness rate to a very high rate as in a hard clutch disc can be instantaneous, rendering a 'stiff' system with a large number of widely split eigenvalues. Therefore, finite element studies can provide a more efficient way of dealing with structural responses of lightly damped driveline pieces.

There are a large number of axial mode orders for cylindrical shells and an almost infinite number of natural frequencies that may be excited into resonance. Therefore, for lightly damped, hollow, cylindrical driveshafts a very high modal density results. This paper investigates the most significant modes of the driveline pieces, using their finite element models. The results of finite element analysis are verified by an experimental rig subjected to shock impulse torques.

2 EXPERIMENTAL RIG

Unlike vehicle shuffle, there are only a few investigations related to drivetrain clonk and much of the work carried out in the industry has been confined to subjective vehicle evaluations and is usually not reported in the literature. However, a recent experimental investigation by Biermann and Hagerodt [**11**] has shown that repeatable clonk conditions can be achieved and measured by experimental rigs that include all elements of a drivetrain system subjected to an instantaneously released preload torque. The authors used their experimental set-up in order to validate their multibody dynamic model. Their experimental technique, carried out for a front wheel drive system, has shown a simple and cost effective way of investigating clonk. Their approach is extended in this paper to the study of the clonk response of a two-piece rear wheel driveline system.

The experimental investigation must ensure the movements of every transmission and structural component along the torque path. As high-frequency vibrations are responsible for the clonk response of the system, monitoring of accelerations of structural elements are required at a multitude of locations.

Figure 1a depicts the experimental static drivetrain rig, representing a typical two-piece light diesel truck drivetrain system. Figure 1b is a schematic representation of the experimental rig. The rig includes the flywheel, clutch, transmission, two-piece driveshaft, differential and rear axle assembly, including hubs and drums. It therefore includes all lash zones in the drivetrain. The transmission is fixed to the ground through compliant mounts, and the bell housing, driveshafts and rear axle are mounted on bedplates fixed to the ground. The driveshaft assembly universal joint angles are set to nominal with no suspension test latitude. The clamped clutch transmits the flywheel impulse torque to the transmission input shaft via the torsional friction disc springs. The transmission is placed in second gear. This configuration renders a static test rig that incorporates all the drivetrain lash zones.

With the engine disconnected, an impulse torque is applied directly to the flywheel. The shock pulse must be simple and well defined, as well as being repeatable, reproducible and representing the actual shock environment experienced by the vehicle. Theoretically, a shock impulse contains energy that spreads over the entire frequency domain. In practice, however, a shock pulse has a finite duration which should be short in order to excite the frequency range of interest. The torsional impulse applied to the drivetrain rig was varied between 80 and 180 ms. The corresponding pulse generated by lash take-up was of the order of 1–2 ms which is sufficiently short to excite the higher-frequency clonk modes. The specified preload torque is applied to the system to take up all lashes and to compress the clutch springs. This preload torque is held on a low-inertia disc brake. The stored energy is then instantaneously released in the form of a ramp sawtooth pulse. The slip release of the preload torque achieves a final steady state value of 150 N m.

Calibrated accelerometers are located and adhered to the various locations shown in Fig. 1b. These accelerometers record tangential acceleration which is then converted into angular components. In order to place an accelerometer pick-up at the ring gear position, the axle was tested without lubrication. Microphones were also placed at the transmission and the rear axle locations. Twelve-channel data acquisition is carried out at a data sampling rate of 12 500 samples per second, corresponding to a time interval of 80 ms.

A series of transient acceleration signals is shown in Fig. 2 for the various pick-up locations shown in Fig. 1b. It can be observed that the signal amplitudes at the transmission flange and the axle differential flange are larger than those from the driveshaft piece since larger lashes exist at these locations. The corresponding torque ramp is also shown in the figure. Figure 3 shows more details of the torque ramp, along with a typical surface structural response of a driveshaft tube. It shows a 1–2 ms impact, followed by approximately 30 ms of driveshaft tube ringing.

The noise monitored by microphones at the transmission and at the rear axle indicates a much higher level at the transmission. Sixteen test runs were therefore carried out, each time monitoring the acceleration output from pick-up location 13 as well as the corresponding noise level acquired by microphone location M1. These are shown in Fig. 4. It is clear that the same

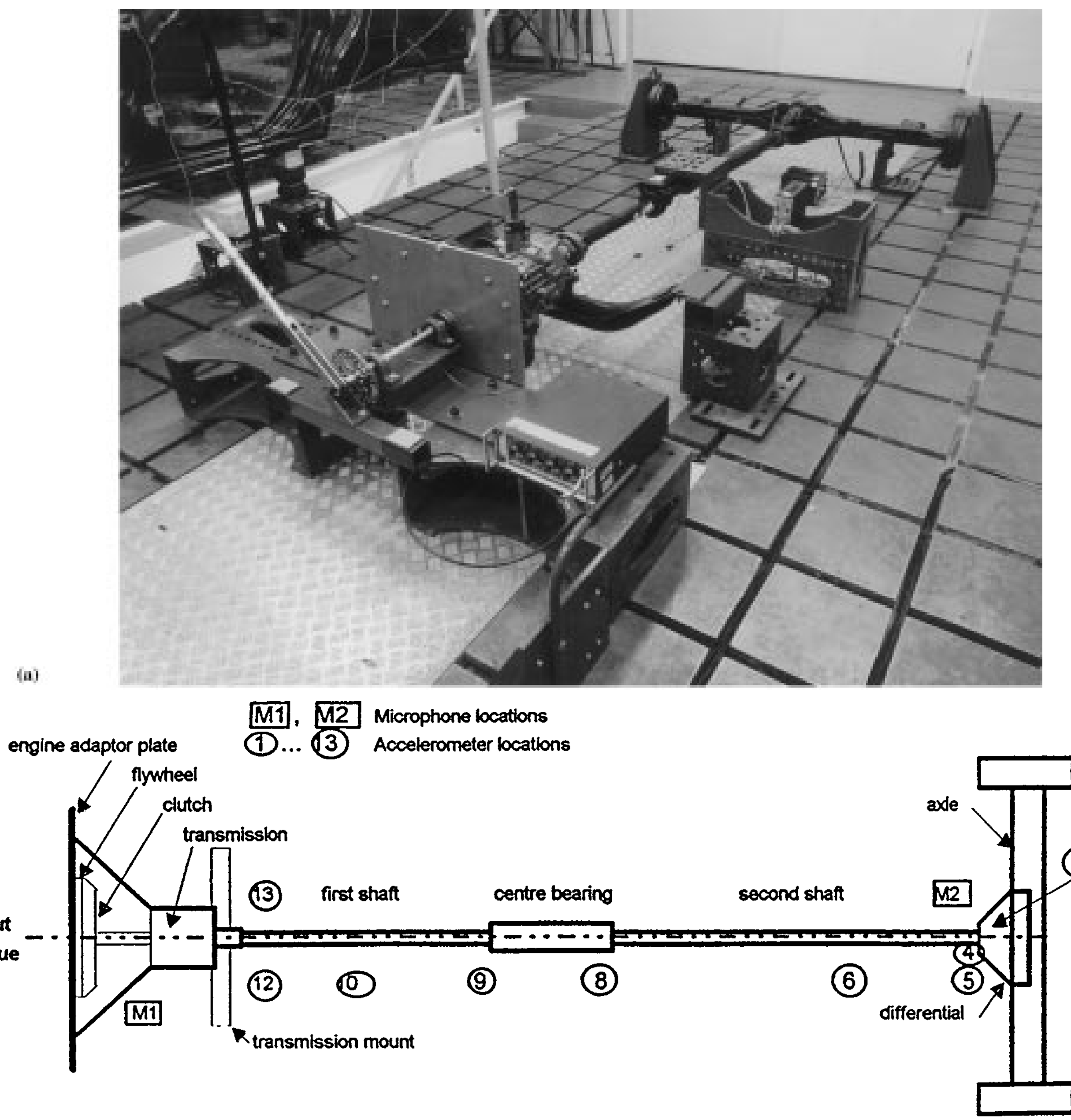
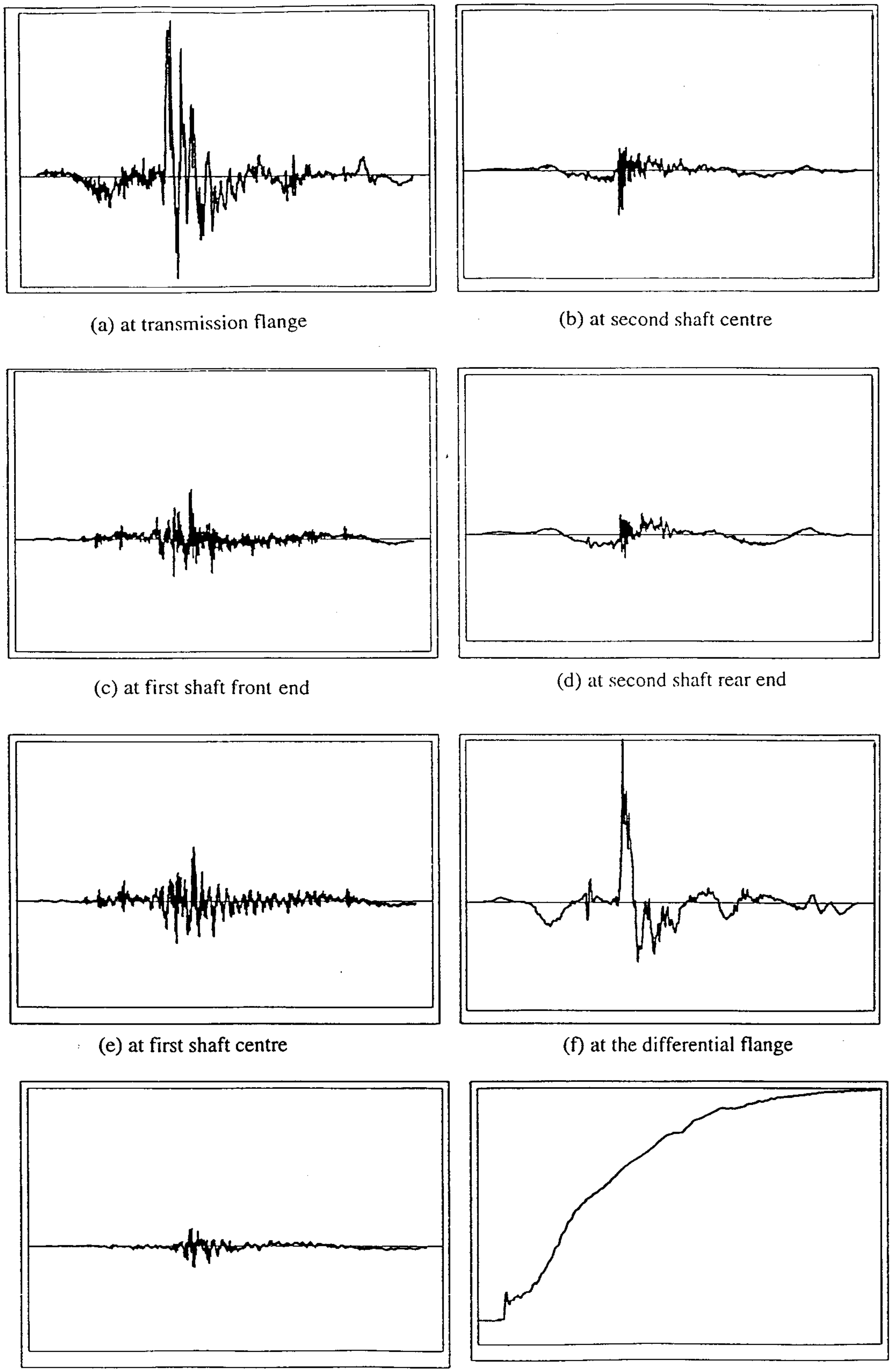


Fig. 1 (a) Photograph and (b) schematic representation of the experimental rig



(a) at transmission flange
(b) at second shaft centre
(c) at first shaft front end
(d) at second shaft rear end
(e) at first shaft centre
(f) at the differential flange
(g) at second shaft front end
(h) the input torque ramp

Fig. 2 Transient acceleration output signals at various locations along the driveline trends are observed for structural response and the radiated ringing noise.

The time response signals for any particular location are averaged over a number of test runs and subjected to fast Fourier transformation. The spectral contents for various locations are thus obtained. Figure 5 shows a typical spectral composition for the driveshaft tubes, indicating high-frequency contents in the range 1.5–5 kHz. In fact, significant contributions occur at 1–1.5, 2.25 and 2.75–3.25 kHz for the front tube and 1.75, 2.5–2.75 and 3.5–3.75 kHz for the rear tube. Note that the figure indicates three different spectra. One is referred to as the baseline response for unfilled hollow tubes. The other two indicate the reduction in amplitudes of vibration when the tubes are either filled with foam or wound with steel strip. It is clearly difficult to construct mode shapes at these various frequencies owing to the very large array of pick-up points that would be required. Therefore, the need for constructing finite element (FE) models becomes evident. The intention will be to create high mesh density models for high-frequency sweep modal analysis to be carried out in the first instance. Once the numerical results are validated against the experimental evidence, specific mode shapes can be obtained and in turn used in a future elasto-acoustic analysis.

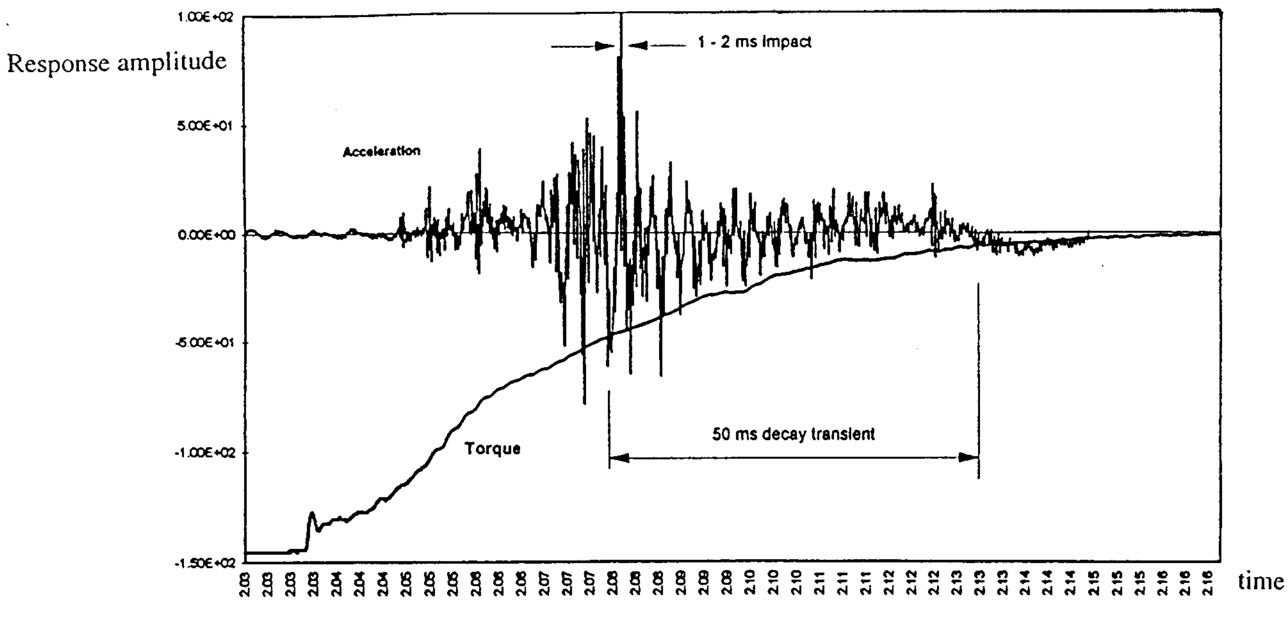


Fig. 3 Typical torque ramp and driveshaft transient response

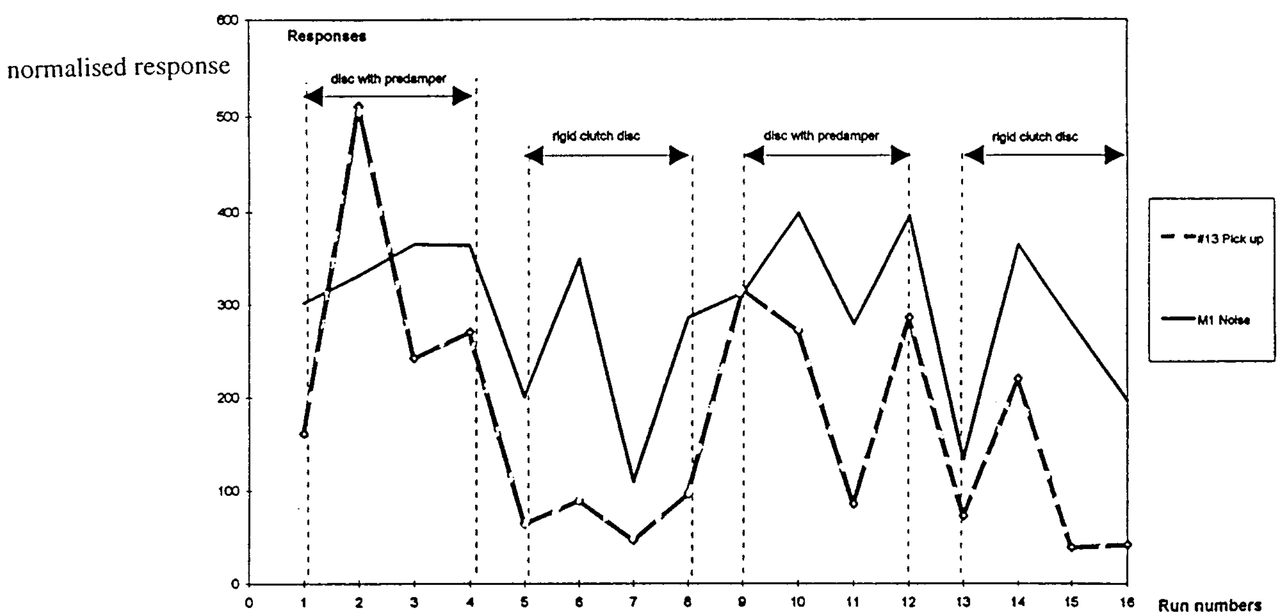


Fig. 4 Vibration and noise response at the transmission flange

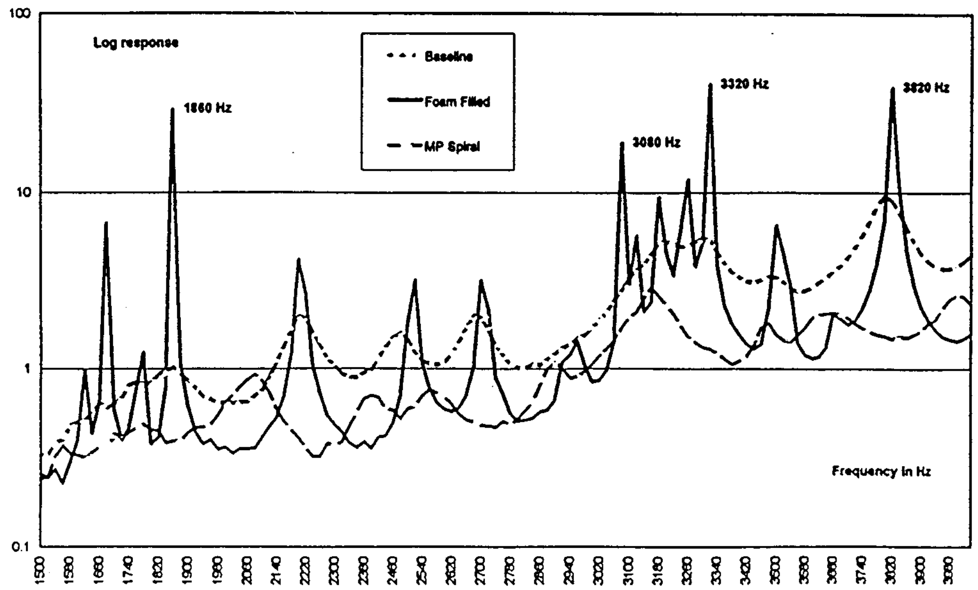


Fig. 5 Experimentally obtained frequency sweep spectra of driveshaft tubes (different constructions)

3 FE MODEL OF HOLLOW CYLINDRICAL DRIVESHAFTS

The driveline consists of two hollow cylindrical driveshafts, both with closed ends. The shorter of the two cylinders is located at the transmission end of the driveshaft. It is 435.5 mm long and of constant diameter (50.7 mm o.d.). The wall thickness for both the cylindrical shell and each of the end walls is 1.65 mm. The second tube is close to the differential and is 958.5 mm long. The diameter of this tube is not constant along its length. At a distance of 172 mm from each end, the outside diameter changes from 75 mm to 90 mm. The wall thickness at all points is again 1.65 mm. The material of construction is steel with a Young's modulus of 210 GPa, a Poisson's ratio of 0.3 and a density of 7.801 kg/m^3.

For both the front and the rear tubes the diameter–thickness ratio is large enough to allow the ABAQUS shell elements S4R5 to be employed as opposed to the three-dimensional elements that are usually employed for modelling of tubes. This type of element is a four-noded curvilinear (doubly curved) shell element with a reduced integration effort. Eight-noded elements can also be employed for improved accuracy, but in this case a large amount of data results owing to the large number of elements needed to capture the high-frequency structural modes. This causes computational problems. Owing to the high-frequency nature of the clonk problem and the large number of modes (i.e. eigenvalues), a large mesh density is found to be necessary. For both models of the front and rear tubes, 106 shell elements are employed along the tube lengths and 32 elements in each cross-section through the thickness of the tube, thus rendering a mesh density of 32×106 elements, to which approximately 100 elements are added at each of the tube ends. Such a large mesh density is required to capture the elastic modal response of each tube for the frequency range 0.5–5 kHz. A major problem with such a high mesh density and a large number of modes to be investigated is the required computational effort. For instance, 160 runs of the ABAQUS model for the front tube were planned, requiring a considerable CPU time allocation. The same mesh density was employed for both front and rear tubes, resulting in larger elements for the rear tube analysis. However, care was taken to ensure computational accuracy with larger elements in the case of the rear tube.

The two end-pieces in each case are made up of a square at the centre and slightly distorted elements around the edges. This is performed since less computing power is required for analysis of quadrilateral elements compared with the triangular ones that would normally be employed to model a circular geometry. The square element at the centre is held still (i.e. constraining all its degrees of freedom) during the analysis. This represents the actual physical constraints in the real system.

Other output control parameters include the number of iterations in the modal analysis and the number of vectors stored during the iteration process [**11**]. The number of iterations ranges from 100 to 150, with the number of vectors ranging from 20 to 30. This renders detailed analyses in the case of both tubes. All models were initially built using PATRAN which has a superior pre-processing capability to ABAQUS. The pre-processed models are then exported to ABAQUS for analysis.

This paper is not concerned with the low-frequency content of driveshaft spectra. However, it is interesting to note that the first rigid modal response of the two-piece driveshaft about the centre bearing occurs at a frequency of 24.4 Hz, which is safely away from the shuffle response of the driveline, in this case occurring at 3 Hz.

Figure 6 shows the spectrum of vibration for the front tube, obtained by a frequency sweep modal analysis for the high-frequency region 1000–4000 Hz. The significant frequency regions in the spectrum are around 1–1.25, 2.25–2.26, 2.75–3.25 and at 3.5 kHz. These findings coincide with some of the experimentally obtained results in Fig. 5. More than 3000 modes are identified in the high-frequency region 1–5 kHz in this analysis. It would be impractical to investigate the modal response behaviour at all of these frequencies. One approach is to obtain the mode shapes at the most significant frequencies in the spectrum of vibrations. The frequencies with highest amplitudes are referred to as pole frequencies. A narrow-band window (pole frequency ± 10 Hz) for modal response investigation around each of these pole frequencies is selected. A frequency sweep is then carried out in each case, determining the maximum response and the exact response frequency. The pole frequencies for the front tube are at 1750, 2450 and 3250 Hz as indicated by the experimental evidence, as well as by the numerically obtained spectrum.

The most interesting modal responses for the front tube occur at frequencies of 2445 and 3257 Hz. Figure 7a shows the circumferential response corresponding to the region around the modal frequency of 2445 Hz with the waveform $n = 2$. The axial mode shape is dominated by a waveform of the order $m = 3$. The numerical results for the axial mode, however, also seem to suggest a further waveform with $m = 26$, modulating with the aforementioned dominant waveform. Such characteristics correspond to a non-linear jump phenomenon, where two amplitudes of vibration exist at the same modal frequency or a transition from a low-torsional mode to a high-frequency response can take place (as in stiff systems). This is shown in Fig. 7b. Such a phenomenon has also been observed in the case of a thin clamped cylinder in reference [**4**]. However, the results may also be misleading as the amplitudes of oscillations are amplified by a factor of 15 in this case. The experimental observations have shown repetitive clonk response at this frequency. It is thought that under these conditions structural instability can occur in the case of the front tube, as described above.

Another significantly troublesome frequency region is 3200–3400 Hz. Figure 8 shows the circumferential modal response at a frequency of 3257 Hz, where $n = 2$ for the circumferential waves.

An interesting feature of the experimental and some of the FEA solutions is that the modal response in the frequency band 1750–2000 Hz points to flexure of the

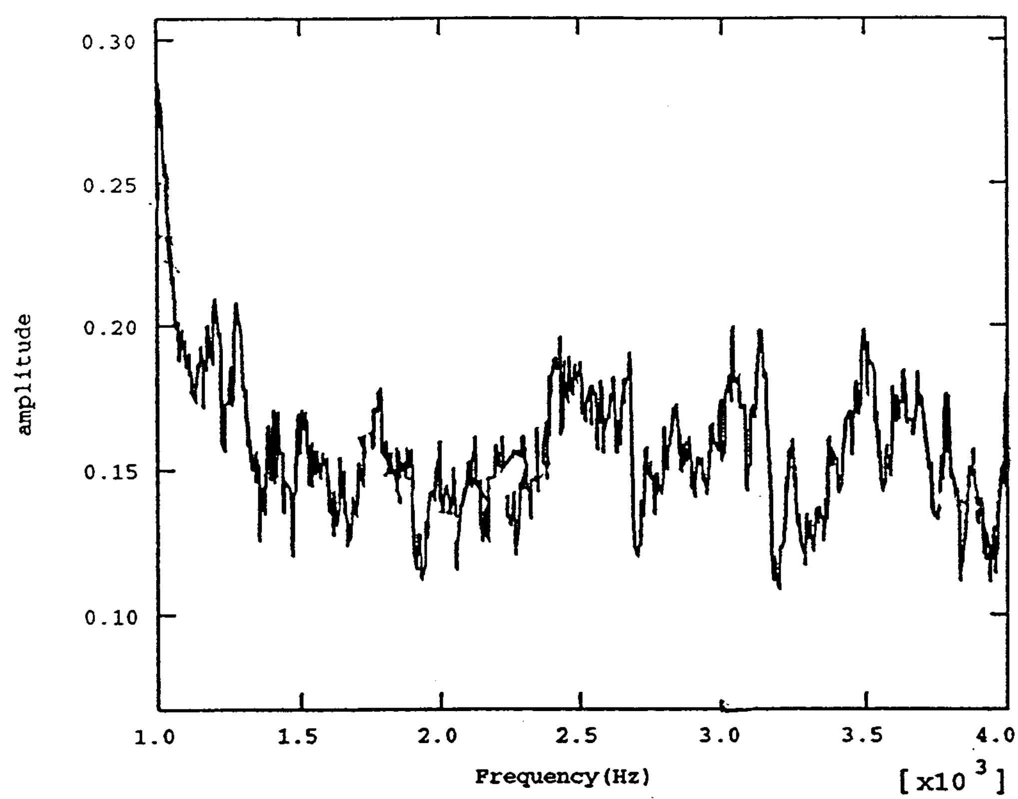
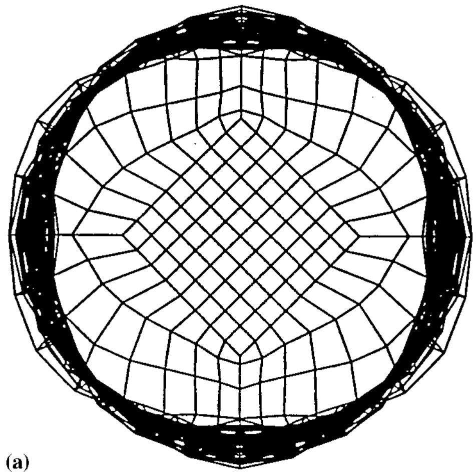
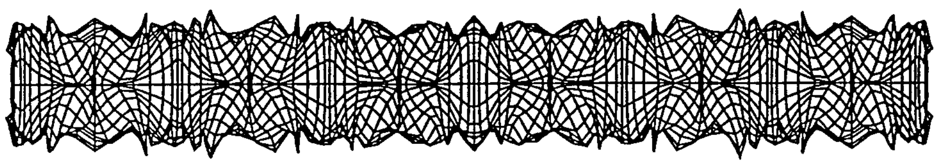


Fig. 6 Front tube frequency sweep spectrum of vibration



(a)



(b)

Fig. 7 (a) Circumferential modal response and (b) axial modal response of the front tube at 2445 Hz

tubes with a large number of axial waves, while in the region 3000-3750 Hz the compression mode becomes progressively dominant. In the intermediate region a mixed mode of response can be observed, nowhere better than in Fig. 7b, where the compressioning effect is beginning to take hold while the axial bending effect is still evident (thus the dual-amplitude ripple response characteristics). Such an effect for clamped response of thin cylinders is also observed in reference [**4**]. This rather interesting classification appears to give credence to the amplitude variations in the frequency sweep experimental spectra of Fig. 5. The baseline spectrum (without damping material) exhibits large amplitudes in all the above-mentioned frequency bands. With foam-filled tubes, the vibration amplitudes are significantly reduced in the lower-frequency bands, as the foam acts as an effective damper against the largely circumferential deformations. At the higher-frequency band, where the compressioning effect is dominant, the damping ability of the foam is rather limited (note that the foam structure introduces little damping in the axial direction). In this region a spiral spring wound structure is more effective as it reduces the rather high axial stiffness of the system and thus the amplitude contribution at high frequencies. This is a significant find and seems to indicate that particular forms of damping should be employed for isolation of structural response in given frequency bands.

4 CONCLUSION

An experimental rig simulating two-piece rear wheel drive light trucks is configured which includes all the drivetrain lash zones. A predetermined preload torque is applied to the system and instantaneously released, recreating clonk conditions that conform well with the vehicle observations. The procedure was found to be simple, repeatable and cost effective for investigation of the clonk phenomenon. The response spectra con-

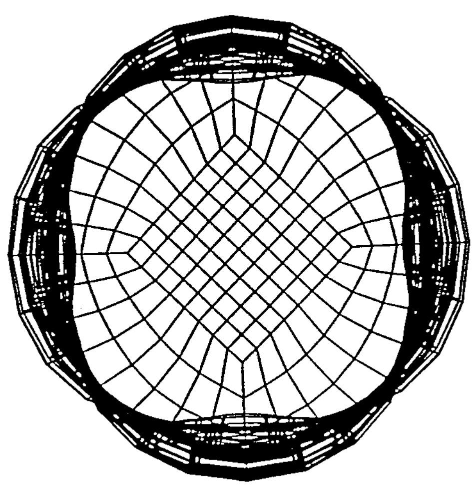


Fig. 8 Circumferential modal response of the front tube at 3257 Hz

structed from the experimental tests and obtained by modal frequency sweep of driveshaft tubes conform closely. Narrow-band modal response investigations of significant response frequencies indicate particular regions of interest where circumferential flexure or axial compressioning become significant. It is found that different forms of damping materials are more effective for given modal responses. This can be significant in the reduction of radiated onomatopoeic noise, referred to as clonk, as the noise monitored by microphone pick-ups has been shown to correlate well with the corresponding vibration amplitudes.

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REFERENCES

1 **Krenz, R.** Vehicle response to throttle tip in/tip out. SAE paper 850967, 1985.
2 **Hrovat, D.** Influence of drivetrain/chassis dynamics on tip in and back out driveability. Ford Motor Company Internal Report, 1989.
3 **Tobler, W.** and **Tsangerides, M.** Dynamic behaviour of a torque converter with centrifugal bypass clutch. SAE paper 850461, 1985.
4 **Tsangerides, M., Tobler, W.** and **Heerman, C.** Interactive computer simulation of drivetrain dynamics. SAE paper 850978, 1985.
5 **Petri, H.** and **Heldingsfeld, D.** The hydraulic torsion damper—a new concept for vibration damping in powertrains. SAE paper 892477, 1989.
6 **Powell, N. N.** and **March, J. P.** Practical applications of dynamic system modelling in powertrain and vehicle refinement. In *Multi-body Dynamics: Monitoring and Simulation Techniques* (Eds H. Rahnejat and R. Whalley), March 1997 (Mechanical Engineering Publications, London).
7 **March, J. P.** and **Amphlett, S. A.** Studying low frequency vehicle phenomena using advanced modelling techniques part 1: Construction of a driveline model. ISATA 94ME043, 1994.
8 **Arnold, R.** and **Warburton, G.** Flexural vibrations of the walls of thin cylindrical shells having freely supported ends. *Proc. R. Soc.*, 1949.
9 **Forsberg, K.** Influence of boundary conditions on the modal characteristics of thin cylindrical shells. AIAA, 1964.
10 **Santiago, J.** and **Wisniewski, H.** Convergence of finite element frequency predictions for a thin walled cylinder. *Computers and Structs*, 1989.
11 **Biermann, J. W.** and **Hagerodt, B.** Investigation of clonk phenomenon in vehicle transmission—measurement, modelling and simulation. In *Multi-body Dynamics: Monitoring and Simulation Techniques* (Eds H. Rahnejat and R. Whalley), March 1997 (Mechanical Engineering Publications, London).