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Root cause identification and physics of impact-induced driveline noise in vehicular powertrain systems

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Abstract: Numerical and experimental investigations shed light on the root causes leading to the emergence and persistence of an acute metallic noise in rear wheel drive light truck drivelines. Sudden demands in engine output torque combined with the presence of lash zones give rise to a phenomenon that is onomatopoeically referred to as clonk. Its multi physics nature requires a comprehensive study, which includes rigid multi-body dynamics, flexible body oscillations, and noise radiation computation. The verification of numerical results is achieved through the design and implementation of a transient dynamic experimental rig, which comprises the complete drivetrain from the engine flywheel to the rear axle. Parametric studies reveal high-frequency contributions in the driveline vibration response of certain structural modes of the driveshaft pieces, which are induced by remote impact of meshing transmission teeth through backlash. The numerically predicted spectrum of vibration is in good qualitative agreement with the experimental measurements. Combined study of the aforementioned results reveals the components that amplify the clonk noise.

Keywords: impact-induced driveline noise, multi-physics, accelerative and ringing response

1 INTRODUCTION

During the last few decades, the continuous trend to reduce vehicle engine radiated noise has resulted in significantly 'quieter' engines, while power output has increased. These two factors are strongly connected to customer demands and, therefore, are considered as imperative priorities by powertrain engineers [1]. However, the aforementioned improvements in vehicle performance have also resulted in a plethora of impact-induced drivetrain noise and vibration, which are amplified in the cabin interior. Moreover, the use of lighter components to decrease vehicle overall weight has unavoidably amplified the structural vibrations and has contributed to new concerns regarding noise, vibration, and harshness (NVH).

In a typical vehicular driveline system, the vibration and noise spectrum contains contributions

from a significant number of components, ranging from a few Hz to several kHz. Clonk is one of the major concerns and it can be described as a short-duration, audible, wide-band (300–5000 Hz) elasto-acoustic phenomenon, that occurs as a result of load reversal in the presence of lashes in the driveline. Through this impulse action, a large number of structural modes of the lightly damped driveline system are excited and the high-frequency content of the modal response propagates to the cabin through the powertrain mountings on the floor or as airborne noise, when the vehicle windows are lowered.

A number of investigators – mainly from the automotive industry – have tried to eliminate this problem using various palliative measures. The tip-in and tip-out drivetrain response (sudden increase/decrease in output engine torque), which is one of the main ways to introduce the driveline metallic noise, was investigated experimentally [2]. The effect of a hydraulic torque converter on the tip-in and tip-out response of a rear wheel drive (RWD) vehicle was studied, both experimentally and numerically [3]. The effect of hydraulic torsional dampers on tip-in/tip-out clonk,

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as well as on clutch operation, which is another important factor in the appearance of high-frequency metallic noise, was investigated [4]. All these studies have concluded that the use of the above palliative methods can improve the drivetrain performance with regard to clonk. However, the high costs for mass production and the complexities of the powertrain system, which could potentially lead to the persistence of other NVH concerns, have prevented the use of these solutions.

It has become apparent that, although there is a reasonable understanding of the conditions leading to clonk, further investigations are needed to clarify its root causes and mechanism of propagation. This requires advanced modelling techniques and the inclusion of system complexities.

Non-linear effects of clutches and transmissions were considered [5], while the complete RWD powertrain system was modelled using lumped masses and direct integration to solve the resulting equations of motion [6]. The parametric studies on torsional vibration were validated by experiments. In an RWD powertrain system investigation that combined experimental and numerical studies [7–9] it was concluded that the driveshaft tubes significantly contribute to the persistence and radiation of the metallic noise. A front wheel drive (FWD) system was examined parametrically and the lumped mass model was verified experimentally [10]. Reduction in the transmission component inertia and angular speeds was proposed to overcome the problem. Some researchers have investigated the applicability of genetic algorithms to the optimization of driveline vibrations [11], while others have employed the multi-body dynamics method to simulate the torsional vibrations of drivelines and explain the effect of specific parameters of interest and the mechanism of vibration transfer paths [12, 13]. Their models consisted of rigid bodies. The hypothesis that the driveshafts contribute significantly to the radiated metallic noise was numerically investigated [14, 15], having introduced the flexibility of driveshafts in their models. Experimental results [8] have confirmed the validity of the latter numerical investigations.

The above investigations revealed the conditions leading to the appearance of clonk and its different forms, and have explained to a certain extent the influence of parameters that affect the severity of clonk. However, the question regarding its root cause identification still largely remains. This is the contribution of the present work, which combines the theory of multi-body dynamics, finite element techniques, and the boundary element method to offer

a generic numerical methodology that can also be used in other NVH investigations. A dynamic experimental rig of an RWD three-piece drivetrain of a light truck was also set up and used for the validation of the numerical work under clutch-actuated clonk conditions. Comprehensive measurements of the driveline vibration response and noise radiation were undertaken. The results have revealed a number of modes observed in both the vibration and noise spectra. Identifying which of these contribute to the metallic clonk noise is the task that leads to the necessary design modification for noise reduction.

2 THREE-PIECE DRIVELINE MODEL

The multi-body dynamic model, built in the commercially available software ADAMS [16] (Fig. 1), is based upon constrained Lagrangian dynamics (more details on the method, the components used, and the constraint types can be found in references [15] and [17]). This is the mechanical model of the drivetrain system. Component flexibility for the high modal density, noise-radiating structures has been included using finite element techniques [18, 19], as shown in Fig. 1. The component mode synthesis (CMS) method was employed to reduce the total number of degrees of freedom (DOF) [15]. Four-noded, two-dimensional shell elements with a pre-specified thickness (1.65 mm) have been used for the three hollow driveshaft tubes. The meshing density is sufficient to capture higher-frequency modes in an accurate manner. Clamped–clamped boundary conditions are applied to the edges of tubes in order to represent the precise assembly conditions in the actual vehicular driveshaft system. A sufficient number of natural modes – including mode shapes up to 12 000 Hz – have been kept during CMS in order to obtain accurate results in the specified frequency area where clonk noise usually occurs (300–5000 Hz). Consequently, each driveshaft super-element includes a variety of modes starting from

Fig. 1 Multi-body model of the driveline mechanical system, flexible components, and examples of mode shapes employed

low-frequency bending modes and leads to complicated shapes that are combinations of bending-axial and high-frequency torsional modes (Fig. 1). The total number of DOF for the reduced model is obtained using the Gruebler–Kutzbach expression as

$$\begin{aligned} \text{Number of DOF} &= (\text{flexible body modes}) \\ &+ [6 \times (\text{number of rigid parts} - 1)] \\ &- \Sigma(\text{constraints}) \\ &= (91 + 60 + 85) + [6 \times (35 - 1)] - 193 \\ &= 247 \end{aligned}$$

The excitation to the model is a torque function of 145 N m maximum magnitude (similar to the vehicle engine output for the same gear engagement), which simulates the load generated during the clutch actuation, depending on the speed of the clutch pedal motion. The function varies from zero to the maximum value in the time given according to the experimental conditions measured in hurried clutch actuation.

The procedure for the calculation of transmission gear forces between the engaged tooth pair(s) during the meshing cycle, taking into account the backlash and varying meshing stiffness, is described in reference [15]. The centres of both gears are constrained against any lateral motion and variations from tooth to tooth are neglected. Owing to the lash zones and the variable number of gear tooth pairs that are in contact simultaneously, the equations of motion become strongly non-linear and can be solved using piece wise linear elements with time-dependent coefficients [20].

Since the response of the dynamic system is calculated for the period of interest, the reactions at the driveshaft extremities are obtained with respect to time. These data are used as an input for a transient analysis with the finite element analysis (FEA) models of the tubes. Then, the surface velocities of the shaft wall nodes are computed for every time step. These eventually become the initial conditions for the acoustic analysis. By using these historical data and applying the indirect boundary element method, the sound pressure fields in the exterior domain can be obtained in a natural way, having previously simulated the in-service events of interest. The boundary element method presents certain advantages compared with other numerical methods for acoustic problems, such as the finite element method (FEM) and statistical energy analysis (SEA). The latter is a time-consuming method for high-frequency problems

(such as clonk), where resonant frequencies are densely packed [21]. FEM is usually limited to low frequencies, because an accurate modelling of a large structure for investigation at high frequencies requires a large number of elements [21]. Additionally, FEM requires the computation of the entire domain and it is rather complicated to apply to noise propagation problems exterior to a hollow cavity such as the driveshaft tubes. On the other hand, the boundary element method only requires discretization of the boundary and can be easily applied to both interior and exterior noise problems.

In the case of driveshafts, the boundary discretization was based on the desired maximum frequency to capture, following observations from on-road tests in vehicles of the same type at the premises of the industrial partner. Therefore, the hollow thin-walled elastic shafts were discretized with sufficiently refined meshes to capture the structural modes of interest, as well as the wavelength of the fluid in the acoustic medium. For the maximum frequency of 4100 Hz, the corresponding wavelength is given by

$$\lambda = \frac{c}{f} = \frac{343 \text{ (m/s)}}{4100 \text{ (Hz)}} = 0.0836 \text{ (m)} \quad (1)$$

This yields an element size of 16.7 mm for exterior noise analysis, where five elements per wavelength were used. Virtual microphones were placed around the driveshaft model, in positions depending on the minimum desired frequency (wavelength) to be captured, which was set to 680 Hz, a value that was also defined on the basis of vehicle tests. Therefore, the distance from the source is calculated as

$$\lambda = \frac{343 \text{ (m/s)}}{680 \text{ (Hz)}} = 0.5 \text{ (m)}$$

Figure 2 shows the discretized model of a tube and the virtual microphones surrounding it.

The propagation of small-amplitude waves can be represented by Helmholtz's equation in the frequency domain

$$(\nabla^2 + k^2)p = 0 \quad (2)$$

where

$$\nabla^2 = \frac{\partial^2}{\partial^2 x} + \frac{\partial^2}{\partial^2 y} + \frac{\partial^2}{\partial^2 z}$$

is the Laplacian and $k = \omega/c$. The Green's function

$$G = \frac{e^{-jkr}}{4\pi r} \quad (3)$$

Fig. 2 Discretized boundary element model of a drive-shaft and the virtual microphones surrounding it

is a solution, providing it satisfies the equation

$$\nabla^2 G + k^2 G = -\delta(x, y) \quad (4)$$

where the Kronecker delta

$$\delta(x, y) = \begin{cases} 0 & \text{when } x \neq y \\ 1 & \text{when } x = y \end{cases}$$

Using the indirect boundary element method, the pressure and velocity on the surface boundary Γ are given as

$$p = \int_{\Gamma} \left(G\sigma - p \frac{\partial G}{\partial \mathbf{n}} \mu \right) d\Gamma \quad (5)$$

and

$$v = \int_{\Gamma} \left(\frac{\partial G}{\partial \mathbf{n}} \sigma - p \frac{\partial^2 G}{\partial \mathbf{n}^2} \mu \right) d\Gamma \quad (6)$$

where $\sigma = (\partial p / \partial \mathbf{n})^+ - (\partial p / \partial \mathbf{n})^-$, $\mu = p^+ - p^-$, and $\mathbf{n}$ is the surface normal vector. The acoustic pressure at a point of interest r_i can be written as

$$p(r_i) = \int_{\Gamma} \left[\mu(r_{\Gamma}) \frac{\partial G(r_i, r_{\Gamma})}{\partial \mathbf{n}_{\Gamma}} - G(r_i, r_{\Gamma}) \sigma(r_i) \right] dS_{\Gamma} \quad (7)$$

where r_{Γ} is a point at the boundary Γ [22, 23].

3 THREE-PIECE DRIVELINE EXPERIMENTAL RIG

The three-piece driveline experimental rig is shown in Fig. 3(a). A variety of factors had to be carefully considered during the rig design and measurement

(a)

(b)

Fig. 3 (a) Three-piece driveline experimental rig; (b) positions of all monitoring equipment

procedure in order to replicate the clutch actuating conditions, caused by hasty driver action, leading to clonk. While the main aim is to gain a fundamental understanding of the physics of the phenomenon, particular objectives of this rig are:

- to carry out a detailed experimental investigation to verify the analytical simulation;
- better to understand drivetrain sensitivity to clonk and impulsive torque excitations;
- to identify the sources that contribute in a significant manner to the propagation of noise and vibration;
- to locate the major clonk noise sources in the drivetrain system.

Furthermore, it is important that experimentation should be carried out in a repeatable manner and under controlled conditions.

3.1 Design considerations

The static, wheel-clamped experimental rig [8] was inadequate in the sense that the transient nature of the phenomenon was introduced without regard to the actual inertial dynamics of the drivetrain system. The standard rear axle hydraulic drum brake mechanism on the vehicle was modified to create the vehicle inertia in a repeatable manner. A hydraulic pump was selected to apply pressure to the piston, which in turn pushed the brake shoes against the

drum. This method is quite straightforward and reasonably controllable.

The combination of an electric motor–inverter provided the necessary specifications to accelerate the drivetrain up to the testing speed of 1500 r/min with a maximum torque of 145 Nm, corresponding to the observed and recorded clonk conditions on the actual light truck, when engaged in second gear. Additionally, it smoothed the motor performance and offered controllable torque rise and fall rates, compared with those of an internal combustion engine. The latter would transmit all the engine orders to the drivetrain system, whose interactions would complicate the investigation of clonk. Moreover, clonk occurs owing to sudden surge or fade in torque and is not an engine order related NVH phenomenon.

The motor is directly coupled to the gearbox through a suitably selected coupling unit to ensure smooth torque transmissibility and limit the transmission of torsional vibrations of the motor itself. The initial idea of using the crankshaft with its engine block was abandoned because of the need to avoid torsional loads that could twist the crankshaft and introduce unwanted vibrations in the system. Furthermore, it was not necessary to fire the engine during the investigation and its use would have required the inclusion of additional components in the virtual prototype, unnecessarily increasing the complexity of the model. The rubber between the two halves of the coupling dampens torsional vibrations and isolates the drivetrain from the motor operational frequencies. One half of the coupling is fixed to the motor output shaft, using a key locking mechanism, while its other half is fixed to the propeller shaft, which is connected to the flywheel. The coupling is fitted with a shear pin in order to protect the electric motor from any reversal or under loading conditions. The propeller shaft is rigidly mounted on a pair of angular contact thrust ball bearings in a back-to-back arrangement for dynamic stability. The alignment in the connection of the gearbox and the motor was achieved by building the gearbox and the block units together on a solid plate as a complete unit, which in turn was mounted to the supporting cross-beams through standard engine mounts.

A standard vehicle hydraulic clutch system has been fitted on the rig as the main actuation mechanism. Provisions have been made for the driveline angles of the system closely to replicate vehicle conditions, as well as for vibration isolation via elastomeric pads inserted between the metallic parts and the floor in order to isolate the rig and minimize vibration transmissibility.

The test procedure with the rig was as follows. Having selected the second gear and built the required speed, the handbrake was applied, if necessary, to replicate any given vehicle laden condition, and then the clutch was abruptly disengaged within 100–300 ms. A faster clutch actuation, while possible with the rig, is usually prohibited by end-stops in the vehicle clutch system. Furthermore, this can lead to engine stalling. This simulates a condition termed as clutch-induced clonk. Alternatively, before building the final speed, a slow backward motion of the motor could be obtained, so that lashes in the various zones are taken up in the reverse direction. This normally results in a more severe metallic noise radiated when the clutch is later disengaged.

3.2 Instrumentation

The positions of all the monitoring equipment are shown schematically in Fig. 3(b). The instruments used in the experimental studies are broadly classified in the following two categories.

1. Instruments that verify/record the input conditions/perturbations. This is necessary in order to ensure repeatability and reproducibility of conditions between various rig configurations under the same testing procedure. Furthermore, the recorded conditions are used for comparison with simulation studies. These include:
 - (a) the accelerometers attached to the clutch pedal to monitor driver behaviour in clutch engagement/disengagement;
 - (b) the controller operating on the closed feedback principle and providing information on the actual rise rate and achieved final speed;
 - (c) the cylinder pressure gauge, which gives the measure of brake force applied to the rear axle.
2. Instruments that monitor the rig dynamic conditions. These fall into three classes:
 - (a) accelerometers located at the mount of the driveline centre support bearings to the chassis, monitoring the vibration transmitted to the vehicle cabin through the structure;
 - (b) non-contact laser Doppler vibrometers (LDVs), which are normally aligned to the rotating axis of the driveshaft tubes in order to measure high-frequency structural vibrations;
 - (c) microphones, positioned at appropriate distances with respect to the driveshaft tubes, transmission bell housing, and the rear axle, in order to capture the desired frequency bandwidth of the radiated noise.

LDVs are technically well suited for measuring vibration on rotating shafts as very effective non-contact alternatives to the use of eddy current probes or traditional contacting vibration transducers. However, the following problem may appear. When using LDVs in vibration measurements of rotating members about an axis normal to the laser beam direction, a fictitious velocity component, $v = \omega r$, owing to the rigid body rotation of the shaft, may corrupt its lateral vibration at the point of measurement, given by $\dot{y}$, where y is the beam direction, ω is the angular velocity of the driveshaft, and r is its outer radius. Thus, the measured velocity will be $v_y = \dot{y} \pm \omega r$. This problem is not significant with high-frequency measurements from a vibrating structure rotating at a relatively low speed. The speed of rotation in the clonk experiments is 1500 r/min, equivalent to 25 Hz, which is considerably below the major clonk frequency range of 1000–5000 Hz. Thus, the use of LDVs is justified.

Free-field microphones are used to measure sound pressure levels during testing. These are set up in vertical orientation and distance to the intended target surfaces of the tubes. The distance was 0.45 m away from the tube surface, which has been calculated using equation (1)

$$c = \lambda f$$

where f is the lowest significant clonk frequency to be observed (in this case, 760 Hz), c is the velocity of sound (343 m/s), and λ is the wavelength (m). The sound pressure recorded by a microphone can be used to obtain the noise level in dB, using the following equation:

$$L_p = 20 \log_{10} \left(\frac{p}{p_0} \right) \quad (8)$$

where p is the recorded pressure (Pa) and p_0 is the reference pressure, given as 20 μ Pa.

The background noise prior to testing was also recorded in order to filter it out of the results at a later stage. The motor, inverter, and surrounding environment induce noise in measurements that can create uncertainties in observations. However, the frequencies of background noise are substantially below the clonk range frequencies. Finally, the lowest frequency band of interest of the reverberant environment has been determined using ISO 3741 : 2000 [24]. The room volume is approximately 200 m³, which defines the lowest one-third octave band centre frequency at 100 Hz and marginally avoids undesirable reductions in the uniformity of the reverberant field at frequencies above 3000 Hz, since this is a

characteristic of rooms with larger volumes [24]. The lowest frequency of interest is well below the clonk frequencies.

4 NUMERICAL AND EXPERIMENTAL RESULTS – DISCUSSION

Figure 4(a) shows a typical clonk signal, obtained by a microphone positioned normal to the surface of the first driveshaft tube at its mid-span. It commences with a short ramp-up period, which is followed by very short impulses of 2–3 ms duration, corresponding to the accelerative noise. Then, a long decaying period of 100–150 ms corresponds to the ringing noise response. These characteristics are typical of clonk signals. The peak pressure is obtained at 2.9 Pa, corresponding to a noise level of 103 dB, typical of annoying noises measured from vehicles on the road, such as light trucks [8].

It is necessary to decompose the recorded signal in order to obtain its spectral content. To avoid aliasing, a sampling rate twice that of the highest frequency of interest is needed according to the Nyquist criterion. Since the highest frequency of interest for this NVH issue is 5000 Hz, it is safe to gather at least 10 000 samples/s. Therefore, a sampling rate of 16 384 samples/s was used. It is clear from Fig. 4(a) that the percentage of samples related to the actual accelerative response (i.e. the impact time) represents about 10 per cent of the acquired record. The problem with Fourier transformation is its averaging nature, which means that, if the analysis is carried out over the entire sample, the result will be unrepresentative of the actual frequency content of the short-lived transient metallic noise. Therefore, it is necessary to window around areas of interest in the signal. This was reported in reference [8], where the auto regression moving average (ARMA) method was used for signal processing. The advantage of ARMA over Fourier transformation is its windowing capability and the fact that, unlike ARMA, Fourier analysis causes side-banding problems. Nevertheless, windowing can be employed with Fourier analysis, as long as a 2" sample size is used with the appropriate sample rate.

In Fig. 4(b), the normalized Fourier window of the clonk peak is shown, where the contribution of various spectral contents is quantified. The signal analysis software Auto Signal has been used for the post-processing of the measurement [25]. The high impulsive action part of the noise signal corresponds to the accelerative response, which lasts for approximately 10 ms. It is important to be able to identify

Fig. 4 (a) Typical clonk noise signal measured at the mid-span of the front driveshaft; (b) Fourier window in the clonk peak; (c) wavelet of the clonk signal

the cause of as many of the spectral contributions as possible in the following two sections of the signal: the accelerative response and the final region of exponential decay, where hollow structures exhibit a

ringing response. The wavelet decomposition provides useful combined time and frequency information to understand the effect of these contributions to the overall spectral characteristics. In the wavelet

spectrum of Fig. 4(c), it can be observed that the main accelerative noise component takes place for a short time period of approximately 5 ms.

The main frequency contributions are at 129, 315, 570, 650, 830, 980, 1080, 1215, 1582, 1630, 1711, 1790, 2325, 2850, and 3370 Hz, as shown in Fig. 4(b). Most of these peaks have been identified using other independent analyses, including impact hammer testing, the model numerical results, and analytical calculations of bearing contributions. The frequency range of 1582–1800 Hz – which makes the highest contribution – represents modal responses of the driveshafts. This has been corroborated by impact hammer testing of the driveline. Table 1 shows the major frequencies extracted by hammer testing on the three tubes with the complete assembly. Some of the other contributions in the spectrum of Fig. 4(b) also agree with the obtained hammer tested modal responses, such as 800, 1080, 2325, and 3370. Clearly, small measurement errors exist either in the microphone pick-up, spectrogram technique, or hammer tests.

With a static test rig (i.e. one that was not driven) having a two-piece driveline, which was used in earlier models of the same truck powertrain [8], an almost identical spectrum of vibration response of the front tube was found to the one used in the present investigation, containing the main peak at 1700 Hz. In this experiment, the spectrogram of the clonk noise obtained by microphones includes signals both local and global in nature. Although care is taken by shielding the bell housing in the present experimentation, some global noise radiated from the surrounding structures also interacts with those intended to be measured from local structures, such as the first driveshaft tube. Nevertheless, the agreement between the findings of reference [8] and the results obtained in the present study is striking. Vafaei *et al.* also observed significant contributions

at 3300 and 3900 Hz, both of which have been obtained here in such a vicinity, either by impact hammer tests or in the spectrum of the clonk-induced conditions. However, in reference [8] these two frequencies have been found as the main power sources, unlike the findings in the present investigation. This difference is caused by the fact that the rear axle in their reported rig was rigidly mounted to the bed-plates. This yielded a more rigid structure, which when subjected to an impulse responds more significantly at these higher modes.

The modal contributions observed also agree with the numerical results that single out the resonating breathing modes (these are efficient noise radiators) as 867, 1198, 1830, 2255, 2312, 2454, 2718, 3115, and 3630 Hz. There are two important observations to be made: a certain amount of error clearly exists with regard to frequency disposition within the experimental and numerical spectra, and not all modal responses of a structure are efficient noise radiators. Figure 5 shows the magnified mode shapes of the main breathing modes observed in numerical results. These natural frequencies have also been verified by hammer testing of the rig.

The contributions at 600 and 650 Hz in the spectrogram of Fig. 4(b) are the gear meshing frequencies owing to transmission error. The transmission input shaft runs at 1500 r/min (25 Hz). The gear meshing frequency is given by the function

$$\omega_m = n_1 \times \omega_1 = n_2 \times \omega_2 \tag{9}$$

where integers n_1 and n_2 represent the tooth numbers and ω_1 and ω_2 are the angular velocities of the pinion and gear respectively. Since the pinion of the fourth-speed gear set has 25 teeth, the frequency of meshing impacts is 625 Hz. In a similar manner, the contribution in the spectrum at 496 Hz is the result of interactions in the meshing pair of the second speed (41 teeth). In this case, the transmission output shaft rotates at 720 r/min, yielding a meshing frequency of 492 Hz.

The as yet not explained contribution in the 126–129 Hz region in the spectrum of Fig. 4(b) is caused by the back-to-back angular contact ball bearing arrangement used in the coupling of the motor to the transmission via a propeller shaft. Each bearing has ten balls. Owing to motor speed (1500 r/min), any unavoidable out-of-balance rotation leads to torsional input at 25 Hz. Even with a nominally balanced horizontal shaft and bearing arrangements, a loaded region develops owing to the effect of gravity in the lower part of bearings. This causes a defined and narrow loaded region in the bearings, which is counteracted to a certain extent by interference

Table 1 Main frequencies observed in drive-shafts during impact hammer testing

Front shaft (Hz)	Middle shaft (Hz)	Rear shaft (Hz)
853	960	862
1100	1602	1165
1640	2317	1819
2286	2481	2493
2335	2729	2525
2412	2858	2668
2460	3482	2780
2480		3218
2715		3743
3068		
3125		
3350		
3553

Fig. 5 Mode shapes of the main breathing modes observed in clonk noise measurements and numerical results

fitting and preloading. However, as the complement of balls rotates, the effective dynamic stiffness of the bearing goes through a cyclic variation. This changing of the effective dynamic stiffness causes a relative movement between the centre of the bearing supports and that of the shaft axis. Even with perfect bearings this problem exists, because it is geometrically inherent. This effect is known as the variable compliance vibration, with a period equal to the time it takes to traverse the distance between two successive balls. Since the outer race is stationary, the frequency of this impulsive action is equal to the surface speed of balls relative to the stationary outer race, and is referred to as the ball-pass frequency [26, 27]. According to reference [27], the ball-pass frequency is one of the primary bearing-induced frequencies and is equal to the number of balls in a cage complement multiplied by the cage speed. For ball bearings, the cage speed is approximately half that of the shaft speed. Thus, for the bearings referred to here, the ball-pass frequency is $10 \times 12.5 = 125$ Hz.

The high-frequency content of accelerating clonk noise is confirmed by the wavelet of Fig. 4(c). It is clearly shown that, in the exponential decay region, the main contribution to ringing noise comes from frequencies lower than 1000 Hz, all of which have been identified. This appears to be the case both in noise and vibration spectra.

Figure 6(a) shows the signal obtained by an LDV facing the mid-span of the front driveshaft tube, orthogonal to the shaft axis, and corresponds to the signal of Fig. 4(a). Both signals are obtained simultaneously. Figure 6(b) shows the Fourier window

of the central region of the clonk signal. The main power sources in the spectrum are at 342, 855, 1632, and 2507 Hz. There are other higher frequencies with much lower amplitudes, simply because the impact energy is insufficient to excite these. Figure 6(c) shows the corresponding wavelet spectrum. It is clear that the contributions of the main structural components reduce significantly after the instance of impact, which is the case in the decaying region. Moreover, the accelerative noise is dominated by the breathing structural mode at 1632 Hz.

The experimental results verify the numerical investigations, which predict the following as main clonk frequencies: 305, 990, 1080, 1750, 2178–2230, and 3366–3730 Hz, which are shown in the vibration spectrum of Fig. 7(b) for the clonk signal peak. Errors of no more than 10 per cent shifting are observed in the aforementioned frequencies. However, a noted difference is that the experiment does not show the same energy content in frequencies around 3500 Hz. The model predicts higher amplitudes at frequencies above 3000 Hz, because it contains no structural damping from the mounting structure, such as the beams and cross-members, which in practice dissipate some of the impulsive energy. The model also does not include damping from universal joints, bearings, and splines. Therefore, it is clear that a larger amount of energy is predicted to remain to excite higher structural modes than in reality happens with the rig. Additionally, parametric studies have revealed that maximum backlash values lead to increased power content in the region of higher frequencies.

Fig. 6 (a) Typical clonk vibration signal measured at the mid-span of the front driveshaft; (b) Fourier window in the clonk peak; (c) wavelet of the clonk signal

In the Fourier window of Fig. 7(a), the numerical results for noise radiation reveal a similar picture qualitatively to the experimental measurements. Here, the main frequencies are in the area 1500–2000 and 2400–2800 Hz. However, there is a quantitative difference in the amount of energy that every

frequency band contains. The reason for this difference is the amount of damping between the rig and the model, which eventually leads to reactions at the driveshaft edges with significant variations in the power spectrum. As mentioned previously in section 2, these reactions are the necessary boundary

Fig. 7 Fourier windows of the numerical results corresponding to the front shaft clonk signal: (a) air-borne noise; (b) structure-borne vibration

conditions for the calculation of the wall surface velocities and the boundary element method implementation. Nevertheless, the advantage that the employed numerical method offers is that it reveals the spectrum and the shape of the main modes in the examined NVH problem, corresponding to particular components. Therefore, it is easier to examine the effect that design changes could bring in the driveline performance.

Figure 8 shows the spectra of both airborne noise and structure-borne vibration, obtained simultaneously for the middle driveshaft tube clonk response, when windowed in the region of high impulsive action. The spectrogram of noise [Fig. 8(a)] shows contributions at 126, 283, 565, 715, 963, 1107, the band 1540–1800, 2381, and 2680 Hz. The explanation for the contribution at 126 Hz has already been provided. Those at 565, 963, 1107, and the band 1540–1800 Hz are modal behaviour of the

tube, as identified numerically and by hammer testing. A higher modal response at 3482 Hz is also found by the impact hammer test, which is not observed. The wavelet diagram of Fig. 8(c) clearly shows that the accelerating clonk noise is dominated by the structural breathing mode at 1600 Hz. Again, the spectral contributions show good qualitative agreement with the numerical predictions [Figs 9(a) and (b)] at 995–1076, 1500, 1755, 2226, 2700, 2812, 3560, and 3780 Hz. Quantitative differences are also observed in the results for the same reasons that have already been provided.

The main remaining contribution at 283 Hz is possibly owing to the variable compliance effect in the needle bearings, supporting either ends of the second driveshaft tube. These bearings have a needle bearing complement of 41 elements. As already described, the variable compliance vibration has a frequency of $41 \times 6 = 246$ Hz, as the driveshaft rotates at approximately 12 Hz and the cage speed of the bearings is approximately 6 Hz. The Fourier window of the accelerometer signal from the front bearing housing shows a contribution at 274 Hz [Fig. 10(a)]. An error of 10–15 per cent is observed for these lower frequencies. The amplitude contribution at this frequency is quite small, unlike the contribution at 125 Hz, since the loaded region in the needle bearings is well spread owing to a large number of needles, the variable compliance effect is usually less significant. The main contributions in the spectrum of Fig. 10(a) are at 392 and 565 Hz, which are structural bending modes of the driveline.

Similar results have been obtained for the rear bearing housing, as shown in Fig. 10(b). The main components are at 85, 274, 384, 499, 590, and 1150 (a breathing mode of the front and rear shafts). The frequency 274 Hz corresponds to the variable compliance effect in the bearing, as already described previously, and those at 384 and 590 Hz are structural modes of front and rear shafts accounting for the transmission of noise into the cabin. The contribution at 85 Hz is likely to be owing to what is termed an 'off-size ball' effect in ball bearings, rarely seen with other types of bearing, such as the needle bearings in this rig. This is found at the speed of rotation of a rolling element, which in effect might be slightly larger in dimension than the others. It is found from

$$f_b = \frac{Df_s}{2d} \quad (10)$$

where f_b is the rotational speed of the centre of each rolling element, D is the pitch circle diameter of the

Fig. 8 Clonk signal of the middle shaft: (a) Fourier window of the airborne noise; (b) Fourier window of the structure-borne vibration; (c) wavelet of the clonk noise signal

bearing, d is the diameter of the needles, and f_s is the shaft speed. Thus, for this bearing

$$f_b = \frac{41.275 \text{ (mm)} \times 12 \text{ (Hz)}}{2 \times 3.175 \text{ (mm)}} = 78 \text{ Hz}$$

This is very close to the spectral contribution at 85 Hz.

The spectrum of vibration for the rear support bearing [Fig. 10(b)] has a richer spectral content than for the front one. This is because, being situated near to the spline joint of the rear shaft, it is subjected to the forces generated, as the spline is another major lash zone in the drivetrain system. The plunging of the spline together with higher reactions generated by the angulation of the nearby universal joint

Fig. 9 Fourier windows of the numerical results corresponding to the middle shaft clonk signal: (a) airborne noise; (b) structure-borne vibration

constitute greater reactions at the rear bearing, which has the spectral contents of both the adjacent driveshafts, as shown in Fig. 10(b).

The corresponding results for the rear driveshaft tube are shown in Fig. 11. Structure-borne vibration is shown in Fig. 11(b), where the main power sources are at 479, 596, 873, 1795, 2230, 3217, and 3673 Hz. Impact hammer tests indicate the main contributions at 863, 1169, 1782, 2679, 3218, and 3700 Hz (Table 1). The energy imparted to the shaft by the impulsive action of the rig, pertaining to the clonk signal, only excites up to the mid-frequency range of its structural modal spectrum. This is the reason for the absence of the higher modes. To obtain these, it would be necessary to introduce a greater impulsive action by clutch actuation, which is not practically possible as it would require actuation times below 100 ms, for which the controller would trip and in a vehicle the engine would stall. However, the

Fig. 10 Fourier windows corresponding to structure-borne vibration: (a) front bearing housing; (b) rear bearing housing

necessary impulsive action was introduced by virtue of the fact that the rear axle was clamped, thus the impulsive action could not be dissipated through rotation of the rear axle half-shafts [8]. This does not represent the real-world situation, but it was used to study extremes of clonk events. The conditions required to replicate vehicle testing, where higher energy modes have been noticed more than on the current rig, but less prominently than for that analysed in reference [8], correspond to the introduction of tyre-road resistance. This requires a more complex experimental rig, including the road wheel on a rolling road chassis dynamometer.

The numerical investigations for the rear driveshaft presented in Fig. 12 indicate the highest contributions at 990–1200, 1600, 2200, 2700, 2780, 3200, and 3600 Hz. These are in very good qualitative agreement with the experimental results and with the impact hammer tests, with an error of 10 per cent

Fig. 11 Clonk signal of the rear shaft: (a) Fourier window of the airborne noise; (b) Fourier window of the structure-borne vibration; (c) wavelet of the clonk noise signal

in frequency disposition. The main observed differences are in the area of 1800 Hz and in the power amplitudes.

An important factor is to ascertain which structural modes may account for the radiated clonk noise. Figure 11(a) shows the main spectral contents for the airborne noise in the area of the rear tube. The

main power source here is at 1794 Hz in the area of accelerative noise. This frequency is also noted in both the hammer test and acquired vibration spectra. Thus, the accelerative nature of clonk noise in the experiment is due to elasto-acoustic coupling at 1794 Hz. This is also confirmed by the wavelet spectrum of noise [Fig. 11(c)]. The short-lived effect

Fig. 12 Fourier windows of the numerical results corresponding to the rear shaft clonk signal: (a) airborne noise; (b) structure-borne vibration

of 1794 Hz can be seen, as it completely ceases in a couple of tenths of a millisecond. However, the effect of the structure mode at 872 Hz continues in both the spectra, indicating its contribution to the ringing noise content. Note that this mode was predicted as a breathing mode by the numerical predictions. These findings indicate the power of combined numerical and experimental investigations in establishing the physics of motion of observed phenomena.

As in the previous noise spectra for the front and middle tubes, there are lower frequencies related to bearings and gear meshing. These are in the bands 85–100 Hz, 250–300 Hz, and 520 and 693 Hz. The bearing-induced vibration is dominated by the variable compliance effect, previously calculated to be at 246 Hz. A needle spin frequency at 96 Hz also falls in the lowest region. This region also has a contribution from gear meshing interactions in the

differential. With a pinion speed of 12 Hz, a gear meshing frequency of 96 Hz occurs, since the pinion has eight teeth. Additionally, multiple meshing frequencies appear in the spectrum, for example, at 192 Hz and at 288 Hz. A key point is that the lower noise spectra are dominated by speed-dependent cyclic impacting phenomena of lower impact energy, such as rattle-type gear meshing problems and bearing-induced vibrations.

Various driveline modes were shown in Fig. 5. It can be observed that for given impulsive action the mode shape of the driveline consists of excited modes of its three tubes. The lower frequencies correspond to less complicated shapes of noise-radiating breathing modes of the driveline pieces. Errors of the order of 0.1–10 per cent have been observed between the experimental and analytical results of this significant spectral content. The modal response of the driveline in the frequency range 2400–2500 Hz coincides with the breathing modes of the three driveshaft tubes, already shown experimentally at 2460, 2481, and 2493 Hz (see Table 1, [8], [28]). This shows a remarkable agreement between numerical predictions and the experimental findings. Similarly, the response in the frequency range 3350–3500 Hz has been identified experimentally as breathing modes of the three shafts at 3350, 3482, and 3218 Hz respectively (see Table 1). The error of prediction is again quite small. However, the main responsible modes for the accelerative clonk noise are in the range 1600–1850 Hz, corresponding again to breathing modes of all the driveshafts (1640, 1602, and 1820 Hz). These can be easily excited with less intense impulsive action and act as noise amplifiers owing to their shape.

5 CONCLUSIONS

A systematic methodology has been developed for the identification of the main components responsible for the noise radiation in powertrain NVH problems. The method presents good agreement with the experimental results, the error in frequency disposition only in a few cases approaching 10 per cent. A difference in the damping quantities between the experimental rig and the model is mainly responsible for this divergence, which is caused by the absence of rig mountings in the model.

In the examined clonk noise case, the main structural modes of the hollow driveshaft tubes that act as loudspeakers have been identified. The elimination of their effects depends on design modifications to the drivetrain system in order to

attenuate the impulsive energy waves, which excite them, before the energy reaches the thin tubes. Changes in the shaft geometric characteristics are unlikely to offer much scope, since they cannot shift the structural modes away from the wide-band spectral content of the impulse energy wave. The addition of components that increase the inertia in the transmission (like the dual-mass flywheel) could be a solution under certain conditions and may be effective for lower spectral phenomena – such as the transmission gear rattle. Such investigations will form the basis of future work.

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APPENDIX

Notation

c	wave speed (m/s)
d	diameter of needle rollers (m)
D	pitch circle diameter of bearing (m)

f	requested frequency of interest (Hz)
f_b	rotational speed of the rolling element centre (Hz)
f_s	shaft speed (Hz)
$\mathbf{n}$	surface normal vector
$n_{1,2}$	number of teeth on pinion and driven gear respectively
p	pressure on the boundary surface (Pa)
p_0	reference pressure in air (20 μ Pa)
r	distance from the source (m)
v	velocity on the surface boundary (m/s)
x, y, z	translational degrees of freedom
λ	acoustic wavelength (m)
ω	wave frequency (rad/s)
$\omega_{1,2}$	angular velocities of pinion and driven gear respectively (rad/s)
ω_m	gear meshing frequency (rad/s)