Integrated investigation of impact-induced noise and vibration in vehicular drivetrain systems

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Integrated investigation of impact-induced noise and vibration in vehicular drivetrain systems

By

Max Mahadevan Gnanakumarr

BEng, MSc

A thesis submitted in partial fulfilment of the requirements for the award of the Degree Doctor of Philosophy of Loughborough University

Wolfson School of Mechanical and Manufacturing Engineering

Loughborough University

November 2004
ABSTRACT

This thesis highlights one of the most significant concerns that has preoccupied drivetrain engineers in recent times, namely drivetrain clonk. Clonk is an unacceptable audible sound, which is accompanied by a tactile drivetrain response. This may occur under several different driving conditions. Many drivetrain NVH concerns are related to impact loading of sub-systems down-line of engine. These concerns are induced by power torque surge through engagement and disengagement processes, which may propagate through various transmission paths as structural waves. The coincidence of these waves with the acoustic modes of sub-system components leads to audible responses, referred to as clonk.

The approach usually undertaken and reported in literature is either purely theoretical or constitutes experimental observation of vehicle conditions. A few research workers have reported rig-based investigations, but not under fully dynamic conditions with controlled and reproducible impulsive action.

The research reported in this thesis combines experimental and numerical investigation of high frequency behaviour of light truck drivetrain systems, when subjected to sudden impulsive action, due to driver behaviour. The problem is treated as a multi-physics interactive phenomenon under transient conditions. The devised numerical method combines multi-body dynamics, structural modal analysis, impact dynamics in lash zones and acoustic analysis within an overall investigation framework. A representative drivetrain system rig is designed and implemented, and controlled tests simulating driver behaviour undertaken.

The combined numerical predictions and experimental noise and vibration monitoring has highlighted the fundamental aspects of drivetrain behaviour. Good agreement is also found between the detailed numerical approach and the experimental findings. Novel methods of measurement such as Laser Doppler Vibrometry have been employed. Simultaneous measurements of vibration and noise radiation confirm significant elasto-acoustic coupling at high impact energy levels.

One of the major finds of the thesis is the complex nature of the clonk signal, being a combination of accelerative and ringing noise, with the latter also comprising of many other lower energy content as observed in the case of transmission rattle and bearing-induced responses. Therefore, the link between rattle and clonk, long suspected, but not hitherto shown has been confirmed in the thesis. Another major find of particular commercial interest is the insignificant contribution of torsional damping devices such as dual mass flywheels upon the accelerative component of the clonk response.

Keywords: Powertrain NVH, Multi-physics numerical analysis, multi-body dynamics, structural modal analysis, elasto-acoustic coupling, the clonk phenomenon, transmission rattle
ACKNOWLEDGEMENTS

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I wish to thanks to my parents, brothers and sisters who have encouraged me throughout the period of research and have provided the best environment for my education and academic success.

Finally, I would like to give special thanks to all my friends and colleagues for their support and encouragement.
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<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
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<tbody>
<tr>
<td>a</td>
<td>Radius of hollow tube</td>
<td>[m]</td>
</tr>
<tr>
<td>b</td>
<td>half length of normal backlash</td>
<td>[m]</td>
</tr>
<tr>
<td>c</td>
<td>speed of sound in air</td>
<td>[m/s]</td>
</tr>
<tr>
<td>C</td>
<td>Clearance in meshing gear teeth (backlash) in chapter 4</td>
<td>[m]</td>
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<tr>
<td>C</td>
<td>Constraint function in chapter 3</td>
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<tr>
<td>$C_h$</td>
<td>Clearance between loose gear and third motion shaft</td>
<td>[m]</td>
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<td>$e_p$, $e_q$, $e_r$</td>
<td>Unit vectors in cylindrical co-ordinates</td>
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<tr>
<td>$E$</td>
<td>Modulus of elasticity</td>
<td>[Pa]</td>
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<tr>
<td>$f$</td>
<td>Frequency</td>
<td>[Hz]</td>
</tr>
<tr>
<td>$f_c$</td>
<td>Critical frequency</td>
<td>[Hz]</td>
</tr>
<tr>
<td>$f_r$</td>
<td>Ring frequency</td>
<td>[Hz]</td>
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<tr>
<td>$F$</td>
<td>Petroff Friction</td>
<td>[N]</td>
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<td>Longitudinal tyre force</td>
<td>[N]</td>
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<tr>
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<td>Vertical tyre force</td>
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</tr>
<tr>
<td>$G$</td>
<td>Gear ratio</td>
<td>[-]</td>
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<td>$h$</td>
<td>Thickness of shell in chapter 3</td>
<td>[m]</td>
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<tr>
<td>$h$</td>
<td>Lubricant film thickness in chapter 4</td>
<td>[m]</td>
</tr>
<tr>
<td>$ijk$</td>
<td>Unit vector set in Cartesian frame of reference</td>
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<tr>
<td>$J$</td>
<td>Mass moment of inertia of the loose gear</td>
<td>[Kgm²]</td>
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<tr>
<td>$[J]$</td>
<td>Jacobian matrix</td>
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<td>$k$</td>
<td>Stiffness</td>
<td>[N/m]</td>
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<td>$K$</td>
<td>Kinetic energy</td>
<td>[J]</td>
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<td>$\ell$</td>
<td>Length of contact between loose gear and third motion shaft</td>
<td>[m]</td>
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<tr>
<td>$L$</td>
<td>Length of gear flank</td>
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<td>$m$</td>
<td>Mass</td>
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<td>$M$</td>
<td>Momentum</td>
<td>[Kgm/s]</td>
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<td>$(m,n)$</td>
<td>Structural wave number</td>
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<td>Symbol</td>
<td>Quantity</td>
<td>Unit</td>
</tr>
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<td>--------</td>
<td>-----------------------------------------------</td>
<td>------</td>
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<tr>
<td>(p,q)</td>
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<td>q</td>
<td>Generalized co-ordinates</td>
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<tr>
<td>r</td>
<td>Radius of curvature of body at the point of contact</td>
<td>[m]</td>
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<tr>
<td>r_{eq}</td>
<td>Equivalent radius of contacting solids</td>
<td>[m]</td>
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<tr>
<td>R_t</td>
<td>Rolling resistance</td>
<td>[-]</td>
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<tr>
<td>R_{1,2}</td>
<td>Gear radii</td>
<td>[m]</td>
</tr>
<tr>
<td>t</td>
<td>Time</td>
<td>[s]</td>
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<tr>
<td>T</td>
<td>Torque</td>
<td>[Nm]</td>
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<tr>
<td>T_e</td>
<td>Engine torque</td>
<td>[Nm]</td>
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<tr>
<td>T_{fb}</td>
<td>Bearing friction torque</td>
<td>[Nm]</td>
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<tr>
<td>T_r</td>
<td>Resistive torque</td>
<td>[Nm]</td>
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<tr>
<td>T_{mean}</td>
<td>Mean torque</td>
<td>[Nm]</td>
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<tr>
<td>\hat{t}</td>
<td>Torque</td>
<td>[Nm]</td>
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<td>u</td>
<td>Speed of lubricant entraining motion</td>
<td>[m/s]</td>
</tr>
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<td>u,v,w</td>
<td>Displacement components in x,y,z directions</td>
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<tr>
<td>v</td>
<td>Velocity of side leakage</td>
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<td>V</td>
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<td>w</td>
<td>Squeeze film velocity: ( \frac{\partial h}{\partial t} ) in chapter 4</td>
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<td>W</td>
<td>Hydrodynamic reaction</td>
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<td>x,y,z</td>
<td>Cartesian co-ordinates</td>
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<tr>
<td>\alpha</td>
<td>Angular acceleration</td>
<td>[rad/s^2]</td>
</tr>
<tr>
<td>\delta</td>
<td>Contact deflection</td>
<td>[m]</td>
</tr>
<tr>
<td>\Delta t</td>
<td>Integration time step size</td>
<td>[s]</td>
</tr>
<tr>
<td>\eta</td>
<td>Lubricant dynamic viscosity</td>
<td>[Pas]</td>
</tr>
<tr>
<td>\rho</td>
<td>Density</td>
<td>[kg/m^3]</td>
</tr>
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<td>\rho(f)</td>
<td>Power spectral density</td>
<td>[W]</td>
</tr>
<tr>
<td>\phi</td>
<td>Angular oscillation in chapter 4</td>
<td>[rad]</td>
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<tr>
<td>\lambda</td>
<td>Lagrange multiplier</td>
<td>[-]</td>
</tr>
<tr>
<td>\mu</td>
<td>Coefficient of rolling resistance</td>
<td>[-]</td>
</tr>
<tr>
<td>\nu</td>
<td>Poisson’s ratio</td>
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</tr>
<tr>
<td>\nabla_a</td>
<td>Gradient with respect to the vector parameter ( a )</td>
<td>[-]</td>
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<td>\omega</td>
<td>Angular velocity of the engine</td>
<td>[rad/s]</td>
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<tr>
<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
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<td>( \omega_n )</td>
<td>Gear meshing frequency</td>
<td>Hz</td>
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<tr>
<td>( r, \theta, z )</td>
<td>Cylindrical co-ordinates</td>
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<tr>
<td>( \psi, \theta, \phi )</td>
<td>Euler angles</td>
<td>rad</td>
</tr>
<tr>
<td>( e )</td>
<td>Denotes engine</td>
<td>-</td>
</tr>
<tr>
<td>( eq )</td>
<td>Means equivalent</td>
<td>-</td>
</tr>
<tr>
<td>( fb )</td>
<td>Corresponds to engine journal bearing friction</td>
<td>-</td>
</tr>
<tr>
<td>( i )</td>
<td>Denotes integration step number</td>
<td>-</td>
</tr>
<tr>
<td>( in )</td>
<td>Refers to the loose gear</td>
<td>-</td>
</tr>
<tr>
<td>( j )</td>
<td>Iteration index</td>
<td>-</td>
</tr>
<tr>
<td>( o )</td>
<td>Refers to initial conditions</td>
<td>-</td>
</tr>
<tr>
<td>( out )</td>
<td>Refers to output gear</td>
<td>-</td>
</tr>
<tr>
<td>( R )</td>
<td>Denotes rolling friction</td>
<td>-</td>
</tr>
<tr>
<td>( s )</td>
<td>Relates to third motion shaft</td>
<td>-</td>
</tr>
<tr>
<td>( t )</td>
<td>Belongs to tyre</td>
<td>-</td>
</tr>
<tr>
<td>( . )</td>
<td>Time rate of change</td>
<td>-</td>
</tr>
<tr>
<td>( .. )</td>
<td>Second time rate of change</td>
<td>-</td>
</tr>
<tr>
<td>( ^\wedge )</td>
<td>Amplitude</td>
<td>-</td>
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# GLOSSARY OF TERMS

<table>
<thead>
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<th>Descriptions</th>
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<tr>
<td>ABAQUS</td>
<td>FEA software package</td>
</tr>
<tr>
<td>ADAMS</td>
<td>Automatic Dynamic Analysis of Mechanical Systems, Trademark of MSC Software</td>
</tr>
<tr>
<td>ARMA</td>
<td>Auto-Regression Moving Average</td>
</tr>
<tr>
<td>BEM</td>
<td>Boundary Element Method</td>
</tr>
<tr>
<td>CAD</td>
<td>Computer Aided Design</td>
</tr>
<tr>
<td>CAE</td>
<td>Computer Aided Engineering</td>
</tr>
<tr>
<td>CMS</td>
<td>Component Mode Synthesis</td>
</tr>
<tr>
<td>CWT</td>
<td>Continuous Wavelet Transform</td>
</tr>
<tr>
<td>DAE</td>
<td>Differential Algebraic Equation</td>
</tr>
<tr>
<td>DFT</td>
<td>Discrete Fourier Transform</td>
</tr>
<tr>
<td>DMF</td>
<td>Dual Mass Flywheel</td>
</tr>
<tr>
<td>DoE</td>
<td>Design of Experiments</td>
</tr>
<tr>
<td>DOF</td>
<td>Degrees of Freedom</td>
</tr>
<tr>
<td>DSMM</td>
<td>Dynamic Stiffness Matrix Method</td>
</tr>
<tr>
<td>EHL</td>
<td>Elasto Hydrodynamic Lubrication</td>
</tr>
<tr>
<td>FEA</td>
<td>Finite Element Analysis</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier Transform</td>
</tr>
<tr>
<td>FRF</td>
<td>Frequency Response Function</td>
</tr>
<tr>
<td>FWD</td>
<td>Front Wheel Drive</td>
</tr>
<tr>
<td>GRF</td>
<td>Global Frame of Reference</td>
</tr>
<tr>
<td>JSAE</td>
<td>Japanese Society of Automotive Engineering</td>
</tr>
<tr>
<td>LDV</td>
<td>Laser Doppler Vibrometer</td>
</tr>
<tr>
<td>LH / RH</td>
<td>Left Hand / Right Hand</td>
</tr>
<tr>
<td>LPRF</td>
<td>Local Part Reference Frame</td>
</tr>
<tr>
<td>MBD</td>
<td>Multi-Body Dynamics</td>
</tr>
<tr>
<td>MDOF</td>
<td>Multi-Degrees of Freedom</td>
</tr>
<tr>
<td>MIRA</td>
<td>Motor Industry Research Association</td>
</tr>
<tr>
<td>MT75</td>
<td>Manual Transmission (5 speed)</td>
</tr>
<tr>
<td>Abbreviation</td>
<td>Description</td>
</tr>
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<td>--------------</td>
<td>-------------</td>
</tr>
<tr>
<td>NASTRAN</td>
<td>FEA software package, Trademark of MSC Software</td>
</tr>
<tr>
<td>NVH</td>
<td>Noise, Vibration and Harshness</td>
</tr>
<tr>
<td>PATRAN</td>
<td>the preprocessor for the ABAQUS FEA package</td>
</tr>
<tr>
<td>PSD</td>
<td>Power Spectral Density</td>
</tr>
<tr>
<td>RWD</td>
<td>Rear Wheel Drive</td>
</tr>
<tr>
<td>SDOF</td>
<td>Single-Degree of Freedom</td>
</tr>
<tr>
<td>SEA</td>
<td>Statistical Energy Analysis</td>
</tr>
<tr>
<td>SFFT</td>
<td>Short Fast Fourier Transform</td>
</tr>
<tr>
<td>TMM</td>
<td>Transfer Matrix Method</td>
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CHAPTER 1

1 INTRODUCTION

1.1 Powertrain concerns in automotive industry

In today’s global economy, importance is placed on delivering products to market quickly. Fast product development cycles allow companies to adopt new technologies and to fulfil customer requirements before competitors.

There are many problems, which the automotive industry has to deal with, noise and vibration being one important aspect. In recent decades vehicle noise levels have gradually declined. The noise quality aspect is becoming increasingly important. The fundamental requirements for Noise, Vibration and Harshness (NVH) in design and development concerns limiting the overall noise levels, through general refinement and striving to remove specific problems. Figure 1.1 shows some of the typical noise generating sources and their respective transmission paths to passenger cavity.

<table>
<thead>
<tr>
<th>Source</th>
<th>Path</th>
<th>Passenger Cavity</th>
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<tbody>
<tr>
<td>Engine</td>
<td>Engine Mounts</td>
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<td>Drivetrain</td>
<td>Drivetrain Mounts</td>
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<td>Tyres &amp; Road</td>
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<td>Exhaust</td>
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<td>Aerodynamic</td>
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Figure 1.1: Typical Vehicle Noise and Vibration Sources and Paths

The level of refinement that customers expect, when they purchase their vehicles is based on their past experiences. So long as their expectations are met, in their new purchase, all is deemed well. Otherwise, they remain dissatisfied and may complain to their dealerships about their new vehicles. Thus, it is a matter of defining and achieving the required standard
Chapter 1

Introduction

during vehicle development. Continual improvements upon the required standards are desirable, however, not at exorbitant costs, in an already highly competitive market place. Thus, if a vehicle shows signs of surpassing the required standards during its development, then there is an opportunity to reduce cost by removing acoustic absorption materials or use lower performing provisions. This can also have knock-on benefits in terms of component weight, resulting usually in enhanced fuel efficiency.

Vehicle drivetrain NVH is difficult to predict due to the complexities arising at the system level, such as non-linearities, redundancies, noise factors, and interactions of various parts of the drivetrain systems (Menday (1997), and Menday (2003)). As a consequence, it is often necessary to resolve NVH concerns by the late and costly addition of palliative measures. Sometimes their contribution is compromised by lack of package space, weight limitations, manufacturing and assembly issues, available time, and other factors. Added value from NVH palliation can be disappointing as a result. It also embodies smooth transitions through transient events such as gear changes, sudden adjustment to the throttle, manoeuvring, etc. The noise and vibration must not stand out at any specific road or engine speed, because of the contrast with the response in the immediate range. Thus, there should ideally be no audible booms, whines, howls, whistles, rattles, clank, or other specific problems regardless of their absolute noise level.

The same basic procedure and tools are available for general development work and for projects requiring the rapid solution of a specific problem (i.e. troubleshooting). However, ideally, development of new vehicles is carried through coordinated, proactive work programmes. Fire-fighting should not be the normal work mode of an NVH department.

Achieving high refinement is thus a combination of good isolation, general refinement, and an absence of specific problems. Isolation for general refinement is designed in through the suspensions, the engine mounting, other mounts and bushes, the acoustic pack and not least, the seals and sealing. Development is necessary to optimise their effectiveness to meet their objective attenuation targets at minimum cost.

Much development work involves comparative or parametric test in which design variants are compared. This may involve different materials, structures, functional properties or
manufacturing processes. This type of “trial and error” engineering can be very effective, if properly implemented through a representative test and measure of effectiveness.

Specific problems are an unwelcome by-product of early designs often due to the existence of resonant modes or the interaction of individually designed components and systems. Specific solutions are required to be developed. It is fairly common for engineers to attempt to solve these problems by trial and error design changes. They may even hit on a solution by using a modification, which was effective in a previous manifestation of the same problem on another vehicle, or simply by luck. However, this approach often wastes a lot of time.

It is better to solve problems by understanding their root causes. First, diagnostic tools are required to identify the mechanisms, causing the problem so that modifications can then be investigated on an informed basis. Initial tests will tend to be aimed at finding out information on the magnitude of noise or vibration concerns at key locations, the frequency content, etc. The work may then take the form of hypothesis testing, which has a sound scientific pedigree. The engineer forms a theory as to the nature of the problem. This tentative idea implies certain things about the behaviour of the system. A test can then be designed, whose outcome can be predicted from this theory or approach. If the test results are not as expected, then the theory is wrong. Otherwise, it remains a good working hypothesis and at some stage the evidence may be strong enough to be conclusive.

All systems with mass and stiffness have natural frequencies. If the problem is due to one or more structural resonances, amplifying the perceived noise or vibration under specific circumstances, then modal analysis can determine the “shape” of the dominant motion, providing insight into how to change the behaviour of the system. In the case of noise, source ranking using rote tracking or “windowing” techniques are very effective starting points, identifying the more prominent sources or specifically strong paths for vibration energy. For example, enclosed spaces like a car interior have acoustical standing wave frequencies, which can result in acoustic resonances if sufficient vibration energy is transmitted into them.

Disconnection tests can also be very effective in identifying paths contribution in straightforward situations, e.g. removing an exhaust hanger. However, care is required not to change interactive dynamic behaviour, which can lead to confusing results. Once the main
elements of problem are understood, modification options can then be selected on an informed basis and tested on the vehicle, using the appropriate standard simulation tests.

The general route to the solution of a specific problem is to:

- specify an appropriate laboratory simulation and objectives
- identify sources, paths and resonances dominating the response, and
- develop solution strategy derived from the understanding of root cause, or causes, of the problem

**1.2 Description of the Vehicular Drivetrain System**

The drivetrain in a vehicle is defined as the torque path from the flywheel through to the wheels, as shown in Figure 1.2.

![Figure 1.2: Shows a Typical Vehicular Drivetrain System](image)

The drivetrain serves two functions: it transmits power from the engine to the drive wheels, and it varies the amount of torque. Multiple ratio transmissions are necessary, because the engine delivers its maximum power at certain speeds. In order to use the same engine revolution at different road speeds, it is necessary to change the "gear ratio" between the engine and the drive wheels. The vehicle has to switch gears in order to move at a wide range of speeds.
There are actually two sets of gears in the drivetrain; the transmission and the differential. The transmission allows the gear ratio to be adjusted, and the differential lets the drive wheels turn at different speeds.

1.3 Problem Definition

Clonk is an unacceptable audible noise, which is accompanied by a tactile drivetrain response. This may occur under several different driving conditions, as follows (Menday (2003)):

- Rapid throttle applied (tip-in) from coast or released (tip-out) from drive condition.
- Rapid engagement/disengagement of clutch at low road speeds. It may also occur after gear selection, if the clutch is rapidly engaged.
- In either case, the resulting torsional impulse to the drivetrain causes a short duration vehicle jerk action and an accompanying clonk noise. It is more noticeable in low gear and low vehicle speeds.
- It is thought that clonk is as a result of a high energy impact (Kelly et al (1999)) frequency range 1000-5000 Hz).

Many drivetrain NVH concerns are related to impact loading of sub-systems down-line of engine. These concerns are induced by power torque surge through engagement and disengagement processes with sudden application of clutch or throttle tip-in and tip-out, as already mentioned above. In such cases, the ramp in input torque is taken up by a lash zone through an impact in meshing gears of transmission, differential or spline joints, or by torsional oscillation of the clutch pre-damper. The generated impact causes elastic structural wave, which may propagate through various transmission paths in the flexible members. Few of these have been identified, which occur at the transmission bell housing and the thin walled driveshaft tubes. The coupled shunt (axial fore and aft motion) and shuffle (torsional rigid body motion of the power train system) (Farshidianfar et al (2000)) motions of the drivetrain system is then accompanied by high frequency elasto-acoustic responses that propagates to the cabin, as well as being air-borne, reverberating from road side boundaries. The problem is quite acute with diesel operated light trucks, which is the subject of this investigation.
1.4 Aim and Objectives

The overall aim of the research is to identify the root causes of high frequency impact induced drivetrain vibration, which is a generic problem in automotive industry. The investigation of the lash zones in the drivetrain system is essential. The specific objectives of the research are:

- Fundamental understanding of the sources and mechanisms of impact loading transmission to light weight structural components of the drivetrain system.
- Fundamental investigation of mechanisms and paths for structural-acoustic coupling.
- Develop state-of-the-art CAE tools for virtual prototype testing and parametric studies of drivetrain high frequency transient dynamic phenomena.

The problem highlighted above has serious repercussions in terms of customer perception of a vehicle. Early investigations have indicated that high frequency elasto-acoustic drivetrain vibrations is not confined to any form of vehicle, but the investigation in this thesis is carried out in the case of light trucks (Menday (2003)).

The development of realistic and practical modelling and simulation tools (as highlighted in chapter 4) would facilitate rapid virtual prototype testing, which would enable design analysis earlier in the drivetrain development work, thus reducing the current “fire-fighting” approach to NVH palliation, which can often lead to very costly vehicle service actions. These CAE tools can also reduce the conceptual design-to-manufacture cycle time, an approach, which has resulted in typical significant reduction in the development lead-time in Japan.

1.5 Overview of Thesis

Chapter 1 is an introductory chapter, which indicates brief overview of NVH concerns in the automotive industry, in particular in the vehicular drivetrain systems. It also defines the problem under investigation here. Aim and objectives of the research have also been highlighted.

Chapter 2 provides an overview of research work carried out elsewhere and the review of literature pertinent to the various aspects related to the contents of this thesis.
Chapter 3 provides the theoretical background: for elasto-multi-body dynamics, as well as for acoustic analysis, which is used later on in chapter 6 to compare with the experimental results.

Chapter 4 is the simulation studies of Clonk for Rattle and also including the effect of lubricant viscosity. The findings are used later on in chapter 6 to explain lower frequencies in the experimental spectra.

Chapter 5 describes the issues that have been tackled in the design, manufacture and assembly of a laboratory-based transient dynamic experimental test rig.

Chapter 6 includes all the results obtained from the experimental rig and the numerical predictions with appropriate comparisons.

Chapter 7 outlines overall conclusions of the research, a critical assessment of the approach undertaken and suggestions for future work.
CHAPTER 2

2 LITERATURE REVIEW

2.1 Introduction

The literature study gives an overview of the subject matter, which is very complex due the many interactions and sources of non-linearity in the drivetrain system. The study of drivetrain problems can be divided into a number of categories of noise and vibration: low frequency rigid body motions, such as drivetrain shuffle and shunt (3-10 Hz) (Farshidianfar et al (2000)) and clutch judder (Centea (1997), Centea et al (1999) and Centea et al (2000)) mid-range frequency vibration problems such as gear rattle (80-250 Hz) (Gnanakumarr et al (2002)), and higher frequency problems with significant acoustic radiation such as whine, howl and clonk.

2.2 Noise and Vibration in vehicular drivetrain system

NVH stands for Noise Vibration and Harshness and is an industry term associated with the treatment of vibration and audible sounds. Harshness usually refers to treatments of transient frequencies or shock. Noise denotes unwanted sound, hence treatments are often to eliminate these sounds and associated vibrations, but occasionally products are engineered to magnify sound and vibration at particular frequencies.

The human ear can generally detect frequencies in the range of 16 to 20,000 Hz (Bosch (1986)). Vibrations above and below this range may not be detectable to human ear, but may still require treatments for improved product performance and longevity.

Some of the terms used to describe various types of noise in automobile power trains (Merritt (1971)) are as follows:

- **Growl**: A continuous noise, of low pitch, often harsh in character.
- **Grunt**: Brief, regularly repeated periods of growl.
- **Groan**: A growl of varying amplitude.
Clonk  The noise accompanying a reversal of torque, when backlash is taken up.
Rattle  The noise of rapidly repeated impacts.
Hum     A uniform note of medium or low frequency, generally regarded as not unpleasant.
Wow     A noise associated with tooth contact and of cyclically varying amplitude.
Whine   An unpleasant note, of high frequency.
Scream  A high-pitched note, distressing to the ear.

These are some of the terms used in the industry.

2.3 Drivetrain Clonk

Krenz (1985) defined clonk as:

A short duration audible transient response, usually the result of a load reversal and in the presence of backlash.

Clonk may occur under different driving conditions. These are broadly (as already mentioned in chapter 1):

- Rapid throttle applied (tip-in) from coast or released (tip-out) from drive condition.
- Rapid engagement/disengagement of clutch at low road speeds. It may also occur after gear selection, if the clutch is rapidly engaged.
- In either case, the resulting torsional impulse to the drivetrain causes a short duration vehicle jerk action and an accompanying clonk noise. It is more noticeable in low gear and low vehicle speeds.
- It is thought that clonk is as a result of a high energy impact (Kelly et al (1999)) frequency range 1000-5000 Hz).

Drivetrain clonk is generic. It is equally likely to cause problems on Front Wheel Drive (FWD) or Rear Wheel Drive (RWD) vehicles, with either petrol or diesel engines. In other words, it is sensitive to many factors. It is also known that the problem demonstrates itself as
a natural variation from vehicle to vehicle, even when comparing vehicles of identical build specifications.

When engine and road inputs are applied to the drivetrain system, rapid changes in the drivetrain response can take place. These rapid changes excite the higher frequency structural modes in the drivetrain, particularly at the driveshaft and transmission bell housing.

If a short duration impulsive torque is applied to a FWD or RWD drivetrain it will excite a response in the system across the frequency range. Impulse torque conditions will occur when the throttle is rapidly applied from coast, or when the throttle is rapidly released from drive. Impulse torque conditions also occur when the clutch is rapidly engaged after a gearshift or when manoeuvring. In either case, the effect of the impulse torque supplied to the drivetrain is to excite a low frequency condition known as shuffle, and a high frequency condition known as clank.

Shuffle is particularly evident at low speeds and in low gear. It is the first torsional mode of the drivetrain and the driver experiences this mode as a longitudinal jerk, referred to as shunt (Farshidianfar et al (2000) and Rahnejat (1998), and Capitani et a (2001)).

The high frequency clank noise may be heard, coincident with the shuffle cycles of vibration, and when backlash in the drivetrain is taken up by the torque surge over a very short time period. Clank may be heard with or without shuffle, and whenever the torque travels through the system backlash. It is particularly noticeable in low gears and at low speeds, when background engine and road noises are minimal.

The high frequency 1000 to 5000 Hz clank response may be clearly seen superimposed on the much lower (3 to 10 Hz) shuffle frequency carrier wave, when it has been taken from an instrumented test vehicle. This high frequency characteristic is common to both the hollow large diameter tubes on RWD and the smaller diameter shafts on FWD vehicles.

In order to study shuffle, a multi-body dynamic drivetrain model is required. In fact, a lumped mass model with 4 or 5 degrees of freedom will clearly demonstrate the low frequency shuffle condition.
A simple two-piece drivetrain model devised in ADAMS, and a corresponding drivetrain test rig, were set up to evaluate shuffle and clonk by Arrundale (1998) and Arrundale et al (1998). Furthermore, a short test was conducted on a vehicle with one rear wheel jacked up and with the handbrake lightly applied (Menday (1997)). The 2nd gear was selected and the clutch was abruptly engaged. The clonk response was subjectively rated to be the same as that heard when driving on the road. It appeared to emanate from the transmission. No data was collected. The purpose of this short static test was to confirm that clonk could be excited with a stationary vehicle. It was, therefore, confirmed that a static drivetrain rig could be expected to behave in a similar manner, with the advantage that experimentation would then be conducted in a controlled environment (Menday (1997)).

The rapid throttle demands are converted to system torsional impulse causing the phenomenon of shuffle to occur. Clonk may be heard on the first cycle of the shuffle response. Krenz (1985) and Tobler (1985) have studied the relationship between audible clonk and shuffle. They have all shown that the first swing on the transient torque can be 2-3 times the steady-state response and is perceived as a short duration jerk, pertaining to clonk. The initial clonk response is then followed by the 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 between 0.25-5 milliseconds, both depending on drivetrain configuration and structural materials.

The shuffle and clonk response are also documented by Biermann and Hagerodt (1999). They report results obtained from a test-rig with those found from a detailed analysis of the drivetrain using a theoretical model. They clearly showed cycles of shuffle accompanied by the reversing clonk characteristics, especially with the transmission engaged in third and fourth gears. They also reported some remedies for the problem, such as reducing the transmission component inertias and matching the clutch damping, the extension of the predamper stage and the damping of the compensation movement in the differential drive.

Tsangarides et al (1985) have also made an important contribution in the torsional analysis of drivetrain, using a lumped parameter model, including the effect of backlash using zero-rated springs. They defined tip-in jerk as the rate of change of vehicle acceleration, which may be accompanied by an audible clonk. Petri et al (1989) have also observed that sudden throttle change transforms into vehicle acceleration/deceleration, which may excite the
system shuffle, accompanied by hard metallic clonk. They observed such acceleration peaks at the input shaft and the differential. Therefore, the NVH spectrum would normally contain contributions from low frequencies, mainly due to the rigid response of the whole system; all the way to the high frequency responses caused by the flexure modes of system components such as hollow drive shaft tubes. The low frequency contents are accounted for by shuffle and drivetrain angulations and these can be obtained by rigid multi-body analysis. 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 et al (1949) have reported that for thin cylindrical shells an infinite number of axial vibration modes (i.e. $m$) are theoretically possible, each with a corresponding number of circumferential waves (i.e. $n$). Therefore, to define a mode, the $m$ and $n$ values must be specified. Forsberg (1964) used an analytic method outlined originally by Flugge in 1934 and concluded 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. Other studies have employed the use of finite elements analysis, for example to compute the natural frequencies of a clamped cylinder by Santiago (1989).

The first swing on the torque transient, which can be 2-3 times greater than steady state torque, is the basic clonk response. This is perceived as a short duration jerk and may be accompanied by a low frequency boom or thud sound. The first may be the noise perceived by the driver when the door window is lowered, and the clonk impact noise is then airborne directly from the contact zones. The thud may be perceived by the driver with the windows closed, and clonk noise is said to be structure borne. Both noise conditions may emanate from the same noise site(s).

Following the initial clonk response, a low frequency torque variation occurs. This is the shuffle response, which is perceived as a fore and aft surge, which may last up to a second.

The drivetrain torque versus time profile should be used to explain this event sequence during throttle tip-in or tip-out (see Figure 2.1)
Figure 2.1: Clonk event with cycles of shuffle

Shuffle is the first torsional drivetrain mode and may be easily identified using a 2 or 3 degree of freedom mathematical model. The lowest shuffle frequency, dependent on drivetrain format, is in the range 3 - 10 Hz that equates to a 300 ms period. It may be fundamentally related to side-shaft torsional stiffness.

2.4 Impact Loading of Static Test Rig

A static (i.e. not rotating) experimental rig was initially devised and set up by Menday (1997) at Ford Research Centre in Dunton, Essex as shown in the Figure 2.2, with all the drivetrain components from the flywheel to the rear wheels. The rear wheel assembly, excluding the tyres were clamped by bed-plates to the ground. The transmission bell housing and the differential were also suitably mounted onto the ground.
A specified preload torque was applied to the system to take up all the lash elements and to fully compress the clutch disc springs. The preload torque was held on a low inertia disc brake. The stored energy was instantaneously released in the form of a ramp sawtooth pulse. The duration of the torsional pulse was varied between 80-180 ms as required, inducing a generated lash take up of the order of 1-2 ms, of sufficiently short period to excite the higher frequency clonk modes. The 1-2 ms impact was followed by approximately 30 ms of driveshaft tube ringing (see chapter 5). The input ramp torque was found to be repeatable and reproducible.
Figure 2.3 shows that a number of accelerometers and microphones were placed at specific locations along the drivetrain rig to monitor radial transverse acceleration and sound pressure levels at a sampling rate of 12,500 samples per second.

### 2.5 Experimental Results of the Static Test Rig

A typical signal obtained by data acquisition is shown in Figure 2.4, in this case from bell housing. The sampling rate was 12,500 samples per second (Menday (1997)).

Signal acquisition and spectral analysis poses a challenge, particularly when windowing around short-lived clonk signals of 1-2 milliseconds (see Figure 2.5). The traditional spectral analysis techniques such as FFT or PSD result in poor frequency resolution and "leakage", where the spectral domain energy in the main lobe of a response leaks into side-lobes, obscuring and distorting other spectral contributions which may be present there (Vafaei et al (2001)). Thus, signal analysis based on an auto-regression moving average (ARMA) method carried out, using the Burg’s recursive algorithm (Burg (1967)) and the Shanks’ method for the determination of the ARMA parameters (Shanks (1967)).

![Figure 2.4: Signal obtained from Bell-Housing (after Menday (1997))](image)
Signals obtained from various positions were subjected to ARMA process, described above. Those obtained at the rear-most torque tube are presented here, as shown in Figure 2.6, which shows the entire spectrum of vibration up to a frequency of 4000 Hz. This spectrum shows much higher contributions at 3000 Hz and 3450 Hz.

Figure 2.5: Impulsive action applied to the static test rig by Menday (2003)

Figure 2.6: ARMA spectrum (after Vafaei et al (2001))
Hollow thin shell tubes, having wall thickness in the region 1.5-3 mm have a very high modal density (see for example, Figure 2.7, where the spectrum of analysis is obtained for a similar tube by a hammer impact test). There are many combined torsional-deflection and compression-axial structural modes.

Figure 2.7: Impact response comparisons of driveshaft tube with different damping methods, in the clonk frequency range (after Menday (1997))

The results of analyses showed that the high frequency modes were a combination of circumferential and axial structural waves that can excite the acoustic modes and drive the adjacent sound pressure field of the cavity. Comparison of these frequencies with pressure wave responses obtained by Arrundale et al (1998) indicated that some of the structural modes are quite close to higher harmonics of the sound pressure waves. The conditions, leading to elasto-acoustic coupling are discussed and investigated in chapter 6.

2.6 Vehicle Shuffle

Shuffle is the first torsional mode of the entire powertrain system (Farshidianfar et al (2000)). This rigid body motion of the powertrain system is noted by the vehicle occupants through a corresponding fore and aft longitudinal oscillation of the vehicle, referred to as shunt. Coupling of the torsional powertrain shuffle and vehicle shunt is affected by the longitudinal stiffness of the tyre in the contact patch and lateral stiffness of the rear axle half-
shafts. Shuffle is a vehicle transient response, resulting from torque reversals that occur during throttle tip-in from coast to drive and throttle tip-out from drive to coast (Krenz (1985)). In the presence of lash zones, defined as very low or zero rated stiffness regions, a torque pulse is generated, which is transmitted to the vehicle via the road wheels as described above.

Definitions of Shuffle/Shunt and Clonk

- **Shuffle/Shunt**: coupled rigid body torsional and axial low frequency oscillations of the drivetrain system
- Observed with sudden demands in torque surge in throttle tip-in/tip out or with sudden clutch actuation
- Can be accompanied by impact torque reversals through lash, leading to high frequency structural vibrations and noise from drivetrain system, referred to as clonk

Shuffle can readily be identified with a simple two or three degree of freedom model, because it is the first drivetrain torsional eigen-frequency. Shuffle is minimised if iteration is used to find the optimum input pulse shape to minimise response and to optimise the active system parameters. This process works only if backlash is absent (Menday (2003)).

Hawthorn (1995) used a lumped parameter model to investigate back-out shuffle.

The model was then used in parameter optimisation studies and he concluded:

- A ramp time 0.3 seconds was critical to minimise shuffle excitation
- There was a lash threshold. Lashes above this threshold caused further increases in the shuffle response. Below this threshold the shuffle was not affected.
- Stiff clutch springs with compliant driveshafts gave a low initial response but a slow decay rate
- Conversely, soft clutch springs with stiff driveshafts gave high initial response and a rapid decay rate.

Rooke et al (1995) created a low order of freedom drivability model, which achieved good shuffle correlation in items of frequency, damping and overshoot.
2.7 Transmission Rattle

Rattle is as the result of impact dynamics of gear meshing teeth in transmission systems, under various loaded or lightly and unloaded conditions (Kamo et al (1996), Karagiannis and Pfeiffer (1991), and Gnanakumarr et al (2002)). It fundamentally differs from clonk or thud conditions that are of a transient nature due to sudden surge or fade in torque demand. However, they both have the lowest common denominator in the action of contact/impact forces through lubricated contacts. The impact nature of the problem is also similar to that caused by transmission errors, such as in whine, with the difference that the latter is a high frequency problem which can also be excited by axle misalignment (Foellinger (2004), and Parkin-Moore et al (2004)). In fact, Foellinger (2004) describes the subtle differences between these impact-induced phenomena. The axle whine problem is exaggerated by the vibrational characteristics of the side shafts, for example in the RWD vehicles, caused by a combination of CV joint vibrations and impact of plunging force, as described by Biermann (2004).

Various forms of rattle have been cited, owing to the mechanism of manifestation and operating conditions (Foellinger (2004) and Dogan et al (2004)):

- **Idle rattle**: with an engaged clutch and transmission in neutral with the engine at idle rpm. This is very audible due to the lack of engine noise and is switched off with clutch disengagement. Brosey et al (1986) refer to this phenomenon as neutral rattle.

- **Creep rattle**: occurs between 1200-2000 rpm in gear and is strongly related to the torsional modes of the drivetrain. Brosey et al (1986) refer to this drive rattle.

- **Over-run rattle**: with throttle off and coasting between 1500-4000 rpm. It occurs at low engine speeds and high loads and when the inertial torque exceeds the drag torque. It is thought that this mode of rattle would be sensitive to bulk lubricant properties such as viscosity.

Some remedial actions are currently undertaken to palliate for these modes of rattle. These are:

- Clutch first stage is used for idle rattle
• Clutch second stage is used for creep-in-gear rattle
• Impact forces from non-torque transmitting teeth are transferred to neighbouring housing.

The following additional observations are also made:
• Rattle is aggravated by ancillaries, when switched on
• Needle type torque pulses result in wide band frequency spectra, thus a connection between rattle and clonk may be initiated.

2.7.1 The rules to reduce transmission rattle

Engineers in the industry have arrived at some empirical rules throughout years. These are not all corroborated by any fundamental studies of physics of motion in meshing contact dynamics.

However, the following items should be considered:

• The first torsional mode of the drivetrain in neutral should be reduced to below the idle rpm for the isolation of rattle. A clutch pre-damper is used for this purpose (Menday (1997) and Dogan et al (2004)). However, this may cause clonk conditions.

• The first torsional mode of the drivetrain in gear can cause shuffle. The side-shaft stiffness is often quoted as a good tuning factor (Biermann (2004)).

• The second torsional mode of the drivetrain (engine and transmission out-of-phase): A dual mass flywheel's additional inertia can pull the resultant resonance below the idle speed. Clutch tuning has found to worsen the tip-in/tip-out and shuffle responses (Dogan et al (2004), and Littlefair (2004)).

• It is thought that reduction of torque rise rate would improve the conditions. This happens to be also favourable in the case of clonk.

• Maximising flywheel inertia is thought to improve the conditions. This can have adverse effects in the case of whoop, should axial float of flywheel be allowed in order to reduce the engine rumble. Whoop is a combined vibration and noise concern
caused by impact of the flywheel into the friction disc and pressure plate during clutch engagement and disengagement actions, as described by Kelly (1999) and Rahnejat and Kelly (1997).

- Minimisation of cycle-to-cycle and cylinder-to-cylinder combustion variation would have a significant effect upon rattle, as the engine order response accounts for the main forcing frequencies in drivetrain systems. Dogan et al (2004) refers to this effect as being particularly significant in high power multi-cylinder diesel engines.

- Minimising teeth lash and maximising lash rate would be beneficial. However, a zero lash is unattainable for obvious kinematic reasons.

- Minimisation of structural mobility is desired, but can have unacceptable cost repercussions.

- Decrease the distance between the output shaft and layshaft (which reduces gear inertia) (Menday (1997) and Dogan et al (2004)).

2.8 Relationship between Clonk and Rattle

The difference between clonk and rattle is shown in Figure 2.8. It can be seen that both phenomena are impact induced, with the difference that the impulsive action is much more severe in the case of clonk conditions. In this case, as the result of a sudden demand in torque caused by driver action, as in throttle tip-in condition or abrupt clutch engagement the impact velocity through backlash is much higher in magnitude. This results in a greater momentum and energy input into the drivetrain system. As a result higher modes of vibration are excited that coincide with many higher modes of vibration of hollow driveshaft tubes and other hollow structures such as the bell housing. These high frequency structural modes can coincide with the acoustic modes of these cavities and propagate as noise.

Figure 2.8 shows that the severity of impact can lead to either a localised deformation effect, often referred to as a Hertzian-type impact, which is described later, or to a global type deformation, caused by propagation of structural waves through the structure. This latter form of deformation is referred to as a St Venant-type impact.
The local deformation takes place against the crushing stiffness of teeth, which is considerably less than tooth stiffness in bending and that in rocking. The latter two give rise to global deformation and propagation of waves onto the surrounding elastic solids such as the input and output shafts, and further onto the driveshaft tubes. Higher stiffness means higher frequency response. Thus, it is clear that localised deformation takes place due to impact, often below the resonant frequency of impacting solids. This, in fact, is the basis of Hertzian-type impact, described below. Therefore, the distinction (as shown in Figure 2.8) between rattle and clonk can be made with respect to the localised versus globalised nature of the deformation of impacting solids. In the case of rattle, the deformation is localised in most cases, and in fact nearly non-existent in the case of very light impact conditions in idle rattle.

2.9 Impact Noise Mechanism

When two bodies impact, sound is propagated by two distinct processes. Firstly, when bodies are solid, and their natural modal behaviour has a high frequency, the nature of impact is often localised, leading to an accelerative type noise, as in the case of the common Newton trolley, shown in Figure 2.9. When the elastic solids are hollow, their natural modal behaviour has lower frequency content. Thus, a localised impact can impart energy levels that reach resonant conditions, and elastic waves propagate through the structure (see Figure 2.10). Therefore, the initial; accelerative noise is followed by ringing type noise with a
plethora of response frequencies. The number of natural modes increases with reduced thickness of the hollow solids.

Figure 2.9: Newton trolley

Figure 2.10: Impact of hollow elastic solids

In practice, the power train system comprises hollow and solid elastic bodies. Thus, a combination of accelerative and ringing noise responses are obtained in impact dynamics,
which may interact with each other, thus making it difficult to identify all the contributing
effects. The nature of these noises are explained in more detail later.

Both figures are in fact good representations of impacting meshing teeth through backlash,
since the contact of teeth, depending on their geometry can be represented by circular or
elliptical point contacts or finite line contact (similar to that made by a roller of finite length
against a flat semi-infinite elastic half-space). Helical involute and hypoid gears used in
vehicular transmissions and differential give rise to elliptical type contact footprints, whereas
spline joints or spur gears form a finite line conjunction.

2.10 Gear Backlash

Drivetrain lash may be described as the summation of lashes at each contact zone in the
drivetrain, including the major lash effect of the clutch disc low pre-damper springs. This
work focused on the contribution of the impacts at each lash point, to the perceived overall
radiated clonk noise.

There are many lash points in a geared system and each lash point has an associated pair of
impacting inertias. The lash zones in a RWD drivetrain are shown in Figure 2.11. In each lash
zone the impact velocity is a function of amount of lash or the gap that the impacting inertia
accelerates through. The larger this gap, the higher the velocity reached prior to impact, if one
ignores the damping effect of the usually present lubricant film. In rattle, the velocity is a
function of engine order vibration, which is most prominent as the second engine order (i.e.
twice the crankshaft angular velocity) in a 4-stroke, 4-cylinder engine. Therefore, palliative
action can be undertaken to reduce the second engine order, such as the use of dual mass
flywheels and flywheel inertial itself. The latter, however, increases the impacting inertia,
when heavier, thus increasing the impacting momentum. Reduction of lash is, therefore, the
optimum solution to the problem. However, the costs associated are not usually permissible
in mass manufactured vehicles. Piece-to-piece variations also have an important effect in this
respect. Dogan et al (2004) discuss the importance of tuning the amount of the driving inertia
up line of the impact zone to reduce the effect of impact, particularly in the case of
transmission rattle. Littlefair (2004) justifies the use of dual mass flywheel for attenuation of
rattle, based upon the reduction of second engine order due to damping effect of grease
incorporated between the primary and secondary inertias.
Other palliations can include the following actions:

- Clutch disc hub spline lash clearance to transmission input shaft
- Clutch disc low rate pre-damper springs the largest single lash source
- Transmission free play between gears in mesh
- Universal joint spline free play
- Final drive lash between pinion and crown wheel
- Halfshaft spline lash

Tsangarides et al (1985) made an important torsional drivetrain analytical contribution, which specifically addressed transient responses to shock inputs in the presence of a lumped lash in a RWD model simulation.

The model was of the lumped parameter type – component inertias connected in series and parallel by stiffness and damping elements and backlash was effectively included by using zero rated springs. Some of the model data was empirical. They defined tip-in jerk as the rate of change of vehicle acceleration which may/may not be accompanied by an audible sound.
2.11 The loudness of impulsive sounds

It is necessary to refer to the perceptive loudness of impulsive sounds. The ear is assumed to be an energy sensitive device, which responds to impulsive sounds by averaging over a certain time (Norton (1989)).

Annoyance is related to loudness and frequency, and intermittent noises are more annoying than steady state noises (Broch (1969)). Annoyance is also related to noise location; it is more annoying if the noise source is mobile or if the source cannot be located.

2.11.1 Sound

Sound refers to audible pressure fluctuations in air. When a body moves through a medium or vibrates, some energy is transferred to the surrounding medium in the form of sound waves. Sound is also produced by turbulence in air and other fluids, and by fluids moving past stationary bodies. In general, sound may be transmitted by solids, liquids and gases.

2.11.2 Sound Propagation

Sound propagation in air can be compared to ripples on a pond (Blake and Mitchell (1972)). The ripples spread out uniformly in all directions, decreasing in amplitude as they move further from the source. For sound in air, when the distance doubles, the amplitude drops to half, or an equivalent of 6 dB. However, this is only true when there are no reflecting or blocking objects in the sound path. Such ideal conditions are termed free-field conditions.

With an obstacle in the sound path, part of the sound will be reflected, part absorbed and the remainder will be transmitted through the object. How much sound is reflected, absorbed or transmitted depends on the properties of the object, its size and the wavelength of the sound. In general, the object must be larger than one wavelength in order to significantly disturb the sound (Norton (1989)). For example, at 10 kHz the wavelength is 3-4 cm, so even a small object such as a measurement microphone will disturb the sound field hence sound absorption and insulation are readily achieved. But, at 100 Hz, the wavelength is 3-4 meters and sound insulation becomes much more difficult (Norton (1989), and Blake and Mitchell (1972)).
2.11.3 Accelerative noise

Rapid change in velocity of the moving parts during the impact gives rise to a pressure perturbation, which in-turn emits accelerative noise (Richards et al (1979a)). This is not dependent on damping or vibration isolation. Accelerative noise arising from impact is a function of impact duration, size factor, area in contact and impact velocity (Richards et al (1979a)). Dominating accelerative noise mechanisms are those, which relate to very short impact times and in association with backlash. The shorter the impact duration the greater the radiated noise energy.

2.11.4 Ringing noise

This is due to sound radiation from vibrating modes of the attached structures. This continues until all the impact energy has been radiated as sound or has been absorbed in the structural damping system (Richards et al (1979b)). Ringing noise is a function of structural damping, the rate of propagation of vibrational energy, Young’s modulus of the structure, radiation efficiency of the structural panels liable to vibrate flexurally, surface areas and velocities and vibration amplitudes. Richards (1979b) has developed empirical formulae for the prediction of noise generated by impact, so that design action could be taken early in a programme to avoid costly palliation later on.

For bodies with sufficiently compact dimensions it was found that acceleration noise and structural ringing noise were of similar order. However, if local resonant flexural modes were excited by the impact, the radiation energy was significantly higher than the energy radiated during the time of impact. The reason for this was the longer time taken for the lightly damped structure to radiate the noise energy by ringing.

2.12 Structure borne and Airborne Noise

Structure-borne waves are vibration that are predominantly low frequency vibrations that may be audible, but are primarily "felt" through tactile perception, such as clutch whoop (Kelly (1997) and Kelly (1999)). Structure-borne vibration is strictly classified as vibration that is transferred through a solid or semi-solid medium from the source to a receiver. These structure-borne frequencies are generally classified as less than 1000 Hz (Norton (1989)). Structural vibrations can be treated in 2 ways; by damping or isolation.
Airborne noise is the kind of sound that most people think of as noise and travels through gaseous mediums like the air. Some people might classify voice as noise, but one would like to think that things like air conditioning or even the hum of computers are better examples of unwanted noise. These vibrations are detected by human ear and may be impossible to detect with sense of touch. There are 2 ways to treat unwanted airborne sound; by erecting barriers or including absorbers.

2.13 Damping and Isolation

Damping is defined as a treatment to reduce the magnitude of targeted vibrations. All damping materials have different characteristics at different frequencies.

Isolation is defined as a method of detaching or separating the vibration from another system or body. Isolation by definition does nothing to reduce the vibration magnitude. It simply separates the vibration from the system one wishes to protect. Some materials may actually perform damping and isolation functions at particular frequencies.
CHAPTER 3

3 THEORETICAL FORMULATION

3.1 Introduction

Vehicular drivetrain systems, defined in this thesis from flywheel to the contact patch of rear wheels are subjected to many sources of excitation, dominated by an engine and road input. Drivetrain systems respond across the spectrum of physics of motion, from large rigid-body dynamics (such as the rotation of transmission input and output shafts) at low frequencies (such as shuffle, shunt and clutch judder) to high frequency structural vibration of thin-walled structures (such as driveshaft tubes and transmission bell housing). Some of the high frequency content of structural vibration impart sufficient energy to the system that excite the acoustic modal response of contained cavities, such as those already mentioned. This coincidence of structural vibration modes with acoustic response of sub-system components is referred to as structure-fluid interaction and specifically as elasto-acoustic coincidence. In industry, this phenomenon is commonly referred to as drivetrain clank.

It is clear that a fundamental and interactive methodology must be developed to deal with the multi-physics nature of the aforementioned phenomena. Clearly, Lagrangian dynamics is the appropriate methodology to deal with multi-body rigid-body dynamics of an assembly of parts, which are constrained together for their functional assurance. This approach forms the basis of the work reported in this thesis. The inclusion of elastic behaviour of certain system components in the dynamics analysis is critical to investigate certain vibration and noise phenomena, such as clank. This can be achieved in a variety of ways, the simplest form being the use of Euler beams to represent elastic behaviour of system components as a first approximation (see Figure 3.1). In this case the mass and inertial properties of a flexible component is discretised as lumped “dummy” parts, interspersed by Eulerian beams. This approach is known as the Transfer Matrix Method (TMM), and its accuracy improves with progressive discretisation. The beam elements are represented by stiffness matrices, providing a relationship between the relative displacements of the ends of the beam in all the six degrees of freedom motion with the applied forces and moments (see Thomson (1976) and Rahnejat (1998)). Determination of structural damping is quite difficult, but are usually
implemented as a percentage of the stiffness coefficients, typically in the range 0.5%-5% (see Rahnejat (1998) and Rahnejat (2000)). Another approach can be by analytical determination of component flexibility by taking into account the various modal behaviours, such as in bending, torsion or axial elongation/compression. The resulting stiffness matrix can then be obtained by concatenation of the various contributing modal responses. This approach has been highlighted by Okamura et al (1995 a, b) as Dynamic Stiffness Matrix Method (DSMM) and successfully applied to engine dynamics problems. It is clear that in the cases of TMM and DSMM, the number of discretised elements are crucial in order to progressively capture higher modes of vibration (these being usually in combined torsional and deflection modes, and referred to as the flexural modes). Therefore, the most suited approach would be to employ finite element analysis, combined with Lagrangian dynamics. This approach may be regarded as elastodynamics.

A large number of elements would normally be required in order to capture high frequency structural modal responses of the system components. This can make for the inclusion of large mass and stiffness modal matrices that render time marching analyses computationally impossible or extremely time and memory intensive. A suitable approach is to undertake finite element modal analysis by inclusion of sufficient number of elements of suitable type (i.e. stress-strain relations) and select the modal responses at frequencies of interest to include within the multi-body analysis. The approach for this is known as Component Mode Synthesis (CMS).

Once the desired modal responses are included in the form of the suitably reduced mass and stiffness matrices into the Lagrangian dynamic analysis the elastodynamic behaviour of the system can be simulated under given applied excitation (see Gnanakumarr et al (2003)). This approach provides both the rigid body dynamics and structural modal response of the system (i.e. elastodynamic behaviour). However, not all structural modal responses of the system culminate in noise radiation, which by their high frequency metallic nature are often quite disagreeable in nature, clonk being the onomatopoeic term given to these.

To ascertain which modes of structural vibration can lead to noise radiation it is necessary to develop analytical models of sound propagation in hollow cavities, in this case in the thin-walled cylindrical driveshaft tubes. The coincidence of the structural wave mode shapes with
these acoustic waves render elasto-acoustic coupling. For acoustically thick structures (as defined later on), finite element or boundary element techniques are suitable.

All the above methodology, their fundamental basis and their developments, is highlighted in this chapter.

![Figure 3.1: An Euler Beam](image)

### 3.2 Multi-body Dynamics Method

The dynamics of many assembled real systems such as vehicle drivetrain systems are quite complex in nature. In particular, such systems often include constraining elements such as joints and non-linear restraining or structural characteristics, such as sources of compliance (as in stiffness and damping). Whilst the inertial dynamics of the multi-body systems is represented by partial differential equations (as in constrained Lagrangian dynamics), the constraint functions and applied forces may be represented by algebraic or differential type equations. The set of equations; the equations of motion, algebraic constraint functions and applied forces/torques require matrix formulation for simultaneous solution. Often the resulting matrices are quite large in dimensions and include many zero entries. The solution methodology, therefore, requires handling of large sparse matrices that cannot be inverted to obtain the solution vector.
Considering the presented constraints, the simulation models should be developed, using simulation software such as Automatic Dynamic Analysis of Mechanical Systems (ADAMS). The following section in this chapter will describe the ADAMS’s modelling methodology.

### 3.2.1 Model Description

The drivetrain model has 5 independent rigid-body degrees of freedom as shown in Figure 3.2. These rigid body degrees of freedom are:

- Rotation of the transmission input shaft
- Rotation of the layshaft
- Rotation of the transmission output shaft and the driveshaft tubes
- Plunging (axial motion) of the driveshaft
- Rotation of the rear axle halfshafts

Other rigid body degrees of freedom are constrained by assembly joints, as described later.

![Figure 3.2: Multi-body assembly of the drivetrain system](image)

To undertake an elastodynamic analysis as described in section 3.1, it is appropriate to make a multi-body dynamic model of the system using a mechanism model, including the use of assembly constraints. For this purpose, ADAMS modelling environment is used.
The multi-body drivetrain model comprises rigid inertial elements (as parts), assembly constraints and force elements. The force elements include applied forces, as well as sources of compliance. The parts in a multi-body model include the ground as the datum to which the fixed global frame of reference is attached. The engine, clutch and the transmission are represented as a lumped inertial member, revolute-jointed to the ground. The torsional stiffness of the clutch system is included in the model as a restraining (resistive) element. The transmission ratio is set for the second gear engagement. A Dirac impact function with a maximum value of 150 Nm is employed for a short duration, representing the impact loading in the transmission through backlash at the onset of gear meshing.

Other parts include the transmission output shaft, revolute-jointed to both the ground and to the engine/transmission inertial block. This shaft is connected to a splined member by a universal joint. The splines allow the axial float of the driveshaft pieces. In this model the driveshaft pieces (i.e. tubes) are treated as elastic members. Other rigid inertial elements include the pinion, the differential and the left and right rear axle half-shafts. These half-shafts are revolute jointed to the differential with imposed coupling ratios. Appropriate bushings are used between the half-shafts and the ground to provide both the torsional and lateral stiffness characteristics of the axle, as well as the longitudinal stiffness of the tyre at the contact patch. The multi-body assembly is shown in Figure 3.2. The list of parts with their inertial properties is given in Table 4.1 and Table 4.2 provides the list of constraints between the parts of the system, including the number of constraints imposed by each joint or joint primitive. The total number of constraints in the system model is obtained in this table as: 235.

Using Gruebler-Kutzbach expression the total number of independent degrees of freedom is, therefore, obtained as:

\[ n_{DOF} = 6(m-1) - \sum \text{constraints} = 5 \]  
(3.1)

where:

\[ \sum \text{Constraints} = n = 235 \]  
(3.2)

Therefore, it is clear that the multi-body model developed here return the same number of degrees of freedom as that intended in Figure 3.2. The initial impact is introduced by the
meshing pair of the transmission input shaft and the lay-shaft. There are two types of formulation that has been incorporated in the model, described later.

### 3.2.2 Lagrange’s Equations for Constraint Systems

The multi-body model is a constrained non-linear dynamics model, incorporating the elastic behaviour of the driveshaft tubes. As previously mentioned, this model is created in ADAMS, a multi-body code developed by MSC Software. The code is based on the automatic generation of equations of motion, for each part of the drivetrain assembly, based on constrained Lagrangian dynamics (Lagrange, 1788), and using generalized Eulerian body 3-1-3 frame of reference. If the generalised co-ordinates are denoted by \( \{q_j\}_{j=1 \to 6} = \{x, y, z, \psi, \theta, \varphi\}^T \), then:

\[
\frac{d}{dt} \left( \frac{\partial K}{\partial \dot{q}} \right) - \frac{\partial K}{\partial q} \frac{\partial V}{\partial q} + \sum_{n=1}^{n} \lambda_n \frac{\partial C_n}{\partial q} = 0
\]  

(3.3)

where: the generalised forces in the Eulerian frame of reference are given as:

\[
F_q = -\frac{\partial V}{\partial q}
\]  

(3.4)

The \( n \) constraint functions for the different joints in the drivetrain model are represented by a combination of holonomic (displacement dependent constraints, such as restriction of motion in a given direction or coupling of pinion rotation to rack travel, for instance in the rack and pinion steering system) and non-holonomic functions as (which are functions of state variable derivatives, such as constant velocity joints) (see Orlandea, 1999):

\[
\begin{bmatrix}
\dot{C}_i \\
\dot{q}_i \frac{\partial C_k}{\partial q_j}
\end{bmatrix} = 0, \ j = 1 \to 6, \ k = 1 \to n
\]  

(3.5)

The compliance of the flexible members is given by stiffness and damping matrices obtained by super-element modal analysis, as described in the next section.
For parts in the model, as listed in Table 4.1, the inertial rigid body motions are described by the first term on the left hand side (LHS) of the Lagrange's equation.

The term \( \frac{\partial K}{\partial \dot{q}} \) is the conjugate momentum in the co-ordinate direction \( q \), since for instance for the translational motion in \( x \), \( K = \frac{1}{2} m \dot{x}^2 \) and \( \dot{q} = \dot{x} \), then \( M_x = \frac{\partial K}{\partial \dot{x}} = m \ddot{x} \). This is also true for the other translational degrees of freedom, where \( q = y \) or \( q = z \). For rotational degrees of freedom, given by the Euler angles \( \psi \) (first rotation about \( z \)-axis), \( \theta \) (second rotation about the new \( x \)-axis) and \( \phi \) (final rotation about the new \( z \)-axis): \( K = \frac{1}{2} I \dot{\psi}^2 \) and \( M_\psi = I \ddot{\psi} \).

The rate of change of momentum (translational or rotational) provides the inertial force. Therefore:

\[
\frac{d}{dt} \left( \frac{\partial K}{\partial \dot{q}} \right) = \frac{d}{dt} (m \dot{q}) = m \ddot{q} \tag{3.6}
\]

where \( q = x, y \) or \( z \) or

\[
\frac{d}{dt} (I \dot{q}) = I \ddot{q} \tag{3.7}
\]

where \( q = \psi, \theta \) or \( \phi \)

Hence, the first term on the LHS of the Lagrange's equation (i.e. \( \frac{d}{dt} (m \ddot{x}) = m \dddot{x} \)) is the inertial force. When the co-ordinates have been chosen properly in-line with the directions of motion, then \( \frac{\partial K}{\partial \dot{q}} = 0 \) always, since kinetic energy is not a function of displacement. The term \( \frac{\partial V}{\partial \dot{q}} \) is the rate of change potential energy, \( V \) with respect to displacement. For rigid inertial parts, \( V \) is the potential energy, whilst for flexible members this represents the stored strain energy. In any case, Euler has shown that: \( F_q = -\frac{\partial V}{\partial q} = -m g \), and the negative sign indicates the direction of gravity. For example, for a spring the stored strain energy is \( \frac{1}{2} kx^2 \), then \( F_q = -\frac{\partial}{\partial x} \left( \frac{1}{2} kx^2 \right) = -kx \), which is the restoring force.

Therefore, the first three terms on the left hand side deal with inertial, body and restoring/restraining forces. The last term gives the reaction forces caused by the constraining
elements. For instance, for a revolute pair, there are 5 constraining functions (i.e. \( C_n \) giving \( n=5 \) equations). The term \( \frac{\partial C_n}{\partial q} \) provides the effect of the constraint in the direction \( q \). The reaction force is given by: \( \lambda_n \frac{\partial C_n}{\partial q} \), where \( \lambda_n \) is the Lagrange's multiplier. However, there may be many contributing reactions in a given direction, therefore the summation mark in the Lagrange's equation.

For example, a revolute joint imposes 5 constraints, or 5 scalar algebraic equations. A revolute joint such as between the transmission output shaft and the ground in the powertrain model, consists of two types of primitive constraint function: point coincidence constraint and orthogonal axes constraint. The former simply refers to the restriction of translational motion of a marker point on a given part relative to another on an adjacent part. If the position vectors of these markers (\( i \) and \( j \)) from the fixed global frame of reference are denoted by \( R_i \) and \( R_j \), then (see Figure 3.3) (Rahnejat 1998):

\[
R_i + r_i = R_j + r_j \tag{3.8}
\]

Therefore, \( R + r = 0 \), where: \( R = R_i - R_j \), and \( r = r_i - r_j \). The position vector \( r \) in the local part frame of reference (LPRF) can be given as:

\[
r = L \times e_q = \{e_p, e_q, e_r\} \times \{0 L 0\}^T \tag{3.9}
\]

Replacing from (3.9) into (3.8) and noting that vector \( r \) has the components: \( x, y \) and \( z \), then:

\[
\{ijk\} \cdot \{xyz\}^T + \{e_p, e_q, e_r\} \cdot \{0 L 0\}^T = 0 \tag{3.10}
\]

Now replacing for unit vectors \( \{e_p, e_q, e_r\}^T \) in terms of the unit vectors \( \{ijk\}^T \), thus:

\[
\{ijk\} \cdot \{x - L(S \psi C \theta C \phi + C \psi S \phi) y - L(S \psi S \phi - C \psi C \theta C \phi) z + LS \theta C \phi\}^T = 0 \tag{3.11}
\]
where here: $S \equiv \sin$ and $C \equiv \cos$

Equation (3.11) provides 3 scalar constraint functions as shown later. The other two revolute joint constraints impose axes orthogonality conditions as:

$$k \cdot e_p = 0$$  \hspace{1cm} (3.12)

$$k \cdot e_q = 0$$  \hspace{1cm} (3.13)

Note that (see Figure 3.3):

$$\{PQR\} = \{e_p e_q e_r\}$$  \hspace{1cm} (3.14)

$$\{XYZ\} = \{ijk\}$$  \hspace{1cm} (3.15)

Thus:

$$k \cdot e_p = 0 \text{ or } \sin \theta \cdot \sin \phi = 0$$

$$k \cdot e_q = 0 \text{ or } \sin \theta \cdot \cos \phi = 0$$

Hence the 5 constraint equations are:

$$\begin{bmatrix}
 c_1 \\
 c_2 \\
 c_3 \\
 c_4 \\
 c_5
\end{bmatrix} = \begin{bmatrix}
 x - L \left( S \psi C \theta C \phi + C \psi S \theta \right) \\
 y - L \left( S \psi S \phi - C \psi C \theta C \phi \right) \\
 z + L S \theta C \phi \\
 S \theta S \phi \\
 S \theta C \phi
\end{bmatrix} = \begin{bmatrix}
 0 \\
 0 \\
 0 \\
 0 \\
 0
\end{bmatrix}$$  \hspace{1cm} (3.16)

These equations provide $C_n$ functions, as used in the Lagrange’s equation, such as:

$$C_1 = x - L \left( S \psi C \theta C \phi + C \psi S \theta \right) = 0$$  \hspace{1cm} (3.17)

It is clear that $\frac{\partial C_1}{\partial x} = 1$ in the $x$-direction, the last term in the Lagrange’s equation gives:

$$\lambda_1 \frac{\partial C_1}{\partial x} = \lambda_1$$

Therefore, $\lambda_1$ is the reaction force in the $x$-direction. The same method can be used to determine all the reaction forces in other degrees of freedom.
Finally, using all the terms in the Lagrange's equation, 6 equations of motion (differential equations) result per part in a multi-body system. There are also a number of constraint equations such as those given above in the matrix equation. The number of these equations, in a multi-body model, is the same as the number of constraints in the system as listed in Table 4.2.

This mix of differential equations (equations of motion) and the algebraic equations (constraint functions) have to be solved simultaneously in each step of time. Such system of equation is referred to as Differential-Algebraic Equation (DAE) set.

![Figure 3.3: Time derivatives of Euler angles](image)

### 3.2.3 Generalised Forces

These elastic restoring forces and other applied forces are transformed to the Euler's (Euler (1767)) generalized co-ordinates by the application of Johannes Bernoulli's principle of virtual work. For the applied forces:

\[
\{\delta W\} = \{\delta q_t\}^T \{F\} + \{\delta \beta\}^T [T_{ij}] \{F\}
\]  

where the small translational and rotational displacements in the Euler frame are given by (whilst \{\delta \beta\} is the infinitesimal rotation about the local frame of reference):

\[
\delta q_t = \{\delta x, \delta y, \delta z\}^T
\]
The transformation matrix from a local frame of reference (indicated by the subscript $i$) to the
global frame of reference (indicated by $j$) is given as:

$$
\begin{bmatrix}
\cos \psi \cos \varphi - \sin \psi \cos \theta \sin \varphi & -\cos \psi \sin \varphi - \sin \psi \cos \theta \cos \varphi & \sin \psi \sin \theta \\
\sin \psi \cos \varphi + \cos \psi \cos \theta \sin \varphi & -\sin \psi \sin \varphi + \cos \psi \cos \theta \cos \varphi & -\cos \psi \sin \theta \\
\sin \theta \sin \varphi & \sin \theta \cos \varphi & \cos \theta
\end{bmatrix} (3.21)
$$

The coefficient of $\{\delta q, 8_r\}$ gives the generalized force, whilst the second term provides the
torque about the centre of mass of the part with the infinitesimal rotation $\{\delta \beta\}$ expressed in
terms of the local part frame of reference. The torque contribution can be given in terms of
the global Euler frame of reference as: $\{\delta q, 8_r\} [e][T_y] \{F\}$, where $[e]$ is the transformation
from the Euler-axis frame to the local part frame:

$$
[e] = \begin{bmatrix}
\sin \theta \sin \varphi & 0 & \cos \varphi \\
\sin \theta \cos \varphi & 0 & -\sin \varphi \\
\cos \theta & 1 & 0
\end{bmatrix} (3.22)
$$

For applied torques, the virtual work is given as:

$$
\{\delta W\} = \{\delta q, 8_r\} [e][T_y] \{F\} (3.23)
$$

And the coefficient of $\{\delta q, r\}$ gives the generalized torque contributions, noting
that: $\{\delta \beta\} = [e]\{\delta q, r\}$.

The elastic forces are set out as:

$$
\{F_{ext}\}_{j=1\rightarrow 3} = \{F\} (3.24)
$$

and:

$$
\delta q, r = \{\delta \psi, \delta \theta, \delta \varphi\}^T
$$
3.2.4 Constraining Reactions

The generalised forces are the applied, restoring and dissipative forces, such as those introduced by the compliant members in the previous section. The constraining elements in a multi-body system also introduce reactions that are included in Lagrangian dynamics by the last term on the left-hand side of the Lagrange’s equation.

The reaction forces and moments, introduced by the imposed constraints, are obtained in the same manner in terms of the Lagrange multipliers, $\lambda_k$ where the following relation for infinitesimal changes should hold:

$$\lambda_k \delta C_k = 0$$

(3.26)

where $C_k$ is a holonomic constraint function. These are the primitive functions that ensure positional or orientation conditions. Certain combinations of these form a physical joint. The most common types are the at-point or point coincident constraint, the in-plane and in-line joint primitives, perpendicular axes and prescribed angular orientation such as the imposition of parallel axes condition. Coupling constraints may also be imposed, relating the position (by a holonomic constraint) or velocity (by a non-holonomic constraint) of parts with respect to each other.

3.2.5 Equations of Motion

The formulated generalised forces in case of body forces, applied forces and the constraining reactions can be implemented in the Lagrange’s equation. The differential-algebraic equation set can now be represented as follows:
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The matrix in the above equation provides the coefficient of the unknown vector \( \{\delta q, \delta \lambda \}^T \) and is known as the Jacobian matrix, \([J]\). Note that for the drivetrain model there are 235 constraints. Therefore, there are 235 of unknown Lagrange’s multipliers to be evaluated in each small time step \( \delta t \). There are also 210 \( \delta q \) values also to be obtained in the case of the current model. This is because ADAMS automatically generates, using Lagrangian dynamics 6 equations of motion per part in the model, and there are 35 parts in the model, see Table 4.1. Thus, there are 481 differential-algebraic equations to be solved simultaneously in each step of time, excluding the flexibility of parts.

The advantage of using constrained Lagrangian dynamics is that these equations are automatically generated by a code such as ADAMS. The disadvantage of the method of automatic generation of equations is that many equations are set, where the given direction is already constrained, resulting in a very large Jacobian matrix, which is further filled with zero entries as shown to arrive at a square matrix for manipulation. This advantage of the method, however, far outweighs this disadvantage, because otherwise equations of motion must be derived analytically through Newton-Euler method, by breaking the system into a number of free-body diagrams and specifying all reactions accurately, a procedure which is prone to possible errors, when a very complex system such as the current model is considered.

3.3 Method of Formulation and Solution

The governing equations of motion for an assembly of rigid and elastic constrained bodies are derived from the Lagrange’s equation for constrained systems in the form:
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\[
\frac{d}{dt} \left( \frac{\partial K}{\partial \xi_j} \right) - \frac{\partial K}{\partial \xi_j} \cdot F_{\xi_j} + \sum_{i=1}^{n} \lambda_i \frac{\partial C_k}{\partial \xi_j} = 0
\]  (3.28)

where

\( \{ \xi_j \}_{j=1 \rightarrow 6} = \{ x, y, z, \psi, \theta, \phi \}^T \) denote the rigid body degrees of freedom

\( \{ \xi_j \}_{j=6 \rightarrow 6+m} = \{ x, y, z, \psi, \theta, \phi, q \}^T \) denote the flexible body degrees of freedom (\( q \) are the modal coordinates and \( m \) their total number)

\( \lambda \) are the Lagrange’s multipliers for the constraints \( C_k \).

The \( n \) constraint functions for the different joints in the drivetrain model are represented by a combination of holonomic and non-holonomic functions as:

\[
\begin{bmatrix}
    C_k \\
    \xi_j \\
    \frac{\partial C_k}{\partial \xi_j} \\
\end{bmatrix} = 0, \ (k = 1 \rightarrow n), \ (j = 1 \rightarrow 6 \ \text{or} \ 1 \rightarrow 6 + m)
\]  (3.29)

The multi-body model of the mechanical system is a constrained non-linear dynamics model, incorporating the elastic behaviour of the driveshaft tubes. The dynamic simulation code ADAMS is based on the automatic generation of equations of motion, using constrained Lagrangian dynamics, formulated in the generalized Eulerian 3-1-3 frame of reference. The compliance of the flexible members is given by stiffness and damping matrices obtained during the creation of superelements by modal analysis, as briefly described in the following sections (Theodossiades et al (2004)).

3.3.1 LU Decomposition (Cholesky factorisation)

For a set of linear equations which has to be repeatedly solved with different inhomogenous terms, the \( LU \) decomposition is recommended (Orlandea et al (1978), Rahnejat (1998)). The \( LU \) decomposition represents the replacement of the \( Jacobian \) matrix by a product of two triangular matrices, known as the lower and the upper triangular matrices. In the lower
triangular matrix all the non-zero elements occupy the triangle on or below the diagonal, whilst in the upper triangular matrix all such terms reside on or above the diagonal.

Therefore:

\[ [L] \cdot [U] = [J] \]  
\[ (3.30) \]

where: \([J]\) is the Jacobian, and the equation of the multi-body system is:

\[ [J] \cdot \{q, \lambda\}^T = \{F_I\} \]  
\[ (3.31) \]

The \(LU\) decomposition can be used to solve equation (3.31) as follows:

\[ [J] \cdot \{q, \lambda\}^T = ([L] \cdot [U]) \cdot \{q, \lambda\}^T = [L] \cdot ([U] \cdot \{q, \lambda\}^T) = \{F_I\} \]  
\[ (3.32) \]

Equation (3.32) can then be represented by a pair of matrix equations. The first step is to solve for \(\{V\}\) vector as indicated below:

\[ [L] \cdot \{V\} = \{F_I\} \]  
\[ (3.33) \]

and afterwards by solving:

\[ [U] \{q, \lambda\}^T = \{V\} \]  
\[ (3.34) \]

This set of matrix equations is sufficient for the solution of a linear system. However, the set of equations in the multi-body systems described in this thesis is non-linear, requiring the use of Newton-Raphson method.

### 3.4 Component Mode Synthesis and Super-Element Creation

To incorporate component flexibility and obtain structural dynamic contributions finite element analysis is the most suitable approach. The introduction of FEA techniques (via use of MSC/NASTRAN code) for the detailed investigation of the behaviour of several parts,
such as the hollow driveshaft tubes of the powertrain model has been considered as necessary, because it achieves two desirable goals:

- It increases the *fidelity* and *accuracy* of the multi-body model

- The realistic *loads - initial conditions* that are required for an external acoustic analysis, may be obtained in a natural way by incorporating an FEA model of a specific component in the mechanical system and simulating the in-service events.

The driveshaft pieces (first, second and third tubes) are major components of the drivetrain, which have significant flexural behaviour due to their thin wall thickness compared to the other parts that present a rather rigid behaviour. By incorporating FEA models instead of rigid bodies for the representation of these components, it is also feasible to determine the *principal stresses* that are generated during the simulation exercise. Furthermore, the initial conditions (displacements, velocities, loads etc.) which are required for an acoustic analysis of the driveshaft tubes can be extracted, which enable determination of sound pressure fields in the exterior domain.

The most important assumption behind this procedure is the consideration of small, linear elastic deformations relative to a local frame of reference, whilst this local frame of reference undergoes large, non-linear motion with respect to a fixed global frame of reference. The discretisation of a component into a finite element model represents the infinite number of degrees of freedom with a finite, but very large number of finite element degrees of freedom. The linear deformations of the nodes of this finite element mode, $u$, can be approximated as a linear combination of a smaller number of shape vectors (or mode shapes), $\phi$ as:

$$ u = \sum_{i=1}^{m} \phi_i q_i $$

(3.35)

where, $m$ is the number of mode shapes. The scale factors of amplitudes, $q$, are the modal coordinates.
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The main concept of *modal superposition* is that the behaviour of a component with a very large number of nodal DOF in a pre-determined frequency area can be captured with a much smaller number of modal degrees of freedom. Thus, the finite element modes can be rewritten in the matrix form as:

\[ u = \Phi q \]  

(3.36)

where, \( q \) is the vector of modal coordinates and the modes \( \phi_i \) are included in the columns of the modal matrix, \( \Phi \). This matrix is the transformation from the small set of modal coordinates, \( q \), to the larger set of physical coordinates, \( u \).

The determination of the modal matrix \( M \) can be achieved due to the Craig-Brampton (Craig and Bampton (1968)) reduction method, which is one of the most general methods for *Component Mode Synthesis* techniques. The summary of this method is described in the following paragraphs.

- A set of *Boundary* degrees of freedom (\( u_B \)) is defined, which is not to be subject to modal superposition and is preserved exactly in the modal basis.
- A set of *Interior* degrees of freedom (\( u_I \)) is defined.

Additionally, two sets of mode shapes are defined, as follows:

- The *Constraint Modes* (\( q_C \)) are static shapes, which are obtained by giving each of the boundary degrees of freedom a unit displacement, while holding all other boundary degrees of freedom fixed. There is a one-to-one correspondence between the modal coordinates of the constraint modes and the displacement in the corresponding boundary degrees of freedom, \( q_C = u_B \).
- The *Fixed Boundary Normal Modes* (\( q_N \)), which are obtained by fixing the boundary degrees of freedom and computing a solution of the eigenvalue problem. These modes define the modal expansion of the interior degrees of freedom. The quality of this expansion is proportional to the total number of modes.
According to the aforementioned, the relationship between the physical degrees of freedom, the Craig Brampton modes and their modal coordinates is expressed as:

\[
\begin{pmatrix}
\dot{u}_r \\
\dot{u}_t
\end{pmatrix} = \begin{pmatrix}
I & 0 \\
\Phi_C & \Phi_{IN}
\end{pmatrix}
\begin{pmatrix}
q_C \\
q_N
\end{pmatrix}
\] (3.37)

where: \( I, 0 \) are unity and zero matrices, respectively.

The generalized stiffness and mass matrices are obtained through the following transformations:

\[
\dot{M} = \Phi^T \dot{M} \Phi = \begin{pmatrix}
\dot{M}_{CC} & 0 \\
0 & \dot{M}_{NN}
\end{pmatrix}
\] (3.38)

\[
\dot{K} = \Phi^T \dot{K} \Phi = \begin{pmatrix}
\dot{K}_{CC} & 0 \\
0 & \dot{K}_{NN}
\end{pmatrix}
\] (3.39)

where, \( \dot{M}_{NN}, \dot{K}_{NN} \) are diagonal matrices and \( \dot{K} \) is a block diagonal matrix.

Since the Craig-Brampton modes are not an orthogonal set of modes, a mode shape orthonormalisation procedure is applied, by solving the eigenvalue problem:

\[
\dot{K}q_m = \gamma \dot{M}q
\] (3.40)

The obtained eigenvectors are arranged in a transformation matrix \( N \), that transforms the Craig-Brampton modal basis to an equivalent, orthogonal basis with modal coordinates \( q^* \), where:

\[
Nq^* = q
\] (3.41)

Thus, the effect on the superposition formula is:
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\[ u = \sum_{i=1}^{N} \phi_i q_i = \sum_{i=1}^{M} \phi_i N q_i^* = \sum_{i=1}^{M} \phi_i^* q_i^* \]

(3.42)

where, \( \phi_i^* \) are the orthogonal Craig-Brampton modes.

*Four-noded, two-dimensional shell elements* with pre-specified thickness (1.65 mm) have been used for the tube walls in the super-element creation instead of solid elements, since the ratio of the tubes’ radii to their thickness is very high (>20). The number of elements and their size across the shaft length and in the perimeter has been kept large enough so that the higher modes with complicated shapes can be captured efficiently. *Clamped-clamped* boundary conditions have been applied to the tube edges to represent the actual assembly conditions in the vehicular drivetrain system. This has been achieved, using appropriate elements in the tube edges so that a master node (in the edge centre) is rigidly attached to many slave nodes (in the edge circumferential) via a type of connection that creates a localised stiffness in the model (RBE2 element, *MSC-PATRAN/NASTRAN* (2001)). Since the master node’s Degrees of Freedom are independent, this node can be promoted to an *attachment point* and used to connect the flexible body to the neighbouring rigid ones of the multi-body model.

A sufficient number of natural modes have been kept in the super-elements creation in order to obtain accurate results in the specified frequency area, where clonk usually occurs (300-5000 Hz). There is no exact rule regarding the number of modes that one should keep in the super-element model, although a general rule of thumb is that a number of modes that cover at least two times the desired frequency area should be kept. The geometric properties of a tube determine its modal density in this area (its upper limit has been determined at 12,000 Hz in this analysis). Consequently, each tube super-element includes a variety of modes, starting from low frequency bending behaviour and lead to complicated shapes that are combination of bending-axial and high frequency torsional modes.

### 3.5 Meshing Cycle Gear Forces

The impulsive action leading to wave propagation and yielding clonk condition is caused by the impact of gear meshing pair under sudden application of throttle or clutch action.
Therefore, a model of impact of meshing gear teeth pair through backlash is required to be included in the elasto-multi-body dynamics model, described above.

The analytical representation and calculation of the transmission gear forces between the engaged teeth pair(s) during the meshing cycle is a critical task, because its inclusion contributes significantly to the successful simulation of the impact conditions. The geometric characteristics of the helical gear teeth affect the dynamics and vibrational behaviour of the mechanical system in a significant way. In the presence of gear backlash, which is either introduced intentionally at the design stages or caused by manufacturing errors and wear. Therefore the equations of motion of such systems become strongly non-linear. Another complication arises from the variable number of gear teeth pairs, which are in contact simultaneously, causing a variation of the equivalent gear meshing stiffness. These two factors introduce serious difficulties in the analysis and can obscure the interpretation of the numerical results. The dynamics of a gear-pair system, involving backlash and time-dependent mesh stiffness \((k(t))\) can be investigated using piece-wise linear equations of motion with time-dependent coefficients. The centres of both gears are constrained against any lateral motion. The meshing stiffness depends on the number and position of the gear teeth pairs, which are in contact at any given instance (see Figure 3.4) and is a periodic function of the relative angular position of the gears (see Figure 3.5), (Zhang et al (1999), Choi and David (1990), and Chen and Tsay, (2002)). In this study, a simple and accurate method has been followed, based on the British Standard ISO 6336. According to this, the meshing stiffness is calculated, using the value of the single stiffness, \(k'\), which is the maximum stiffness normal to the helix of a single tooth pair, representing the requisite load over a 1 mm face-width directed along the line of action to produce in-line with the load deformation amounting to 1 \(\mu\)m. It is given by the function:

\[
k' = 0.8 \times k'_h \times C_R \times C_B \times \cos \beta
\]

(3.43)

For details regarding the calculation of the theoretical single stiffness, \(k'_h\), gear blank factor, \(C_R\) and basic rack factor, \(C_B\), one can go back to the ISO 6336. The total meshing stiffness is given by the function:

\[
k = \text{total length of contact lines} \times k'
\]

(3.44)
If the tooth-to-tooth variations (i.e. pitch errors, tooth finishing) are neglected, the fundamental frequency of both of these quantities equals the gear meshing frequency:

\[
\omega_m = n_1 \omega_1 = n_2 \omega_2
\]  

(3.45)

where the integers \( n_1 \) and \( n_2 \) stand for the teeth number and \( \omega_1, \omega_2 \) are the angular velocities of the pinion and gear, respectively. The force developed between the pair of gears is given by the product: 

\[ k(t)h(x), \]

where, \( x(t) = R_1 \phi_1(t) - R_2 \phi_2(t) \), \( R_1 \) and \( R_2 \) represent the contact point radii of the gears, \( \phi_1(t) \) and \( \phi_2(t) \) are the two torsional coordinates (rotation angles of the gears) and

\[
h(x) = \begin{cases} 
  x - b, & x \geq b \\
  0, & |x| < b \\
  x + b, & x \leq -b 
\end{cases}
\]  

(3.46)

where \( 2b \) represents the total normal backlash between the gear teeth (Theodossiades et al. (2000)).

Figure 3.4: Zone of Contact and Contact Elements (3 Teeth Pairs in Contact).
3.6 Acoustics

The study of acoustic emission is critical in the understanding of the clonk phenomenon. Not all the structural modes of vibration of the thin-walled driveshaft tubes result in acoustic emission. It is important to identify the troublesome structural modes, usually in the range 1-5 KHz, which coincide with acoustic cavity modes and impart sufficient energy to cause noise radiation.

3.6.1 Structural analysis of circular cylindrical shells

Arnold and Warbuton (1949) have defined thin cylindrical shell vibration modes, given by combination of $m$ axial vibration and $n$ circumference vibration modes. For cylindrical shells a co-ordinate system; $r, \theta, z$ can be used. The unit vector directions are $e_r$ in the radial direction at any cross-section, $e_\theta$ in the circumferential direction and $e_z$ in the axial direction as shown in Figure 3.6. Then, the displacements in these directions are denoted by $w$, $v$ and $u$ respectively. The equations of motion for such a thin cylindrical shell can be obtained from the basic theory of extensional vibration of thin plates as described by Love (1944). In the
theory developed by Love (1944) the effect of transverse shearing stress on the equilibrium of forces in the circumferential direction is considered to be negligible (Krauss (1967)). Love (1944) noted that for extensional vibration of such thin plates "the form of equations shows that there is a complete separation of modes of vibration involving transverse displacements or flexure from those involving displacement in the plane of the plate". This means the frequencies of the extensional modes will be independent of the plate thickness, whilst those in transverse vibration and flexure are proportional to the thickness. In deriving the equations of motion for the vibrating thin cylindrical shells, the acoustic medium is assumed to be inviscid and the loading due to this is assumed to act normally to the surface of the cylindrical shell. Therefore, for a thin elastic cylindrical shell Junger (1986) obtained the equations of motion, making the following assumptions:

- The thickness of the shell is small compared to the smallest radius of curvature (i.e. $h<<a/10$).
- The deformation is small compared to the shell thickness.
- The transverse normal stress, normal to the neutral surface, is negligible.
- Strain normal to neutral surface is negligible.

The equations of motion are:

\[ \frac{\partial^2 u}{\partial z^2} + \frac{1 - \nu}{2a^2} \frac{\partial^2 u}{\partial \theta^2} + \frac{1 + \nu}{a} \frac{\partial^2 v}{\partial z \partial \theta} + \frac{\nu}{a} \frac{\partial w}{\partial z} = \frac{1}{c_p^2} \frac{\partial^2 u}{\partial t^2} \]  \hspace{1cm} (3.47)

\[ \frac{1 + \nu}{2a} \frac{\partial^2 u}{\partial z \partial \theta} + \frac{1 - \nu}{2} \frac{\partial^2 v}{\partial z^2} + \frac{1}{a^2} \frac{\partial^2 v}{\partial \theta^2} + \frac{1}{a^2} \frac{\partial w}{\partial \theta} = \frac{1}{c_p^2} \frac{\partial^2 v}{\partial t^2} \]  \hspace{1cm} (3.48)

\[ \frac{\nu}{a} \frac{\partial u}{\partial z} + \frac{1}{a^2} \frac{\partial v}{\partial \theta} + \frac{w}{a^2} + \beta^2 \left( \frac{a^2}{a^2} \frac{\partial^4 w}{\partial z^4} + 2 \frac{\partial^4 w}{\partial z^2 \partial \theta^2} + \frac{1}{a^2} \frac{\partial^4 w}{\partial \theta^4} \right) = \frac{p_a(1-\nu^2)}{Eh} - \frac{1}{c_p^2} \frac{\partial^2 w}{\partial t^2} \]  \hspace{1cm} (3.49)

where

\[ \beta^2 = \frac{h^2}{12a^2} \]  \hspace{1cm} (3.50)
Note that: $p_a = p_a(\theta, z)$ is the external load, acting normal to the cylindrical surface of the shell, and: $c_p = \frac{E}{\sqrt{(1-v^2)}} \rho$ is the velocity of compressive waves in plates, and $\rho$ is the density of the elastic material.

The boundary conditions used are:

For a simply supported cylinder,

At $z = 0$ and $z = L$ (ends of the cylinder):

$v = 0$, $w = 0$, $\partial^2 u/\partial z^2 = 0$

For clamped ends:

At $z = 0$ and $z = L$ (ends of cylinder):

$U = v = w = 0$ and $\partial w/\partial z = 0$

The displacement components are:

$u = \sum_{m,n} U_{mn} \cos n\theta \sin k_m z$ (axial displacement)

$v = \sum_{m,n} V_{mn} \sin n\theta \cos k_m z$ (circumferential displacement)

$w = \sum_{m,n} W_{mn} \cos n\theta \cos k_m z$ (radial displacement)

where: $k_m = (2m-1)\pi/2L$, $m = 1, 2, 3, ...$
$U_{mn}$, $V_{mn}$ and $W_{mn}$ are displacement amplitudes in the axial, circumferential and radial directions respectively. $m$ is the number of axial half-waves (see Figure 3.7) and $n$ is the number of the circumferential full-waves (see Figure 3.8).

The motion is a combination of standing waves in both the circumferential and longitudinal (axial) directions. The substitution of the solutions in the equations of motion and incorporating the boundary conditions yields three simultaneous equations, the eigen-values of which determine the natural frequencies of the shell. The eigen-values are proportional to the square of the natural frequencies, $\xi \propto \omega^2$.

For each combination of circumferential and longitudinal modes, there are three natural frequencies, which are the roots of the cubic energy equation. Soëdel (1981) has provided a detailed work on the solution of the equation of motions. These natural frequencies are:
• $\omega_t = a k_m \sqrt{(1-v^2)/2}$, natural frequency with predominant torsional component.
• $\omega_a = a k_m \sqrt{1-v^2}$, natural frequency with predominant axial component.
• $\omega_r = \sqrt{1+\beta^2 k'm^2 a^4}$, natural frequency with predominant radial (flexural) component, which is normally the lowest natural frequency.

The effect of external forces on the equations of vibration of the thin cylindrical shells can also be included in the equations (3.47)-(3.49), provided that these forces are normal to the perfectly cylindrical surface. However, in the case of driveshaft tubes investigated here, such external forcing does not exist. In other applications both external driving forces may be involved, as well as modal behaviour of imperfect cylindrical shells, as in the case of vibrating gyroscopes made of cylindrical shells with some degree of ovality, as described by Fox (1999). Then, given boundary conditions, such as clamped-free arrangement, one may use harmonic representation of displacements: $u$, $v$ and $w$ in terms of generalised co-ordinates and set the equation of motion in a generalised form through Lagrange’s equation, as shown by Fox (1999). The advantage of this approach is also in the inclusion of the rigid body rotational degrees of freedom of the system as well. In this thesis (see chapter 4) the same approach is undertaken, but the modal behaviour of the shell is incorporated into the Lagrangian dynamics via use of Super-element finite element analysis and component mode synthesis.

### 3.6.2 Coincidence

Higher order acoustic modes propagate with phase speeds that vary with frequency (these are regarded as dispersive waves), whereas plane waves propagate at a constant speed (Norton (1989)).

Sound waves in the axial direction inside a cylindrical shell demonstrate a continuous variation of axial wave number with frequency. Standing structural waves in circumferential direction have constant circumferential wave numbers, while those in the axial direction have discrete values of axial wave number components.

Coincidence between internal sound waves and resonant structural (flexural) modes of cylinder wall requires an exact wave number (wavelength) and frequency matching in both
axial and circumferential directions (coupling). This is called complete coincidence. However, in practice: There is exact wave number and frequency coupling in the circumferential direction, if: \( n = p \).

There is exact wave number, but not frequency coupling in the axial direction, since a cylinder has discrete values of structural natural frequencies, as described above. When structural and sound waves have different frequencies, yet equal wave numbers (i.e. \( k_x = k_m \)) at the cylinder wall, the coincidence is termed wave number coincidence. In general terms, the coincidence occurs between the \((m,n)\) structural modes and the \((p,q)\) acoustic modes.

### 3.6.3 Sound radiation

Ring frequency, \( f_r \), is defined by Wang (2000 a, b) as the frequency at which the wavelength of extensional (axial) waves in the shell becomes equal to the shell circumference, obtained as: \( f_r = (1/2\pi a)\sqrt{E/\rho} \). This definition is an alternative to that of Forsberg (1964). It is important to note that the analysis carried out by Junger (1986) to obtain the equations of motion for the thin cylindrical shells, described above, based on the bending of slender beams apply as an approximation when the modal responses are below this ring frequency. Therefore, for driveshaft tubes, described later which are combinations of axial and radial modes (usually with \( n=0 \) see Figure 3.8) are efficient radiators of noise. Another assumption made is that the tubes under investigation are free of any imperfections in wall thickness or geometry. In practice, this is not true of any structure, leading to some deviations in modal behaviour as shown by Hwang et al (1999) and in particular for circular thin rings by Fox et al (1999).

For the driveshaft tubes: \( a = 37.5 \) mm, made of Steel, leading to a ring frequency of \( f_r = 21.74 \) KHz.

The critical frequency, \( f_c \), is defined by Wang (2000 a, b) as the frequency at which the acoustic wavelength in the medium is equal to the acoustic wavelength in the shell structural material, or it is the frequency at which the bending wave velocity in the structure equals the speed of sound in the fluid (Norton (1989)): \( f_c = (c^2/2\pi h)\sqrt{12\rho(1-v^2)/E} \).
For the driveshaft tubes in the current investigation, \( h = 1.65 \text{ mm} \), made of Steel. Also: \( c = 340 \text{ m/s} \). Thus: \( f_c = 7.19 \text{ KHz} \).

A shell is described as acoustically thin, if: \( f_r / f_c < 1 \), and acoustically thick, if: \( f_r / f_c > 1 \).

Therefore, although the driveshaft tubes are geometrically thin, they are actually acoustically thick. This enables the use of finite element modal analysis. For acoustically thin structures the FEA technique is not recommended. A more representative approach will be Statistical Energy Analysis (SEA).

The modal density is defined as the number of vibration modes per unit response frequency (Norton (1989)).

The acoustic efficiency is the ratio of the radiated sound power to the sound power that is radiated from a plane wave with an equivalent surface area and mean square surface velocity (Fyfe and Ismail (1989)).

Statistical analysis is used when the acoustic properties and resonant frequencies are densely packed and it is not practical to analyse them individually (Wang (2000a, b)). The analysis is valid at high frequencies.

Deterministic analysis considers, at each frequency, individual vibration velocity distributions, which could be affected by the geometry and boundary conditions. This type of analysis is significant in the region of lower frequencies (Fyfe and Ismail (1989)).

Shell curvature increases flexural wave phase velocities and subsequently increases sound radiation efficiency and reduces density of natural frequencies. If the effects of curvature are negligible, the cylinder could be described in terms of an equivalent plate. The equivalent plate is a flat panel that has the same dimensions as the cylinder. The smaller \( f_r / f_c \), the smaller the effect of curvature near the critical frequency will become, and an equivalent-plate model could be used as an approximation (Manning (1964)).

Radiation efficiency was found to have a peak at the ring frequency (i.e. at a high efficiency) (Wang (2000a, b)). Above the ring frequency, curvature effects disappear and the shell
vibrates like an equivalent flat plate and the radiation efficiency falls, only to rise again as the frequency approaches the critical frequency (Fahy (1985)). The statistical approach is applicable as this type of shell has a high modal density. In acoustically thin shells, radiation is independent of length, and radiation efficiency is independent of the boundary conditions. The radiation efficiency for such types of shells depends on the acoustic behaviour of each vibration mode, therefore, geometry and boundary conditions of shell are significant. For instance, radiation differs with different lengths. Acoustically thick shells have low modal density. The radiation efficiency of acoustically thick shells reaches unity at high frequency, where elasto-acoustic coupling should be considered. Their acoustic behaviour is influenced by the nature and type of excitation. For example, road input and the sudden engagement of the clutch could produce different responses in the driveshaft tubes. A shell could be geometrically thin \((a >> h)\), but acoustically thick.

When the structural wave number is greater than the corresponding (at the same natural frequency) acoustic wave number, the mode is described as subsonic or an acoustically slow mode. The subsonic modes do not radiate sound efficiently. When the structural wave number is less than the corresponding acoustic wave number, the mode is said to be supersonic or acoustically fast. Supersonic modes have high radiation efficiency and are usually suspected for such problems as clonk. Below the critical frequency, subsonic and supersonic modes co-exist and the dominance of either would determine the overall efficiency (Wang (2000 a,b)).

In order for the radiation efficiency to be determined in the region below the critical frequency (Wang (2000 a,b)) noted that sub- and supersonic modes have to be determined first.

Now, if supersonic modes dominate, then modal efficiency is unity. Boundary conditions and excitations have to be considered as the modal-averaged efficiency depends on them. Changes in the former could change a supersonic mode into a subsonic one and vice versa. The latter determines the modal amplitude.

An index, \(\Delta L\), is defined as follows (Norton (1989) and Wang (2000 a,b)): \(\Delta L = k_s - k = \frac{\omega}{c} - \sqrt{(k_m^2 + k_s^2)}\) (3.51)
where: \( k_x \) is the acoustic wave number, \( k \) is the corresponding structural wave number at the same natural frequency, \( \omega \) is the natural frequency, \( k_m \) is longitudinal structural mode, and \( k_n \) is the circumferential structural mode.

If \( \Delta L \geq 0 \ (k_x \geq k) \), and the investigated mode is supersonic.
If \( \Delta L < 0 \ (k_x < k) \), and the investigated mode is subsonic.

Figure 3.9, Figure 3.10 and Figure 3.11 shows the separation between supersonic and subsonic modes for each tube of the three-piece drivetrain. The quantity: \( \Delta L(m,n) = k - k_x \), separates these modes, where, \( k \) and \( k_x \) are the acoustic and structural wave-numbers, respectively.

![Figure 3.9: Front tube subsonic and supersonic modes](image)

The results shown in Figure 3.9-3.13 are listed in *****. In this table all modal frequencies are indicated by wave number frequencies, with a identity number for those which are found to be supersonic. These numbers are also shown in the figures.
Chapter 3 Theoretical Formulation

The coincidence between the \((m,n)\) and \((p,q)\) modes can be found when \(n=p=l\), as defined by Wang (2000 a, b) for the definition of ring frequency. Therefore, for example, when \(l=2\), then \((m,2)\) structural waves for any value \(m\) can be plotted on the same graph as \((2,q)\) for acoustic waves. Then, where these line cross, a coincidence is noted. For the middle driveshaft tube with the simply supported closed ends, the coincidence is shown for structural modes \((m,1)\)

Figure 3.10: Middle tube subsonic and supersonic modes

Figure 3.11: Rear tube subsonic and supersonic modes
with all the acoustic modes \((1,q)\) (see Figure 3.12) and for \((m,2)\) and \((2,q)\) (see Figure 3.13). Theses structural modes, \((m, n)\) are obtained by the solution of a complicated cubic energy equation, as this described by Arnold and Warburton (1949).

The results for supersonic modes shown in Figure 3.9 – 3.13 are listed in table 3.1. All modes are indicated by wave number and frequencies.

![Figure 3.12: Coincidence of \((m, 1)\) structural tube modes and internal acoustic modes (centre driveshaft tube)](image-url)
Figure 3.13: Coincidence of \((m, 2)\) structural tube modes and internal acoustic modes: Centre driveshaft tube

<table>
<thead>
<tr>
<th></th>
<th>Wave Number (m)</th>
<th>Wave Number (n)</th>
<th>Frequency (\omega_{m,n}) (Hz)</th>
</tr>
</thead>
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<td>1st Tube</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>3</td>
<td>2205</td>
<td></td>
</tr>
<tr>
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<td>1</td>
<td>3210</td>
<td></td>
</tr>
<tr>
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<td>3333</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>4</td>
<td>4517</td>
<td></td>
</tr>
<tr>
<td>2nd Tube</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>2941</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>2</td>
<td>3930</td>
<td></td>
</tr>
<tr>
<td>3rd Tube</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>3</td>
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<td></td>
</tr>
<tr>
<td>1</td>
<td>4</td>
<td>4220</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.1: Modal frequencies and its wave number
4 MODELLING AND SIMULATION STUDIES

4.1 Introduction

The prototypes are expensive and time intensive periods within the development cycle of a product. Many vehicle manufacturers are reducing the number of hardware prototypes by virtual prototypes (i.e. the use of computer models to enable accurate representation of system behaviour). Computer models are simplified representations of real systems, usually embodying some assumptions. Therefore, virtual prototyping depends on the particular application and on the appropriate and justified use of adopted assumptions. Virtual prototyping simulations enable scenario building investigations at much reduced costs, when compared to actual prototypes. Alternative designs can be evaluated, and the necessary time to bring a new product to the market may be shortened by the elimination of many iterations of laboratory testing and prototype fabrication.

In particular, in complex investigation of vehicle NVH studies, there is a demand for analysis tools to be sufficiently detailed and allow the broadest spectrum of users to employ their capabilities effectively. Virtual prototyping tools are in the form of powerful computer-based software packages that provide modelling and simulation environments for their users. This means that the model input parameters should represent real physical data such as geometrical dimensions, masses, moments of inertia, stiffness and damping coefficients.

Sophisticated modelling environments are now available as virtual prototyping tools, such as MSC-ADAMS, DADS, SIMPACK, VAMPIRE, and others. The work highlighted in this thesis is a model of a vehicular drivetrain system developed in ADAMS, which is an acronym for Automatic Dynamic Analysis of Mechanical Systems (Ryan (1990), Ryan and Steigerwald (1988)). The modelling procedure in ADAMS is based on constrained Lagrangian dynamics and is outlined in the previous chapter.

Unlike linear system dynamics problems, for complex non-linear multi-body systems, where many interactions between the parts may take place (for example, in vehicular drivetrain...
systems) the number of factors affecting the design process may be quite significant. Hence, for such systems, it would not be cost effective to carry out "what if" scenarios on full size physical prototypes, since a large number of such prototypes might be required in order to obtain the desired goal. The reasons for using such powerful virtual prototyping tools described above have been highlighted by Ambrossi et al (1989).

4.2 Multi-Body Dynamic Model

Multi-body dynamics (MBD) is the physics of interaction of an assembly of rigid or flexible inertial bodies/parts, held together by some form of constraints or restraints like joints, couplers, gears, bearings, bushings. Furthermore, other forms of constraints arise from pre-specified motions that ensure inertial components in an assembly of parts follow a pre-defined type of motion. The overall behaviour of a multi-body system is, therefore, influenced by its individual inertial elements which may exert forces upon each other, and/or when exposed to external force excitation (Rahnejat, 1998). Since the combination of constraints, restraints, applied forces/torques, and the inertia of the parts, govern the overall motion and response of the multi-body system, it is necessary to devise a method of formulation and solution in order to obtain and understand the resulting dynamic behaviour. The methodology is based upon constrained Lagrangian dynamics, and the solution method is adopted for a mix of differential-algebraic equation set (DAE). The equations of motion are partial differentials, whereas the constraint functions are usually algebraic expressions.

The results obtained from multi-body dynamic analyses are usually in the form of displacements, velocities, accelerations and reaction forces/torques (Orlandea et al 1984). These time-domain outputs can be converted into frequency signals to obtain the spectra of individual vibration of parts in the system. Based on this, the designers can make modifications to multi-body mechanisms to guard against resonant conditions, when prevailing operating conditions may coincide with the fundamental natural frequencies of the overall system or its components. Furthermore, the reaction force output from the system allows for better component design in terms of reducing stress levels, thus increasing the life of parts under cyclic loading variations.
4.2.1 Description of the Drivetrain model

A multi-body dynamic model, built in ADAMS (MSC-ADAMS/View (2002)) (see Figure 4.2) is based upon constrained Lagrangian dynamics. This is the mechanical model of the drivetrain system. Component flexibility for the high modal density - radiated noise structures has been included through the use of finite element technique and component mode synthesis method, using NASTRAN (MSC-PATRAN/NASTRAN (2001)), so that an extension of this study will permit the calculation of radiated noise from the mechanical parts of interest, using either a suitable finite element approach or boundary element method.

The model comprises all the components of the drivetrain system, starting from the transmission input shaft up to and including the rear axle. The inertial properties of all the parts, the constraints introduced by their assembly (modelled with combination of idealised joints), the compliances-restraints of the model and the applied external forces are described in following sections.

4.2.2 Drivetrain modelling procedure in ADAMS

The following procedure is used to create an elasto-multi-body dynamics model of the drivetrain system:

- As the first step in modelling procedure the reference coordinate system and the correct units system should be defined. Then each part should be defined with the correct mass, centre of mass location, mass moment of inertia and the orientation. Each part possess certain number of markers including aforementioned properties and these markers move with the part. In every model there is a non-moving part possess with global coordinate system called “ground” and this can be regarded as the 3-D space MSC-ADAMS/View (2002).

- After defining all these parts with appropriate properties, the required motions and constraints will be introduced. Now the model is restricted to move in a way, which complies with these constraints and motions. The predefined joint primitives in ADAMS such as revolute, planar, cylindrical etc. use to reduce the number of DOF of the system accordingly. Motions in ADAMS such as translational, revolute etc. will be used with defined movements.
• Import the tube parasolid CAD file into NASTRAN: Creation of the tube FEA model using 2D shell elements (6 DOF for every node). This is justified, because the cylinder wall thickness is very small compared to the radius of the hollow tube. The ratio is 23:1 (0.0375/0.00165)

• Cylinder material properties (steel) are used:
  Young Modulus, $E=206 \, \text{GN/m}^2$
  Density, $\rho=7850 \, \text{kg/m}^3$
  Poisson ratio, $\nu=0.3$.

• Tube boundary conditions: Fully restrain the 3 orthogonal displacements of all nodes at the closing diameter. The imported cylinder tube is open-ended (see Figure 4.1).

• Set the grid mesh such that the number of elements along each normal axis is of the similar order for more accurate results.

• Set the number of nodes along the tube length such that the higher frequency modes and their shapes from 1000 to 5000Hz may be captured and defined by the modal analysis.

The exact procedure carried out is outlined on chapter 6 section 6.6.1.

Figure 4.1: The computational mesh density for a hollow tube
4.2.3 Degree of freedom in the drivetrain model

ADAMS uses Gruebler-Kutzbach equation to calculate the degrees of freedom associate with the multi-body drivetrain system. Interconnections between bodies are often complex. The DOF can be obtained, using the expression as:

\[
\text{Number of DOF} = \text{Flexible Bodies Modes} + 6 \times (\text{number of rigid parts} - 1) - \Sigma (\text{constraints})
\]

\[
= (142 + 102 + 122) + 6 \times (32 - 1) - 193
\]

Hence, the drivetrain model has 359 degrees of freedom. The choice of DOF is as described in Chapter 3.4.

The next step is to make sure that these degrees of freedom found by the model are realistic representation of the physical system. This is a crucial step in the modelling and analysis of drivetrain system.

The 5 degrees of freedom for the 3-piece rigid drivetrain model are as follows:

- Transmission input shaft and main shaft rotation
- Transmission counter shaft rotation (the layshaft)
- Rotation of the transmission output shaft, three-piece driveshaft assembly and the pinion
- Translational motion of the slip spline (i.e. the plunging motion of the drivetrain)
- Crown wheel rotation and all subsequent rotational parts to the wheels (i.e. the rear axle half-shafts).
Figure 4.2: Multi-body dynamic model of the vehicular drivetrain system in ADAMS
4.2.4 Parts description

Table 4.1 shown the list of 35 component parts of the drivetrain model, and their respective masses and moments of inertia.

4.2.5 Constraints

Table 4.2 shown the list of 35 component parts and constraint types in the drivetrain model.

It can be noted that the total number of constraints = 235

4.2.6 Compliances and restraints

A restraint resists movement by its compliance, i.e. it is not rigid.

Table 4.3 defines the 10 compliant component parts of the drivetrain model, and their respective stiffness and damping characteristics.

4.2.7 Applied forces and torques

A force or torque may be externally applied to a MBD system. It may also be considered as an internal force or torque arising from internal interactions.

Table 4.4 shown the list of 4 force and torque locations in the drivetrain model, both external and internal. Also shown is the magnitude and duration for each torque.
<table>
<thead>
<tr>
<th>Reference Number</th>
<th>Part Name/Description</th>
<th>Mass (kg)</th>
<th>Moments of Inertia (kg·m²)</th>
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<td>$I_{xx}$</td>
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<td>8.11E-04</td>
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<td>1.25E-03</td>
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<td>2.76E-03</td>
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<td>8</td>
<td>First driveshaft output flange</td>
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<td>5.27E-04</td>
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<tr>
<td>9</td>
<td>Flange yoke</td>
<td>0.85</td>
<td>7.58E-04</td>
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<tr>
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<td>Spider</td>
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<td>1.74E-03</td>
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<td>13</td>
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<td>0.97</td>
<td>5.27E-04</td>
</tr>
<tr>
<td>14</td>
<td>Flange yoke</td>
<td>0.85</td>
<td>7.58E-04</td>
</tr>
<tr>
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<td>Spider</td>
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<td>1.37E-04</td>
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<tr>
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Table 4.1: Parts in the 3-piece drivetrain model

All values stated here are provided by Ford Motor Company as typical data.
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Table 4.2: Constraints in the 3-piece drivetrain model

All values stated here are provided by Ford Motor Company as typical data. Part I and J are those connecting parts described in columns 2 and 3 joined in the manner as described in column 4.
### Table 4.3: Restraints and compliances in the drivetrain model

All manufacturing data are provided by Ford Motor Company

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<tr>
<th>Reference Number</th>
<th>Parts/Areas of Application</th>
<th>Characteristics</th>
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<td>Superelement - 142 Modes</td>
</tr>
<tr>
<td>2</td>
<td>Second Driveshaft tube</td>
<td>Superelement - 102 Modes</td>
</tr>
<tr>
<td>3</td>
<td>Third Driveshaft tube</td>
<td>Superelement - 122 Modes</td>
</tr>
<tr>
<td>4</td>
<td>Total Normal Backlash ~ fourth gear set</td>
<td>Experimental Data - 63μm</td>
</tr>
<tr>
<td>5</td>
<td>Total Normal Backlash ~ second gear set</td>
<td>Experimental Data - 75μm</td>
</tr>
<tr>
<td>6</td>
<td>First Driveshaft Tube Angle (around y axis)</td>
<td>Manufacturing Data - 4.5°</td>
</tr>
<tr>
<td>7</td>
<td>Second Driveshaft Tube Angle (around y axis)</td>
<td>Manufacturing Data - 4.9°</td>
</tr>
<tr>
<td>8</td>
<td>Third Driveshaft Tube Angle (around y axis)</td>
<td>Manufacturing Data - 2.8°</td>
</tr>
<tr>
<td>9</td>
<td>Third Driveshaft Tube Angle (around z axis)</td>
<td>Manufacturing Data - 1.5°</td>
</tr>
<tr>
<td>10</td>
<td>Axial and Torsional Coupling Stiffness</td>
<td>Experimental Data</td>
</tr>
</tbody>
</table>

Table 4.4: Internal and external applied forces

<table>
<thead>
<tr>
<th>Reference Number</th>
<th>Type</th>
<th>Position</th>
<th>Magnitude</th>
<th>Duration</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Ramp Torque</td>
<td>Transmission Input Shaft</td>
<td>145Nm built up during typical clutch engagement - disengagement (0.1-0.5s from lowest to highest point of clutch pedal)</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Fourth Gear Impact Force</td>
<td>Gear Meshing Cycle</td>
<td>Hertzian - Bending</td>
<td>Impact</td>
</tr>
<tr>
<td>3</td>
<td>Second Gear Impact Force</td>
<td>Gear Meshing Cycle</td>
<td>Hertzian - Bending</td>
<td>Impact</td>
</tr>
<tr>
<td>4</td>
<td>Bearing Force</td>
<td>Central Bearing</td>
<td>Hertzian</td>
<td></td>
</tr>
</tbody>
</table>

4.2.8 Assumption were made in the simulation studies

The following assumptions were made, when considering the impact of transmission helical gears in mesh:

- Backlash is defined as the circumferential free play on the pitch circle diameter.
• The simulation starts with the driving gear tooth spaced equidistant between the two opposite adjacent teeth.

• The model is set to increment in extremely small time (2μs -10μs) intervals through the contact condition, as the tooth contact moves from the heel to the toe.

• At each contact point, the circumferential movements of each gear are calculated and the resultant velocity is used to evaluate the local elastic deformation (once the lash had been taken up).

• The local contact force is evaluated from the deflection and stiffness values and used to calculate gear torque.

The following transmission gear mesh model assumptions are also made:

• No flexural deflection of the transmission main shaft and counter shaft takes place, whilst transmitting the torque. Hence no dynamic increase in backlash occurs due to shafts’ separation.

• No gear lateral tooth vibration takes place under impact, and no forced response results.

• Dry tooth contact is considered. Elastic contact is assumed under Hertzian conditions. This is justified in chapter 3.

• Only torsional loadings are considered.

• However, global tooth bending deflection under load has been included: the teeth are considered to be elastic cantilevers.
4.3 Numerical approach for the transmission Rattle

Rattle is as the result of impact dynamics of gear meshing teeth in transmission systems, usually under various loaded conditions. It fundamentally differs from clonk or thud conditions that are of a transient nature due to sudden surge or fade in torque demand (Merritt (1971), Menday et al (1999) and Vafaei et al (2001). However, both rattle and clonk have the lowest common denominator in the action of contact/impact forces through lubricated contacts. Some investigators have created rattle conditions by modelling the behaviour of unselected loose gears, and found reasonable qualitative agreement with observations from devised experimental rigs (Biermann and Hagerodt (1999)).

Various forms of rattle have been cited, owing to the mechanism of manifestation and operating conditions. These include idle rattle, with an engaged clutch and transmission in neutral with the engine at the idling rpm. These have been described in chapter 2. Idle rattle is very audible due to the lack of engine noise and is switched off when the clutch is disengaged. Idle rattle has become a quality issue in the customer perception of a vehicle. It is mistakenly perceived as tappet noise by the laymen, and considered as an unrefined engine response. Therefore, there is an impetus in the automobile industry to eliminate or mask the effect of gear rattle. Creep rattle occurs between 1200 – 2000 rpm, when in gear, and it is strongly related to the torsional modes of the drivetrain system. The over-run rattle takes place when the throttle off and coasting between 1500 - 4000 rpm. It occurs at low engine speeds and high loads and when the inertial torque exceeds the drag torque. It is thought that this mode of rattle would be sensitive to bulk lubricant properties such as viscosity. This is one of the solutions envisaged by many industrial investigators and their academic partners, such as Dogan et al (2004).

Some remedial actions are currently undertaken to palliate for these modes of rattle. These are: Clutch first stage is used for idle rattle, clutch in the second stage is used for creep-in-gear rattle, and the impact forces from non-torque transmitting teeth are transferred to neighbouring housing (Menday (2003) and Foellinger (2004)).

During the current experimental investigation (reported in chapters 6, it was noted that the conditions promoting clonk conditions include excitation of lower frequency phenomena, as expected. Rattle characteristics can, therefore, be present in the response spectra of noise
radiation. This gives credence to the interplay between rattle and clonk, and indicates that the study of one in isolation from the other is not prudent, particularly when root cause solutions or possible palliative measures are sought. Indeed it is long surmised in industry that some palliative actions against one of these phenomena may lead to the exacerbation of the other. Therefore, it was decided that a transmission rattle model should also be developed and simulations carried out. However, in this case the model need not be created in a code such as ADAMS, and more emphasis should be placed upon the impact conditions through a lubricated conjunction, as well including the frictional characteristics of conforming contact between loose gears and layshafts. This approach has not hitherto been reported in open literature, and forms the basis of work reported in this section.

4.3.1 Inertial Dynamics

The equations of motion for a loose gear mounted on an unselected layshaft, impacted by its conjugate mounted on the output shaft can be written as follows (see Figure 4.3):

\[ J\ddot{\theta}_{i,j} + F_{i,j}R_m = W_{i,j}R_m \]  
\[ J_i\ddot{\theta}_i = T_r - T_p - T_h \]

Note that for a loose gear, the applied torque is as a result of repetitive impacts between teeth of loose gear pairs. The contact load, \( W_{i,j} \), is obtained in terms of the relatively lightly loaded impact of the two bodies in mutual approach, in this case a pair of teeth, one being on the driven transmission output shaft and the other on the unselected third motion shaft, which rotates at the speed of the input shaft (see Figure 4.4). The loose gear on this shaft is ideally held in place by the tractive action of the hydrodynamic film. However, the gear can move relative to the shaft, since the tractive force introduces a friction torque, which is insufficient to guard against the small oscillations of the loose gear. The action of the hydrodynamic traction can be considered as the restoring compliance of the system at such low applied torques. This form of lightly loaded contact is further reinforced by low speeds of entraining action, which result at low engine speeds (such as in creeping rattle at 1200 – 2000 rpm). However, even at such low contact velocities (entraining and squeeze action), a lubricant film is formed between the contiguous bodies in contact. When the impact load is low, this leads to
the formation of a thin hydrodynamic film, in the absence of elastic deformation. At moderate to higher loads the elastic deformation of the contacting solids promotes lubricant film formation under elastohydrodynamic regime of lubrication. However, in a lubricated contact, as is the case in vehicle transmission system, the load is carried by the combined effect of lubricant layer and the elastic deformation of the contacting solids. The mechanism of lubrication, which supports the load is known as elastohydrodynamics. For loose gear rattle, the prevailing conditions at low loads are hydrodynamic.

Figure 4.3: Force in the loose gear

Figure 4.4: View of the transmission
Using equations (4.1) and (4.2), one can obtain the angular acceleration of the loose gear, $\dot{\phi}_{i,j}$, and that of the third motion shaft, $\dot{\phi}_s$, when the lubricant reaction and the restoring friction torque are known. Now the instantaneous angular velocity of the loose gear and its angle of twist can be obtained by step-by-step integration, using a time marching method, such as the Newmark's average acceleration method (Newmark (1959) and Timoshenko et al (1974)). This method is unconditionally stable for small values of $\Delta t$. Its accuracy deteriorates when the step size is increased, thereby the acceleration in such a time step increases significantly and the condition of the linearity of acceleration variation within a step is abrogated.

The following relations are from linear acceleration method:

\[
\begin{align*}
\phi_{i,j} &= \phi_{i-1} + \frac{1}{2}(\phi_{i,j} + \phi_{i-1})\Delta t \\
\varphi_{i,j} &= \varphi_{i-1} + \frac{1}{2}(2\phi_{i-1} + \phi_{i,j})\Delta t + \frac{1}{6}(2\dot{\phi}_{i-1} + \dot{\phi}_{i,j})\Delta t^2
\end{align*}
\] (4.3)

The contact reaction for a pair of impacting teeth for a loose set of gears can be obtained by the solution of the instantaneous contact conditions. First, the approach or separation of a pair of loose meshing teeth during the step size $\Delta t$ is obtained as (see Figure 4.5):

\[
h_{i,j} = C - (r_{in}\varphi_{i,j} - r_{out}\varphi_{out})
\] (4.4)

![Figure 4.5: Determining lubricant film thickness](image)
Chapter 4

Modelling and Simulation Studies

The term \( h_{i,j} \) is the film thickness, whilst the term in parenthesis is the mutual approach or separation of the impacting solids. To obtain the approach of the solids, it is necessary to calculate the angular motion of the output shaft, \( \varphi_{out} \). This is obtained as:

\[
\varphi_{out} = \dot{\varphi}_{out} \Delta t
\]  

(4.5)

where: \( \dot{\varphi}_{out} \) is the angular velocity of the transmission output shaft.

The iterative procedure within a time step of integration, \( i \), is indicated by the subscript, \( j \). This procedure must satisfy a suitable convergence criterion, before the time step of analysis is advanced. The convergence criterion is set on the variations in the gap as:

\[
|\varphi_{i,j} - \varphi_{i,j-1}| \leq 10^{-4}
\]  

(4.6)

Of course the output shaft is subjected to oscillations of angular velocity during throttle action, in addition to the perturbations in the combustion cycle. Rattle can be investigated in isolation from other drivetrain noise and vibration concerns, such as clonk (Menday et al (1999), and Biermann and Hagerodt (1999)) or whoop (Rahnejat et al (1997), and Kelly et al (1998)), if one is to assume steady-state conditions for the case of engaged inertias of the system and study the transient dynamics of loose impacting gears. Under such conditions, the angular motion of the third shaft (i.e. the layshaft) can be obtained in terms of the applied engine torque, the bearings friction torque and the rolling resistance torque.

The bearing friction torque is obtained as the hydrodynamic traction in crankshaft engine bearings and has the following form:

\[
T_b = T_{scm} + \tilde{T} \sin 2\alpha \xi
\]  

(4.7)

where \( T_{scm} = 57.5 \text{ Nm} \), \( \tilde{T} = 2.1 \text{ Nm} \), where: \( \omega = 40\pi \text{ rad/s} \). Note that the dominant contribution is at the second engine order for this case of 4-cylinder, in-line, 4-stroke diesel engine. The steady maximum engine torque, \( T_e = 145 \text{ Nm} \). Now referring back to equation
(4.4), the resistive torque due to longitudinal tyre force, as the rolling resistance is obtained as:

\[ T_r = F_t R = \mu F_t R_s, \text{ where: } F_t = \frac{1}{4} mg \]  

(4.8)

ϕ\text{out} is obtained by the gear ratio of the engaged gear in terms of ϕ\text{s} as:

\[ \dot{\phi}_{\text{out}} = G \dot{\phi}_s \]  

(4.9)

4.3.2 Lubricated Impact

The lubricated conjunction between the impacting teeth of loose gears is a lightly loaded counterformal contact (these are non-conforming contacts of contiguous solids), which may be regarded to be of an infinite line configuration (because the semi-half width of the contact is assumed much smaller than the length of the contact, extending along the width of the tooth flank). The relative motion of a pair of teeth in a loose gear is a combination of mutual approach or separation, inducing squeeze film action upon the intervening film of lubricant as well as the reciprocating entraining motion due to oscillation of the loose gear. As the output shaft rotates, repetitive impact is made between the mounted gear on this shaft and the conjugate loose gear on the unselected shaft. The hydrodynamic force operates in any lightly loaded conjunction that an approach is made between the contiguous surfaces. The force is contributed by the combined entraining and squeeze film action of the lubricant film as (Rahnejat (1985)):

\[ W_{i,j} = \frac{2Lu_{i,j} \eta r_s}{h_{i,j}} - \frac{3\pi L\eta r_s}{\sqrt{2h_{i,j}^{3/2}}} \frac{\partial h_{i,j}}{\partial t} \]  

(4.10)

This entraining motion is the component of the velocity vector tangential to the surface of the approaching solids, \( u_{i,j} \) (see Figure 4.6). There is also a velocity of normal approach or separation of impacting teeth, normal to the surface of the contact; \( w_{i,j} = \frac{\partial h_{i,j}}{\partial t} \) given by the rate of change of film in small computation steps \( \Delta t \). Note that the second term in equation
(4.9) operates for negative values of squeeze velocity (i.e. approaching surfaces). The contact of a pair of teeth is represented by an equivalent roller of radius $r_e$ against a flat, where:

$$\frac{1}{r_e} = \frac{1}{r_m} + \frac{1}{r_{out}}$$  \hspace{1cm} (4.11)

The entraining velocity of the lubricant in the contact domain is given as the average velocity of sliding surfaces as:

$$u_{i,j} = \frac{1}{2} [r_{out}\dot{\phi}_{out} + r_m\dot{\phi}_m]$$  \hspace{1cm} (4.12)

The second term is the same as the rate of change of the rigid squeeze action, which is given as: $w_{i,j} = \frac{\partial h_{i,j}}{\partial t}$. This is the same as the rate of change of equation (4.6).

![Figure 4.6: Surface speeds of contacting soild](image)

### 4.3.3 Restoring tractive torque

The oscillations of the loose gear are resisted by the tractive torque set up by the hydrodynamic film between it and the rotating unselected shaft, upon which it is mounted. The hydrodynamic force due to relative rolling motion in this conformal contact (i.e. the two bodies snuggle closely to each other) is given by a Petroff's bearing with a $\pi$ film as:
where the speed of entraining action here is given as:

\[
v_{i,j} = \frac{1}{2} \left[ (R_s + C_b) \dot{\phi}_m + \frac{1}{2\pi} R_s \dot{\phi}_s \right]
\]  

(4.14)

The restoring torque is, therefore, given as: \( F_{i,j} R_m \) as given in equation (4.1).

### 4.3.4 Numerical Procedure

The numerical procedure undertakes the following steps:

- Equations (4.1) and (4.2) are used to obtain the angular acceleration of the loose gear: \( \ddot{\phi}_{i,j} \), and the third motion shaft \( \dot{\phi}_s \).

- The angular velocities and displacements of the loose gear and the third motion shaft are calculated by successive integration, using a marching step-by-step integration algorithm, in this case the linear acceleration method, developed by Timoshenko et al (1974), based upon the Newmark (1959) \( \beta \) integrator (i.e. equations (4.3)).

- The speeds of entraining motion in all contacts are obtained using equations (4.12) and (4.14).

- The speed of normal approach is obtained as: \( \frac{\partial h_{i,j}}{\partial t} \).

- The lubricant film thickness is calculated from equation (4.4).

- The value of load \( W_{i,j} \) is calculated from equation (4.10).

- The restoring hydrodynamic force is obtained, using equation (4.13) and the radius of the unselected third motion shaft, supporting the loose gear.

- The solution is checked against the convergence criterion.

- If convergence is not found, then: \( j = j+1 \), and the procedure is repeated in the same step of time.

- If convergence is obtained, then the time counter is upgraded: \( t = t+1 \), and the simulation time is advanced by a step size \( \Delta t \).
Clearly, initial conditions are necessary to commence the numerical procedure.

4.3.5 Initial Conditions

The following initial conditions have been applied:

The initial angular velocity of the output shaft, when the transmission is engaged in second gear. This is determined by the engine torque and the gear ratio in second gear, and is given by equation (4.9).

The values for the angular velocity and displacement of the loose third gear at \( t=0 \) are considered as: \( \phi_0 = \phi_e = 0 \).

The initial value of angular acceleration at time step, \( i=1 \), is obtained from equation (4.1).

4.3.6 Results and Discussion

A simulation study was carried out for a light truck drivetrain system (the same as the one investigated for clonk conditions in this thesis), with a maximum engine torque of 145 Nm, when engaged in second gear with the transmission ratio of 2.08, with the differential ratio of 4.63. The loaded tyre radius is 345 mm in laden condition. The total inertia of the components is 1.3575 kgm\(^2\). The engine lubricant has a viscosity of 0.0345 Pas at the assumed bulk oil temperature and at atmospheric pressure. Due to low contact loads, thus low hydrodynamic pressures, the lubricant is considered as iso-viscous.

The simulation has to be carried out in very small integration step size due to short impact times, as well as the need to satisfy the linearity of acceleration variation required by the linear acceleration method within each step of time. The integration step size used was 1 \( \mu \)s. The simulation was carried out for a period of 0.1 seconds, or in other words for 100,000 time steps. Rattle is a transient event, due to the nature of the torque variation (see Figure 4.7). The simulation was carried out on a Pentium IV 1.8 GHz machine in the Linux environment, with a CPU time within a minute.
Figure 4.7: Applied torque variations

Figure 4.8: Angular velocity of third loose gear

Figure 4.8 shows the oscillatory behaviour of the angular velocity of the third loose gear, the mean value of which is the same as that of the engine. The oscillation values, although small are as the result of the impact hydrodynamic force and the fluid film traction, as described above. The value of $\varphi_{i,j}$ continues to increase in a clockwise sense, with similar small perturbations (i.e. there is no motion reversal of the loose gear). The corresponding hydrodynamic reaction force between pairs of impacting teeth is shown in Figure 4.9. It can be observed that as any pair of impacting teeth approach each other (indicated by the negative squeeze effect in Figure 4.10), the second term in the hydrodynamic restoring force (equation (4.7)) dominates. This decreases the lubricant film thickness, as shown in Figure 4.11.
Therefore, sharp rises, indicated by a position slopes in Figure 4.9 correspond to mutual approach of the teeth. The lubricant film approaches its minimum value. Subsequently, separation of bodies occur and the effect of squeeze film reduces, resulting in a corresponding decrease in the value of load. Each cycle of load; the initial sharp rise and subsequent fall in the contact load represents an impact of small duration. Note that in a lubricated impact, because of the existence of a film the contact is maintained at all times with a value of load which only increases due to sudden approach of the bodies.

![Figure 4.9: Hydrodynamic reaction](image)

![Figure 4.10: Squeeze film velocity](image)
This is shown by the existence of a finite minimum load value in Figure 4.9. The thinner the film thickness, the stiffer the contact stiffness becomes, thus the impact time gradually increases. This can be noted by differentiation of equation (4.10) with respect to film thickness to give an indication of lubricated contact stiffness non-linearity as: \[ \frac{\partial W}{\partial h} \propto \frac{1}{h^2}. \]

Referring to Figure 4.9 the severity of impact is reduced as the applied torque tends to drive condition. This means that the applied torque reaches near balance condition with the tractive restoring torque applied between the loose gear and the third motion shaft, represented in Figure 4.12, by the traction force.

Figure 4.13 shows the frequency spectrum of the oscillations of the impact load. The spectrum is obtained using FFT, which would only give a reasonable indication of the spectral contents of a transient signal. The fundamental frequency is at 40 Hz, with its higher multiples at 80, 120 and 160 Hz. These must be due to contact stiffness non-linearity of the lubricated conjunction, although one may initially suspect that the oscillatory behaviour of the bearing friction torque, comprising second engine order contribution at 40 Hz may be responsible for the introduction of a forcing frequency into system dynamics.

![Figure 4.11: Lubricant film thickness](image-url)
Various authors have proposed that rheology of transmission fluid may be engineered in a manner to attenuate the gear rattle problem by increased squeeze film action of the lubricant (Dogan et al (2004), Menday (2003)). However, Dareing and Johnson (1975) and Mehdigoli et al (1989) have shown that fluid film damping is very small for thin films of the order obtained in the current analysis. Therefore, in the current analysis the lubricant viscosity is increased to see the effect of rheology upon propensity to rattle.

In order to resolve this issue another simulation is carried out with an increase in lubricant viscosity of an order of magnitude at 0.345 Pas. This has the effect of increasing the lubricated contact stiffness by at least an order of magnitude, not accounting for increased stiffness due to reduced film variation. This can be seen in Figure 4.15, when comparing the oscillatory behaviour of the film with increased lubricant viscosity.
Figure 4.13: Spectrum of load for low viscosity

Figure 4.14: Spectrum of load for high viscosity
Figure 4.14 shows that the response frequency has shifted to the higher region of the spectrum, confirming the hypothesis that impact response in the lubricated conjunction is due to contact stiffness non-linearity, and not because of the applied forcing frequency. Referring back to Figure 4.15, one can observe that the period of oscillations with higher lubricant viscosity is about one third of that corresponding to the fundamental frequency of response for the case with lower viscosity lubricant.

One key observation of this analysis is that increased lubricant viscosity enhances the Petroff friction and lubricant reaction, this reducing the effect of impact in lubricated contact. This means that rattle conditions due to engine order effects are reduced, but still remain at the fundamental contact frequency, determined by the lubricant film stiffness (Gnanakumarr et al (2002)). This frequency remains in the spectrum of noise emitted from the bell housing, as noted in the experimental results of chapter 6.

 Whilst improvements can be seen in the case of loose gear dynamics, the increased lubricant viscosity can pose a serious problem in the engaged pairs by increasing the drag torque and reducing engine efficiency. This has almost intuitively (i.e. not through fundamental understanding) led the industry to introduce a secondary viscous damping effect by the introduction of dual mass flywheels. In these cases the primary inertia engages the secondary inertial via contact with a coil spring, which is attached to the latter. The coil spring sits in a
groove in the secondary inertial component and slides within it, with contact shoes separated from the groove by a film of grease (see Figure 4.16) (Littlefair (2004)). The grease introduces damping in the motion of the spring against the secondary inertia, and the spring may be tuned to reduce the main engine order response (second engine order for 4-stroke, 4-cylinder engines). However, the problems of shoe wear, grease degradation due to temperature and coil spring surge remain. A fundamental study of rattle, using a model as developed here, and incorporating a dual mass flywheel is long overdue, and is proposed as future work in this thesis.

Figure 4.16: Typical dual mass flywheel (after LuK (2002))
CHAPTER 5

5 EXPERIMENTAL METHODOLOGY

5.1 Introduction

In order to validate the drivetrain model, the simulated conditions using the model described in chapter 4 have been compared with experimental findings in chapter 6. Model validation is an important step in any numerical analysis work, as it imparts confidence to the users in employing the model for design of experiments that reduce the development time of future generations of vehicles.

Additionally, laboratory testing in Noise and Vibration is to provide a means of investigating one aspect of refinement at a time, eliminating other influential sources. Also, where possible, it is desirable that the facilities provide a basis for investigating individual transmission paths and design parameters within the drivetrain system.

This chapter mainly describes the design of a dynamic test rig in order to study the transient response of a vehicle drivetrain system. The experimental rig supports a Ford Transit Van’s drivetrain system, on which the clonk phenomenon has been reported. The main aims and objectives of this rig are:

- To gain a fundamental understanding of the physics of the concern, namely clonk
- To carry out detailed experimental investigation of the concern
- To study the structural responses of the system under impacting conditions
- To better understand drivetrain sensitivity to clonk and impulsive torque excitations
- To identify remedial actions in a fundamental, as opposed to a “fire-fighting” manner
- 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
- To use the experimental data to verify the analytical simulation
- To measure responses, which truly reflect the effects of parameter changes and not those of other peripheral factors

Furthermore, it is important that experimentation should be carried out in a repeatable manner and under controlled conditions. This gives the major advantage, where the behaviour of sub-
systems and components can be closely examined and at the same time compare the experimental results with analytical predictions. The piece-to-piece variations in manufacture and assembly of drivetrain components are unavoidable, particularly when dealing with tight tolerances and backlash in geared and splined components. These problems, and variations, in driver-to-driver behaviour makes vehicle testing rather subjective for fundamental physical study is concerned. However, rig-based studies also have some limitations, including imposed boundary conditions and to some extent the lack of "real world" noise factors. This makes rig-to-vehicle correlation studies a critical issue from the automotive manufacture's viewpoint, since the outcome of the rig-based studies should lead to practical solutions for customer concerns.

5.2 Design of a Dynamic Test Rig

To capture the transient behaviour of the clonk phenomenon, it is essential to introduce inertial dynamics of the drivetrain system under conditions that promote impact loading in the various lash zones. The experimental rig described in this chapter is, therefore, rather inadequate in the sense that the transient nature of the phenomenon is introduced without regard to the entire inertial dynamics of the drivetrain system, in terms of weight transfer in fore and aft shunting of the vehicle during shuffle. This section describes the many issues, which are encountered, when designing a dynamic test rig. They include provisions for alignment of the drivetrain system to closely replicate vehicle condition, as well as vibration isolation from environmental effects, and from those that are introduced by actuation mechanisms, other than those that are encountered under similar vehicle testing conditions. Method of actuation of the rig and imposition of impulse, such as throttle action or clutch activation is of paramount importance. Finally, proper provisions have to be made with regard to repeatability of the actuating signals, their conformance to driver or vehicle behaviour and the reproducibility of the actual variations. Figure 5.1 shows the design of the experimental rig.

The following sections deal with these issues in some detail, providing justification for use of given configurations.
Figure 5.1: The CAD drawing of the experimental rig
5.2.1 Dynamic Rig Design Issues

The operation of the experimental rig should be identical to the actual vehicle or achieved through number of components, whose specifications needed to be chosen carefully in order to replicate the actual clonk conditions.

5.2.2 Introducing Vehicle Inertia

The realistic replication of the conditions that lead to the clonk behaviour of the drivetrain system require the introduction of the vehicle inertial effects on the drivetrain. Thus, the following methodology has been developed in order to calculate the appropriate inertia that has to be introduced to the system combined with their right positioning.

The equations of motion for the body (\(\ddot{x}\)) of the van and the rear axle assembly (\(\dot{\phi}\)) respectively are as follows:

\[
M\ddot{x} = F \tag{5.1}
\]
\[
J\dot{\phi} = T - FR \tag{5.2}
\]

where \(M\) represents the mass of the vehicle body, \(F\) is the total friction force on the wheels of the rear axle, \(J\) is the total inertia of the rear axle assembly (shaft-wheels), \(T\) is the transmitted torque to the rear axle and \(R\) is the wheel radius. For pure rolling conditions:

\[
\ddot{x} = R\dot{\phi} \tag{5.3}
\]

By substituting (5.4) in (5.2) and the resulting equation in (5.3):

\[
J\dot{\phi} = T - MR\dot{\phi}R \tag{5.4}
\]

which finally becomes:

\[
(J + MR^2)\dot{\phi} = T \tag{5.5}
\]

This means that the body of the vehicle can be simulated as an equivalent amount of inertia of magnitude \(MR^2\) added to the rear wheel assembly.
In the case of Ford Transit (3500 Kg mass, approximately in laden condition, and wheel radius of 365mm) this inertia is calculated as 467 kgm\(^2\), so each half of the rear half-shafts should undertake about 233.5 kgm\(^2\). This is a considerable inertia, which requires an enormous mass mounted on each half of the rear axle.

To simulate these conditions, there are three possible ways:

1. The standard rear axle hydraulic drum brake mechanism on the vehicle can be used to represent the vehicle inertia. However, this needs some modification to the system that can affect the repeatability of the experiment. The main advantage is that such a mechanism provides the option to control the developed brake force and thus to simulate different conditions, e.g. an unladen vehicle, or a vehicle in different laden states.

2. The construction of a mechanism that includes pulleys and gear pairs in order to achieve a high final ratio and, thus, the required amount of inertia can be mounted sensible on each half of the rear axle. The advantage of this mechanism is the low cost, but the major disadvantage is that this requires extremely careful consideration from health and safety reasons, because of the high rotational speed of the mounted masses.

3. To locate portable dynamometers in wheels to introduce appropriate resistance.

Therefore, hydraulic drum brake was deemed as a good option, which is also cost effective.

### 5.2.3 Hydraulic Drum Brake

The Ford Transit axle was used in this experiment, which already has a standard drum brake system. There were two methods that could be used to apply the brake. One of them is to pull the hand-brake cable which would apply brake, but the disadvantage was seen as lack of repeatability. In other word, one could not ensure the exact force being applied. The second method was to apply pressure to the piston, which in turn applies the brake. This method is quite straight forward and reasonably controllable.
Hydraulic pump was selected to apply the pressure to the piston, which in turn pushes the brake shoes against the drum. This is shown in Figure 5.2. This introduces the resistance as in a real vehicle on the road. However, this simple implementation cannot include the transient effects of vehicle inertia, nor it is intended to when clutch clonk conditions are investigated.

![Figure 5.2: The hydraulic braking arrangement](image)

### 5.2.4 Method of Actuation

There were two alternative methods to actuate the rig: one by the electric motor and other by an internal combustion engine. There are some advantages and disadvantages between these. The advantages of the electric motor are: smooth output torque, easy installation and maintenance. One of the major disadvantages of an internal combustion engine was seen as the difficulty of its operation in the open space laboratory. Also the IC engine would transmit all the engine orders to the drivetrain system, which is very difficult to investigate in a specified area of the research.

Therefore the electric motor was seen as the best option. In particular, an appropriate coupling mechanism can be developed to inhibit the transmission of motor vibration to the drivetrain system. This is discussed later. The motor is also mounted on its own foundation, isolated from the structure upon which the powertrain system is mounted. It should also be noted that clonk occurs due to sudden surge or fade in power and is not an engine order
related NVH phenomenon. Thus, the use of an appropriate motor with adequate programmable rise rate is justified.

5.2.5 Electric motor selection and specifications

It was decided to use an electric motor, as this provides the opportunity to apply fully controllable torque conditions compared to those of an internal combustion engine. This motor must have the appropriate specifications in order to be capable of accelerating the drivetrain up to the test clonk condition speed of 1500 rpm. In this case—in the real vehicle—the maximum torque applied by the engine is 145 Nm at 1500 rpm (in 2nd gear), which corresponds to conditions of the transmission engaged in the second gear. Thus, the required power of the motor is calculated by using the following equation:

\[
P = T \times \omega
\]

\[P = 145 \text{ (Nm)} \times (1500 \times 2\pi/60) \text{ (rad/s)} = 22.78 \text{ kW}
\]

Based on these specifications, which provide the required operating condition for the experiment rotational speed and transmitted torque a three phase, four pole electric motor was chosen. Figure 5.3 and Table 5.1.

![Figure 5.3: Shows the chosen Electric Motor](image-url)
5.2.6 Motor Controller

The main power components of an inverter have to able to supply the required level of current and voltage in a form suitable for the motor use. The smoothness in the motor’s performance is achieved by using an inverter with appropriate characteristics like the IMO Jaguar VX 2200 –P-EN. The speed setting alteration on the inverter produces torque rise or fall rates. This is important in order to take up the lash in the system in a very short period analogous to the first swing in the cycles of shuffle, referred to as jerk, described in chapter 2. The chosen inverter and the complete controller unit are shown in Figure 5.4. The main features of the unit are:

- Low noise: reduces interference with devices such as sensors and load cells
- Quiet motor, when driving with higher carrier frequency settings. Noise radiated from the motor can affect the signal acquired by the microphones set in place to obtain clonk noise.
- Selectable control meter outputs (analogy/pulse changeover)
- On-line tuning to continuously check for variation of motor characteristics during running for high-precision speed control
- Various frequency setting methods
5.2.7 Coupling

Since the motor is directly coupled to the flywheel and through this connection transmits the required torque to the drivetrain, a suitable coupling unit had to be chosen that ensures the smooth torque transmissibility. This coupling must not allow torsional vibrations to be transmitted to the drivetrain and interfere with the data recorded and must also isolate the 50 Hz operational frequency of the motor as well as its multiples. The initial idea of using the crankshaft with its engine block as a means of torque transmission was abandoned, because it would create torsional loads that would result in the twist of the crankshaft.

The final method chosen was to use a simple flexible rubber coupling in-line between the motor and the input shaft of the gearbox. The rubber between the two halves of the coupling helps to dampen torsional vibration and to isolate the drivetrain from the operational frequencies of the electric motor. Table 5.2 shows the coupling unit specifications. One half of the flexible coupling is fixed to the motor output shaft, using key locking mechanism, whilst 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 over-loading conditions. The propeller shaft is mounted rigidly on a pair of deep groove ball bearings, discussed later.
Table 5.2: Shows the Coupling Unit Specifications

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Size (mm)</strong></td>
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</tr>
<tr>
<td><strong>Mass (kg)</strong></td>
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</tr>
<tr>
<td><strong>Inertia (kgm²)</strong></td>
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<td><strong>Dynamic Stiffness (Nm/°)</strong></td>
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<tr>
<td><strong>Nominal Torque (Nm)</strong></td>
<td>950</td>
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<td><strong>Maximum Misalignment (mm)</strong></td>
<td>Parallel – 0.4</td>
</tr>
<tr>
<td></td>
<td>Axial – 1.1</td>
</tr>
</tbody>
</table>

5.2.8 The Shear Pin

Due to the extreme test conditions, one of the major aspects is to protect the electric motor from any arising over-loading conditions. Therefore, a threaded shear pin is placed in the coupling, which connects the electric motor and the transmission (see Figure 5.5). The threaded shear pin shears off at the specified torque of 160 Nm and also acts as a mechanical locking device for vibration resistance. The shear pin is screwed through the coupling into the propeller shaft.

Figure 5.5: Shows the shear pin fitted in the coupling
5.2.9 Bearing Specifications

In order to support the propeller shaft and the weight of the coupling unit, it is necessary to use the bearings. Therefore two self-lube cast iron pillow block units with deep groove ball bearings were selected. Both bearing are mounted into the bedplate with motor and the transmission. The units are shown in Figure 5.6.

<table>
<thead>
<tr>
<th></th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ball rotational frequency</td>
<td>66.63</td>
</tr>
<tr>
<td>Ball rotational frequency inner ring</td>
<td>147.675</td>
</tr>
<tr>
<td>Ball rotational frequency outer ring</td>
<td>102.315</td>
</tr>
<tr>
<td>Cage pass frequency – inner ring</td>
<td>14.775</td>
</tr>
<tr>
<td>Cage pass frequency – outer ring</td>
<td>10.23</td>
</tr>
</tbody>
</table>

Table 5.3: The bearing frequencies

Another important issue was the alignment in the connection of the gearbox and the motor, which had to be achieved in the most simple and stable manner. For this purpose, it was decided that the gearbox and the block units had to be built together on a solid plate. The complete unit is shown in Figure 5.10.
5.2.10 Clutch actuation system

The standard vehicle hydraulic clutch system has been fitted into the test rig see Figure 5.7. The design of clutch actuation mechanism is based on the concept that the experiments must be characterised by repeatability under fully controllable conditions. This means that the controlling and altering clutch engagement-disengagement time is required, in addition to a recording device for this actuation time. The recorded action should also be introduced in the model in order to simulate the exact test conditions.

However, it was decided that manual actuation of clutch will be sufficient, if the pedal acceleration could be monitored by an accelerometer. The observation of which would indicate the degree of repeatability of the manual action. Therefore a Brüel & Kjaer accelerometer (type 4393) is mounted on the clutch pedal with output signal conditioned by a Nexus charge amplifier.

Figure 5.7: Hydraulic clutch Pedal Box
5.3 Installation Issues - Construction of the Rig

Figure 5.10 and Figure 5.11 shows a plan view of the experimental set up. In the following paragraphs a few details regarding installation and construction issues are mentioned.

The main supports of the rig are two 5m (3.5m + 1.5m) long I-Section beams of 210mm height, bolted to the laboratory floor by rawl-bolts. Elastomeric pads are interference fitted between the beams and the floor in order to isolate the rig and avoid transmission of vibration to the rig from the motor and other laboratory machinery through the floor (see Figure 5.9). Their surfaces have patented moulded recessed offset cells to allow flow of the elastomer when under load, while maintaining lateral stability. This design eliminates the shape factor usually associated with elastomeric materials and provides a positive grip to the rigs' supporting beams and floor under load. Two layers of Fabcel 25 pad (see Figure 5.8) are used (properties are listed in Table 5.4). These pads are resistant to most oils, water, steam and chemicals. The maximum working temperature limit for continuous exposure is 65°C.

![Fabric padded pad](image)

Figure 5.8: Shows the Fabcel elastomeric pad

<table>
<thead>
<tr>
<th>Fabcel 25 Pad</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shore “A” Durometer Hardness</td>
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<td>Damping C/Cc (avg.)</td>
</tr>
<tr>
<td>Density (Avg. value)</td>
</tr>
<tr>
<td>Set and Creep</td>
</tr>
<tr>
<td>Maximum Load</td>
</tr>
<tr>
<td>Thickness</td>
</tr>
</tbody>
</table>

Table 5.4: Mechanical Properties of Fabcel 25 Pad
There are 4 Box-Section (100x100x 10 mm wall thick), 1.37m long and 2 I-section (200x100mm) cross-beams, 1.37m long are bolted onto the two 5m long I-Section beams (210x135 mm). The box-section cross-beams support the motor and the transmission. The I-Section beams are support the driveshaft joints.

For better alignment of the motor-coupling-bearings-flywheel-clutch-gearbox assembly, the transmission bell housing and the coupling system have been fixed to a bed-plate, which in-turn has been mounted to the cross-beams through 6 standard Ford Transit engine mounts.

The Inverter is mounted separately and in isolation from the rig to a portable trolley with locked wheels.

The two centre journal bearings of the three-piece drivetrain are mounted on the cross-beams. More specifically, consideration has been given to the angles that these shafts assemble together and with the rear axle.

The rear axle is hung from the mounting brackets that are also mounted on the I-Section beams.

Figure 5.9: Shows the installation between floor and beam
Figure 5.10: Shows the construction detailed of the experimental rig
Figure 5.11: Shows the actual test rig included beside as photograph
5.4 Safety Considerations

Since there are many rotating parts in the test rig, safety is a paramount issue. Rigid metal caging has been installed around all the exposed rotating or moving parts, such as the coupling, driveshaft, and rear wheel assembly area. One of the disadvantages of the safety cages is their contribution to noise generation due to their low rigidity.

5.5 Operating procedure

1. Make sure that all the safety cages are in place and properly secured.
2. Make sure that all the electrical connections and emergency switches are visible and in working order.
3. Start the motor and build up the required speed having already selected the appropriate gear. This is achieved by developing an appropriate control programme in the IMO controller, providing the rise rate (i.e. ramp) and the required final angular velocity.
4. Apply the hand brake if required for replicating any given vehicle laden condition and at the same time disengage the clutch. This simulates a clonk condition referred to previously and termed clutch-induced clonk.
5. Release handbrake and reduce motor speed.
6. Engage clutch and restart building speed, then continue from step 3.

Prior to all tests, with either configuration, the background vibration and noise is monitored, using all the microphones and the laser doppler vibrometers. This provides a good measure of environmental comparison between the various test conditions. Spectra of these background vibration and noise are obtained. These also provide a good picture of vibration and noise generating sources that exist for all the experiments. The spectra contain contributions from out-of-balance vibration, bearing-induced vibrations, IMO controller noise, mains and its harmonics. Therefore, comparison of these with the spectra under actual testing conditions is useful in two ways. Firstly, they identify some of the contributory causes to spectral components, as already described in the previous section. Secondly, the agreement of these background spectra provide an assurance of maintenance of identical conditions for comparative tests carried out.
5.6 Instrumentations

There are number of instruments used in the experimental studies in order to gather various data. These are in two broad categories. The first category of instruments verify or 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 as inputs to the model for comparative simulation studies. The instruments recording input conditions include the accelerometer attached to the clutch pedal to monitor driver behaviour in clutch engagement/disengagement. Another actuation monitoring device is the IMO controller itself, which operates on closed feedback principle and provides information on the actual rise rate and the achieved final speed. This is taken to be the nominal speed of the flywheel. The last monitoring device is the cylinder pressure gauge, which gives the measure of brake force applied to the rear axle.

The second category of instrumentation deals with monitoring of rig dynamic conditions. This falls into three classes.

The first one deals with vibration transmitted to the vehicle cabin through the structure, being at the mount of the driveshaft bearings to the chassis. Two accelerometers are used for this purpose.

The second class of vibration measurement devices are non-contact Laser Doppler Vibrometers, which are appropriately positioned to measure high frequency structural vibration of the rotating driveshaft tubes.

The final class of monitoring devices are microphones, positioned in appropriate locations to measure sound pressure radiated from various structures, including driveshaft tubes, the bell housing and rear axle.

These are all described in following paragraphs.

5.6.1 Structure-borne transmitted vibrations

Accelerometers were used to measure the response of the structure during the actual test. There are four Brüel & Kjær accelerometers Type 4393, used in this experiment. Two of
them were mounted onto the clutch pedal, which measure the pedal acceleration and its velocity by direct integration (see Figure 5.14).

The third accelerometer was mounted on the centre bearing, between first driveshaft and second driveshaft, which measures the response of the driveshaft mount. The fourth accelerometer was mounted on the centre bearing between second driveshaft and third driveshaft, which also measures the response.

5.6.2 Non-contact vibration measurement

For measuring vibration on a rotating driveshaft, laser vibrometers are technically well suited. The laser vibrometer is an effective non-contact alternative to the use of a Eddy current probe or traditional contacting vibration transducer. The principle of the Laser Doppler Vibrometry (LDV) relies on the detection of a doppler shift in the frequency of coherent light scattered by a moving target, from which a time-resolved measurement of the target velocity is obtained (Bell and Rothberg (2000 a, 2000 b).

A pair of Polytec OFV4000 rotational vibrometers are used configured to measure the velocity of the lateral oscillations of the structures. These are two beam lasers, one of which is not required. Thus, (by reflecting one beam directly back into the instrument with a “single point adaptor” lens cap, the required configuration is obtained. There exists usually a problem with LDV measurement of members that rotate about an axis normal to the beam direction. This is caused by a fictitious velocity component due to the rigid body rotation of the shaft, being: \( v = \omega r \), where the \( \omega \) is the angular velocity of the driveshaft and \( r \) is its outer radius. This corrupts the lateral vibration of the shaft at the point of measurement, given by \( \dot{y}, y \) being the direction of the beam. Thus: \( v_y = \dot{y} \pm \omega r \). This is a known problem with LDV. However, it is not significant with high frequency measurements from a vibrating structure, which is rotating at a relatively low speed. The speed of rotation in the clonk experiments is 1500 rpm, equivalent to 25 Hz, which is considerably below the clonk frequencies: 1000-5000 Hz. Thus, the use of LDV is justified.

The two lasers are placed orthogonal to the surface of the first and the second driveshaft tubes. The distance to the target was set at 400mm.
Additionally, a Polytec OFV3000 Single point laser beam was set orthogonally to the axis of the third driveshaft tube. The principle of measurement is the same as those previously discussed.

The directed beams along the surface normal to any given cross-section of the tubes capture their transient behaviour during the passage of the clonk wave. Retro-reflecting tapes are adhered to surface of the driveshaft tubes in thin strips. This is to ensure that the incident beam is reflected with a minimum of dispersion. Although the system is operating and spectral information is obtained, mode shape reconstruction is being hotly debated as this type of use of laser under fast transient conditions has not hitherto been carried out from a rotating target, subject to high modulating modal density. Prior to setting up of the dynamic (transient) rig or the static (short impulsive) rig, it was envisaged that a scanning laser vibrometer would be able to obtain modal response of a structure in a single pass. This is indeed is the case for vibrating panels of car bodies subject to continues excitation. However, it should be noted that the passage or reflective wave behaviour in clonk traverses any of the 3 tubes in less than 5-10 ms, with the best of scanning systems requiring 50 ms to scan the same length, even without regard to pauses required for data collection. An important lesson is therefore learnt, that scanning vibrometry is unsuited to capture of modal behaviour caused by short-lived transient signals. However, one can keep LDV the beam stationary and observe the passage of the total modal behaviour during the clonk phenomenon. This, in fact, is all that is required in the capture and decomposition of the transient clonk signal.

The set up of the LDVs against the driveshaft tubes is shown in Figure 5.12.
5.6.3 Sound pressure measurements

Microphones are used to measure sound pressure level during the testing. Three microphones (Type 4155 with charge amplifier) are set up in correct orientation and distance to the intended target surfaces of the tubes (see Figure 5.13). The distance was 450mm away from the tube surface and 570mm from the laboratory floor, which has been calculated using equation (5.7).

\[ V = \lambda f \]  

(5.7)

where: \( f = \) lowest frequency (760Hz), this being assumed to be the lowest significant clonk frequency, \( V = \) velocity of sound (340 m/s), and \( \lambda = \) wavelength (m)
The microphone outputs are directly connected to the Brüel & Kjær PULSE (type 7109) data acquisition system.

The sound pressure recorded by a microphone can be used to obtain the noise level, using the following expression:

\[ L_p = 20 \log_{10} \left( \frac{P}{P_r} \right) \]  \hspace{1cm} (5.8)

Where \( P \) is the recorded pressure in Pascal, \( P_r \) is the reference pressure, given as:

\[ P_r = 20 \mu\text{Pa} \]  \hspace{1cm} (5.9)

\( L_p \) is the noise level in dBA.

Figure 5.14 shows a schematic of the positions of all the measuring instrumentation.

**Figure 5.13: Microphone set-up**
Figure 5.14: Shows all the instruments set-ups and were used in experiment
5.7 Data collection

The Brüel & Kjær PULSE (type 7109) ten channel data acquisition system has been used to collect the test data (see Figure 5.15). The system is connected to a Pentium IV 2 GHz laptop computer. Raw data were transferred into various data analysis software such as the Visual Fortran version 6.0 and AutoSignal version 1.7. Laser doppler vibrometers are configured to measure surface velocity. The signal from these is differentiated over very small time steps to obtain acceleration. It is important to select suitably small sampling time (in this case about 60 μs) in order to make sure that a good acceleration signal with minimum amount of noise results.

All the measurement devices were calibrated according to manufacturing specifications. These are given in Table 5.5.

Figure 5.15: Shows the data collection set-up
### Table 5.5: Calibration details for various measurement devices

<table>
<thead>
<tr>
<th>Channels</th>
<th>Hardware Set-up Sensitivity</th>
<th>Software Set-up Sensitivity</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1&lt;sup&gt;st&lt;/sup&gt; Shaft Laser</td>
<td>6000°/S/V 0.10°/V 10Hz-10kHz Tracking-FAST</td>
<td>2 V m/s 7Hz (high pass)</td>
</tr>
<tr>
<td>2&lt;sup&gt;nd&lt;/sup&gt; Shaft Laser</td>
<td>2000 mm/S/V 0.10°/V 10Hz-10kHz Tracking-FAST</td>
<td>2 V m/s 7Hz (high pass)</td>
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<td>3&lt;sup&gt;rd&lt;/sup&gt; Shaft Laser</td>
<td>1000 mm/S/V 5kHz Tracking-FAST</td>
<td>2 V m/s 7Hz (high pass)</td>
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<tr>
<td>1&lt;sup&gt;st&lt;/sup&gt; Bearing Accelerometer</td>
<td>100mV/m/s&lt;sup&gt;2&lt;/sup&gt; (Nexus)</td>
<td>1 V m/s&lt;sup&gt;2&lt;/sup&gt; DC (high pass)</td>
</tr>
<tr>
<td>2&lt;sup&gt;nd&lt;/sup&gt; Bearing Accelerometer</td>
<td>100mV/m/s&lt;sup&gt;2&lt;/sup&gt; (Nexus)</td>
<td>1 V m/s&lt;sup&gt;2&lt;/sup&gt; DC (high pass)</td>
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<td>1 V m/s&lt;sup&gt;2&lt;/sup&gt; DC (high pass)</td>
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<td>Pedal Accelerometer (Vel.)</td>
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<td>3&lt;sup&gt;rd&lt;/sup&gt; Shaft Microphone</td>
<td>49 mV/Pa 22.4 Hz (high pass)</td>
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</tr>
</tbody>
</table>

Test Date: 27 May 2004
Motor Speed: 1500 rpm
Solid Flywheel
5.8 Impact Testing

During the set up of various parts of the test rig, and after assembly it was deemed necessary to obtain the modal behaviour of all components and structures, with the aim of minimising the chance of resonant behaviour of these under impact conditions affecting the true representative characteristics of the drivetrain system in clonk. This is difficult to achieve, because the clonk conditions occur across a broad spectrum of frequencies (1000-5000 Hz). However, use of thick sections such as the supporting beams and isolation of the powertrain from the supporting structure, using Neoprine pads minimises the chance of the structure affecting the monitored signals. It is also important to isolate the measuring equipment from the vibrating structures. Thus, the LDV systems and the microphones were mounted on their own tripods and isolation supports.

Impact tests are used to determine the dynamic characteristics of mechanical structures. The test measures force as structural input (excitation) and acceleration as structural output (response) (Ewins (1994)). In general, the transfer function from the input force to the output acceleration is the mobility function. The mobility function in the impact test is given by the Frequency Response Function (FRF), which is termed: the accelerance or interance of the system, which is simply the ratio: acceleration/force. The accelerance function can be used to determine the resonances in the structure. A resonance is characterised by an associated resonance frequency, damping value and residue (scaling factor). These parameters can be derived from the measurement. An impact hammer is used for the excitation. In impact testing, the excitation is applied to the structure using a hammer, where the force transmitted to the structure is measured. The response of the structure is measured using an accelerometer. Excitation was fixed point and response was measured various point of the structure or components of drivetrain system. Table 5.6 shows the various modal frequencies of different structure obtained by impact hammer testing.
<table>
<thead>
<tr>
<th>Front Trans</th>
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<th>End Trans</th>
<th>1st Tube</th>
<th>2nd Tube</th>
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<th>Diff</th>
<th>Left Half</th>
<th>Right Half</th>
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<td>50</td>
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<td>84.38</td>
<td>338.3</td>
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Table 5.6: shows the various modal frequencies of different structure obtained by impact hammer testing
6 SIMULATION & EXPERIMENTAL RESULTS

6.1 Introduction

This chapter contains both the experimental results and the numerical predictions in this thesis, except the numerical predictions for rattle conditions, described in chapter 4.

The experimental results relate to the initial impact tests, as well as the main results for transient conditions with the recorded and analysed clonk phenomenon, obtained under various conditions and different rig configurations. These various conditions and rig configurations are used with the dual purpose of parametric studies and ascertaining the effectiveness of certain design features in the powertrain system against clonk conditions.

The numerical predictions consist of a number of simulation studies carried out to gain a fundamental understanding of the root causes of the clonk problem, in concert with the experimental observations, as well as those used under exact rig conditions for the purpose of validation.

6.2 The Experimental Procedure

With the experimental rig, clonk conditions could be obtained in a number of ways. Firstly, the IMO controller could be programmed to initiate motion of the motor in the anti-clockwise direction and at suitably low speed of revolution in order to take up all the lashes in the powertrain system in all the lash zones. This process was very similar to that reported by Menday (2003) with his static test rig, where a torque was applied through a low inertial disc brake to preload the system, by taking up all the system lashes. With the current rig, whilst the lash is taken up, at the same time resistance can also be applied to the rear axle by operating the hydraulic force (as described previously).

In the case of the static test rig, Menday (2003) released the preload with a trigger acting in a short period 80-180 ms. In the case of the current transient dynamic rig, the IMO controller
enables programming of torque reversal over a short period of time, and according to a certain torque rise rate, referred to as being equivalent to the first swing in cycles of shuffle response (i.e. jerk). The application of this action led to clonk response, with measured noise levels of maximum 85 dBA. Similar conditions reported by Menday (2003) in-situ in vehicle tests had indicated noise levels in the region of 105 dBA. Investigations carried out showed that owing to the rigidly mounted rear axle in the rig no weight transfer from rear to the front takes place, which would be expected in similar conditions, being analogous to throttle tip-in, which is accompanied by vehicle dive. Therefore, it was deemed that an alternative method should be used to acquire aggravated clonk conditions.

The alternative way of achieving clonk conditions is to replicate clutch clonk at low road speeds, in second gear, emulating the behaviour of an aggressive driver. The procedure for such a test with the current rig is to drive the system at a constant speed of 1500 rpm, with transmission in second gear and actuate the clutch abruptly within 100-300 ms. It should be noted that faster clutch actuation, whilst possible with the rig, is usually prohibited by end-stops in the clutch system. Furthermore, it may lead to engine stalling. In the case of the rig, excessively fast clutch actuation can lead to tripping control in the IMO. The procedure, thus described is known in the industry as side-slipping off the clutch and is regarded as unacceptable, but is known to be prevalent with impetuous van and light truck drivers. Such has become the culture of vehicle customers that all kinds of behaviour should be tolerated by the system, and any mal-function is regarded as a system fault.

The procedure highlighted above can be termed as clutch induced clonk, which has been a major source of concern in the automotive industry. Even with moderate clutch actuation speed, at low speed manoeuvres such as in car parks, even small saloons are prone to clonk conditions, particularly when negotiating small speed bumps. The realisation of this fact, and the intense competition in the small saloons’ market has made clonk a major quality issue, as recently highlighted by Richard Parry-Jones, Vice President of Ford Global Developments (2004). Therefore, clutch-induced clonk is, in fact, the most commercially relevant mode of investigation, and adopted with the developed rig.
Therefore, the final adopted procedure was to run the rig at 1500 rpm in second gear and actuate clutch repeatedly, at different actuation speeds. When this procedure was applied, it was noted that the clonk response correlated well with the actual road conditions.

A number of experiments were carried out with the adopted procedure. For all conditions reported here at least 16 nominally identical tests were carried out, and a representative result reported in this chapter. These included the use of different drivetrain configurations. Two configurations were essentially used. In one case a standard 3 piece-drivetrain system was set up with a solid mass flywheel (SMF). In the other case, the same standard 3 piece-drivetrain was used with a dual mass flywheel (DMF). The reason for the use of these different configurations was to ascertain the effect of DMF on the clonk response of the system. This has been a contentious issue in industry. DMF was developed as an alternative to clutch pre-dampers to attenuate the effect of engine order vibrations (discussed in chapter 2), which has been shown to be the case by, for example, the analysis in chapter 4 for rattle. However, not surprisingly clutch manufacturers, who are usually the developers of pre-dampers and DMFs as well, claim that DMF also contributes to the attenuation of clonk behaviour. The powertrain specialists in the automobile manufacturers, who are the end-users of this technology are devided as to the benefits of the DMF in clonk. Since clonk is not an engine order related problem, at least by definition and through observation of vehicle test data, it appears that any benefits accrued can be best regarded as coincidental. Testing of alternative configurations was deemed to shed a definitive light upon this issue.

The use of different rig configurations meant that provisions had to be made for quick disassembly and assembly of parts, particularly that the position of the flywheel alters according to its type.

During the initiation of the test, the driver behaviour is recorded by an accelerometer placed on the clutch pedal. Furthermore, the rig rotational input speed and any variations are recorded by the IMO controller. The background noise prior to the test is also recorded in order to filter it out of the test results at a later stage. Environmental noise, and that of the IMO and the motor are substantially below the clonk range frequencies, but can modulate with these and create uncertainties in observations. Many of the tests were carried out at night, where other machinery nearby were switched off, all lightings and air conditioning systems,
etc were also turned off. Most tests were carried out a number of times to ensure that the results are reproducible and statistically significant. Representative results of many tests are reported in this thesis.

Once the test procedure is carried out, as described above, resulting in clonk response, data is recorded by bearing mounted accelerometers, LDVs and microphones simultaneously. These are correlated with each other to ensure the integrity of tests carried out.

6.3 The Clonk Signal

The clonk signal can be obtained in a number of ways. Menday (1997) and (2003) attached miniature piezo-accelerometers to the driveshaft tubes. These ultra-light weight accelerometers suppose to not significantly alter the modal behaviour of the thin walled hollow tubes. However, with the continuing trend in the reduction of wall thickness (currently at around 1.65 mm), it is clear that addition of any mass can significantly affect the system response. Therefore, use of accelerometers is only justified at locations, such as the mounting bracket of center-piece bearing housings' of the driveshaft to the vehicle chassis. However, these locations are only suitable to obtain a measure of structure-borne vibration transmission to the passenger cabin. The signal at these locations is somewhat attenuated, and it is not representative of the severity of the conditions that lead to acoustic emission. In other words, the structure-borne transmitted effect to the cabin is not regarded as clonk, but as thud, which is heard by the occupants, when the windows of the vehicle are closed. When these are lowered, the faster wave motion on the tubes cause sound radiation, which bounces back off the kerb or parked vehicles at much higher dB, and is referred to as clonk. Nevertheless, due to low structural damping of the drivetrain, the frequency content of thud and clonk are very similar. Thus, accelerometers were place on bearing brackets as another form of the verification of the frequency content of the signal. Furthermore, bearing induced frequencies can also be monitored best by accelerometers positioned at these locations.

Figure 6.1 shows a typical clonk signal, obtained by a microphone positioned normal to the surface of the first driveshaft tube at its mid-span and at a distance of 45 cm. Note that the signal commences with a ramp up period of approximately 100 ms. Then a very short impulse is recorded of the duration 0.25-2 ms, corresponding to the accelerative noise, followed by a
long decaying period of 80-150 ms, corresponding to the ringing noise response. These characteristics are typical of clonk signals. In this case, the peak pressure is obtained as 2.91 Pa. Equation (5.8) can be used to obtain the noise level, yielding 103 dBA in this case. The measure noise level here is rather typical of annoying noises measured from vehicles on the road, such as 80-85 dBA reported by Hachenbroich (1994) for heavy trucks, even for low-noise transmission with engine encapsulation.

![Figure 6.1: A typical clonk signal](image)

The response in Figure 6.1 corresponds to the rig configuration, employing a single mass solid flywheel and an aggressive clutch actuation of a duration of approximately 80 ms, as shown by the pedal accelerometer recording, zoomed in Figure 6.2. The peak acceleration recorded is around 0.9 m/s². Note that the peak acceleration is at the instant of release. Prior to this the disengagement time shows driver action in 120 ms, which is representative of slide-slipping off the clutch pedal. The operating range for clutch actuation is 100-300 ms.
6.4 Physics of the clonk phenomenon

Noise spectra

The driver behaviour in clutch actuation, both in rate of disengagement and the manner of release changes the impulse input signal into the drivetrain. In other words, the rise rate of pedal acceleration, such as that shown in Figure 6.2 (in this case from 0.1 m/s² to 1 m/s²) and clutch pedal return determine the magnitude of the impulse as well as its energy content (the width of the impulse is proportional to energy content). Therefore, the larger the peak acceleration and the shorter the duration of the peak pulse, the worst the clonk conditions. Therefore, it is important for powertrain designers to make provisions to fool-proof the system against such clutch-induced impulsive actions, whilst ensuring no loss of clutch function. One may be to limit the travel, which may have adverse effects on clutch idealized functions. The other may be to introduce damping effects through use of viscous dampers or pre-dampers. On the other hand, these solutions will add to the inertia up-line of transmission, which has been regarded as excessive in comparison to those of the drivetrain system by some
authors (Dogan et al (2004)). One should remember that such detailed analysis were not carried out in the initial development of powertrain systems, which have largely evolved through piece-meal solutions, and the trend of using lower inertial hollow driveshaft tubes is a recent development, in response to the demands for improved fuel efficiency, without regard to NVH problems. Therefore, the behaviour of the broad spectrum of drivers must be fundamentally understood, before appropriate palliative actions are introduced.

Referring back to Figure 6.1, it is necessary to decompose the recorded signal in order to obtain its spectral content. This enables identification of spectral contents with high output power. When compared to those obtained by LDV for structural vibration, one can then pinpoint coincidence between structure and acoustic modes. One form of palliation would be to change physical or geometrical attributes of the driveshaft tubes or other potential resonating structures, such as the bell housing in order to avoid excitation of identified troublesome structural modes.

Fourier transformation of the signal requires a sampling rate that captures the highest spectral composition of interest. Furthermore, to avoid aliasing, one needs at least a sampling rate twice that of the highest frequency of interest, according to the Nyquist criterion. Since, the highest frequency of interest according to the observation of vehicle data is 5000 Hz, it is safe to gather 10,000 samples per second. However, it is prudent to take a larger record. Thus, a sampling rate of 16,384 samples per second was used. For Fourier analysis the number of samples should be an integer multiple of 2, as: $2^n$. For the chosen sampling rate: $n=14$.

Referring back to Figure 6.1, it is clear that the percentage of the samples related to the actual accelerative response (i.e. the impact time) represents less than 10% of the acquired record. The problem with Fourier transformation is its averaging nature. Thus, if an FFT or PSD analysis is carried out over the entire sample, the result will be unrepresentative of the actual frequency content of the signal. Therefore, it is necessary to window around areas of interest in the signal. This was reported by Vafaei et al (2001), who used Auto-Regression Moving Average (ARMA) method for signal processing. The advantage of ARMA over FFT or PSD is its windowing capability and the fact that unlike ARMA, Fourier analysis causes side-banding problems, as described by Vafaei et al (2001). Nevertheless, one can employ windowing with Fourier analysis, so long as a $2^n$ sample size is used with the appropriate
sample rate as described above. This action is termed Short Fast Fourier Transform (SFFT), leading to a spectrogram, which is employed in three regions of interest with the acquired clonk signals in this thesis.

The regions of interest are: the initial ramp up, prior to the second region, being the accelerative response, followed by the final region of exponential decay, where for hollow structures a ringing response results. This approach not only provides the power sources in the overall signal, but it also gives an indication of the timing of the each contribution, similar to wavelet analysis, which is also used in this thesis. The importance of combined time-frequency analysis becomes clear in attributing spectral contributions to given causes and effects, thus enabling possible palliative actions to be investigated. There are difficulties in exactly determining the region as indicated above to window around. This can lead to changes in the spectral composition obtained from the raw signal. This effect has been noted in the current analysis, but alters the results by a small percentage in the case of the higher clonk frequencies of interest (i.e. 1-5 kHz).

The high impulsive action part of the noise signal in Figure 6.1, monitored by a microphone from the mid-span of first driveshaft tube corresponds to the accelerative response under impulsive action. This takes place approximately at the time of 1.21 s for a duration of 40 ms. There are 1024 samples in this region, which enable spectral analysis to be carried out. Figure 6.3 shows this power spectrum in normalized dB, which is the contribution at each frequency relative to the maximum dB output. Therefore, the contributions nearest to the zero value constitute the most significant power sources in the spectrum. It is important to be able to identify the cause of as many of the spectral contributions as possible. This task is performed by the aid of a number of other independent analysis, including hammer impact testing results (reported in chapter 5), the numerical results (reported later in this chapter), and analytical calculations of bearing contributions, and the impact of lightly loaded loose gears (rattle results in chapter 4).

Wavelet decomposition of a signal provides combined time and frequency information, which is very useful to understand the effect of various contributions to the overall spectral characteristics in a transient event. The original idea for wavelet decomposition is now more than half a century old. However, only in recent times systematic applications have been
undertaken and the method has begun to be established Daubechies (1990) and Newland (1993).

![Fourier Frequency Spectrum](image)

**Figure 6.3: A spectogram of noise monitored from the 1st tube**

Wavelet analysis decomposes a given signal into its components, which are not necessarily simple harmonic functions, as is the case in the Fourier analysis. Wavelets can be functions of different possible families of orthogonal local basic functions (Newland (1993)). Wavelets allow the changing spectral composition of a non-stationary signal to be measured and presented in the form of a time-frequency map. The signal is broken down into a series of local basic functions called wavelets. Each wavelet is located at a different position on the time axis and is local in the sense that it decays to zero, when it is sufficiently far from its centre.

The time-frequency distribution obtained through Wavelet transform enables one to observe the contribution of different frequency components over the full spectrum from one instant to the next.
It is possible to decompose any arbitrary signal \( f(x) \) into its wavelet components. The approach is the same as for harmonic analysis except that, instead of breaking a signal down into harmonic functions of different frequencies, the signal is broken down into wavelets of different scale (different level) and different positions along the x-axis. A finite length known signal \( f(x) \), occurring over the interval \( 0 \leq x < 1 \) can be represented as a constant level (i.e. for all the wavelets of level less than zero) plus wavelets of all levels above zero. Thus:

\[
f(x) = a_0 + a_1 w(x) + a_2 w(2x) + a_3 (2x - 1) + a_4 w(4x) + a_5 w(4x - 1) + a_6 w(4x - 2) + a_7 w(4x - 3) + a_8 w(8x) + a_9 w(8x - 1) + a_{10} w(8x - 2) + \ldots \ldotsanumber{6.1}
\]

where: \( 0 \leq x < 1 \)

In this thesis the commercial code AutoSignal version 1.7 is used to carry out wavelet analysis of clonk signals. The code uses the Morlet wavelet function, which is a continuous wavelet transform. The Continuous Wavelet Transform (CWT) is used to decompose a signal into wavelets, small oscillations that are highly localized in time. This makes the Morlet wavelet particularly suitable for highly localised nature of the clonk transient signal, as viewed by an LDV. Whereas the Fourier transform decomposes a signal into infinite length sines and cosines, effectively losing all time-localization information, the CWT’s basis functions are scaled and shifted versions of the time-localized mother wavelet. The CWT is used to construct a time-frequency representation of a signal that offers very good time and frequency localisation.

Figure 6.4 shows the wavelet spectrum obtained for the same impact region of the clonk signal in Figure 6.1 as that used for the spectrogram of Figure 6.3. It is clear that the high frequency content of the signal occurs in the same 30-40 ms region as that obtained by the spectrogram of Figure 6.3. This conformance of the results of the wavelet analysis with the SFFT approach, justifies the Fourier analysis technique employed here.
Figure 6.4: A wavelet spectrum of noise monitored from the 1st tube

In the wavelet spectrum of Figure 6.4, the bright red and orange areas represent the high frequency content of the clonk signal. It can be observed that the main accelerative noise component takes place for a duration of approximately 5 ms.

It is important to identify the possible underlying causes for all the significant spectral contents. Here the spectrogram of Figure 6.3 is more useful, because it “quantifies” the contribution of the various spectral contents. Referring back to Figure 6.3, it can be observed that the main contributions are at: 129, 315, 563, 650, 800, 1631, 1700, 2325 and 3375 Hz. Most of these peaks have been identified. The band of frequencies in the range 1630-1700 Hz represent one of the modal responses of the front tube. Referring back to the normalised dB power spectrum of Figure 6.3, one can note that the highest contribution comes from this band of frequencies as dB Nom → 0. This has been corroborated by impact hammer test of the assembled structure at the mid-span of the front tube, providing significant spectral contents at 850, 1100, 1640, 2335-2550, 3100, 3350 and 3500 Hz (see Figure 6.5). Some of the other contributions in the spectrum of Figure 6.3 also agree with the obtained hammer tested modal responses, such 800, 2325 and 3375. Clearly, small measurement errors either in the microphone pick-up, spectrogram technique or hammer tests account for the slight differences obtained.
Figure 6.5: Spectrum of impact hammer test for the 1st tube

With a static test rig (i.e. one that was not driven), having a two-piece drivetrain, but used in earlier models of the same truck powertrain as being investigated here, Vafaei et al (2001) found spectrum of vibration response of the front tube (being almost identical to the one used in the current investigation) to contain the peak at 1700 Hz as the main power source, using ARMA for the analysis of the clonk signal. Their spectrum was somewhat cleaner than that obtained here, because windowing in ARMA eliminates the side-bands more effectively than a spectrogram. Another problem in the spectrogram of the clonk noise, obtained by a microphone is the nature of data recording with such a device. Microphones tend to acquire signals that both local as well as global in nature. Although care is taken by shielding the bell housing in the current experimentation, some global noise radiated from the surround structures also interact with those intended to be measured from a local structure, such as the first driveshaft tube in this case. Nevertheless, the agreement between the findings of Vafaei et al (2001) and the results obtained here is striking. They also found that other significant contributions at 3.3 and 3.9 KHz, both of which have been obtained here in such vicinity, either by impact hammer tests or in the spectrum of clonk conditions. However, they found these two frequencies as the main power sources unlike the findings in the current
investigation. The explanation for this difference is, however, quite simple, because the rear axle in their rig was rigidly mounted to bed-plates. This yielded a more rigid structure, which when subjected to an impulse responds more significantly at these higher modes.

The modal contributions found here also agree with the numerical results, reported later in this chapter, as well as with finite element modal analysis work carried out by Menday et al (1999), who found the front tube contributions at 1700, 2445 and 3257 Hz. The current numerical results (reported below), singles out the resonating breathing modes (these being efficient noise radiating modes) as: 867, 1830, 2250, 2454 and 3115 Hz. There are two important observations to be made. Firstly, these modes agree reasonably well with the current findings, although a certain amount of error clearly exists with regard to frequency disposition within the experimental and numerical spectra. Secondly, not all modal responses of a structure are efficient noise radiators, as described in chapter 3.

It is, therefore, essential to monitor the structure-borne vibration, using either mounted miniature accelerometers (as in the case of investigations reported by Vafaei et al (2001), Menday (2003) and Menday et al (1999)) or eddy current probes or laser doppler vibrometers (as in the current investigation). With a rotating rig used here the first option is not appealing, since slip rings should be used to mount the accelerometers upon, and these are prone to generation of spurious noise. The fast transient nature of the clonk signal also makes the use of eddy current probes not suitable. Thus, LDVs are used, as described in chapter 5.

Before proceeding to report the findings with the use of LDV, there are some other significant findings with the application of SFFT to the transient noise signal, which should be addressed here. With the windowing technique developed, it is possible to ascertain the transient nature of clonk generation from the ramp up region of the signal to the accelerative impact region (already described above) and onto the final decaying ringing noise region.
Figure 6.6: Spectrum of noise from the 1\textsuperscript{st} tube during the ramp up in the clonk signal

Figure 6.6 shows the spectrum of the clonk noise signal from the first driveshaft tube in the region prior to the impact (i.e. in the ramp up region). This shows that the spectral contributions in the lower frequency region dominate the spectrum. These are at 25, 126, 306, 496 and 600 Hz. Note that the contribution at 306Hz is almost the same as that observed in the previous spectrogram at 335 Hz, both of which are very close to the bending mode obtained by hammer test at 315 Hz. The contribution at 126 Hz was also observed in the previous spectrum, and is discussed later, whilst that at 25 Hz is the rigid body rotational speed of the transmission input shaft, which is heard, owing to the fact that perfect idealized alignment of the motor-propeller shaft-flywheel assembly is not possible in the real world.

The contributions at 600 Hz here, and at 650 Hz in the spectrogram of Figure 6.3 are gear meshing frequencies due to transmission error. They are obtained as the result of meshing of the gear mounted on the input shaft and its counterpart on the layshaft. The input shaft runs at 1500 rpm or 25 Hz, and has 25 teeth. Therefore, the frequency of meshing impacts is 625 Hz. This can also be obtained by alternatively noting that the layshaft rotates at 18.94 rev/sec, with 33 teeth, again resulting in the repetitive impact event at 625 Hz.
The other contribution in the spectrum of Figure 6.6 at 496 Hz is as the result of similar interactions between the meshing pair of the second gear. In this case the transmission output shaft rotates at 12 rev/sec, with 41 teeth, yielding a frequency of 492 Hz.

Wavelet analysis of the microphone signal for the same pre-impact region shows that all the spectral contributions of any significance occur below the frequency of 1000 Hz. Although some authors have reported clonk frequency range between 300-5000 Hz, the results here show that no sudden impact at high torque demand is necessary to excite frequencies up to 1000 Hz in light truck drivetrain systems. Therefore, the correct frequency range for clonk response should be defined as above a frequency of at least 1000 Hz, as has been noted in this thesis. All frequencies in the pre-impact region are related to rigid body out-of-balance motions and low to moderate repetitive impacts due to transmission errors, similar to those experienced under various forms of rattle, explain in chapters 2 and 4. These lower contributions in the spectrum of vibration are excited by the clutch engagement at any rate; fast or slow, since they are inherent properties of the steady state dynamics of the powertrain system. The clutch actuation merely transforms a translational input into a torsional impulse through the action of the clutch friction disc torsional springs. The speed of actuation merely affects the final sudden impact due to the surge of these springs, increasing or decreasing the velocity of impact of gear teeth pairs, thus the transfer of momentum into a corresponding impulse.

![Wavelet Spectrum](image)

**Figure 6.7:** A wavelet spectrum of noise during ramp-up from the 1st tube

Now, the as yet not explained contribution at 126-129 Hz region in spectra of Figure 6.3 and Figure 6.6 can be explained. This contribution can be as a result of a number of coincident
causes. Firstly, the back-to-back deep groove ball bearing arrangement used in coupling of the motor to the drivetrain via a propeller shaft have a complement of 10 balls each. The motor speed was chosen at 1500 rpm to represent the engine speed of the light truck, when the transmission is engaged in second gear. Therefore, 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 due to the effect of gravity in the lower part of bearings. This causes a defined and narrow loaded region in bearings, which is counteracted to a certain extent by interference fitting and preloading. However, as the complement of balls rotate the effective dynamic stiffness of the bearing goes through a cyclic variation (i.e. it repeats itself as the balls’ configuration repeat its shape (i.e. their relative disposition). This changing of 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 a 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 outer race, and is referred to as the ball-pass frequency (Rahnejat and Gohar (1985), Vafaei and Rahnejat (2002) and Lynagh et al (1999)). According to Lynagh et al (1999), 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 10×12.5=125 Hz.

This is of course one of the contributing factors at this frequency, this being the reason for its significant contribution to all the spectra of noise. The other main contributory factor is the loose gear impacts (i.e. rattle) at the same frequency as shown in chapter 4, where the engine order contributions were suppressed by the choice of appropriate transmission fluid viscosity, but the impact frequency due to lubricant hydrodynamic stiffness remains at 125 Hz. Therefore, the torsional oscillations at ball-pass frequency coincide with the loose gear rattle frequency, culminating in a significant contribution at 125 Hz.

It is important to verify the hypothesis that the main modal response of the first tube at 1630 Hz, present in the central region of the clonk signal is indeed due to high energy impact, thus
an accelerative response. To do so, a spectrogram of the decaying ringing region should not contain any significant contribution from this mode. The spectrogram for this region is shown in Figure 6.8, with the equivalent wavelet spectrum in Figure 6.9. Both these spectra confirm the hypothesis. It is interesting to note that the decaying ringing noise spectra only contain contributions below 1000 Hz, all of which have already been identified. It is this finding which gives credence to the suggestions that a dual mass flywheel, usually active at these lower frequencies may indeed have some role to play in the attenuation of ringing noise, post a clonk response.

![Fourier Frequency Spectrum](image)

**Figure 6.8:** Spectrum of noise from the 1st tube during the final decaying ringing in the clonk signal

![Wavelet Spectrum](image)

**Figure 6.9:** A wavelet spectrum of noise during decaying from the 1st tube
6.4.1 Vibration spectra

As already discussed in the previous section, it is important to ascertain which of the structural vibration modes of driveshaft tubes coincide with those obtained in the decomposition of the clonk noise signal, reported above. This ideally leads to the identification of efficient noise radiating modes, which are usually the breathing modes (as described in chapter 3). Since all the structural modes are as the result of geometry (radius to length ratio, wall thickness), material properties and method of assembly, then it is conceivable that alternative arrangements may be made to avoid either the excitation of these modes or eliminate them by redesigning actions. To achieve this aim, it is necessary to obtain the spectrum of vibration of structure-borne response by a fast non-contacting measurement system. In the case of investigations here, a series of LDVs were employed.

Figure 6.10 show the signal obtained by an LDV positioned, facing the mid-span of the first tube and orthogonal to the tube axis. This corresponds to the signal obtained by the microphone at the same position. Both signals are obtained simultaneously.
Figure 6.11 shows a spectrogram of the central region of the clonk signal obtained initially by an LDV, and differentiated in small time steps to obtain acceleration data. The main power sources in the spectrum are thus obtained. These are at 342, 855, 1632 and 2457 Hz. There are other higher frequencies with much lower amplitudes, simply because the impact energy is insufficient to excite these.

A number of observations can be made. Firstly, The sharp definition of the spectral components and absence of the usual side-bands show the success of the devised SFFT technique, which has yielded an uncluttered spectrum which is usually achieved by more complex ARMA technique. Secondly, all the significant peaks agree reasonably with those in the spectrum of noise in Figure 6.3, and correspond to the structural modal behaviour as obtained by the impact hammer tests (see Table 5.6). Finally, the absence of lower out-of-balance and rattle frequencies is noticeable, indicating that their effect is not local, but global, thus the reason for their presence in the noise spectrum, and not in the structure-borne spectrum here. There are, of course some spectral contributions which may not be present in all the equivalent spectra.

Figure 6.11: Spectrum of vibration monitored from the 1st tube
Figure 6.12: A wavelet spectrum of vibration monitored from the 1st tube

Figure 6.12 shows the corresponding wavelet spectrum to the spectrogram of Figure 6.11. Very good agreement is observed. Additionally, it is clear that the contributions of the main structural components reduce significantly after the instance of impact. This can be seen more clearly in spectrogram of the signal decaying region and the corresponding wavelet spectrum of that region in Figure 6.13 and Figure 6.14.

Figure 6.13: Spectrogram of decay signal region of the structure-borne response at the mid-span of the 1st tube
Figure 6.13 shows that all the previously mentioned spectral contributions still retain their presence in the spectrum, but except for the contribution at 838 Hz, the others, particularly the one at 1630 Hz exhibit an order of magnitude reduction. This confirms the supposition that the contribution at 1630 Hz is an accelerative response, whilst the other at 838 Hz is the main contribution to ringing noise, as it appears both in the spectra of noise and vibration and persists for all the record length of the signals, as indicated by the wavelet spectrum of Figure 6.14.

![Wavelet Spectrum](image)

**Figure 6.14: Wavelet spectrum of decaying structure-borne response of the 1\textsuperscript{st} tube**

The analysis carried thus far for the acquired signals from the first tube can also be applied to those obtained from other structures, such as the second and third driveshaft pieces, bell housing and driveshaft support bearings. The observations made for the results described above are equally applicable to the other mentioned structures. Thus, for the sake of brevity the results for the others are described only for specific features that deviate from those already explained above.

Figure 6.15 shows the spectra of both airborne noise and structure-borne vibration obtained for the second driveshaft tube's clonk response, when windowed in the region of high impulsive action. These were obtained simultaneously with those for the first tube, discussed above. The spectrogram of noise in Figure 6.15(a) shows contributions at: 126, 283, 565, 715, 963, 1107, the band: 1540-1660, 2381 and 2680 Hz. The explanation for the contribution at 126 Hz has already been provided. Those at 565, 963, 1107 and the band: 1540-1660 Hz are
modal behaviour of the tube, as identified by the impact hammer test as: 595, 954, 1145, 1605, 2310 and 2670 Hz.

(a) Spectrogram from 2\textsuperscript{nd} tube accelerative noise

(b) Structure-borne vibration from 2\textsuperscript{nd} tube
A higher modal response at 3410 Hz is also found by the impact hammer test, which is not observed in the spectrograms of Figure 6.15. Therefore, the spectral contributions show good agreement with the established structural modal responses of the tube, as well as their numerical predictions (reported later) at: 1034, 2294, 2456, 3060 and 3410 respectively. These are the predicted acoustically efficient breathing modes of this tube. The only main remaining contribution at 283 Hz is possibly due 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 12 rev/sec, and the cage speed of the bearings is 6 rev/s. A spectrogram of the accelerometer signal from the bearing housing shows a contribution at 274 Hz (see Figure 6.16). One should note that an error of 10-15% is likely in the SFFT method used here for the lower frequencies. The

Figure 6.15: Noise and vibration spectra for 2nd tube
amplitude contribution at this frequency is quite small, unlike the contribution at 125 Hz, since the loaded region in needle bearings is well-spread due to a large number of needles, thus the variable compliance effect is usually negligible. The main contributions in the spectrum of Figure 6.16(b) are those at 386 and 560 Hz, which are the structural modes of the first and second tubes. This means that the perceived *thud* noise in the cabin is at these frequencies.

![Figure 6.16: Response from the accelerometer on the 1st bearing housing](image)

Similar results have been obtained for the second bearing, as shown in Figure 6.17. The main components are at 85, 236, 274, 384, 499, 590, 1150 (a structural mode of the second tube, mentioned above) and 1210 Hz. The contribution at 1210 Hz has been found to be a bending mode of the third driveshaft tube by numerical modal analysis, but oddly not found in any of the experimental spectra, presumably because of its proximity to the 1150 Hz. Very close frequency contributions tend to merge together in Fourier analysis. The frequencies 234 and 274 Hz correspond to the variable compliance effect in the bearing as already described above, and those at 384 and 590 Hz are the structural modes of the second tube that account for the transmission of *thud* noise into the cabin in these light trucks. The contribution at 85 Hz is likely to be due to what is termed as “ball size” effect in ball bearings, and rarely seen with other types of bearings, such as the needle bearing here. This is found as the speed of rotation of a rolling element, which in effect might be slightly larger in dimension than the
others. It is found as: \( f_b = \frac{Df_i}{2d} \), where \( f_b \) is the rotational speed of the centre of each rolling element, \( D \) is the pitch circle diameter of the bearing, \( d \) is the diameter of the needles and \( f_i \) is the shaft speed. Thus, for this bearing: \( f_b = \frac{41.275 \times 12}{2 \times 3.175} = 78 \text{ Hz} \), which is very close to the spectral contribution at 85 Hz.

Figure 6.17: Response from the accelerometer on the 2nd bearing housing

The spectrum of vibration for the rear support bearing has more spectral contents than the first bearing. This is because, being situated near to the spline is subjected to forces generated here, 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 (i.e. the Cardan) joint here constitute greater reactions at the rear bearing, which has the spectral contents of both the adjacent driveshaft tubes, as shown in Figure 6.17.

The corresponding results for the third driveshaft tube are shown in Figure 6.18. Those shown in Figure 6.18(b) are the structural modes, obtained by an LDV positioned at the tube’s mid-span. The main power sources in this spectrum are at: 479, 873, 1795, 3217, 3673 Hz. The impact hammer test indicates the main contributions at: 863, 1169, 1782, 2679, 3218 and 3700 Hz (see Figure 6.19). It can be observed that higher the magnitude of the accelerance corresponds to the structural mode with a larger energy content. However, the energy imparted to the tube by the impulsive action of the rig, pertaining to the clonk signal only excites up to the mid-frequency range of the structural mode spectrum of the tube. This is the
reason for the absence of the higher modes in the spectrum obtained by the LDV. To obtain the higher spectral content, one would need to introduce a greater impulsive by clutch actuation, which is not practically possible as it would require actuation times below 100 ms, for which the IMO controller would trip and in a vehicle the engine would stall. However, as shown in Figure 2.6, Vafaei et al (2001) were able to introduce the necessary impulsive action, by the virtue of the fact that their rear axle was clamped, thus the impulsive action could not be dissipated through rotation of the rear axle half-shafts. This does not represent real world situation. The conditions required to replicate vehicle testing, where higher energy modes have been noticed more so than the current rig, but less prominently than that analysed by Vafaei et al (2001) correspond to the introduction of tyre-road resistance. A more complex experimental rig is required for this purpose, perhaps mounting the rear axle, including the road wheel on a rolling road chassis dynamometer. This represents one of the suggestions for future work, proposed in this thesis.

The numerical investigations, reported below, predict the efficient sound radiating modes of this tube to be at: 892, 1155, 2520, 2569 and 3522 Hz. These are in general agreement with the hammer test results and for the modes less than 3000 Hz with the findings of the experimental rig.

An important factor is to ascertain which structural mode under the testing condition may account for the radiated clonk noise. In other words, which structure-borne mode coincides (i.e. couples with) the acoustic modes of the tube. Figure 6.18(a) shows the main spectral contents for the clonk noise. It is immediately clear that the main power sources here are at 872 and 1794 Hz, the contribution of the latter being dominant. This frequency is also noted in both the hammer test and the LDV acquired spectra. Thus, it is clear that the accelerative nature of the clonk noise in the experiment is due to elasto-acoustic coupling at 1794 Hz. This is confirmed by the wavelet spectra of both noise and vibration (see Figure 6.18(c) and (d)). In the noise spectrum, the short- lived effect of 1794 Hz can be seen and completely ceases after 1.24 seconds. However, the effect of the structure mode at 872 Hz continues in both the spectra, indicating that this is a contributor to the ringing noise. Note that this mode was predicted as a breathing mode by numerical prediction. These findings indicate the power of combined numerical and experimental investigations in establishing physics of motion of observed phenomena.
(a) Spectrogram from 3\textsuperscript{rd} tube accelerative noise

(b) Structure-borne vibration from 3\textsuperscript{rd} tube
(c) Wavelet decomposition of clonk noise signal in the impact region of 3\textsuperscript{rd} tube

(d) Wavelet decomposition of the central region of clonk structure-borne signal of 3\textsuperscript{rd} tube

**Figure 6.18** Noise and vibration spectra for 3\textsuperscript{rd} tube

**Figure 6.19**: Spectrum of impact hammer test for the 3\textsuperscript{rd} driveshaft tube
As in the previous noise spectra for the front and middle tubes, there are lower frequencies, related to bearings and gear meshing. In the case of Figure 6.18(a), these are in 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 (i.e. 85-100 Hz). This region also has a contribution from gear meshing interactions in the differential. With the pinion speed of 12 Hz, and with 8 teeth, a 96 Hz contribution exists. With a number of teeth in mesh at any instant of time, multiples of this frequency also contribute to the spectrum, for example, at 192 Hz and at 288Hz, the latter of which falls in the noted region: 250-300 Hz. A key point of these discussions is that the lower noise spectra are dominated by speed dependent cyclic impacting phenomena of lower impact energy, such as rattle-type gear meshing problem, and bearing-induced vibrations.

### 6.5 Effect of Dual Mass Flywheel (DMF)

Dual mass flywheel is a device, which incorporates a conventional flywheel, referred to as the primary inertia, constituting the main inertial element. It also comprises a secondary member, which has a much lower inertia, being essentially a shell. A torsional spring conforms to the shape of this secondary member and seats inside it. The contact between this spring and the shell is effected via a number of mounted members, referred to as shoes. The cavity formed by the arrangement is filled with grease, often doped with metallic compounds, which improve grease stability at high generated temperatures. The primary inertia is attached to the spring and compresses it when subjected to torque and the arrangement resists torsional input. The spring stiffness is chosen to dampen the second engine order in 4-stroke engine, this being the main torsional vibratory signal. The drag introduced by the grease also plays a significant role (Littlefair (2004)). The arrangement is shown in Figure 4.16.

Since transmission rattle is due to the oscillation of loose gears, it is essentially excited by engine order vibration, particularly the idle or neutral rattle. For a 4-stroke, 4-cylinder engine with high output power, such as a diesel engine, the significant engine order contributions are the even order; 2nd, 4th, 6th and the 8th engine order, with the 2nd order being the dominant vibratory signal (Rahnejat (1998), and Gupta (2003)). Therefore, it has been shown that the dual mass flywheel attenuates transmission rattle. It also affects the other cyclic vibration
modes due to impact of gear teeth in meshing contact due to the presence of transmission error and backlash. These exhibit higher frequencies due to greater impulsive action, which may lead to exciting of lower structural modes of structural resonators such as the bell housing and the driveshaft tubes. These contributions have already been observed in the various spectra reported in the previous section.

One of the as yet not established effects of dual mass flywheel is its effectiveness in transient impulsive conditions that lead to clonk. It is expected that the normal velocity of impacting teeth should be attenuated by the action of any device on the driving inertia that introduces a form of torsional damping, such as a DMF. However, this effect may be rather limited due to the high spectral content of such sharp impulsive action. In other words, the DMF acts as a counter (i.e. a filter) against certain narrow band of frequencies, where as the clonk signal is effectively broad-band in nature (i.e. contains frequencies in a wide spectrum of response).

Most modern drivetrains in diesel engines now employ DMF, primarily as a safeguard against transmission rattle, with manufacturers of the device claiming partial effectiveness against the clonk phenomenon. The device is expensive (on average £200 per powertrain), which constitutes a significant cost in high volume models. Transmission rattle may be countered by other more cost effective measures, including backlash eliminators mounted on loose gears (see Figure 6.20 (a)) or shims introduced on gear flanks to minimise the effect of impact (Figure 6.20 (b)) or alternatively chamfering the sharp edges of teeth to reduce impact at teeth root or tip (Rivin (2000), and Fudala et al (1987)). These palliative measures are considerably more cost effective than the dual mass flywheel, which additionally has tribological problems of grease degradation and excessive shoe wear. Furthermore, sudden demands in torque variation can also lead to surge effect in the torsional spring which adversely affects the engine management system, which is triggered to response by flywheel oscillations. This recent problem has been noted with DMF (Menday (2004)). The effectiveness of DMF has been questioned by some in the automobile industry, and its retention in the future is seen as being dependent on its performance against clonk as well as rattle.
The experimental rig, initially configured with a solid mass flywheel (SMF), the tests results of which are described in the previous section, was reconfigured to accommodate a DMF instead of the SMF. The same experimental procedure, namely a clutch-induced clonk test was carried out. In order to obtain a fair comparison between the use of DMF and SMF all input conditions and environmental factors were monitored in order to make sure that identical conditions were achieved.

Prior to all tests, with either configuration, the background vibration and noise is monitored, using all the microphones and the laser doppler vibrometers. This provides a good measure of environmental comparison between the various test conditions. Spectra of these background vibration and noise are obtained. These also provide a good picture of vibration and noise generating sources that exist for all the experiments. The spectra contain contributions from out-of-balance vibration, bearing-induced vibrations, IMO controller noise, mains and its harmonics. Therefore, comparison of these with the spectra under actual testing conditions is useful in two ways. Firstly, they identify some of the contributory causes to spectral components, as already described in the previous section. Secondly, the agreement of these background spectra provide an assurance of maintenance of identical conditions for comparative tests carried out. Figure 6.21 shows a typical spectrum of the background vibration measured using an LDV, in this case for the rig configuration with a SMF. The low power level indicates that testing has been carried out under suitably controlled conditions.
Most of the contributions are related to the out-of-balance effect, which is clearly very low indeed, but the effect increases under impulsive action. The main contributions are at 24, 51, 75, 99, 150, 163, 299, 323 and 353 Hz. With the motor angular speed of 25 rev/sec, any small out-of-balance results in a contribution at 25 Hz, and its multiples: 50, 75, 100, 125, 150, etc. These are mostly noted in the spectrum. Interestingly, with steady state running of the engine, and no impulsive action gear meshing and transmission error frequencies at 496 and 625 Hz are not noted.

![Fourier Frequency Spectrum](image)

**Figure 6.21: Spectrum of background vibration measured by LDV from first tube under steady state condition**

Also noteworthy is that the background vibration from the IMO controller is at the mains frequency and its multiples, which in fact coincide with the out-of-balance contributions from the propeller shaft at 50, 100, 150 Hz, etc.

Environmental conditions, aside from noise of motor, IMO controller and other nearby machinery also play a role, particularly in the noise measurement by the microphones, such as the air temperature which slightly affects the speed of wave motion in air. Testing periods were scheduled in times that all environmental conditions: air temperature and humidity could
be considered as almost identical. In fact, all tests were carried out at the room temperature of 25°C and humidity level of 50%. The speed of sound in air under these conditions is 347 m/s.

6.5.1 The clonk signal with DMF

Figure 6.22 shows a clonk signal obtained from the first driveshaft tube of the experimental rig, using a DMF. The peak pressure at the instant of impact is 2.36 Pa, corresponding to the sound level of 101 dBA. This is 2 dBA lower than that with the SMF. Given the possible errors in measurement repeatability, many tests under similar conditions were carried out, and an improvement of 2-3 dBA was noted for the tests with a DMF when compared to those with the SMF.

![Figure 6.22: A typical clonk signal with DMF](image-url)
Figure 6.23: Typical clutch action

Figure 6.23 shows the clutch actuation signal for the test with the DMF, corresponding to the clonk signal obtained in Figure 6.22. The actuation time is similar to that in Figure 6.2 (i.e. in the range 100-120 ms).

The procedure both in the acquisition and the processing of the signals is the same as those already reported in the previous sections.

Figure 6.24 (a) shows the spectrogram of the power sources from the front driveshaft tube for the windowed region around the accelerative (i.e. the impacting region) of the clonk signal. The main spectral contributions are at: 137, 333, 372, 524, 865, 967, 1584, 1638, 2349, 2388, 2496 and 3328 Hz. This condition is comparable with the spectrogram of Figure 6.3. The contribution at 137 Hz is due to bearing-induced vibration, noted in Figure 6.3 as 129 Hz, with its actual analytical value being 125 Hz. The band 333-372 Hz is the bending structural mode, obtained as 315 Hz in Figure 6.3. The 524 and 865 Hz contributions are the main structural contribution of the ringing noise, noted with SMF at 520 and 863 Hz in Figure 6.3. The main accelerative noise contribution is in the band 1584-1638 Hz, also noted at 1631 Hz in Figure 6.3. The higher modes at 2349-2496 band and 3328 Hz were also observed with the
SMF configuration at 2325 and 3375 Hz respectively. Thus, the frequency composition of the signal is almost identical between the alternative contributions at the central part (i.e. the impact region) of the clonk signal. The only difference is the significant attenuation of the contribution due to meshing teeth at 625 Hz, not noted here. This can be significant, since the DMF suppose to suppress the cyclic dynamic effects, such as this type of contribution.

(a) Spectrogram from 1st tube accelerative noise
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(b) Structure-borne vibration from 1st tube

(c) Wavelet decomposition of clonk noise signal in the impact region of 1st tube

(d) Wavelet decomposition of the central region of clonk structure-borne signal of 1st tube

Figure 6.24: Noise and vibration spectra for 1st tube (DMF)
Figure 6.24 (b), (c) and (d) show the power spectrum of vibration and the corresponding wavelet spectra of noise and vibration signals respectively. The contributions at mid and higher range frequency (i.e. in 1000-2500 Hz) are similar to those noted for the SMF-based drivetrain arrangement shown and discussed in the previous sections. The significant observation from these graphs is the presence of a number of lower frequencies, not noted in the case of SMF. This, in the first instance, appears to be rather surprising particularly vis-à-vis the claims made by the DMF manufacturers that it attenuates or removes lower frequencies induced by the engine torsional vibrations (i.e. the engine orders). This is the reason behind the use of DMF as a palliative measure against transmission rattle, described in the previous chapters (see also Littlefair (2004)). However, the analysed noise signal in the previous section indicates at least 2-3 dBA attenuation of the overall noise level from the equivalent SMF-based drivetrain under same testing condition.

To obtain a clearer picture one should note the following:

1- The power spectra for noise and vibration based on the normalised dB for the alternative arrangements (i.e. with SMF and DMF) cannot be directly compared due to these being relative contribution of spectral components within the given spectrum itself. Thus, spectra that pertain to provide absolute rather than relative power contributions should be obtained to compare the results of the alternative powertrain configurations.

2- The wavelet spectra are also not directly comparable as the coloured contours are chosen automatically as the relative intensity of different spectral contributions within a given spectrum, making quantitative comparison between various spectra impossible. However, those for the DMF indicate dominance at lower frequencies, which persist for longer durations.

If the supposition that the lower frequency content in impulsive action are more significant with the DMF arrangement than the traditional SMF is held true, the implications for the continued use of DMF appears to be bleak. Precisely because of this it was found necessary that many tests should be carried out in order to either repudiate this hypothesis or confirm it, based on a significant number of trials and impulsive conditions to provide statistical
significance. For each condition, depending on clutch actuation speed 16 tests were carried out, and noise and vibration data were collected from all the driveshaft tubes and support bearings in each case, as well as power and wavelet spectra obtained for all of these. This was a major and lengthy undertaking. Below Figure 6.25 and Figure 6.26 show other representative results with the DMF arrangement. They also show the same trend as that described above (i.e. with DMF lower frequency content is much more active than the comparable spectra with SMF, all given in the previous section under identical conditions).

(a) Spectrogram from 2nd tube accelerative noise
(b) Structure-borne vibration from 2\textsuperscript{nd} tube

(c) Wavelet decomposition of clonk noise signal in the impact region of 2\textsuperscript{nd} tube

(d) Wavelet decomposition of the central region of clonk structure-borne signal of 2\textsuperscript{nd} tube

Figure 6.25: Noise and vibration spectra for 2\textsuperscript{nd} tube (DMF)
(a) Spectrogram from 3\textsuperscript{rd} tube accelerative noise

(b) Structure-borne vibration from 3\textsuperscript{rd} tube
To find a quantitative comparison between the two drivetrain configurations; with SMF and with DMF, one method is to evaluate the overall sound pressure level directly from the peak clonk signal. This has been carried out above, and it was noted that a modest gain of 2-3 dB is attained through the use of DMF. Considering that DMF installation has an additional cost of 200-300 Euros, on any perceived measure of cost effectiveness such as cost per dB attenuation no justification can be made, particularly that evidence in the field indicates insertion of sound absorbing media, such as foam filling of the tubes (Menday (1997)) or insertion of a piece of cardboard is almost as likely to achieve the same gain with almost negligible associated costs.

Pure perception of noise, often used as a subjective measure in industry shows preference for NVH characteristics with DMF-based configurations. In Ford, for example, a Vehicle Evaluation Rating (VER) system is used, where a harsh response is given a lower number
(such as 4 or 5) against a higher number for a refined system (such as 8 or 9). Test drivers or customer clinics are used to arrive at an overall preferred characteristics. However, with the methodology used here one can compare spectral contributions of the two alternative configurations, not only by perception, but in a more quantified manner. When in the vicinity of the test rig, under clonk condition, it is clear that the nature of noise is less metallic with the DMF. The higher frequencies promote a more severe metallic noise.

Figure 6.27 shows two comparable power spectra of vibration of first driveshaft tube obtained by laser doppler vibrometers for: (a) the SMF-based and (b) the DMF based alternative. Note that the spectrum for the case of the DMF configuration shows a 20% attenuation in the main structural mode around 945-970 Hz, with little change in amplitudes at the higher modes. The attenuation here appears to have transferred power to the higher frequency modes of vibration in Figure 6.27 (b). The reason for this is that the DMF absorbs some of the impulsive energy through viscous dissipation (drag) by the grease and some by combined viscous shearing and boundary lubrication between the shoes and the groove in the secondary inertial member. This means that less energy is transmitted to initiate impact in transmission system. Less impact energy means a greater proportion of vibration at lower frequencies. Since these lower frequencies are well below the efficient sound radiating breathing modes of the driveshaft tubes, the spectrum of noise with DMF has a lower spectral content in terms of frequency disposition. This can be seen by a comparison of Figure 6.28(a) for the SMF and 6.28 (b) for the DMF. It is noted that some attenuation has taken place in mid frequency contributions to the noise output and some at the higher frequencies, in expense of increased noise levels at the lower frequencies. However, as already noted above the overall sound level is only marginally changed, although the nature of sound is different because of the differences in the spectral dispositions. Thus, a more of a muffled clonk noise of a thud nature (analogous to gear teeth impacting pairs, as described by Merritt (1971)) is noted with the DMF drivetrains, whilst the higher frequency content with SMF results in more of a metallic noise. This explains why the subjective ratings by individuals of the clonk event favours the drivetrains with DMF, as human annoyance factor is greater for metallic noises.
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Figure 6.27: Vibration spectra for SMF and DMF 2nd tube
Figure 6.28: Noise spectra for SMF and DMF 2nd tube


6.6 Numerical Prediction

It is clear that numerical prediction with the model described in chapter 4 enables identification of efficient noise radiating modes of individual tubes. Furthermore, the driveshaft, as an assembly has its own overall modal behaviour, which may coincide with any of the efficient modes of the tubes, thus leading to clonk conditions. To ascertain under what operating conditions, such as impulsive action, these modes may become excited it is necessary to use an elasto-multi-body dynamic model as described in chapter 4, and introduce the impulsive action identical to those introduced in the experimentation for the dual purposes of: 1- validation of the model, and 2- to gain a more fundamental understanding of the underlying physics of the problem. It should be noted that the experimental results and their interpretation pertain to observation and surmising of likely causes and mechanisms which lead to an event, whilst numerical simulation is based upon modelling of established physics, such as inertial dynamics, structural vibration and sound propagation. Therefore, if the results of numerical predictions coincide with experimental findings one can then conclude that the physics contained within the model, implemented by the analyst indicates correct understanding of the phenomenon. This is the approach undertaken in this thesis. Furthermore, this eliminates the trial and error basis of investigation by putting it on a scientific basis.

The procedure adopted here can be stated as follows:

1- Finite element analysis is used to mesh the hollow driveshaft tubes and undertake modal analysis. This leads to the identification of many modal responses of the driveshaft pieces, and in particular the breathing modes of the tubes, which are efficient noise radiators.

2- These efficient modes such as the breathing and flexure responses are made up of interaction of lower modes, for example, some bending (deflection) and torsional harmonics. Therefore, the noise radiating modes and those contributing to their formations are selected out of a large number of mode shapes and included into a multi-body dynamic model of the drivetrain system. This procedure is known as mode reduction technique, based on synthesis of selected modes, thus the term component mode synthesis (CMS). Once these modes are imported to the overall drivetrain
model, impulsive actions at different levels can excite the overall drivetrain modal
behaviour, which constitutes the individual tubes' responses.

3- Now imposing impulsive action similar to those applied to the experimental rig, and
typical of driver behaviour, such as sudden clutch actuation or throttle tip-in and
throttle tip-out, the response of the drivetrain system can be predicted.

4- If this response leads to clonk conditions and agree with identically controlled
experiments, as described in the previous sections, then the fundamental nature of the
phenomenon in both cause and mechanism of transmission is established in a
deterministic and scientific manner. If not, the underlying physics embodied in the
model should be revisited.

6.6.1 Structural modes of driveshaft tubes using FEA

The mode shapes of the driveshaft tubes can of course be determined using the analytical
method, described in chapter 3. However, a larger number of harmonics \((m,n)\) have to be
employed to include their modal response with high frequency behaviour for clonk
conditions. When a large number of harmonics are required the problem is better tackled by
finite element modal analysis. In this approach the perfect cylindrical shells are discretised by
a mesh of shell elements of the thickness of the shell, in the case of different tubes by an
appropriate number of elements in the axial direction of the cylindrical generator and in the
circumferential direction. These were chosen as follows for each of the tubes in the 3-piece
driveshaft:

1- For the first tube: 41 X 14 elements (first in the axial direction, and second in the
circumferential).

2- For the second tube: 29 X 14 elements

3- For the third tube: 37 X 14 elements

Figure 6.29 shows a typical meshed cylindrical shell.
Thin cylindrical shells have a high number of modes, from low frequency rigid body motion to low frequency torsional or bending modes to higher combined torsion-deflection modes, known as flexure or breathing modes. The structural modes of some of the tubes obtained by finite element modal analysis are shown in Figure 6.30. These modes of driveshaft tubes are dependent on the imposed boundary conditions. The tubes have closed-ends and are considered fixed, as they are rigidly connected to universal joints or splined to the transmission output shaft for the first tube and to the pinion shaft for the case of third (or rear) tube.
6.6.2 Dynamic modes of drivetrain under impulsive action

Once the driveshift tubes are assembled within the multi-body environment, different form (i.e. ramp, dirac or triangular) impulsive functions with different durations excite various modes of the assembly (i.e. the drivetrain). These modes are a combination of excited modes of driveshift pieces, and contain higher or lower spectral contributions according to the severity of the impulsive action.

Various drivetrain modes are shown in Figure 6.31. It can be observed that for given impulsive action the mode shape of the drivetrain consists of excited modes of its three tubes. The lower frequencies, such as that shown in Figure 6.31(a)-(b) correspond to bending modes of driveshift pieces, and result in small angulation of the drivetrain universal joints. At some higher frequencies, for example at 1035 Hz, some of the efficient noise radiating breathing modes are excited, in this case corresponding to the middle tube. Note that this mode has already been observed experimentally at 970 Hz in Figure 6.15. This shows a small error of the order of 6.7% in prediction of this significant spectral content. The modal response of the drivetrain at 2545 Hz coincides with one of the main breathing modes of the first driveshift tube, already shown experimentally at 2457 Hz (see Figure 6.11, also found by Vafaei et al (2001) and Arrundale et al (1999)). This shows a prediction error of 0.12%, which indicates remarkable agreement between numerical predictions and the experimental findings. The next illustrated case in (e) is at a frequency of 3348 Hz, which has been found as a breathing mode.
experimentally at 3218 Hz (see Figure 6.18). The error of prediction is again quite small at 4%.

(a) at 302 Hz

(b) at 551 Hz

(c) at 1035 Hz

(d) at 2454 Hz

(e) at 3348 Hz

Figure 6.31: Some of the predicted mode shapes of the drivetrain under impulsive conditions
6.7 Comparison of the experimental findings and numerical prediction

In the numerical analysis a throttle tip-in condition is chosen as the impulsive action. This, in effect, applies the sudden torque as a ramp of given duration, from a zero value to a magnitude of 145 Nm in 120 ms. This condition is identical to the clutch actuation process used with the experimental rig. As in the case of the experiments, the transmission is engaged in second gear and the input shaft speed is chosen as 1500 rpm.

The section above has provided a means of comparison of drivetrain modal behaviour with the experimental findings, for particular conditions that lead to propagation of noise. The agreement is found to be very good. A more global picture of comparison between experiment and predictions can be obtained by undertaking spectral analysis of the numerically obtained clonk signal at the same locations from the driveshaft tubes as those monitored by the deployed LDVs.

With the numerical predictions, it is quite simple to increase the number of integration time steps as multiples of $2^n$ for Fourier analysis, which enables capturing of the higher frequency content of the spectrum. This necessitates larger number of samples (identical to those of simulations) to be analysed from the experimental data. The experimental results reported in the previous sections make use of record sizes of 1024 or 2048 samples for the impact region of the spectrum. In this section 4096 or 8192 samples are used. This means that the step size is between 12-24 μs. The results presented here are in fact in agreement with those reported in Figure 6.11, Figure 6.15 and Figure 6.18. However, better frequency resolution and more accurate amplitude may be assumed as the inherent averaging process with the Fourier analysis is carried out over a larger number of samples for the same duration of first swing in the signal (pertaining to the jerk response, as described previously).

For the case of the first driveshaft tube the enhanced experimental spectrum still contains the main contributions at 334, 850, 1631, 2200-2300, 2500 Hz, with lower contributions elsewhere (see Figure 6.32). The numerical results show good agreement with these in most
cases, indicating main power sources at 302, 750, 1000, 1750, 2200, 2800 and a band 3300-3728, which also exist at much lower amplitude in the experimental spectrum. The main source at 1631 Hz, has been noted at 1750 Hz, not only here, but also for similar tubes invariably between 1650-1800 by other research workers, including Vafaei et al (2001), Menday (1997, 2003). This in fact is the largest noted error of 7.2%. The model predicts higher amplitudes at frequencies above 3000 Hz, because it contains no structural damping from mounting structure, such as the beams and cross member, 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 exists with the rig. Another major mode described above is at 2457 Hz, predicted numerically at 2500 Hz, with very small error.
Figure 6.32: 1st shaft
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Figure 6.33: 2nd tube
Figure 6.34: 3rd tube
The same trend is also observed in Figure 6.33 and Figure 6.34 for the spectra related to second and third tubes respectively. In the case of the second tube an error of 15% is noted for the major power source at 934 Hz, previously discussed. This is seen at 1076 Hz. Finite element modal analysis predicts this mode at 1035 Hz. This indicates that the errors observed are due to signal processing using Fourier transformation, and are not significant. With the third tube, the mode at 3348 Hz in Figure 6.31 is shown at 3360 Hz in the numerical spectrum and at 3672 Hz in the experimental equivalent. The percentage error is quite small.

In conclusion good agreement has been found between the numerical predictions and the experimental findings, in particular with respect to the frequency content of the spectra. This agreement also extends to a lesser extent to the relative modal contributions within the corresponding spectra in terms of amplitudes. However, the agreement cannot be claimed for the actual amplitude contributions, because of a number of reasons. Firstly, the inherent averaging nature of Fourier analysis does not yield reliable amplitudes. Secondly, identical record lengths for given synchronised events cannot be ensured for numerical and experimental investigations. This is one of the main reasons for not obtaining quantitative comparisons.

As regards the mechanism of clonk behaviour, the impact-induced, wave propagation and elasto-acoustic coincidence hypothesis, embodied in the model is verified by the concordance of predictions with the experimental findings.
CHAPTER 7

7 CONCLUSIONS & FUTURE WORK

7.1 Overall Conclusions and Contributions to Knowledge

The thesis contains a number of major conclusions. These include the mechanism and means of propagation of the clonk phenomenon. It is shown that the mechanism responsible for clonk is any form of perturbation that induces impulsive action in the drivetrain system. It is usually in the form of sudden application of torsional impulse, which is usually introduced by throttle action or hasty clutch actuation. Both these actions introduce sudden variation in input torque to the system, and occur regularly in driving conditions. Therefore, driver behaviour can be a determining factor in the initiation of impulsive action. The propensity of the drivetrain system to clonk is dependent on the lash zones in the system, as well as the structural damping in the system. In general, the drivetrain system comprise many lash zones, such as meshing pairs in transmission, differential and the splinted joints, as well as low stiffness elements in the clutch system. Furthermore, the drivetrain system consists of structural members with low damping.

The first of these two features (i.e. backlash in the lash zone) have significant piece-to-piece manufacturing and assembling variations. This makes the severity of clonk to be significantly different between nominally identical vehicles. For example, the extent of backlash between the pinion and the ring gear in the diferential can range between 1-9 degrees. It is clear that with larger backlash, but the same impacting inertia, the energy would be significantly higher. Since this energy is hardly dissipated by low system damping, large number of structural modes are usually excited, some of which are efficient noise radiators. The thesis has identified all these major modes of drivetrain noise and vibration, both via numerical prediction and experimental investigation. These findings are in accord with the work reported by other investigators, such as Krenz (1985), Biermann and Hagerodt (1999), Menday (1997 and 2003).
Clonk, as described above, is a short-lived transient phenomenon, for which very few experimental investigations have been reported (for rear wheel drive vans and light trucks by Menday (1997), Vafaei et al (2001) and for front wheel drive saloons by Biermann and Hagerodt (1999) and Menday (2003)). In all these cases either in-situ vehicle tests or non-rotating (i.e. static) rigs have been used, except the case of the work reported by Biermann and Hagerodt (1999), which is more inclined to the investigation of rattle at lower frequencies. This thesis contains experimental work with a fully dynamic test rig with transient impulsive action, which is regarded as a novel approach in combined investigation of rattle and clonk and their interactions. One of the main contributions of the thesis is the interactions between rattle and clonk. These phenomena are usually regarded to be distinct and non-interactive. However, the experimental results show that both are different manifestations of impulsive action caused by a torsional input to the system. The existence of rattle frequencies, verified by the developed analytical technique in the measured decaying ringing noise spectra gives credence to interactions of the two phenomena.

Another source of contribution to knowledge, reported in this thesis, is the identification of specific spectral contents of noise and vibration signals in both accelerative and ringing responses of the system, using short FFT spectrograms. This approach is supplemented by use of wavelet decomposition to verify the transient nature of the accelerative response, as well as the decaying nature of the ringing response.

Additionally, the experimental rig was designed and developed in order to investigate the effect of introducing additional damping into the drivetrain system, by the inclusion of a dual mass flywheel to ascertain its effectiveness. This device has been used increasing in industry to attenuate transmission rattle, but its effectiveness against clonk response of the system remained unclear. The thesis shows that its effect is marginal in the case of the ringing noise content, and not significant in the case accelerative response. This constitutes one of the major finds of thesis with significant commercial implication.

Unlike other rig and vehicle based investigations of the clonk phenomenon an assortment of instrumentation was deployed in this investigation to monitor noise and vibration signals simultaneously, both for structure-borne transmission to the cabin, known as thud and airborne noise as clonk. For the latter, sound pressure levels were obtained at the same time as
structural wave propagation in the driveshaft tubes, using non-contact Laser Doppler Vibrometry. This is the first time that LDV technique has been used for monitoring of high frequency structural vibration of rotating thin driveshaft tubes. Coincidence was observed between certain structural and acoustic modes of these structures, which were further verified by the numerical simulations.

A detailed multi-physics numerical approach is developed and reported in the thesis, comprising constrained Lagrangian inertial dynamics of the drivetrain assembly, structural modal vibration of thin hollow elastic tubes and impact dynamics of meshing pairs. This approach represents a methodological improvement in drivetrain modelling through inclusion of stiff characteristics of such non-linear systems, which are not often included in many simplified linearised models, reported in literature such as those by Biermann and Hagerodt (1999) and Dogan et al. (2004). A stiff system is defined as one, where under a given condition there exists a or a number of active eigen values with other modal responses remaining dormant, until a sudden change such as an impact excites them, thus changing the modal response of the system dramatically over the transient period. Linearised models fail to adequately capture such a transient behaviour, thus not suitable for studies of the clonk phenomenon.

The combined experimental and numerical approach in the study of the clonk phenomenon with detailed conformance between observation and model outcome has not hitherto been reported in literature, and is regarded as one of the main contributions of the thesis to the advancement of knowledge.

7.2 Achievement of Aims and Critical Analysis of Approach

The main aims of the investigation are broadly highlighted in chapter 1. These included the following:

- To gain a fundamental understanding of the sources and mechanisms of impact loading transmission to light weight structural components of the drivetrain system.
• A fundamental investigation of mechanisms and paths for structural-acoustic coupling.
• Develop state-of-the-art CAE tools for virtual prototype testing and parametric studies of drivetrain high frequency transient dynamic phenomena.

Chapters 3-5 describe the methodologies employed to achieve the above objectives, whilst chapter 6 outlines the results of the investigation. Specifically, the requirements of the first two objectives were met, with the following conclusions:

• The mechanisms responsible for clonk are any forms of perturbation that induce impulsive action in the drivetrain system. This is usually an impact in the various lash zones, including: transmission, differential, clutch system and all the splined connections.
• The method of actuation, inducing clonk is a torsional impulse, usually caused by driver action in the form of throttle action or sudden clutch actuation.
• Means of transmission of the impact energy is usually of a localised nature, in many cases in line with the Hertzian impact theory or its variants. These conditions are observed at lower levels of impact and do not lead to propagation beyond local fields, such as in idle rattle. Higher impact velocities or component inertia can lead to sufficient energy in impact zones to cause structural modal response of components, leading to wave propagation in line with St Venant-Von Helmholtz theories.
• Materials/Structural factors play an important role in the propagation paths of noise and vibration. Low structural damping and lower stiffness characteristics can lead to larger modal composition of the response. Note that short duration impact signals contain frequencies up to 5 KHz. Thin hollow structures have many modes within this range. These include the driveshaft tubes and the bell housing.

Furthermore, the multi-physics methodology was devised to create elastodynamic models of drivetrain systems, based on the constrained Lagrangian multi-body dynamics with the inclusion of modal behaviour of flexible components. In addition, an impact model, based on lash characteristics of impacting pairs was developed and included in the overall model. The resulting model and the overall methodology was verified by experimentation. The validated model can now be used for protracted examination of system behaviour, not only for existing drivetrain configurations, but also for projected developments due to its generic nature. This
approach, often referred to as virtual prototype testing is very cost effective in new vehicle development programmes, as it reduces the time to market and potential fire-fighting of the NVH problems thereafter. In fact, this approach has already been deployed in the new Transit powertrain programme.

As in all other investigations one could have set other aims of scientific or technical importance with the benefit of hindsight. A number of enhancing features could have been included in the current investigation. On the numerical side, the current methods include the use of finite element analysis and component mode synthesis to represent component flexibility into the multi-body model of the drivetrain system. This approach is quite suitable for combined rigid body and structural dynamics, but precludes the determination of acoustic modes of the thin hollow structures. These were obtained using the analytical model described in chapter 3 or by experimentation as reported in chapter 6. Alternatively, boundary element method can be used to obtain both structural and the acoustic modes.

Additionally, the impact models are for dry meshing solid pairs, except for the case of investigation of rattle reported in chapter 4. Lubricated contacts tend to increase the impact time, as well as alter the nature of deformation (see for example Dowson and Wang (1994) and Al-Samieh and Rahnejat (2002)). Although this effect has been found to be not very significant for small backlash, but its inclusion will improve the accuracy of predictions. However, the effort is required warrants a specific research programme of its own.

Another main achievement of the work has been the development of a substantial test rig, with a modular and flexible architecture, allowing the use of various drivetrain configurations and types of flywheel. The rig can be used for physical prototyping of drivetrain systems with concept components, such as the DMF, used in this thesis. It can also accommodate other newly developed flywheel systems, such the pendulum type devices. Additionally, various clutch systems can be deployed, such as those with pre-dampers or various clutch spring characteristics with various lash rates. Menday (2003) has investigated these features to a certain extent. With commercial viability of palliation in mind for large volume manufacture and tight profit margins, one aspect of investigation not carried out in this thesis, is foam filling or insertion of sound absorption material such as card-board pieces into the driveshaft tubes. Such attempts can form part extension of the current work in the future.
7.3 Suggestions for Future Work

The above highlighted omissions from the current investigation, mostly due to the limited availability of time, form the basis for suggested extensions to the work reported here. These can be sub-divided into two main areas of numerical analysis and experimental investigations.

The mechanisms of actuation and propagation of the clonk phenomenon is now well established due to the work reported here and those of Menday (2003). What remains to be performed is enhancement of the numerical model to include within it prediction of sound pressure level around and inside the hollow thin shell cavities in a single analysis. This constitutes the extension of what one might regard as the existing elasto-multi-body dynamic model to an elasto-acoustic/multi-body model.

On the experimentation side, the existing rig may be used to test various driveshaft tubes with changes to their material of construction, geometric alterations or insertion of sound absorbing media. These may be regarded as alternative palliation methods, with varying degrees of success. For instance altering the geometry of the tubes will change their modal behaviour, and the clonk type response may become more significant at difference spectral contributions. Filling in the tubes with industrial foams can be hazardous, as some of which have already been banned under Health and safety Regulations. Foam filling can lead to attenuation of some of the spectral modal responses and not all, as found by Arrundale (1998). Partial filling is more cost effective, and it is claimed that if it is carried out in certain locations has the most benefit in absorption of acoustic waves. These claims and others in industry, and chiefly by component suppliers do not seem to be based on any fundamental study, similar to the one carried out in this thesis. Insertion of card-board pieces into the drive shaft tubes is yet another recently claimed palliation method. One would expect that some attenuation of the clonk response would take place with such an arrangement, but equally certain undesired effects may also ensue. One such undesired effect would be increased inertia of the tubes, as they are quite light and any significant card-board thickness can add significantly to their inertial properties. This has the adverse effect of increased rigid body inertial dynamics, which is mainly due to unavoidable imbalances in assembly and alignment of driveshaft
pieces. Therefore, increased mechanical losses will decrease drivetrain efficiency. One should note that hollow driveshaft pieces were progressively introduced in the past two decades in order to reduce the unbalanced problems with solid shafts of earlier constructions.

These palliation methods can form the basis of future investigations in a fundamental manner. This would involve the determination of absorption coefficients and inclusion of these into boundary element analysis. In general, a root-cause solution may be found in modification of impacting geometries, which would required in-depth contact mechanics analysis, a fundamental study, worthy of a PhD study of its own. Such studies have been attempted in the case of transmission rattle to a certain extent, for example, by Rivin (2000).
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References


Conclusions & Future Work
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APPENDIX

PUBLICATIONS


