PHD

Application of hydro-elastomer technology to vehicle suspensions

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Avon Vibration Management Systems specialises in the development and production of elastomeric and hydroelastic engine mountings (hydramounts and hydrabushes), and elastomeric suspension bushes. There is increasing interest in the development and use of hydroelastic suspension components with complex interlinked powertrain / suspension mounting systems.

The primary objective of this work is to improve the secondary ride (4 - 20 Hz) region of passenger vehicle response by means of passive wheel suspension development. The human body is most sensitive in this frequency range, especially 4 to 10 Hz, due to the resonance of internal organs. Little work has been done to fundamentally improve vehicle performance in this field, though primary ride has been studied extensively. This investigation involves the modification of parts and the development of new components for ride improvement.

This work includes an investigation of the current state of the art, analysis of the merits and shortcomings of existing systems and components, and development of a hydroelastic strut mounting and complete suspension unit concept. A computer model, written specifically for secondary ride analysis, is used to investigate the potential of these and other ride improvement schemes; these predictions are compared with measurements taken from practical trials for a conventional vehicle and one fitted with the hydroelastic strut mounting both in component form and as complete systems.
A strut mounting component concept with frequency dependent damping is developed successfully, using the results of simulation and practical trials to explain the function, develop a design methodology and determine design parameters and limitations. The benefits of the system are demonstrated, showing reduction of secondary ride vibration.

A complete suspension system replacement, based upon the same technology, is investigated. Simulation results are used to show the potential ride benefit of such a system, but also reveal an inherent problem with the control of the body roll mode.

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### Nomenclature

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<th>Symbol</th>
<th>Variable</th>
<th>Units</th>
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<tr>
<td>Ac</td>
<td>Cross sectional area of the hydramount channel</td>
<td>m²</td>
</tr>
<tr>
<td>Accc</td>
<td>Acceleration of fluid in the channel</td>
<td>m/s²</td>
</tr>
<tr>
<td>Ae</td>
<td>Effective Area of Hydramount working chamber</td>
<td>m²</td>
</tr>
<tr>
<td>areap</td>
<td>Working area of the damper piston</td>
<td>m²</td>
</tr>
<tr>
<td>areavalve</td>
<td>Plan area of the damper valve head</td>
<td>m²</td>
</tr>
<tr>
<td>areaV</td>
<td>Cross sectional area of the valve opening</td>
<td>m²</td>
</tr>
<tr>
<td>C_crit</td>
<td>Critical Damping</td>
<td>Ns/m</td>
</tr>
<tr>
<td>Cd</td>
<td>Sum of Discharge coefficients for channel flow path</td>
<td>-</td>
</tr>
<tr>
<td>c</td>
<td>Reference road roughness constant dependent on w</td>
<td></td>
</tr>
<tr>
<td>circvalve</td>
<td>Circumference of the damper valve head</td>
<td>m</td>
</tr>
<tr>
<td>E</td>
<td>Modulus of material</td>
<td>N/m²</td>
</tr>
<tr>
<td>e</td>
<td>Deflection of component</td>
<td>-</td>
</tr>
<tr>
<td>F</td>
<td>Force</td>
<td>N</td>
</tr>
<tr>
<td>F₀</td>
<td>Force at point of reversal</td>
<td>N</td>
</tr>
<tr>
<td>F_damper</td>
<td>Force generated by the damper</td>
<td>N</td>
</tr>
<tr>
<td>F_fluid</td>
<td>Force generated as a result of the fluid pressure acting on the working chamber walls</td>
<td>N</td>
</tr>
<tr>
<td>F_n</td>
<td>Natural Frequency of wheel hop mode</td>
<td>Hz</td>
</tr>
<tr>
<td>F_spring</td>
<td>Force Due to spring</td>
<td>N</td>
</tr>
<tr>
<td>f(X)</td>
<td>Value of spring force from interpolation of load / deflection data points</td>
<td>N</td>
</tr>
<tr>
<td>K</td>
<td>Stiffness parameter of rubber model</td>
<td>N/m</td>
</tr>
<tr>
<td>K₁</td>
<td>&quot;stiffness&quot; parameter of rubber model dependent on n</td>
<td></td>
</tr>
<tr>
<td>K₂</td>
<td>Increase in stiffness</td>
<td>N/m</td>
</tr>
<tr>
<td>K_d</td>
<td>High frequency dynamic stiffness of the hydramount</td>
<td>N/m</td>
</tr>
</tbody>
</table>
Kdiaph  Volumetric stiffness of the in-channel diaphragm
(where present) \( N/m^5 \)

Kp  Puffing (volumetric) stiffness of the working chamber \( N/m^5 \)

Kp(vol)  Value of pressure in working chamber due to interpolation of pressure/volume data points \( N/m^2 \)

Ks  Static stiffness of the hydramount \( N/m \)

Kt  Stiffness of Tyre \( N/m \)

Kv  "Stiffness" parameter of rubber model dependent on \( p \)

kd  Dynamic stiffness of damper and damper bush \( N/m \)

ks  Stiffness of the suspension spring \( N/m \)

Lc  Length of the hydramount channel \( m \)

Me  Effective mass of the hydramount channel - the mass required for an equivalent dynamic absorber \( Kg \)

Mw  Unsprung Mass \( Kg \)

m  mass \( Kg \)

n  Index representing rubber hysteresis -

P1  Pressures acting on ends 1 and 2

P2  of the channel \( N/m^2 \)

Pchamber  Pressure generated in the working chamber of the hydramount \( N/m^2 \)

Pdamp  Pressure generated by the damper valve \( N/m^2 \)

Pdiaph  Pressure generated by the in-channel diaphragm (where present) \( N/m^2 \)

Phyd  Pressure loss due to hydrodynamic effects \( N/m^2 \)

Pnet  Net fluid pressure acting on the fluid in the channel \( N/m^2 \)

Pvisc  Pressure loss induced in channel by viscosity \( N/m^2 \)

p  Index representing rubber hysteresis -

pcomp  Pressure generated by the 'compression' valves \( N/m^2 \)

pext  Pressure generated by the 'extension' valves \( N/m^2 \)

psd  Power Spectral Density of road profile \( m^3/cycle \)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rhyd</td>
<td>Hydraulic radius of the channel</td>
<td>m</td>
</tr>
<tr>
<td>stiffspr</td>
<td>Volumetric stiffness of the damper valve</td>
<td>N/m^5</td>
</tr>
<tr>
<td>t</td>
<td>time</td>
<td>s</td>
</tr>
<tr>
<td>Vchan</td>
<td>Mean velocity of flow in channel</td>
<td>m/s</td>
</tr>
<tr>
<td>Vole</td>
<td>Additional volume into working chamber due to the sum of the channel flows</td>
<td>m^3</td>
</tr>
<tr>
<td>vdamp</td>
<td>Relative velocity across the damper</td>
<td>m/s</td>
</tr>
<tr>
<td>velflow</td>
<td>Velocity of flow through the damper valve</td>
<td>m/s</td>
</tr>
<tr>
<td>vol</td>
<td>Total change in volume of working chamber, absorbed by volumetric compliance</td>
<td>m^3</td>
</tr>
<tr>
<td>volchan</td>
<td>Volumetric displacement of the fluid in the channel</td>
<td>m^3</td>
</tr>
<tr>
<td>volspr</td>
<td>Volume of fluid accepted by opening valve</td>
<td>m^3</td>
</tr>
<tr>
<td>Wn</td>
<td>Resonant Frequency of the hydramount channel</td>
<td>rads/s</td>
</tr>
<tr>
<td>w</td>
<td>Index representing wavelength dependency of road roughness</td>
<td></td>
</tr>
<tr>
<td>X</td>
<td>Deflection of spring</td>
<td>m</td>
</tr>
<tr>
<td>x</td>
<td>Deflection of rubber component from reversal position</td>
<td>m</td>
</tr>
<tr>
<td>x_o</td>
<td>Deflection at reversal</td>
<td>m</td>
</tr>
<tr>
<td>e</td>
<td>Strain</td>
<td></td>
</tr>
<tr>
<td>\theta</td>
<td>Loss angle of the damper and damper bush</td>
<td>degrees</td>
</tr>
<tr>
<td>\mu</td>
<td>Viscosity of fluid</td>
<td>N/m^2 s</td>
</tr>
<tr>
<td>\nu</td>
<td>Wave Number</td>
<td>cycles/m</td>
</tr>
<tr>
<td>\rho</td>
<td>Density of the fluid</td>
<td>Kg/m^2</td>
</tr>
<tr>
<td>\sigma</td>
<td>Stress</td>
<td>N/m^2</td>
</tr>
</tbody>
</table>

Definition of Terminology in Appendix A.
1 Introduction

1.1 Introduction

The current state of vehicle ride understanding in the literature is generally limited to primary body control only; documentation of secondary ride is minimal, and limited to the theoretical analysis of wheel hop. Much of the existing work is dedicated to the study of handling developments, especially tyre models, braking and steering control systems.

1.1.1 Primary ride

Primary ride describes modes of vibration dominated by the rigid body motion of the shell - these are typically less than 3 Hz and predominantly combinations of bounce, pitch and roll of the vehicle as a whole. These phenomena are well documented in varying levels of complexity and have been the subject of practical as well as theoretical testing, for example the "flat ride" investigations by Olley (ref. 1) including the K² rig, a vehicle modified to produce a wide range of mass distributions. Olley (ref. 1) describes in some detail the effects of mass distribution and suspension stiffness leading to the "flat ride" concept (fig. 1) in which the natural bounce frequency of the front of the vehicle is less than that of the rear such that the rear "catches up" when traversing a bump, thus reducing subjectively undesirable pitch. Best (ref. 2) continues this analysis with discussion of wheelbase filtering effects, in which wavelengths related to the vehicle wheelbase produce local minima in pitch and bounce response (fig. 2) and the composition of road input.
A wide range of passive, semi-active and active suspension systems have been investigated with simple 1, 2 and full car 1 to 5 mass models (literature review, Bernard, Vanderploeg and Shannan (ref. 3)) and also practical vehicle testing programmes, attempting to minimise the low frequency movement of the vehicle.

1.1.2 Secondary ride

Secondary ride describes modes of vibration dominated by the rigid body motion of wheels and powertrain, typically 4 Hz - 20 Hz. The influence of wheel hop and particularly bounce, pitch and roll of the engine are not well documented, being mentioned only occasionally in the literature in an idealised form. From a ride point of view this area of the vehicle response is equally, if not more important than the primary ride due to the human body subjective response (fig. 3), which is most sensitive to bounce input in the 4 - 10 Hz region - see BS 6841 (ref. 4). Some authors have wrongly attributed all secondary ride to wheel hop & tyre influence (ref. 5), where in truth the powertrain is also a significant contributor, as described by Kafetzis & Lucas (ref. 6) and Demic (ref. 7). The power unit is suspended on compliant mountings, usually with a resonant frequency of approximately 8 Hz in order to isolate significant firing and mechanical vibration (typically a minimum of 20 Hz for 4 cylinder engines) whilst avoiding coincidence with either body or wheel vibration modes. If the engine movement is lightly damped, this can introduce very significant resonance of the engine which is transmitted to the body. In addition, this isolation tends to reduce the contribution of the engine to the mass resisting the ride input from the suspension due to wheel hop modes.
1.1.3 Road input

Typical road input characteristics are well documented; there is an inverse relationship between wavelength and amplitude of input, reported by Anderton (ref. 8), Best (ref. 2), and Ammon (ref. 9), often described in terms of Power Spectral Density as

\[
\text{psd} = c \cdot \nu^w
\]

where,

<table>
<thead>
<tr>
<th>Type</th>
<th>c</th>
<th>w</th>
</tr>
</thead>
<tbody>
<tr>
<td>Major road</td>
<td>1.5e-7</td>
<td>2.7</td>
</tr>
<tr>
<td>Minor road # 1</td>
<td>1.1e-6</td>
<td>2.5</td>
</tr>
<tr>
<td>Minor road # 2</td>
<td>1.0e-6</td>
<td>3.0</td>
</tr>
</tbody>
</table>

\[
\text{psd} = \text{power spectral density of the road profile (m}^3\text{/cycle)}
\]

\[
\nu = \text{wavenumber (cycles/m)}
\]

These values of c and w (fig. 4), are provided by Gordon (ref. 10). The phase between adjacent tracks is also related to wavelength as shown by Ammon (ref. 9), greater coherence occurring for larger wavelength, due to the road building process and natural features (fig. 5).
Most road data are presented statistically in this way, but some is also shown as an actual road measurement history (fig. 6) and occasionally a constant 0.1m/s velocity characteristic is used. This is a first approximation to the road spectra; the rate of decay with frequency is not the optimum, but it provides a good rule of thumb for approximate calculations. Some authors also consider performance over discrete events (pot holes, sleeping policemen etc.); these tend to be most noticeable on short journeys, and consequently have a significant influence on the results of most subjective assessment techniques.

1.2 Fundamental Suspension Requirements

The basic design objectives of a vehicle suspension system are usually specified as "to minimise the vertical acceleration in the body to increase passenger comfort, minimise the dynamic tyre load to provide maximum road holding while cornering and braking, and react to load changes on the body that occur during manoeuvring and when the passenger load changes" (ref. 11). It is also worth including body pitch and roll motion as sources of discomfort, as implied by Moulton (ref. 12), Best and Middleton (ref. 13), and the frequency of vibration is an important consideration, illustrated by BS 6841 (ref. 4). In addition, there are the practical limitations of packaging the suspension and room for it to function, usually described as working space or rattle space.

A more comprehensive list of requirements might be; To minimise bounce, pitch and roll acceleration of the body, particularly at frequencies to which the human body is sensitive, to minimise dynamic tyre load, to minimise response to steady state and low frequency load changes, to minimise suspension package and working space. These requirements interact and are conflicting, leading to complex trade offs between the various performance measures.
The main consideration for this work is the minimisation of body acceleration, maximising ride performance, however evaluation of any changes made must consider the effect on the body control, package space and handling requirements.

1.3 Passive Suspensions

A conventional suspension system consists of a spring, usually a steel coil or leaf, and a damper unit in which a fluid is forced through small orifices to provide a damping action. The design and construction of dampers vary depending upon the application, and their characteristics can be tuned by the manufacturer, however all dampers exhibit an increasing dynamic stiffness with frequency. In order to produce acceptable noise and harshness performance, it is common practice to fit elastomer bushes in series with the damper unit, thus limiting the maximum stiffness attained by the suspension. These bushes progressively reduce the effectiveness of the damper with increasing frequency of excitation.

Figures 7, 8 show the effect that varying the stiffness of the damper bushes has on the ride performance of a vehicle. Stiff bushes that force the damper unit to deflect are advantageous for primary ride performance, maximising the effect of the damper and controlling the body resonance, but increase the transmission of vibration over most of the rest of the frequency range including secondary ride and noise regions, producing higher levels of vibration and noise. Soft bushes tend to deflect themselves, in preference to the damper unit and especially at high frequency, providing improved isolation at the expense of primary ride control. This compromise is an inherent limitation of the conventional suspension.
Besides the problem of finding suitable damper bush rates, the ride engineer must also consider the damper characteristic and its effect (ref. 2). The critical damping required for the two major modes of vibration, body bounce and wheel hop are, for a typical vehicle, 2KNs/m and 4KNs/m respectively. As a first approximation, a road profile may be considered as a constant velocity spectrum, thus for a particular road, the damper operates at a similar point on its force - velocity characteristic for all frequencies of excitation. Under these conditions it is impossible to damp both modes equally; either the wheel hop mode will be poorly damped, or the body mode excessively damped, or both. The effect of the damper bushes, by reducing the effect of the damper with increasing frequency, makes this shortcoming worse by further reducing the damping for the wheel hop mode.

Here follows a review of components influencing vehicle ride, typical characteristics and methods of operation.

1.3.1 Springs

Vehicle springing may be provided by various methods as described by Wildig (ref. 14) and Greenslade (ref. 15) - coil, torsion and leaf springs are usually made of steel but can also be composites (ref. 16); air springs, in which a trapped volume of pressurised air supports the load; and hydropneumatic springs, in which a fluid chamber or system, pressurised by a gas volume supports the load, for example Citroen's hydropneumatic system (ref. 17) and Moulton's Hydragas (ref. 12).
The load versus deflection characteristic may be tuned as a function of load and deflection in various ways depending on the spring type and construction (figs. 9 - 12). The main function of the spring is to support the static payload, but the stiffnesses selected will influence the primary ride frequencies, pitch and roll response and isolation above the resonance frequency. An additional consideration is the laden to unladen change in suspension deflection, though this may be partially addressed by using levelling systems at additional cost. These requirements are conflicting, since softer springs improve isolation but increase the change in static deflection for a given payload increase and allow greater movement under pitch and roll moment loading.

1.3.2 Dampers

Wildig (ref. 14) describes a typical automotive damper as a piston and piston rod assembly attached to the body via a compliant mount, which is constrained to move inside an inner cylinder, surrounded by an outer cylindrical casing attached to the wheel hub or suspension link, usually with a bush (fig. 13). Bump (compression) and rebound (extension) damping are caused by viscous flow through valves in the base of the damper and piston respectively, and may be tuned independently (fig. 13). This twin tube configuration is most popular for passenger cars, though motorcycle dampers tend to be mono tube designs with an enclosed high pressure gas volume to allow damper rod intrusion and prevent cavitation (fig. 14). Normally, the damper settings are chosen to provide greater damping on the rebound stroke, and to incorporate "knee" points, reducing the damping rate for high velocity, to limit harshness (fig. 15). This configuration is often cited as minimising the body disturbance for negative (upward) road events, but neglects the poor performance for a positive event (fig. 16). Damper characteristics are also frequency dependent (see 3.3 dampers), though this has not been documented in the literature.
The function of the damper is to reduce the amplitude of resonance, though this function must be weighed against the increase in transmissibility that occurs for excitation frequencies above the resonance as the damping rate is increased (fig. 7). Wallentowitz & Holdmann (ref. 18) and Kafetzis & Lucas (ref. 6), have described this as, 'Increasing damping levels tends to reduce the sprung mass resonances and, therefore, improves primary ride. This has the disadvantage of reducing attenuation of the body response in the secondary ride region'. Switchable and adjustable rate dampers of various construction and operating strategies are also in limited use; Wallentowitz & Holdmann (ref. 18), Seiffert & Walzer (ref. 19).

Ultimately the resistance within the damper causes it to become stiff, the motion is then accommodated by a damper bush, providing an additional compliance to soften the suspension when friction or high frequency excitation causes the damper to 'lockup'. This additional compliance improves the isolation at high frequency by virtue of the reduced effective stiffness (fig. 8). In many suspension systems it also accommodates suspension articulation and / or provides a geometrical function or compliance. The inclusion of a compliant element tends to reduce the effectiveness of the damper, therefore a compromise exists between primary control and isolation, similar to that with damper settings. A stiff bush provides better primary control at the expense of harshness and secondary ride, conversely a soft bush reduces the harshness and improves secondary ride, but primary control suffers (fig. 8). Hydro-elastic developments of these damper bushes are beginning to appear in patent applications (refs. 20, 21), though their effects are not documented.
1.3.3 Hydramount

Hydro-elastic mountings are becoming increasingly commonplace in the suspension of powertrains in passenger cars. They exhibit a performance advantage with respect to conventional elastomer mounts in terms of idle vibration isolation and control of engine motion, while offering significant cost savings compared with active mounting systems.

"In an increasingly competitive market the pursuit of cars with low fuel consumption and manufacturing costs has resulted in a trend to lower weight construction. This has increased the difficulties in achieving acceptable levels of noise and vibration at a time when consumers are expecting the standards to be continually improved." (West, ref. 22)

A typical engine mounting system consists of the powertrain, supported by three or four mounts that must address the following:

i) Reaction of static and low frequency dynamic forces, namely the weight, lateral and longitudinal inertial loadings, and drive torque. This is essential to restrain the power unit within the vehicle and provide tractive effort.

ii) Control of powertrain resonances, especially vertical, pitch and roll modes. These modes are excited by random road and periodic wheel inputs to the body, and are perceptible as engine 'shake'.

iii) Isolation of high frequency vibration from engine imbalance and firing forces. This high frequency excitation can cause both noise and vibration within the passenger compartment. If a cavity resonance coincides with the excitation a 'boom' condition occurs, producing a marked increase in cabin noise.
Appropriate positioning of the mounts and selection of stiffnesses can achieve great improvement in this compromise (fig. 17), however a trade off always exists; a soft, lightly damped mounting system will isolate vibration but the engine will be free to resonate on the mounts, typically at a frequency of 8 - 12 Hz. This vibration is detrimental to the secondary ride of the vehicle, as described by West (ref. 22) and Rivin (ref. 23), and large excursions from the static position will occur. A stiff mounting provides better ride and smaller engine movement, by virtue of the increased control, but has a greater transmissibility for idle vibration and noise.

The recent trends in passenger car design serve to make an acceptable compromise more difficult to achieve with elastomer mountings. An increase in suspended powertrain mass, as encountered with diesel engines or front wheel drive vehicles and an increase in drive torque, again arising from front wheel drive or from more powerful engines, tend to require mountings with higher static stiffness. The control of powertrain resonances, requiring damping forces, dictate the use of high hysteresis rubber, which stiffens remarkably with both a reduction in amplitude and an increase in the frequency of vibration. The acceptable isolation of high frequency vibration, however, requires low dynamic stiffness.

Hydramounts offer a fundamental improvement over conventional elastomer mountings. For a given static stiffness, a far greater level of damping is possible, over a limited frequency range, than with elastomer properties alone and with a smaller 'penalty' in terms of the high frequency stiffness. Thus, the conflict between static stiffness, damping and high frequency stiffness is reduced.
A hydramount (fig. 18) is a fluid filled engine mounting, with a relatively soft, lightly damped elastomer spring for isolation of engine vibration; but heavily damped around one or more of the resonance frequencies of the engine on its mounts to minimise the effect on vehicle ride (ref. 22). Thus the compromise still exists but the overall effect is much improved. The 'damping' is usually effected by virtue of resonant fluid column (ref. 24), acting as a tuned absorber, with an amplified effect due to the hydraulic amplification achieved within themounting. A wide range of characteristics may be achieved, as described by Hartel & Hofmann (ref. 25), and some active (ref. 26), switching and amplitude decoupled (ref. 27) mounts are available to further reduce the compromise between isolation of idle vibration and minimal detriment to ride (figs. 19 - 22). Section 3.4 describes the operation, characteristics and simulation of the hydramount in more detail.

1.3.4 Tyres

For vehicle dynamics analysis, the tyre characteristics are a critical part of the vehicle system. Considerable effort has been, and continues to be, expended in accurately modelling the forces and moments generated by slip at the contact patch during manoeuvring of the vehicle for analysis of handling performance, however relatively simple models continue to be used for representing the vertical characteristics of the tyre. Similarly, analysis of the high frequency vertical characteristics associated with high speed rolling on a smooth surface has been considered (refs. 28, 29) both practically and with Finite Element analysis. 

The effect of the tyre characteristics on vehicle ride performance is equally marked. Tyre stiffness is a dominant factor affecting the wheel hop mode, and must be modelled correctly for accurate ride prediction.
Tyres have a complex construction aimed primarily at handling and road holding functions in wet and dry conditions (ref. 30). Much effort is expended in tread compounding, pattern design, bracing etc. for handling performance. The wet grip, durability, lateral response; involving the determination of lateral stiffness, cornering stiffness and frequency response, Loeb, Guenther, Chen and Ellis (ref. 31) and Willumeit & Bohm (ref. 32), and more recently the rolling resistance for fuel economy and minimal noise generation for drive by noise assessment, Nash (ref. 33) are the primary considerations.

The handling related areas of tyre performance are well documented and modelled in numerous ways. Most commonly a 'lookup table' technique or curve fit is used, for example the Pacejka 'magic' tyre models (fig. 23), described theoretically by Pacejka and Bakker (ref. 34) and for practical use and determination of the parameters by Van Oosten and Bakker (ref. 35). These models include the nonlinear limit handling operating region. Where this is not required, the simpler 'cornering stiffness' approach is often used as described by Hales (ref. 36) and Loeb et al (ref. 31).

The vertical characteristics of tyres are not well documented; handling models often use a simple spring and dashpot model to determine the tyre normal force. This is probably sufficient for the relatively low frequencies encountered in handling analyses, but is inadequate for ride modelling due to the visco-elastic nature of rubber, as described by Menderes & Konter (ref. 37) and Coveney & Turner (ref. 38) (fig. 24). The small excitation amplitudes and frequencies involved vary over a range in which the viscoelastic properties of the rubber are significant. Some work has been done investigating the effects of road roughness on handling, by Xiding, Jude and Yongxin (ref. 39), which interestingly does not appear to include an improved model for tyre normal force calculation, though the smoothing effect of the contact patch is accounted for.
1.3.5 Novel passive systems

These are characterised by their storage and dissipation of energy and fixed component operating characteristics, though the more complex hydroelastic systems incorporate load levelling which requires a power supply and are therefore, strictly speaking, semi-active.

Hydro-elastic suspension systems combine a gas spring and fluid force transmission medium in a variety of arrangements. Usually the fluid is used to interconnect between wheel stations, allowing more freedom in the design of the suspension.

The Hydragas system (fig. 12) is a self contained example which also incorporates damping valves, replacing the conventional damper units. Its chief advantage is the ability to select the stiffness and damping independently for bounce and pitch input, as described by Moulton & Best (ref. 12), due to the combination of hydraulic, taper, parasitic and drop angle induced contributions to the wheel rates (fig. 12). This additional flexibility in the design allows a low pitch frequency to be obtained, reducing the response to pitch excitation in the secondary ride region. The addition of a fluid reservoir and pump allow a basic load levelling function, and with suitable sensors and control valve, a slow active system can be produced (ref. 40).

Finally, self pumping leveller systems utilise the suspension movement over road undulations to effect steady state levelling. The unit is usually continuously pumped by the suspension movement, obtaining the necessary power supply from the road irregularities, and incorporates a relief valve that discharges fluid when above the desired ride height, thus maintaining the set trim heights.
Meller (ref. 41) describes both single and twin chamber designs based upon automotive damper designs that use the pressure build up to support the load (fig. 25). The ability to maintain the desired ride height, regardless of payload, enables the vehicle designer to utilise softer springs, thus improving ride isolation for excitation frequencies above the primary ride resonance. It is interesting to note that due to its tendency to raise the body, the damping characteristics of these systems must necessarily be biased towards greater bump rates than rebound, the opposite to conventional dampers, apparently without detriment to the overall ride quality, thus calling into question the thinking behind differential bump / rebound rate settings.

1.4 Semi-active and Active Suspensions

1.4.1 Semi-active suspensions

An analysis of the operating conditions of fully active suspension systems show that although active suspensions require a power input, on average they dissipate more energy than they absorb. It therefore seems reasonable to suggest that suspension performance equivalent to active performance could be achieved without the need for auxiliary suspension power (ref. 42). As a consequence, it follows that the performance of a suspension capable only of varying the rate of energy dissipation should approach that of an active system at a fraction of the cost. Semi-active suspensions attempt to achieve this, requiring a minimum of external power supply to operate switches, solenoids etc. which alter the characteristics of the components, for example a switchable rate damper, in accordance with a predetermined set of control laws.
In the simplest case, a switchable damper is set either hard or soft depending upon the vehicle position and velocity, usually obeying a clipped Karnopp control law (fig. 26), which attempts to emulate a sky hook damper, minimising body movement. This is achieved by switching to maximum damping when the resulting force tends to reduce absolute body velocity, and minimum damping when the force tends to increase absolute body velocity; for example, the body is falling at a greater rate than the wheel, therefore a stiff setting is chosen, since the damper force tends to slow the body movement (ref. 43). Further refinements include continuously variable rate dampers, which are difficult to produce with solenoid operated valves, but can be achieved with piezoelectrically operated valves (ref. 44) or by using an electrorheological damping fluid (ref. 45). It is also possible to temporarily optimise the damping for handling manoeuvres by consideration of steering and braking input (ref. 18).

The 'PAWS' system (fig. 27), presented by Wilkinson and Crolla (ref. 42), is a hydropneumatic suspension with a pair of damper valves and accumulators. The two stage system provides frequency dependent damping, the intention being to optimally damp both wheel hop and body bounce modes. This appears to provide performance approaching that of an active suspension in both model predictions and prototype trials. The experimental results presented by Wilkinson and Crolla (ref. 42) show similar secondary ride performance for PAWS and a conventional passive suspension, but with a significant improvement in primary ride for the PAWS suspension, indicating an improvement in the primary / secondary ride compromise. This system is intended for, and demonstrated on, commercial vehicles.
Variable leverage systems use a passive spring and damper element, but vary the load by adjusting the lever ratio through which the forces act (fig. 28). If the actuator movement is perpendicular to the suspension forces very little power is required, sufficient only to overcome friction in the mechanism. The passive element may be a conventional spring and damper, as used by Venhovens, Van Der Knaap and Pacejka (ref. 46) and acting through a variable leverage linkage based upon a 'cone mechanism', or a more unusual approach, described by Leighton & Pullen (ref. 47) which uses a 'zero rate' buckling spring to provide a 'constant' force, again driven through a variable leverage system in accordance with an 'active' control strategy.

1.4.2 Active suspensions

In most theoretical analyses, an active suspension actuator is considered to be capable of providing whatever force is demanded by the controller in attempting to maintain the vehicle attitude. In practice the maximum forces are limited by the hardware and power availability. Active suspensions require a significant external power source to operate the actuators, which usually support at least part of the vehicle weight, although it is desirable to minimise this in the interest of reducing power consumption. There are a multitude of actuator types, hydraulic and air being the most popular, but also electrical and magnetic systems.

The simplest systems provide an enhanced self levelling function with a bandwidth less than 1 Hz, which levels the body in response to sustained deflection only. Slow active systems with bandwidth 1 to 4 Hz (refs. 11, 48) attempt to control the primary body movements only, retaining conventional passive spring and damper components in series with the actuator to provide control of higher frequency input (fig. 29).
This arrangement allows a cheaper control valve and reduced power consumption because of the limited bandwidth, however the benefit is gained since the passive components can be optimised for wheel control, whilst the active element controls the body. In full active systems, with bandwidth typically greater than 25 Hz, the actuator is connected directly between the body and wheel, controlling both primary and secondary ride. This arrangement allows active control of the wheel motion, as analyzed from a control point of view by Thompson (ref. 49). This analysis predicts significant improvement for the body vibration across the frequency range, a result not apparent in the practical experiments conducted by Goran, Bachrach and Smith (ref. 50) and Aoyama, Kawabata, Hasegawa, Kobari, Sato and Tsuruta (ref. 51), who report poor ride performance above 2.5 Hz and 15 Hz respectively.

There are, however, additional disadvantages of active suspensions; as the primary ride performance improves the power consumption (ref. 52), hardware cost and high frequency harshness increase, particularly for the full active systems due to the limited frequency response of the valve actuators. Goran et. al. (ref. 50) quote a peak consumption of 14.5 KW for their broad bandwidth active suspension, and an average consumption of 9 Kw. Practical systems have limited bandwidth, and consequently transmit more energy at high frequency; both Goran et. al. (ref. 50) and Tillback and Brock (ref. 53) report a significant increase in transmitted vibration above 25 Hz (the upper bandwidth limit in both cases) for their systems (fig. 30). Besides the technical performance, there are package and cost considerations for the hydraulic circuit, the actuator installation and accumulators. In terms of reliability the broadband active systems also have the significant disadvantage of actuator locking or collapse in case of control failure or pressure loss. Consequently, slow active and semi-active suspensions are a more attractive proposition for production vehicles (refs. 44, 48).
An attempt to provide most of the advantage of active systems with much reduced power consumption, cost and high frequency harshness is the concept of active roll control. For simple systems the load levelling function of the active suspension is lost but the flat cornering response, which is subjectively very beneficial (refs. 54, 55), and can simplify suspension design, shows substantial benefit. These systems usually incorporate a pair of linear or rotary actuators that are capable of biasing the front and rear anti roll bars in response to lateral acceleration to minimise body roll characteristics, but with otherwise conventional suspensions (fig. 31).

Darling, Dorey and Ross-Martin (ref. 56), Sharp and Pan (ref. 57) and Lang and Walz (ref. 58) analyze systems similar to that described, and all report improved subjective and measured handling response by virtue of the reduction in suspension movement during vehicle manoeuvres. In addition, Hickson and Darling (ref. 59) report ride performance improvement when leakage through the valve is introduced. Further improvement may be possible when using softer anti roll bars, though additional actuator travel would be required. Usually, the conventional spring and dampers are retained to control bouncing motion, however an additional low power levelling system may restore much of the attitude control benefit that has been lost; a combination of a low frequency levelling system and high frequency roll control being especially desirable, as produced by Citroen (refs. 17, 60, 61).
1.4.3 Comparison of systems

The conventional passive suspension is established and proven. It is a reliable and relatively low cost option. The novel passive systems discussed offer varying improvement over the conventional systems, but are more expensive. The cost, reliability and high frequency performance issues raised above make fully active systems unattractive at this time. Barak (ref. 62) states that the relative performance indices for conventional passive, semi-active and active suspensions are 4.5, 1.5 and 1.0 respectively and concludes that 'semi-active suspension is superior to the passive one in contrast to the fully active in that the semi-active suspension requires no other power supply than that for signal processing and valve activation'. This simplified statement of the relative merits ignores the full range of systems under investigation, but indicates the relative performance of the main classifications. Thus semi-active suspensions, active roll control and novel passive systems are the most attractive alternatives to a conventional passive suspension.
1.5 Vehicle Layouts

This section describes some of the features that, while not included in the primary area of study, have an impact on the objectives and in particular, the achievement of the practical objectives.

1.5.1 Front Suspensions

For modern passenger cars, front suspensions are invariably independent systems with an anti roll bar. The most commonly used design is the MacPherson strut, followed in popularity by double wishbone or multilink systems. Occasionally, an axle is found on 4x4 vehicles. The suspension design will influence the package space, both length and cross section, available for spring and damper units and will also affect the forces applied to the mounting points. For the work carried out on these developments, a double wishbone or multilink system is preferable, since the longitudinal and lateral loading experienced by the spring and damper mounts is minimal in comparison to the MacPherson strut, though adoption of a wishbone or multi link arrangement will result in a higher suspension ratio and greater loading of the suspension components as a result. The current popularity of front wheel drive vehicles makes it likely that any vehicle chosen for test work will be of this configuration.
1.5.2 Rear Suspensions

Rear suspensions are occasionally similar to those used on the front of vehicles, however it is usual to use a simpler system for the rear of the vehicle, steering is generally not required, and the impact of wheel attitude is less pronounced. Besides those systems already mentioned, trailing and semitrailing arms are common, though H frame arrangements, in which trailing arms are connected by a beam, contributing lateral control and some anti roll function, are popular for small to medium sized vehicles. Generally, the package space available for rear suspension is greater than that for the front, the linkages are simpler, without steering mechanisms, and the suspension ratio is usually close to unity.

1.5.3 Engine Mounting Systems

Engine mounting systems can be broadly divided into categories according to the engine orientation, driven wheels and number and type of mountings (fig. 17). Longitudinal powertrain installations are usually used with rear wheel drive and three or four mounting points, two 'engine' mounts supporting the bulk of the mass of the powertrain, and one or two 'gearbox' mounts, primarily controlling pitch of the unit (fig. 17a). Transverse powertrain installations are used on front wheel drive vehicles. The powertrain mass is supported on 'engine' and 'gearbox' mounts, with one or two mounts used to control pitch motion. For the three mount systems, the mass carriers also transmit driving torque. The mounts controlling pitching motion may be mountings similar to the other units (fig. 17b), in which case they will influence the bounce and roll modes, or 'tierods' (fig. 17c), which control the pitching motion only. The exception to these general arrangements is that used by Audi for both front and four wheel drive, with a similar layout of the mounting points (fig. 17d).
1.5.4 Seating

For the passenger car, seating consists of a foam cushion that is usually supported upon metal springs, the assembly providing a comfortable seat under static conditions. The foam and spring, combined with the mass of the passenger tend to produce a system with its vertical resonance near to 4 Hz. This frequency is influenced by the impedance of the occupant, amplitude of vibration and, obviously, the seat characteristics (ref. 63). Inevitably, however, the effect is to amplify the vibration experienced by a passenger considerably in the 4 Hz region, isolating with increasing success any higher frequency vibrations (fig. 32). This isolation does not diminish the desirability of reducing seat rail vibration in the secondary ride region; any improvement will be reflected in a proportionate, though attenuated, drop in vibration transmitted to a passenger, in addition to which, the occupant's feet and hands will still be contacting the controls or body shell.

Other vehicles, notably commercial vehicles, tractors, lift trucks and the like exhibit suspension or body - on - tyres modes in the region of 4 Hz, and for such vehicles suspension seats, providing additional isolation of the occupant, are commonplace. These seats commonly exhibit resonance near 2 Hz, isolating the passenger from the vertical vibration at higher frequencies. In most cases, the intention is to reduce the risk of injury due to prolonged vibration in 'work - vehicles'. At present, these seats are not used in passenger vehicles, primarily due to cost and adverse customer reaction to a 'mobile' seat; the unsettling feeling of significant movement between the seat and floor often outweighs the measured ride benefit. Figure 32 shows typical transmissibilities for rigid, conventional foam and spring, and suspension seats.
1.6 Objectives of the work

1.6.1 General Objectives

• To investigate the operating environmental factors influencing ride.

• To investigate the vehicle components influencing ride, particularly wheel suspension components, their characteristics, features and limitations.

• To identify and pursue possible means of ride improvement, particularly the application of hydroelastic devices.

• To develop and demonstrate the potential of such devices.

1.6.2 Simulation Objectives

• To develop and validate a vehicle simulation model suitable for primary and secondary ride analyses, including conventional suspension components, engine and mounting system.

• To extend this model so that it is capable of predictive work, indicating the performance of proposed novel suspension systems with hydroelastic components of various configurations.
1.6.3 Strut Mounting Objectives

- To improve the secondary ride of a vehicle by utilising a hydroelastic damper bush that, by modifying the characteristic of the damper will overcome some of its limitations and increase the damping of the wheel hop mode.

- To show this performance improvement with the ride model and also to produce and fit the device to a vehicle, demonstrating the advantage practically, with objective and subjective measures.

1.6.4 Complete Suspension Objectives

- To build on the strut mounting work, developing a hydroelastic suspension device to improve the ride of a vehicle by utilising a frequency dependent damping regime, damping body and wheel hop modes.

- To show this performance improvement with the ride model and to investigate further the feasibility of this concept.
2 Modelling The Vehicle

2.1 Models

2.1.1 Current models

Bernard et al. (ref. 3) describe the simulation models most commonly used for vehicle suspension performance prediction (figs. 33 - 37).

The simplest body and primary spring vehicle models (fig. 33) are suitable only for the most basic primary ride investigation, as used by Colinot, Hernet and Jarri (ref. 64). Due to the lack of wheels, engine, and often pitch & roll freedom, they are only suitable for low frequency body control. These are used to some extent for active suspension work as single degree of freedom quarter vehicle models, providing a relatively quick and easy means of comparing the effectiveness of suspension systems and control strategies. They are often extended to predict 3 - 6 Hz, which is of questionable value because of the increasing influence of secondary ride effects with increasing frequency.

Body and wheel quarter vehicle models (fig. 34) are the "next best" models, incorporating a rudimentary wheel hop mode, as used by Sharp and Crolla (ref. 65) to produce a qualitative comparison of passive, semi-active and active suspensions. This represents a significant improvement, since body isolation, suspension working space and dynamic tyre load may be considered, but there is still no engine, which is an important feature to achieve accurate secondary ride simulation. Demic (ref. 7) develops a quarter vehicle model with wheel, powertrain and human body, for which the response is significantly different to the simpler models, illustrating their importance (fig. 37).
The bicycle models (fig. 35) incorporate pitch motion; this is significant for primary ride and also influences the engine movement when it is included. This additional pitch movement is a further improvement since pitch is considered important in subjective assessments.

Full car models allow at least pitch, roll and bounce motion, and often full movement in 6 axes, and thus provide a more detailed representation of vehicle performance (fig. 36). The addition of roll freedom can be very significant when modelling single wheel or offset inputs, since the vehicle roll inertia is relatively small, and this movement is therefore important.

Linear steady state models enable analysis in the frequency domain, and more commonly have engine, wheels etc. included because of the reduced computational requirements. Nonlinear systems are difficult to model in this way, and normally the characteristics are linearised about a local mean to enable a perturbation analysis. In addition, the models are limited to steady state and statistical analyses.

Bosworth & Fursdon (ref. 66) have developed a means of modelling the simpler types of hydraulic engine mount characteristics in a similar manner to the Pacejka magic tyre model, allowing analysis of the engine mounting system, and Ochsner & Bernard (ref. 67) report a component mobility method that utilises the frequency response functions for individual components. Again, this assumes that the components have linear characteristics. For ride analysis, the transient response to discrete or random input is often desirable, and the nonlinearities of the system can be critical.
2.1.2 Limitations

Models that exclude the wheels are suitable only for the most basic body control work since no secondary ride resonances can occur; models including wheels and powertrain can provide a more accurate model of secondary ride. A quarter vehicle model can allow some appreciation of the wheel hop modes and engine bounce but neglects pitch, which is important subjectively, and roll, which is significant if the system or excitation are asymmetric.

A bicycle model is better, as it allows a pitch response; full car models can provide the most accurate simulation, but require considerably more vehicle parameters and computing resources. The more complex the model the more sensitive they become to inaccuracies in the data.

Linear modelling is totally unsuitable for modelling systems with highly nonlinear hydramount and damper characteristics and also precludes important transient response analysis. The changes made to linearise a model can introduce serious inaccuracy. In addition, single events and random input are particularly significant for comparison with subjective assessment; a subject tends to notice single events during a relatively short test, and the overall response to random input is noticeable on longer journeys.
2.1.3 Models Used

The model system consists of an input 'deck', solver and post processors (fig. 38). The input 'deck' provides component characteristics, their connectivity and output data requirements. The solver uses this data and road input data files, producing a time history of desired variables to allow transient, random, sinusoidal and measured road data to be used as excitation for the model. Post processing may be direct plotting of the time history or further processing for graphical representation, fourier transforms, averaging, peak hold, etc.

The model operates in the time domain, allowing the analysis of transient inputs, time history plotting and the incorporation of nonlinear characteristics. The integration technique used is an extremely simple stepping routine originally used for hydramount modelling (ref. 68). This is not necessarily the most efficient solver but can cope with the abrupt changes in hydramount characteristics which are often encountered and does not get 'stuck' when dealing with the nonlinearities.

Figure 39 shows the function of the program schematically. The model first calculates forces within the model using the compliant element characteristics, the model position and velocity based upon the previous step or initial conditions. These are reduced to a single force and moment vector applied to each mass and inertia in the model. A matrix method is used to calculate linear and angular accelerations of the masses, including gyroscopic effects, and these are integrated to find the new positions and velocities. This procedure is repeated for the time specified in the input deck to complete the analysis. At each step, critical variables, again specified in the input file, are stored as output.
The modular approach allows the same solver software to be used for models of many vehicle configurations and also many test conditions, including sinusoidal, random and single event excitation of the same model. This is important because of the variety of changes to the vehicle which are to be considered. The results processing flexibility allows a multitude of averaging, smoothing, fourier, peak spotting, etc. analyses using in house processing routines or proprietary software.

All the masses in the model are treated as rigid bodies and move freely in the full 6 axis system, with gyroscopic forces included. This complexity is not strictly necessary for the initial developments, but may be significant for more detailed work. The 'spring' elements, including damper components, can be connected between masses and each other to model simple spring and dashpots, more complex series spring arrangements with massless nodes, and may have different nonlinear characteristics along each axis.

Hydraulic chambers are provided for hydramount and hydroelastic element modelling, they are placed as per the springs, and may be active along any single axis, allowing, for example, damping of a fore aft vibration. Fluid link passages may run between chambers or to compliant 'reservoirs', and act as the resonant masses required in the hydramount modelling.

Four input "platens" are available for excitation in three axes - a simple linear tyre model is used for lateral and longitudinal forces, lateral response based on slip angle and a cornering stiffness, as described by Hales (ref. 36), and longitudinal input based upon tyre load and the angle of contact with the road, calculated from the road profile. These simplified models are currently available for handling and longitudinal compliance investigation.
They are considered sufficient at this stage since the main interest is ride - hence any lateral input is likely to be relatively small and in the linear range of tyre response; Lechner & Perrin (ref. 69) show that for normal driving the linear range is seldom exceeded. The simple spring and dashpot model originally used for the vertical characteristics was not sufficient. This is the subject of investigation in section 3.2.

2.2 Vehicle Tests

2.2.1 Rig testing

Rig testing provides an accurate, repeatable environment for investigation which is not influenced by weather, road safety etc. It also allows testing which is impractical on the road or test track, for example pure bounce, pitch or roll inputs. The test rig can also be used to reproduce road or random inputs. Conventional laboratory instrumentation can be used, eliminating the dependency upon battery based power supplies and reducing earthing problems. These tests can be directly related to modelling results and used for validation since the same excitation can be applied repeatedly to both model and test vehicle.

2.2.2 Road testing

Road testing is the most realistic means of evaluating a vehicle when subject to the actual operating environment. Vehicle modifications are limited and traffic, weather and driving variability make the analysis more difficult. It is also necessary to consider the input spectra, transient response and effort expended in identifying "good" test roads that excite the problem under investigation.
2.2.3 Rig / road testing discrepancy

The differences presented above necessitate a combination of rig and road testing for most development programmes. The rig provides repeatable testing to rank possible solutions and is capable of providing known inputs for model correlation work. Subjective road testing is most often the final assessment technique on which decisions are based, though these tests are much less repeatable and open to individual interpretation.

When attempting to correlate rig and road tests, in addition to an allowance for wheelbase filtering effects, a major concern is the difference in characteristics between rolling and non rolling tyres. The suspension geometry (especially if considerable scrub is present) and lateral stiffness of a static tyre adds stiffness to the system under test, whereas for a rolling tyre, the rolling effect of forward motion translates this as a slip angle, which is speed dependent and produces a similar lateral "stiffness" effect, but much less than that of the non rolling tyre. An additional consideration is the change in rubber characteristic with operating condition, which is discussed in section 3.2.4. Reducing tyre pressure is a controversial method of reducing the tyre lateral stiffness used by some researchers; the static tyre may also be placed on a laterally compliant platen.
2.2.4 Ride assessment techniques

The vast majority of vehicle ride assessments are performed on a subjective basis. The correlation between measured parameters and subjective assessment is generally poor for many combined vibration environments, though the method of reducing multiple input data recordings to a single 'NASA' value, as developed by Leatherwood & Barker (ref. 70), is used by some vehicle manufacturers. This is by no means a simple process, involving several stages of manipulation. The suitability of this method for passenger vehicle ride evaluation is sometimes questioned because the weightings used are derived from high vibration environments (helicopters, rockets etc.).

2.3 Validation

2.3.1 Single, two and three mass systems

Single mass, quarter vehicle and bicycle models (figs. 33 - 35) were used to demonstrate good correlation of numerous simulation results with analytical solutions from both hand calculations and finite element analysis. This gave confidence in the correct modelling of the response of masses, basic linear spring and dashpot elements and their interactions (figs. 40, 41). The simulations predicted both trends and absolute values of the response successfully. This confirmed that the fundamental calculations performed by the simulation software were functioning correctly.
2.3.2 Monocycle, bicycle and full car models

The results of initial vehicle simulations were less successful. The response of monocycle models with and without the engine predicted the correct trends for parameter variation studies, but the fit to vehicle test data was poor. The fit to measured data was not improved when bicycle and full car models were examined (fig. 42), these also predicted trends successfully, but failed to produce an accurate response. An investigation of the sensitivity of the model to variation of the vehicle parameters demonstrated that it was not possible to improve the fit to measured data by simply considering the effects of component tolerance (fig. 43). Measurements of weights, dimensions, damper bushes and hydramount characteristics confirmed that these were within specification, leaving tyre and damper characteristics as the most significant unknowns. The investigation of tyre and damper characteristics and development of enhanced models for these components is discussed in section 3.
3 Modelling The Components

This section describes the component models developed as a part of this work. The rubber, tyre and damper models were developed because initial vehicle simulation results did not correlate well with vehicle tests. The hydramount model was similar in form to those in the literature, however the model is extended and the revised governing equations are presented.

In the general case, the component was fitted in to a hydraulic test rig, loaded with the correct preloads and excited at a range of frequencies and amplitudes representing the operating conditions to determine the characteristics and their dependency on amplitude, velocity, frequency etc. Normally static stiffness and the characteristics of dynamic stiffness and loss angle with frequency were measured. These test results were used as a reference to assess the accuracy of the model under similar conditions.

3.1 Rubber

Vehicle suspension systems usually contain numerous rubber components, some of which directly affect the ride quality of the vehicle, for example damper end bushes. It is important, given the impact of damper bush stiffness on ride, that these components be modelled accurately, particularly their amplitude dependency. Turner (ref. 71) describes a triboelastic model which has been used, with some minor adjustments, in this work.
3.1.1 Rubber Characteristics

'Rubber is normally described as having viscoelastic properties. However, vulcanised rubber at temperatures considerably higher than the second - order transition temperature shows mechanical properties that are much more strain dependent than frequency dependent.'

Turner (ref. 71) subsequently described this behaviour as 'Triboelastic'. This strain dependency is significant, Turner quotes a factor of 4 change in dynamic stiffness for a 5 - fold increase in strain for a typical tyre compound. There is also some change in stiffness with frequency, though this is much less significant, a factor of 2 over the range 0.5 - 100 Hz at most.

3.1.2 Rubber Model

The rubber components were initially simulated using the triboelastic model developed by Turner (ref. 71) which consists of a spring and nonlinear Maxwell type model in parallel (fig. 44). The basic equation for the triboelastic model is;

\[ \sigma = E \cdot \varepsilon = -K_1(d\varepsilon/dt)^n + K \cdot \varepsilon \]

(2)

with suitable values of \( E, K_1, K, n \) to match experimental data (fig. 45).
with manipulation, this becomes;

(3) \[ F = K_1 \left[ \left( \frac{F_0}{K_1} \right)^{(1-1/n)} - t(1-1/n) \right]^{1/(1-1/n)} + K \cdot x \]

where,

\begin{align*}
F &= \text{force (N)} \\
K_1 &= \text{‘Stiffness’ (units dependent upon n)} \\
K &= \text{Stiffness (N/m)} \\
F_0 &= \text{Force at reversal of direction (N)} \\
t &= \text{time (s)} \\
n &= \text{index representing hysteresis}
\end{align*}

This equation is used to determine the effect of stress relaxation effects on the load. A simple spring model is added in parallel with this model to provide linear static stiffness. This model still exhibits limitations, the stiffness characteristic is difficult to match to data and correlation is possible over a limited range only.

The rubber model described by Turner (ref. 71) exhibits some of the requirements of a model for vehicle secondary ride, but also incorporates some limitations; it is accurate over a limited range only and the fit to loss angle characteristics is not good (fig. 46). The absolute upper limit on stiffness appears to be the major problem. A revised model, based upon the same triboelastic theory, provides a better fit and has been used for most of the modelling work.
The basic governing equation is:

\( F = K_v \cdot x^p \) 

with suitable values of \( K_v, p \) to fit experimental data.

The actual code used is more complex,

\[ F = F_0 + K_v \cdot (\text{ABS}(x-x_0)^p) \cdot \text{SGN}(x-x_0) \]

where,

- \( F \) = Force generated by the component (N)
- \( F_0 \) = Force at a reversal of direction of travel (N)
- \( K_v \) = 'stiffness' of the component (dependent upon \( p \))
- \( x \) = deflection (m)
- \( x_0 \) = deflection at a reversal of the direction of travel (m)
- \( p \) = index representing the degree of hysteresis

This model can be unstable, since the change in direction must be identified and care is needed to do this accurately. These problems are particularly acute for stiff bushes, which are difficult to model satisfactorily without undue penalty in run-time.
3.1.3 Discussion

Modelling of rubber components is a complicated problem, with a considerable body of work in the literature as a reference (refs. 71, 38, 72, 73). Many of these consider detailed material models for use in FE packages, however what is required for this work is a model that represents the characteristics of a component. The primary feature of the rubber components used in vehicles is their amplitude dependent nature, stemming from the material characteristics (ref. 71, 38), and this is modelled satisfactorily with the model presented (Equations 3,4). Figure 47 shows correlation of model and experiment. The stability of this model is still a problem, limiting its use, however this is offset by the simple tests required to deduce the model parameters, the simplicity of the model itself and the very obvious indication of instability, should it occur.

3.2 Tyres

Tyre testing is usually arranged to produce data for pseudo - static characteristics only, and at maximum rated pressure, not the true working conditions (refs. 31, 35, 74) (figs. 48, 49). The emphasis is placed upon the steady state handling and grip capabilities of the tyre, though some handling tests for lateral response are performed at higher frequencies (ref. 32). There is also much tyre testing on vehicles, both subjective and objective, again concentrating on handling and grip characteristics as reported by Frankel & Howard (ref. 30). A tyre test fixture was designed and manufactured for testing the vertical characteristics of a tyre, as required for modelling work and validation.
This test work may incur criticism regarding the non rolling tyre testing conditions, however the measured vehicle data that is being used for validation purposes was collected from 4 poster tests, also performed with non rolling tyres, hence the model will be consistent with the tests, so this is a secondary consideration at this stage. Dunn and Olatunbosun (ref. 75) describe some measurements of tyre vertical characteristics made on a rolling tyre which correlate with the traditional spring and dashpot model, however the results are not convincing, especially the damping characteristics (fig. 50).

3.2.1 Tyre Test Procedure

The tyre was fitted to a standard vehicle wheel. The wheel was fitted into a servohydraulic test machine (fig. 51), and preload, representing the static loading of the vehicle, was applied. The hub was supported and excitation was introduced through a flat aluminium plate, larger than the contact patch, attached to the hydraulic ram. Sinusoidal frequency sweeps were carried out at range of velocity amplitudes, representing the normal operating range. The load and deflection were recorded and processed to produce static stiffness, dynamic stiffness and loss angle characteristics. These tests were performed at a range of preloads and inflation pressures.
3.2.2 Tyre Results

Manufacturer's data supplied for ride modelling work typically consists of a load deflection curve for the tyre at maximum rated inflation pressure, and occasionally other, lower inflation pressures. Figure 48 shows these characteristics for a tyre at three pressures.

The data measured for this investigation consists of dynamic stiffness and loss angle values for the frequency range 1 to 30 Hz with amplitudes 0.005 - 0.3 m/s. A data set was measured for five inflation pressures (20 - 40 Psi) at 2500 N preload and seven preloads (1000 - 4000 N) at 30 Psi inflation pressure.

Figure 52 shows the variation in stiffness with preload for a range of frequencies, each showing a gently increasing trend, note that the difference in stiffness is typically 15% for a fourfold increase in preload. This is surprisingly small and suggests that the contribution of the air as a spring may be much smaller than anticipated. Figure 53 shows the variation in stiffness with inflation pressure for a range of frequencies, again showing an increasing trend, but much less marked than expected. Figure 54 shows the variation in stiffness with amplitude. This shows a slight softening of the tyre with increasing amplitude, consistent with the behaviour of rubber compounds. Figure 55 shows the most dominant trend, up to 50% increase in stiffness with frequency over the range tested. Note the initially steeper curves, particularly for small amplitudes. This is also consistent with carbon black filled rubber compounds, as used in tyre construction. Finally, figure 55 shows a small, but consistent loss angle for the tyre.
This evidence points towards the carcass of the tyre supporting a significant portion of the applied load when inflated, and contributing a large portion of the dynamic stiffness. Discussion with tyre engineers (ref. 76) tended to confirm that the tyre construction was the major contributor to tyre vertical stiffness, the inflation pressure being used to fine tune the completed tyre. This is consistent with the rubber-like response of the tyre, the loss angle measured is consistent with that of the compounds usually used in tyre construction; the complete characteristic of the tyre with air is not much removed from the behaviour of rubber.

3.2.3 Established Tyre Models

The simplest model of tyre vertical characteristics is a spring. Tyres exhibit relatively little damping, so occasionally this can be neglected completely. This model is suitable for time domain and frequency domain modelling, provided the correct value of stiffness is used for the frequency range of interest.

The spring and dashpot model attempts to include some damping, though the viscous damping model commonly used is a poor match for the viscoelastic characteristics of rubber components. This model can also be implemented in time and frequency domain models; again, the stiffness must be selected for the range of interest.

Hysteretic damping resembles the damping found in rubber much more closely than the viscous model. The damping is usually represented by a frequency dependent damping coefficient. It is easily incorporated into a frequency domain model, but time domain implementations are more problematic.
3.2.4 Discussion

The tests were performed with a non-rolling tyre, at room temperature and with the tyre contacting a smooth aluminium plate. For the intended purpose, confirmation of the tyre characteristics when testing a vehicle on a four-post rig for subsequent model validation, these conditions are an accurate replication of the 'operating' conditions. When compared with the real operating environment, the introduction of rotation, heat build up and road roughness will impact upon the model accuracy.

Probably the most significant is the rolling motion, which will superimpose a relatively large strain cycle on the small strain road inputs experienced by the tyre as it passes through the contact area. This large strain motion will induce an effective softening of the rubber, provided there is not a complete reversal of motion locally, as reported by Coveney & Turner (ref. 38), and thus produce a softer response from a rolling tyre. The second most significant factor is the temperature; close to its glass transition temperature, viscous effects become increasingly strong, affecting the rubber behaviour. This effect will be significant for colder operating conditions.

3.2.5 Triboelastic Tyre Model

Since the tyre response is dominated by the rubber characteristics, with reinforcement and air pressure providing only secondary effects, the enhanced tyre model is based upon the same triboelastic model as that used for rubber components. The governing equations and implementation of this model are similar to that described in section 3.1, with the addition of a contact patch algorithm as used by Xiding et al. (ref. 39). This routine produces a weighted average of the road profile below the contact patch to be used as the road input.
As with rubber components, this model can be unstable, since the change in direction must be identified and care is needed to do this accurately. For initial investigations, it became advantageous to use a simple spring model with stiffness constant appropriate to the dynamic stiffness of the tyre, using the more complex and time-consuming model for fine tuning work. For more detailed work however, the improvement in the model is significant, figure 56 shows a good correlation for most of the frequency range for both stiffness and loss angle traces and the amplitude dependency trend is similar.

3.2.6 Tyre Model Conclusions

The investigative work has highlighted the unexpected dominance of the rubber construction in the measured characteristics of an automotive tyre. The effect of the internal pressure appears to be secondary, provided the tyre is reasonably well inflated. This discovery has prompted the development of a new tyre model, based upon a model for rubber, which correlates better with experimental data in the range of frequencies encountered in secondary ride modelling. This is a significant step in gaining confidence in the vehicle model, since the agreement between the model and experiment is improved. (fig. 56)

In addition, these findings shed some light on the problem of the non-rolling tyre and the affect that this has on rig measurements. This is by no means a complete explanation, and further work conducting similar tests with a rolling tyre will be required to assess the full impact of the triboelastic behaviour of the tyre in service.
3.3 Dampers

The damper characteristics supplied by manufacturers are often based upon damper valve tests only, with the results manipulated to predict the performance of an assembled unit. There is also a large tolerance on the specification of most dampers, nominal force ± 40% is not uncommon for low velocities (fig. 15). Where testing of complete units is carried out, the test cycle generally concentrates on large amplitude and relatively low frequency testing (ref. 77). Switchable damper characteristics are also tested in a similar manner, though more testing is required, and a series of force-velocity characteristics are produced as shown by Petek (ref. 45). In addition, the frequency response of the switching mechanism is tested. For a variable rate damper this may take the form of a performance envelope, with minimum and maximum force capability.

3.3.1 Damper Test Procedure

Dampers and damper and bush combinations for Audi S8, A4 and VW Golf vehicles were tested from 1 to 30 Hz, at 0.01, 0.02, 0.03, 0.05, 0.07, 0.1, 0.15 m/s amplitude to determine the frequency dependency and overall characteristics in the normal range of ride operation. These characteristics were then compared with the data provided by the vehicle manufacturer to assess the ability of the model and chosen parameters to represent the damper. The force and deflection characteristics were recorded and processed to provide dynamic stiffness and loss angle in the same manner as the results of the tyre test. Note for the dampers used, the suspension lever ratio = 1.4 and therefore the testing is equivalent to up to ± 0.21 m/s movement at the wheel; this is typically the maximum value achieved for a 0.1 m/s amplitude road input at the wheel hop frequency, where maximum amplification occurs. This operating range is consistent with 'A' road test data for the vehicle used at approximately 80 KPH.
3.3.2 Damper Results

Dampers were tested to confirm their characteristics, in an attempt to determine the cause of the discrepancy between the vehicle simulation and tests. The results indicated that the correct magnitude of forces were being predicted for both bump and rebound, but the phase in relation to the excitation was different, the damper having significantly more stiffness than expected, particularly at high frequency with small amplitude excitation (figs. 57, 58). This unexpected increase in elastic stiffness can partially explain the discrepancy between the calculated and modelled wheel hop frequency predictions, and that measured on the vehicle (fig. 42).

In theory the wheel hop frequency is given by,

\[
F_n = \frac{1}{2\pi} \sqrt{\frac{K_t + k_s}{M_w}} = 13.2 \text{ Hz}
\]

(6)

However, including the damper stiffness effect,

\[
F_n = \frac{1}{2\pi} \sqrt{\frac{K_t + k_s + k_d \cos(\theta)}{M_w}} = 18.1 \text{ Hz}
\]

(7)
where,

\[
\begin{align*}
F_n &= \text{Natural frequency of wheel hop mode (Hz)} \\
K_t &= \text{Tyre vertical stiffness (N/m)} \\
K_s &= \text{Suspension stiffness (N/m)} \\
K_d &= \text{Dynamic stiffness of damper and damper bush (N/m)} \\
M_w &= \text{Unsprung mass (Kg)} \\
\theta &= \text{Loss angle of damper and damper bush (degrees)}
\end{align*}
\]

This is much closer to the measured frequency of 17.5 Hz.

3.3.3 Enhanced Damper Model

The results of the damper testing were used as guidance to produce a revised model. In addition, a damper was dismantled to provide insight into the internal components and function (fig. 13). Duym, Steins and Reybrouk (ref. 78) report similar test results and develop a series of damper models. These do not replicate the damper characteristics measured, neglecting the important stiffness features.

The new model is more complex, simulating the operation and flow dynamics of the internal damping valves, and requires more parameters than the original nonlinear viscous damper to adequately describe it. Figure 59 shows the process used to simulate the damper. The model reproduces the construction of a typical automotive damper, with two separate valves for bump and rebound motion. The valve model consists of a spring, controlling the area of a port between fluid chambers, and introducing compliance as in the real damper. The flow through the port is resisted by viscous and orifice damping terms.
The equations used for each valve are:

\[(8) \quad P_{damp} = volspr \cdot stiffspr\]

where,

- \(P_{damp}\) = pressure generated across the valve (N/m\(^2\))
- \(volspr\) = volume of fluid accepted by the valve (m\(^3\))
- \(stiffspr\) = stiffness of the volumetric compliance formed by the valve and spring (N/m\(^5\))

\[(9) \quad volspr = \int areaP \cdot v_{damp} - v_{flow} \cdot areaV \quad dt\]

where,

- \(volspr\) = the volume of fluid absorbed by the valve due to deflection of the valve and spring (m\(^3\))
- \(areaP\) = area of damper piston (m\(^2\))
- \(v_{damp}\) = velocity experienced by damper (m/s)
- \(v_{flow}\) = velocity of flow through valve (m/s)
- \(areaV\) = cross sectional area of the valve opening - this is dependent on both the size of the valve, and the deflection of the spring, which corresponds to the valve lift (m\(^2\))
where,

- **circvalve** = circumference of valve (m)
- **areavalve** = plan area of valve (m²)

These equations produce two opposing pressure values, one for each valve. These are combined to produce the net force.

\[(11) \quad F_{\text{damper}} = (p_{\text{comp}} - p_{\text{ext}}) \cdot \text{areap} \]

where,

- **pcomp** = pressure generated by 'compression' valve - refer back to pressure equation for calculations (N/m²)
- **pext** = pressure generated by 'extension' valve (N/m²)

Currently it is necessary to determine the parameters for the valve spring rates and port sizes from experimental data, tuning the model to reproduce the damper characteristics. Generally this will be the case for development work; the damper will be available for testing, but its components will not.
3.3.4 Discussion

With appropriate values, this model gives an accurate representation of an automotive damper (figs. 60, 61). In some cases it has been necessary to include a seal friction model, based upon a Maxwell model (fig. 44), for the dampers used, and other, gas pressurised dampers will require an additional spring element, though this type of damper has not been encountered in this work. The level of complexity in the model is minimised by the use of single volumetric stiffness values for the valve springs and adoption of a single piston area. In reality, the spring characteristics are nonlinear to produce the knee points found on damper characteristics and the effective piston areas are different for compression and extension.

These simplifications are reasonable; experimental measurements of operating conditions indicate that the damper functions beyond the knee point of the characteristic on rare occasion only, and the model is parametric rather than attempting to reproduce exactly the conditions within the damper. A final consideration when simulating a damper is the effect of temperature on the viscosity of the fluid. The viscosity can easily be doubled when operating in low ambient temperatures, and this will have a marked effect on both primary and secondary ride characteristics. Duym et al. (ref. 78) confirm this tendency, but neglect the effects of temperature in both testing and simulation. The effects of temperature are outside the scope of this work.
3.4 Hydramounts

3.4.1 Introduction

This section describes the hydramount in detail and presents a modular technique for modelling the sub components, such that mountings of various configurations may be represented.

A typical hydramount comprises five significant features (fig. 18): A main elastomer spring which supports the static load, displaces fluid when deflected and provides some volumetric compliance. A bellows that, together with the main spring, contains the fluid within the mount. A plate that divides the mount into a working chamber and a compensation chamber. An inertia track or channel, often incorporated in the divider plate, which joins the working and compensation chambers. Most hydramounts also include a diaphragm or amplitude decoupler to introduce a nonlinear modification to the volumetric compliance of the main spring.

3.4.2 Mount Operation

The principle of operation is similar to that of a helmholtz resonator or tuned mass absorber, using the hydraulic components to amplify the effect. The frequency to which the mount is tuned is governed by the combined volumetric compliance of the working chamber, the dimensions of the channel and the fluid density. Relative motion between the engine and body fixings of the mount cause the main spring to deflect, modifying the volume of the working chamber. At low frequency the displaced fluid passes through the channel to or from the compensation chamber.
The velocity of fluid in the channel is much greater than that measured across the mount because of the large ratio between the effective area of the main spring and the cross sectional area of the channel. As the excitation frequency approaches the tuned frequency, the mass of fluid in the channel begins to resonate on the spring provided by the volumetric compliance, causing pressure fluctuations in the working chamber that are phase shifted with respect to the excitation. These pressure fluctuations act on the large area inside the main spring, producing a damping force. When the excitation frequency is increased beyond the resonance, the movement of fluid in the channel diminishes, and displaced fluid is absorbed by the volumetric compliance, creating an in phase pressure variation and contributing to the dynamic stiffness.

The diaphragm or amplitude decoupler element is used to modify the volumetric compliance of the main spring. It may be nonlinear because of its geometry or increasing contact with snubbing surfaces as it is forced to move. A rising rate characteristic is used to provide more volumetric stiffness when the excitation or response amplitudes are large, thus producing damping for excitation in the tuned frequency region, and lower volumetric stiffness for small amplitudes. This is intended to provide low stiffness for improved isolation of engine vibration and noise; with the decoupler stiffness becoming significant, and thus producing damping, only when large road induced vibration is encountered.

3.4.3 Characteristics and Formulae

Hydramounts are typically tested from 1 to 30 Hz at amplitudes of 0.1, 0.3, 0.5, 0.7, 1.0 mm for ride characteristics, and 10 to 200 Hz at 0.1 mm amplitude, or 50 to 600 Hz at 1G amplitude for evaluation of noise performance. This represents normal operating conditions and provides stiffness, damping, decoupling, cavitation and snubbing data.
Semi-active switchable versions are tested in each of the operating conditions. The typical characteristics of hydramounts are well documented (refs. 25, 26, 7)(figs. 18 - 22).

The hydramount operation results in a characteristic as shown in figure 18, in which a static stiffness and a dynamic stiffness can be clearly identified. In the region between these points, the stiffness dips below the static value, then peaks before falling back to the dynamic stiffness. Simultaneously, the loss angle (phase angle between deflection and force) rises to a maximum consistent with the mid point of the frequency rise and tuned frequency. Bosworth and Fursdon (ref. 66) derive equations for the dynamic stiffness, the tuned frequency and effective mass as follows;

\[
\begin{align*}
K_d &= K_s + K_2 \\
K_2 &= Ae^2 \cdot K_p
\end{align*}
\]

where,

- $K_d$ = High frequency dynamic stiffness (N/m)
- $K_s$ = Static stiffness (N/m)
- $K_2$ = Increase in stiffness (N/m)
- $Ae$ = Effective piston area of the main spring; fluid volume displaced per unit linear deflection (m$^2$)
- $K_p$ = Puffing stiffness; the volumetric compliance of the working chamber defined as the change in pressure per unit change in volume (N/m$^5$)
\[(14) \quad W_n = \sqrt{\frac{K_p \cdot A_c}{\rho \cdot L_c}}\]

where,

- \(W_n\) = Resonant frequency (rads/sec)
- \(A_c\) = Cross sectional area of channel (m\(^2\))
- \(\rho\) = Density of fluid (Kg/m\(^3\))
- \(L_c\) = Length of channel (m)

\[(15) \quad M_e = \rho \cdot A_c \cdot L_c \left(\frac{A_e}{A_c}\right)^2\]

where,

- \(M_e\) = Mass of equivalent dynamic absorber (Kg)

Figure 62 is based upon the simplified linear model proposed by Seto, Sawatari, Nagamatsu, Ishihama and Doi (ref. 79) and illustrates the action of each part of the mount and the total effect (ref. 68). Note that the initial action of the channel motion is equal and opposite to that of the top, balancing the volume of the working chamber and resulting in static stiffness only since fluid displaced by the top simply flows through the channel into the bellows. As the excitation frequency is increased, the influence of the channel motion becomes larger and shows an increasing phase shift relative to the excitation.
At the resonant frequency, the channel movement is a maximum, and produces significant damping forces. Finally, the channel movement diminishes, leaving only the elastomer spring and the in phase top pumping action, which produces the dynamic stiffness. The figure also illustrates the condition at a particular frequency, in this case 10 Hz. The channel motion is significant, but below the maximum achieved at resonance as the excitation frequency is still below resonance. The overall effect is shown in terms of $K_d$ and $\theta$, more commonly presented on a Bode plot of performance.

When a mount with nonlinear characteristics is considered, the value of $K_p$ becomes amplitude dependant and consequently the resonant frequency, $W_n$, stiffness rise, $K_2$, and damping effect also vary widely with amplitude (figs. 64, 65).

3.4.4 Component Modelling Technique

The model developed in this work operates in the time domain, enabling nonlinear and transient analyses of the mounting, and thus avoiding the limitations of linearized modelling methods (refs. 79, 80).

To allow modelling of a wide range of mount and mounting system types, the hydramount sub components were modelled separately and combined to produce a complete hydramount unit. This modelling technique enables a single modelling system to be used for simple hydramounts as demonstrated here, mountings with multiple channels, interconnected systems and so on. The methodology is similar to that used in the Bath FP modelling environment (ref. 81), in that the sub components can be combined in any configuration that ensures compatible inputs and outputs between linked models (fig. 66).
For the hydramount, separate models are produced for the main spring, working chamber, channel and compensation chambers. In all cases, the fluid is considered incompressible. This is justified by the low pressures at which the unit operates (±1 Bar), the low compressibility of the operating fluid in comparison with the gaseous elements and the high volumetric stiffness of the channel walls.

### 3.4.5 Models and Nonlinear Features

The main spring model is an idealised spring with nonlinear stiffness, achieved by interpolating between points on a load vs deflection curve.

\[
\text{Fspring} = f(X)
\]

where,

- \( F_{\text{spring}} \) = spring force (N)
- \( f(X) \) = value of spring force from interpolation of load vs deflection points (N)
- \( X \) = deflection of mount (m)

The working chamber is modelled as a piston with an effective area equal to that of the main spring, an additional volume term to provide the connection to channel elements, and a volumetric compliance, accounting for the combined main spring, diaphragm and amplitude decoupler elements. This compliance is nonlinear and is achieved by interpolating between points on a characteristic curve in a similar way to the nonlinear stiffness.
(17) \[ P_{\text{chamber}} = K_p(X \cdot A_e - V_{\text{ole}}) \]

(18) \[ F_{\text{fluid}} = P_{\text{chamber}} \cdot A_e \]

where,

- \( P_{\text{chamber}} \) = Pressure in working chamber (N/m\(^2\))
- \( V_{\text{ole}} \) = Additional volume term due to fluid transfer into the channel(s) (m\(^3\))
- \( F_{\text{fluid}} \) = Force due to fluid pressure acting on the inside of the main spring (N)
- \( K_p(\text{vol}) \) = Value of chamber pressure interpolated from pressure vs volume points; The value of chamber pressure is determined by linear interpolation between values from a lookup table which represents the non linear 'compliance' of the chamber. (N/m\(^2\))

In addition, a check is performed to limit the minimum value of \( P_{\text{chamber}} \) to a user defined value. This feature simulates cavitation in the working chamber when the pressure in the working chamber approaches absolute zero.

The channel model includes inertial mass, viscous and hydrodynamic loss effects. The model requires values for the pressure applied at each end of the channel. As the fluid is considered incompressible, it is treated as a solid 'slug' within the channel (ref. 82). The viscous loss is calculated for laminar flow; \( Re < 2300 \). This is consistent with the operating conditions.
\[ P_{\text{visc}} = 2 \frac{L_c \mu V_{\text{chan}}}{(R_{\text{hyd}})^2} \quad \text{(19)} \]

\[ P_{\text{hyd}} = \frac{\rho}{2} C_d V_{\text{chan}}^2 \quad \text{(20)} \]

\[ P_{\text{net}} = P_1 - P_2 - P_{\text{visc}} - P_{\text{hyd}} \quad \text{(21)} \]

\[ A_{\text{cc}} = \frac{P_{\text{net}}}{\rho L_c} \quad \text{(22)} \]

where,

- \( P_{\text{visc}} \) = Pressure loss due to viscosity and channel cross section characteristics (N/m\(^2\))
- \( \mu \) = Viscosity of fluid (N/m\(^2\)s)
- \( V_{\text{chan}} \) = Velocity of fluid in channel (m/s)
- \( R_{\text{hyd}} \) = Hydraulic radius of channel (area / perimeter) (m)
- \( P_{\text{hyd}} \) = Pressure loss due to hydrodynamic effects (entry, exit, flow path) (N/m\(^2\))
- \( C_d \) = Discharge coefficients
- \( P_{\text{net}} \) = Net pressure differential acting on the channel (N/m\(^2\))
- \( P_1, P_2 \) = Pressures acting on opposite ends of the channel (N/m\(^2\))
- \( A_{\text{cc}} \) = Acceleration of the fluid slug in the channel (m/s\(^2\))

Later in the development programme, it became necessary to add a 'diaphragm stiffness' term to the channel model, representing the effect of a rubber diaphragm within the channel (fig. 67).
The equations become,

\begin{align*}
(23) \quad P_{\text{visc}} &= 2 L_c \mu \frac{V_{\text{chan}}}{(R_{\text{hyd}})^2} \\
(24) \quad P_{\text{hyd}} &= \frac{\rho}{2} C_i V_{\text{chan}}^2 \\
(25) \quad P_{\text{diaph}} &= K_{\text{diaph}} \cdot \text{volchan} \\
(26) \quad A_{\text{cc}} &= \frac{P_{\text{net}}}{\rho \cdot L_c} \\
(27) \quad P_{\text{net}} &= P_1 - P_2 - P_{\text{visc}} - P_{\text{hyd}} - P_{\text{diaph}}
\end{align*}

where,

- \text{volchan} = \text{volume of fluid displaced in the channel (m}^3\text{)}

The compensation chamber model operates in the same way as the working chamber model. The only difference is that the 'x' term, relating to mount deflection is always zero, hence the model simplifies to;

\begin{align*}
(28) \quad P_{\text{chamber}} &= K_p (V_{\text{ole}})
\end{align*}

where,

- \text{Pchamber} = \text{Pressure in compensation chamber (N/m}^2\text{)}
- \text{Vole} = \text{Volume of fluid transferred from the channel(s) (m}^3\text{)}
Generally, the parameters required to simulate the hydraulic elements are derived from finite element models, analysis of test results or deduced from data related to similar parts. The model can also be used to determine the characteristics of the components, assisting with component design problems.

3.4.6 Model Prediction and Prototype Comparison

Hydramounts were tested from 1 to 30 Hz at amplitudes of 0.1, 0.3, 0.5, 0.7, 1.0 mm to characterise their performance. This represents normal operating conditions and provides stiffness, damping, decoupling, cavitation and snubbing data. The model was used to simulate response to the same test conditions. Figures 63 - 65 show comparisons of model predictions and the corresponding mount characteristics.

Mount #1 (fig. 63) has a linear diaphragm and main spring, and the channel is relatively large, resulting in minimal resistance to flow. This produces a 'peaky' mount with large loss angle and little change in the resonant frequency with amplitude. The high frequency stiffness varies only slightly with amplitude. This response is also predicted by the simulation.

Mount #2 (fig. 64) also has a linear diaphragm and main spring, but contains a smaller, more restrictive channel. As a result, the mount becomes more strongly amplitude dependant, since the larger amplitude resonance is less well developed. This produces a mount with smaller loss angles and some reduction in frequency with increased amplitude. Again, the simulated result follows this trend. In addition, the rubber compounds used in this mount have more hysteresis, producing a greater increase in high frequency stiffness with reducing amplitude.
Mount #3 (fig. 65) has a linear main spring, but the diaphragm is replaced by an amplitude decoupler. This produces a mount that is strongly amplitude dependant, producing little damping for small amplitude excitation, but greater loss angle at increasing frequency as the amplitude is increased. The action of the amplitude decoupler also serves to reduce the high frequency stiffness of the mount with reducing amplitude, resulting in a mount with better noise performance. The amplitude dependency of the decoupler is also accurately represented by the model.

3.4.7 Discussion

The use of a hydraulic means to amplify the effect of a small dynamic absorber gives the hydramount its primary advantage over conventional mountings and those incorporating traditional tuned masses; a few grams of fluid provides a similar effect to several kilograms applied at the mounting points. In addition, nonlinear features offer further advantages in tuning mountings for a particular application. Modelling the hydramount as a series of separate components is intended to allow a wider range of mount configurations to be represented with a single modelling system. This can be used as a product development tool, and in addition, the modular nature of the model lends itself to incorporation in a vehicle model, and thus predictions of the effect of hydramounts on the ride characteristics of a complete car. The inclusion of nonlinearities is essential to accurately model hydramounts now in common use on passenger cars.
3.5 Roads

3.5.1 Road Surface Characteristics

Road surfaces encountered by passenger vehicles possess a combination of features, each of which cause the vehicle to respond differently, with different subjective effects. The most common features are the random surface of the road, periodic features, whether by wear of the surface or the manufacturing process, and single events, pot holes, repairs, service covers etc. These distinctions are often overlooked, but are particularly important when considering ride performance, since different strategies are applicable to dealing with each type of feature.

The random surface provides a continuous broadband input, exciting all frequencies with some opportunity for large resonant modes to develop. The amplitude of the input is speed dependent. The periodic surface produces inputs at particular frequencies, dependent upon the vehicle speed and road surface, which allows greater opportunity for resonances to develop, provided the excitation frequency is appropriate. Finally, the single events are akin to step or ramp inputs, and lead to a series of superimposed decaying pseudo-sinusoidal responses.

The published literature tends to concentrate on the continuous random input. La Barre, Forbes and Andrew (ref. 83) present a series of results for approximately 50 road profiles in the UK, France, Belgium and Germany, considering major, minor, trunk and high speed roads. The overriding feature of these measurements is the difference between road surface character not only between the different types of road and their location, but also the variability in surface within a particular classification of road.
This seems to correlate well with industry practice, each manufacturer identifying certain favourite stretches of road for particular tests depending on the problem under consideration.

3.5.2 Road Models Used

The three road features described have been modelled in several ways; firstly, sinusoidal frequency sweeps with various wheel to wheel phase were used, representing in a simplified way the periodic behaviour of the road surface and also providing a means of reproducing rig tests with sinusoidal input for bounce, pitch and roll excitation. A range of constant velocity amplitudes were considered.

Secondly, random signals with constant velocity profiles were produced to replicate the road surface. Two different conditions have been considered, that where the rear wheel inputs are time-shifted copies of the front wheel input, as experienced by a real vehicle, and that where the front and rear inputs are uncorrelated to provide a response profile without wheelbase filtering effects. In both cases the left and right hand wheel tracks are correlated as per figure 5. The inverse FFT method, described by Cebon and Newland (ref. 84) was considered, but was rejected as a suitable method for two reasons; first, the method requires sufficient FFT data to generate the road section, and if the profile is not to be repeated, the data required is equal in size to the road profile data to be produced, which was not available. Second, the method described produces continuous input at the average amplitude for each frequency, rather than a discontinuous, variable amplitude signal, as witnessed for real roads (fig. 68). These two problems were better addressed by using pseudo random data, producing random amplitude and phase within the signal, a simple and quick method requiring minimal measured data.
Finally, single step events, both positive and negative, with varying spacing between the steps have were to represent the discontinuous road features. These take the form of step and 1-cos bump profiles of varying height and length (figs. 69, 70).
4 Validation of the Revised Vehicle Model

4.1 Monocycle model with / without engine

The model showed the correct characteristics, reproducing the vehicle response accurately, and was useful for certain applications where engine pitch is not significant. This model has been used for the early strut mount modelling and investigation of some engine to suspension interconnection systems. (fig. 34)

4.2 Bicycle model with engine

The modelling was much more complex for this system, and initially did not correlate well with the vehicle in certain areas (fig. 42). The revised model predicted vehicle response with much greater accuracy, but it was still impossible to fit the response for the whole frequency range (fig. 71). Finally, the problem became apparent; the suspension stiffness quoted by the manufacturer was incorrect. With an accurate value the models matched test results, with the exception of the seat rail data which shows some evidence of chassis bending modes (fig. 72). This inaccuracy is to be expected since all masses were treated as rigid bodies.
4.3 Full car model with engine

The full car model suffered from similar problems to those experienced with the bicycle model. The same problems were responsible for the discrepancies. Figures 72 - 80 show a comparison of the responses for the left-hand seat rail, right-hand engine mount and right-hand front wheel with the corrected vehicle data. The prediction of the wheel response and engine mounting point is excellent in all three cases, following the actual test results closely. The seat rail prediction is good for lower frequencies, but is poor around 22 Hz. Subsequent testing on a the vehicle showed considerable bending of the floor pan in this frequency range, both longitudinally, the first body bending mode, and laterally, resonance of the floor. This accounts for the difference in response.

4.4 Validation Conclusions

The modelling system as described is suitable for creating a range of models, with varying complexity in both the definition of the vehicle configuration and the construction of the components. At one extreme, a simple linear quarter vehicle model, with one degree of freedom can be produced, suitable for the most basic primary ride analysis. At the other, a complete full car vehicle model can be developed with engine, subframes and other lumped masses, and complex nonlinear mountings, dampers, tyres and, as required, hydroelastic components with interconnections if desired. It has been shown that these models are capable of simulating the vehicle response accurately. The vehicle parameters used for further simulation work are presented in Appendix B.

The most commonly used models are the quarter vehicle and bicycle models, with only the features necessary for assessment of the proposal under investigation.
5 Strut Mount Predictive Modelling

The strut mount concept arose from analysis of the damping ratio provided by a conventional damper for the body and wheel modes, discussed in section 1.3. This analysis identified the wheel mode to be considerably under damped. The strut mount was conceived as a replacement for the upper damper bush, positioned between the damper and body, and exhibiting strongly frequency dependent damping using hydramount principles to increase the damping of the wheel hop mode only. The discovery of the actual damper characteristics, discussed in section 3.3, showing progressively reduced damping with increasing frequency further strengthens the proposal.

5.1 Introduction

Hydro-elastic mounts are becoming commonplace in powertrain mounting systems, but their characteristics make them suitable for solving other vibration problems in vehicles. This section describes the development of a hydroelastic mount that improves secondary ride when used as a strut mount, or damper top bush, on suspensions with telescopic dampers. The performance limitations of conventional bushes are discussed, leading to an explanation of the rationale behind the hydroelastic strut mount. (ref. 85)

What becomes apparent from figures 7, 8 is that the ideal suspension would be relatively soft, with heavy damping around the resonant frequencies only, thus providing control of the body and wheel modes, but isolating the higher frequencies and the important secondary ride region between the main modes, in the frequency range 6 - 12 Hz.
With the conventional system the first of these requirements, control of the body mode, can be achieved satisfactorily, but increasing the damping to suppress the wheel hop mode introduces a trade-off between isolation and control of the wheel. The frequency dependent properties of the hydramount will allow the introduction of additional damping for the wheel hop mode.

5.2 Modelling

A vehicle model was produced to investigate the replacement of the conventional damper bushes with a hydroelastic strut mounting. A quarter vehicle model was used, incorporating the wheel and body only. The effect of powertrain motion was omitted to improve the clarity of the prediction, but it was also felt that the strut mount would be equally applicable to front and rear suspensions, and so the powertrain effects would not necessarily be significant in all installations.

The wheel and body were modelled as point masses, the tyre as an undamped spring, and the main suspension as a constant rate spring. The damper was modelled as an idealised viscous damper with a nonlinear force-velocity profile, incorporating a bump/rebound bias and knee points. The strut mount was modelled as a lightly damped spring with a hydramount unit to provide increased damping. The hydramount model was a simplified version of Avon's hydramount prediction software, incorporating the main nonlinear geometrical and fluid flow effects, as described previously. Further details of hydramount modelling techniques are given in references 66, 68 and in section 3.4. A time stepping routine was used with a Newton-Cotes integration scheme to model the response in the time domain, using a Fourier analysis to generate the frequency response plots.
5.3 Components

A range of strut mount characteristics were investigated and the simulated performance compared to determine an optimum mount condition. The intention was to preserve the primary ride and noise performance, and to maximise the improvement in the control of the wheel hop, thus reducing secondary ride vibration. In order to achieve this, both the static stiffness and the high frequency dynamic stiffness chosen were made equal to the values for the existing component. The remaining features, tuned frequency, channel and pumping chamber dimensions were optimised by an iterative process of full factorial experiments for parameter variations. The objective was to minimise body vibration, ideally with improvement across the whole frequency range, though the biggest improvements were inevitably in the secondary ride region to which the mountings were tuned. Figure 84 shows the final optimised requirements.

5.4 Results

Figure 81 compares the model prediction for the original condition with elastomer bushes with that of the hydroelastic strut mount equipped vehicle. The model predicts a 25% reduction in the wheel hop vibration transmitted to the body, using the optimised strut mount characteristic (fig. 84).
5.5 Conclusions

The modelling work predicts up to 25% reduction in the wheel hop vibration experienced by the passengers (Fig. 81), with little or no trade off in primary control and noise transmission since the static stiffness and high frequency dynamic stiffness are similar to those of the existing bush. This is a significant improvement, desirable for a vehicle manufacturer.

This favourable result, combined with the predicted physical requirements of the part (fig. 84), which are reasonably easy to achieve practically, led to continuation of this work, producing a prototype development programme. Early prototype simulators, based upon a production mount, were produced and rig tested with dampers to gain confidence (figs. 82, 83). Parts suitable for vehicle tests were produced and tested on a vehicle provided by the customer.
6 Strut Mount Investigation and Validation

Prototype parts were produced for vehicle testing, based upon the optimised prediction of the model, with a range of characteristics, different tuning, stiffness, etc. A full list is given in Appendix C. These were tested, initially on the front suspension only, but later on both the front and rear suspension, of a test vehicle provided by Audi (fig. 85). The objectives of this practical test work were to provide further validation of the vehicle model, extending its capability to novel vehicle configurations and to demonstrate the effectiveness of the strut mount in controlling wheel hop.

6.1 Vehicle Tests

Benchmark measurements were made with the standard vehicle for sinusoidal sweep rig tests and objective and subjective road tests. The front hub, damper rod, strut tower, floor and seat rail were instrumented with accelerometers for the initial trials, with the rear hub and rear corner of the sill being added during the investigation (figs. 86 - 93).

The first tests with hydroelastic mounts were performed exclusively on a four-poster rig using constant velocity sinusoidal sweep inputs to gain an insight into the response of the vehicle with the strut mounts. These tests confirmed the model predictions to be reasonably accurate in broad terms. Subsequent tests were carried out both on the rig and a selected stretch of road at a range of test speeds. Both subjective and objective measures were used to optimise the vehicle ride performance.
6.2 Design of Parts

The mounts were designed to fit the existing vehicle with minimal modifications, and initially to be fitted to the front suspension only. The package requirements necessitated a departure from the usual hydramount design (fig. 18), finally producing a pressurised, double acting design as shown in figure 94. The pressurisation was necessary to provide adequate resistance to cavitation, the twin working chamber arrangement to both reduce the pressure requirement and balance the end load that would otherwise occur. The rubber sections were analyzed with finite element models to gain confidence in the durability of the design, and also to ensure that the stiffness and volumetric compliance properties were appropriate.

6.3 Manufacture of Parts

The next stage of development was to manufacture prototype mounts for vehicle trials. A range of mounts were produced from the basic design with varying stiffness and damping characteristics, nominally the optimum condition predicted by the model with positive and negative variations achieved by changes to rubber compounds and channel length. Additional parts were produced for later development of the rear suspension.
6.4 Component Tests

Each component was individually tested after assembly to verify its characteristics. Appendix C contains full details, but figure 95; the 'nominal' condition, shows typical characteristics. Comparison with figure 84 indicates the problems associated with producing the parts, the actual dynamic stiffness is higher than expected, a combination of the rubber compound being slightly too stiff, and the increased amplitude dependency of the harder rubber compounds. This also contributes to the increase in the 'spread' of the damping peaks with amplitude, though the main problem here is that of the channel; problems were encountered in sealing the channel path when parts had been assembled repeatedly. Soft tolerance rings were used to minimise this effect.

6.5 Installation

Installation in the vehicle involved dismantling and removing much of the suspension, especially for the front, where the concentric damper and coil spring were removed as a unit and dismantled separately. At the front the dampers themselves were modified to allow variation in length, to compensate for the additional height of the hydroelastic bush, and two sets of suspension castings were used, one set machined to accept the new component. The rear suspension modifications were less severe; the dampers were modified, as per the front, and adapters were made to mate the new bush to the existing suspension mounting points. The adjustable length dampers were an important part of this work; it would have been far easier to permanently shorten a second set of dampers, but the difference in damper characteristics would influence the behaviour, affecting the comparison with the original condition. Figures 96 - 101 show the stages of the installation procedure.
6.6 Test Procedures

Initially, tests were performed on the four-post rig only, providing data for controlled conditions. This was used to provide a comparison with the model data and to carry out initial tuning work.

6.6.1 Optimisation - front suspension

The rig tests continued in parallel with objective and subjective road tests to optimise the system, with the aim of minimising body acceleration, as per the simulation work and maximising ride comfort. For the road tests, a specific section of road was used, and tests were carried out at a range of speeds to assess the effect of wheel base filtering and relative phasing of the inputs to the front and rear wheels. Care was taken to ensure minimal change in the fuel load of the vehicle. The results of these investigations were promising, although the impact of the hydroelastic mounting was less than expected. The effect of the changes due to the strut mount diminished as the measurement location moved away from the front of the vehicle. This is not surprising, since there is a progressively increasing contribution from the standard rear suspension and a reduction in the contribution of the front suspension. As a consequence, the benefit as measured at the strut tower was considerable, but at the seat the effect was much smaller and difficult to detect subjectively.
6.6.2 Optimisation - front and rear suspension

In order to demonstrate the full potential for ride improvement, more prototypes were produced and fitted to the rear suspension in addition to the front. This configuration was also optimised through rig and road testing, and using the knowledge gained from the work on the front of the vehicle. The same testing procedures were followed, with additional accelerometers fitted to the rear of the vehicle.

6.7 Results

6.7.1 Rig tests

Figures 102, 103 show a summary of some of the seat rail responses, measured on the four-poster rig. Responses are shown for the standard benchmark condition, for a vehicle fitted with the optimised front hydroelastic strut mounts only and for a vehicle with ride comfort optimised mounts front and rear. The front only condition shows some improvement across the whole secondary ride region, from 3 to 30 Hz, the greatest effect occurring around 17 Hz. This is consistent with the tuning of the mounting used and the model prediction. It was found that none of the hydroelastic mountings tested made the secondary ride performance worse than that of the standard vehicle, however the static stiffness of the bush had an effect on the primary ride, and the dynamic stiffness influenced the high frequency response, as predicted. Varying levels of improvement were possible, depending on the exact tuning of the parts.
The result for mounts fitted front and rear shows further gains in secondary ride performance (fig. 103), since the improvement is now present front and rear; the standard ride performance of the rear suspension does not dilute the improvements made at the front. A reduction in the vibration of one third is achieved across a broad range of frequencies from 15 to 25 Hz. The effect on primary ride is not so good, however, exhibiting a worsening of the body bounce mode. In fact this is not due to the damper bush, but can be explained by the modifications made to install hydroelastic mounts in the rear suspension. It was necessary to remove the rear bump stops to fit the mountings, and their absence allowed much greater free travel of the rear suspension.

6.7.2 Objective road tests

Figures 104, 105 show the road test results for the same optimised mounting conditions as figures 102, 103. Once again seat rail response is used as a measure of the suspension performance. The benefit of the mounts is less marked in these tests, but an improvement over the benchmark condition can still be seen for both hydroelastic mounting configurations. The effects are limited to the secondary ride region, above 15 Hz.

The main differences between the rig and road tests are the tyre condition, longitudinal input and excitation content. For the rig tests, the tyre is not rolling; the effect of sidewall stiffness and filtering due to the contact area are changed relative to the normal operating conditions on the road. The four-poster rig applies vertical excitation only, there is no facility for replicating the longitudinal inputs experienced on the road. Finally, the rig test utilises a sinusoidal input at a single frequency and the response is allowed to settle, while the road excitation is almost exclusively random, producing predominantly transient response from the vehicle. With these considerations, the ability to identify similar trends in both rig and road tests is encouraging.
Subjective assessment of vehicle ride is considered to be an important technique for quantifying vehicle ride performance. The acid test for any proposed ride-improving product is that its effects be noticed by passengers in the vehicle. To this end a group of ride engineers made subjective assessments of each set of strut mounts fitted to the vehicle. Figure 106 shows a summary of the average ratings awarded to the same optimised configurations as presented in figures 102 - 105. The assessment panel were asked to rate the vehicle performance against a standard 1 to 10 scale (fig. 107) for noise, primary and secondary ride and general vibration.

The noise assessment was intended to ensure that the effects of high frequency stiffness were considered. This revealed a similar small increase in the noise experienced by passengers for both test installations; front only and all wheels. The primary and secondary ride characteristics were judged to have been progressively improved by the addition of front and rear hydroelastic mountings, and the general vibration, used to assess rattles, trim vibration etc. was also considered to be marginally better. The overall effect, then, was perceived to be an improvement in the primary and secondary ride of the vehicle and a small reduction in the general vibration.
6.7.4 Comparison with Model Prediction

Figure 108 shows a comparison of the model prediction and actual performance improvements measured in the vehicle tests. The improvement is presented as the vibration measured with hydroelastic mounts as a percentage of the vibration in the standard condition. The model prediction shows similar performance below 12 Hz, with a reduction in vibration between 12 and 20 Hz. This trend tallies with the results for the rig tests, the performance of the front only condition is very similar and the front and rear installation shows considerable improvement. This suggests that the mounts have been more effective than expected, though this may be attributable in part to the damping produced by the prototype being slightly greater than that predicted by the model. There is also some discrepancy around 10 Hz, since the model does not include the powertrain effects. Considering these differences, the model correlation is good.

6.8 Conclusions

It is known that the conventional suspension system exhibits design conflicts that result in a compromise in the primary and secondary ride performance of passenger cars. However it is also shown, though not explicitly, by Wallentowitz and Konik (ref. 86), Best and Middleton (ref. 13) and others that it is possible to partially resolve these conflicts with frequency dependent damping (figs. 7, 8). In this case this has been achieved by using a hydroelastic strut mounting in place of the conventional elastomer bush to increase the damping of the wheel hop mode.
This development is capable of improving the vehicle ride when fitted to the front of a vehicle only, but maximum benefit is obtained with mountings fitted to all the wheel stations. The front and rear of the vehicle interact at the frequencies associated with secondary ride, hence the mountings for front and rear suspensions must be tuned as a system for maximum benefit.

The improvement is measurable both objectively and subjectively at the front and rear of the vehicle, the improvement achieved in this study amounting to approximately half a point for primary ride and one point in secondary ride, measured on a subjective scale of 1 to 10. This is a significant gain in performance.
7 Strut Replacement Simulation

7.1 Introduction

The strut mount project is a stepping stone to the strut replacement concept. Information from the strut mount analysis, with predictive modelling, helped to indicate the feasibility of this development. The intention is to replace the conventional main spring and damper with a single hydroelastic unit performing both spring function and frequency dependent damping in a similar manner to the existing hydramount designs rather than the intended 'frequency independent' damping of a conventional suspension. It is necessary to provide two damping peaks, corresponding to primary and secondary ride resonances (figs. 7, 8), requiring a hydramount device containing a channel for each damping peak. The fact that the resonance that induces damping requires several cycles to develop is hoped to be an important feature; for a discrete event, excessive damping can make the body response worse, however damping is required when a sustained cyclic input is encountered. With suitable tuning, this could be optimised to provide increased damping of sustained input, only at critical frequencies, the remainder of the input vibration being isolated.

7.2 Modelling

The complete strut concept was initially simulated using the same quarter vehicle without engine model used for the strut mount investigation, with the modified suspension system. Further in the development, bicycle and full car vehicle models, again without engine, were used to investigate pitch and roll performance. A range of sinusoidal discrete event and continuous random input signals were used as road input.
Two different configurations were investigated, employing the same full factorial investigation and optimisation techniques used for the strut mount work. Again, the objective was to minimise body vibration, maximising the secondary ride comfort, but, ideally, to improve the primary response also, given the tuneable nature of the suspension device.

7.3 Components

The strut mount development gave an initial indication of the values for the stiffnesses and channel sizes that would be required. These were used as a starting point for the investigation of the suspension characteristics. The optimisation procedure was carried out several times for each of the concepts, and from these results it was possible to estimate the required physical dimensions of the components for each 'optimum' configuration. This provided an indication of the package requirements of the design, another important factor in the design of the suspension.

7.4 Results

The first configuration considered was that shown in figure 109. Although the performance of this arrangement was demonstrated using simulation to be an improvement over the standard suspension and presented considerable flexibility, the analysis of the parameters used to generate this configuration indicated very quickly that the practical implementation would be far too large for use in a vehicle. The major limiting factor appeared to be the use of three relatively soft compensation chambers, resulting a large volume for each. In addition, the channel dimensions required to achieve the correct frequencies were also considered difficult to implement practically.
The second configuration (fig. 110) is a considerable improvement over the first, with only two, stiffer compensation chambers of reduced volume. The addition of the diaphragm in channel configuration incurs a small increase in package requirement, and the channel dimensions were found to be more suitable. This arrangement is the subject of a patent application.

Two different sets of results were produced, representing vehicle systems optimised for minimum body acceleration for both sinusoidal and random road input conditions.

7.4.1 Sinusoidal Sweeps

A range of sinusoidal sweep inputs were used with amplitudes representing a range of road input conditions. The simulation investigation produced an optimised set of component characteristics (fig. 111), leading to the vehicle response shown in figure 112. For this input condition it was not possible to improve on the primary ride of the standard suspension system, and in fact the performance is slightly worse. There is, however, massive improvement in the secondary ride and noise regions of the body response. The wheel motion is degraded in this condition, but this is not considered to be significantly worse.

7.4.2 Random Inputs

The model was also optimised for random inputs, and in this case two 'optimum' conditions were identified. These represent a secondary ride optimised solution, and one in which the secondary ride has been compromised to improve primary response (fig. 113). In both cases, the wheel motion is degraded (fig. 114), and may begin to affect performance.
7.4.3 Discussion

The optimised characteristics for sinusoidal and random excitation are significantly different (fig. 115). This is because the most appropriate response to a continuous sinusoidal input is a high level of damping, which will take time to develop due to the growth of the channel resonance over several cycles. Similarly for a random input, a lower level of damping is acceptable due to the transient nature of the excitation, but a quicker response is required. In order to assess the impact of these differences, the 'optimised' models were rerun for sweep, random and discrete event inputs to provide direct comparison of the performance under various conditions. Figures 116 - 118 summarise the vehicle response predictions for each system, compared with the standard vehicle.

The body response to sinusoidal input clearly shows the improvement in secondary ride with all of the hydroelastic systems, however the primary ride is also seen to suffer in all cases. The sweep optimised condition appears most favourable, followed by the random 'good ride' configuration. The wheel response indicates a similar trend, though the difference between the sweep optimised and random optimised conditions is more apparent.

For random input, the trends are slightly different. The primary ride response of both the sweep optimised and random good ride configurations are an improvement over the standard vehicle, though the random good ride condition exhibits poor performance around 2.5 Hz. For secondary ride response, the random optimised systems are clearly an improvement over the conventional and sweep optimised systems, the latter being particularly bad at wheel hop. In all cases, the high frequency performance is much better than the conventional system.
Finally, a step input was used to demonstrate transient performance. Figure 118 shows the body and wheel response respectively. In all cases, the response of the hydroelastic systems are much more oscillatory than the conventional suspension, though the initial shock response is reduced.

Given these very different 'optimal' solutions, attempts were made to combine the desirable features of each, to produce an improved version. This was not successful. Two problems were identified.

Firstly, it became apparent that, due to the configuration of the device (fig. 110), the various parts of the response could not be attributed to individual parts of the system as previously expected. The 'secondary' channel contains the diaphragm whose stiffness dominates the 'primary' channel resonance. If the secondary channel is made large and free flowing, like that in the random good ride condition, then the effect of the primary channel is improved since the resistance to the fluid motion is reduced, which generates more damping from the unit. This improves the primary ride, especially for sinusoidal input, and the lower portion of the secondary ride region, to 12 Hz, due to the reduction in flow restriction.

Secondly, for any given excitation there is an optimum size for the channel controlling the resonance;

If the channel is too small, for example the primary channel for the random optimised condition, then flow through the channel will be restricted, the internal resonance within the mount will be suppressed and there will be insufficient damping to control the resonance effectively, resulting in poor primary ride performance.
More significantly, if the channel is too large and the resonance within the component is over developed, for example the random optimised conditions, then the resonance peak is 'split' into two. This occurs when the part exhibits a rate of stiffness increase with frequency which is too large, resulting in the actual stiffness and that required to produce a resonance becoming close or crossing at more than one point, as shown in fig. 119. Fig. 119 compares the actual stiffness of both conventional and hydro-elastic suspensions with the stiffnesses required to produce body and wheel resonances over a range of frequencies. Considering the wheel hop mode, for example, the conventional suspension and required stiffness coincide at only one point, resulting in a well defined wheel hop frequency. For the hydroelastic suspension, however, the actual stiffness is close to the required stiffness over a wide range of frequencies. In the central region, the system is well damped by the hydramount damping characteristic, resulting in a controlled response, but the outlying regions are not well controlled, since the hydramount damping is confined to a narrow band of frequencies, and produces a double peak, two poorly damped resonant regions separated by a well damped one.

Both the 'under damped' condition, when the channel is too small, and the 'over damped' condition, when the channel is too large, can occur within the same part, if different excitation conditions apply. This may be a different character of excitation, the change from sinusoidal to random input, or change in the amplitude of excitation, a rough or smooth road. Obviously a vehicle will encounter road input consisting of a wide range of characteristics and amplitudes; this then is a major limitation of this technology.
Simulations were also performed to investigate the pitch and bounce response of a vehicle fitted with the hydroelastic suspension units. Figure 120 shows the pitch and bounce response of the vehicle to a random road input. The main problem occurs with the roll response of the body. The natural frequency of the roll mode is significantly higher than the bounce or pitch modes, and as such it falls outside the region in which the suspension units produce their maximum damping. As a consequence, the roll mode is not well controlled, building up considerable motion from even a small input. It is not possible, using the dynamic absorber technology described, to broaden the scope of the damping peak to cover all the body modes.

7.5 Conclusions

The concept presented demonstrates the excellent potential for improving vehicle ride using suspension components with frequency dependent damping characteristics. The lower secondary ride and noise frequency ranges show enormous improvement over the standard suspension.

It also highlights problems with the proposed system, namely the sensitivity to the form and amplitude of excitation, the limited frequency range of the damping region and the detrimental effect of the stiffness rise associated with the damping. An extension of the analysis by Bosworth and Fursdon (ref. 66), who show that peak damping is proportional to the stiffness rise, indicates that this problem will become progressively worse as the natural frequency of the system is reduced;

\[(29) \quad \omega_n = \sqrt{\frac{K_s}{m}} \quad (30) \quad C_{\text{crit}} = \sqrt{\frac{K_s}{m}} \]
hence $K_2$, stiffness rise due to damping is given by,

\[(31) \quad K_2 \propto W_n \cdot m\]

hence,

\[(32) \quad \frac{K_2}{K_s} \propto \frac{1}{W_n}\]

where,

- $C_{crit}$ = Critical Damping (Ns/m)
- $m$ = Mass (Kg)
- $K_2$ = Increase in stiffness (N/m)
- $K_s$ = Static Stiffness (N/m)
- $W_n$ = Natural Frequency (rads/s)

It follows that, for a given proportion of critical damping, the associated stiffness rise must be a greater proportion of the static stiffness as the resonance frequency is reduced and hence, a greater spread of the resonance peak or peaks will occur. This is certainly a problem which will affect all efforts to control primary ride with dynamic absorbers.

The roll response of the vehicle has also been highlighted as a problem, and a separate means of controlling the roll mode will be required for any practical system.
The second configuration exhibits an additional compromise between the primary and secondary channel functions, as described earlier. This reduces the scope for tuning the device, resulting in either good primary control or good wheel hop isolation on random surfaces. This compromise is not applicable to the first configuration with separate channel and compliance components, however this system is not practical because of package considerations.

Finally, the simulated response to transient input is poor for both conditions, oscillations persisting for some time after the excitation.
8 Conclusions

8.1 Vehicle and Component Testing

8.1.1 Rig Testing

Rig testing a complete vehicle using the four post test facility proved to be a useful technique, allowing rapid progress in a controlled environment. The simple sinusoidal testing is too basic for confident prediction of the effects of fine tuning on secondary ride performance on the road, but allows the assessment of a change to the vehicle provided it is reasonably large. This was useful in quickly finding a configuration close to the desired characteristic for further tuning on the road. The major obstacles to further use of the rig appear to be the non-rolling tyre, reproduction of road input and limitations of rig response. All these can be compensated for, at least partially, with more sophisticated control systems for the test rig.

8.1.2 Road Testing

As expected, road testing was more variable than the rig test work, with the majority of the variation accounted for by the road chosen and vehicle speed. The results were found to be very repeatable provided these influences were well controlled. The actual operating environment produced slightly different results to the rig tests; large differences in vehicle condition were ranked in the same order of preference, however smaller changes did not necessarily follow the same ranking. This confirmed the view that rig testing is a valuable tool for rapid progress, but does not dispense with fine tuning work carried out on the road.
8.1.3 Subjective testing

This is the most variable test procedure used in the evaluation of the vehicles tested, and also the one used by many car manufacturers for final approval of a vehicle. In addition to the physical experimental variation between successive tests, it is also influenced by personal preference, build, posture and physical condition of the test subject. It is necessary to use a large group of trained assessors, aware of the issues involved. 'Blind' testing is harder for the subjects, but removes much of the psychological bias produced by 'knowing the right answer'. Frequent use of a control condition helps the analyst to compensate for 'drift' in the data as the subjects tend to become more critical as the optimisation progresses. Subjective opinion can be very useful, especially if the subject relates more than a simple 'marks out of ten' grade. Knowing which body part or parts are affected, approximately the frequency this represents and an indication of its source is very valuable. These responses will vary from subject to subject.

8.1.4 Components

It is important to use the correct range of amplitudes, particularly when testing components which have strongly nonlinear characteristics. It has been useful, where possible, to use measurements of the standard vehicle to obtain an estimate of the operating conditions of the components concerned. In the future it will be desirable to test the response of components to composite excitation in addition to the sinusoidal sweep tests performed in this work as a means of investigating the nonlinear response. Verification of the individual components was also beneficial in validating the whole vehicle model, serving as a means of eliminating the initial simulation problems.
8.2 Simulation

8.2.1 Vehicle

This work has shown that a very good representation of a vehicle can be produced with accurate vehicle and component data. Small errors in the model parameters are capable of causing a large discrepancy in the simulated response. With good data, it is possible to simulate the vehicle accurately up to 20 Hz using the techniques described. The limiting factor in this work has been the rigid body assumption for the vehicle structure. The first bending mode of the chassis in this case is approximately 20 -25 Hz, corresponding well with the frequency at which the seat rail response predictions become inaccurate. Other parts of the vehicle, not influenced by body bending, are still represented accurately to higher frequencies. The scope of this model is sufficient for the work carried out here, but will require improvement for simulation of high frequency response.

Accurate simulation of the vehicle on a road profile remains difficult; primarily due to the lack of road data. This may be partially addressed in the future using improved four post rig control systems to replicate road input from road load data, the road input signals can then be used for rig testing and simulation purposes alike.
8.2.2 Components

Accurate representation of components is vital in achieving a good simulation of the whole vehicle. Both stiffness, damping and transient response are important features. It is also important to check everything as far as possible; even the most basic data supplied by vehicle manufacturers is not necessarily reliable. The models used are often complex when all the characteristics of the vehicle and components are included. This leads to considerable time and computation. It is acceptable to use simpler models of both vehicle and components for approximate work, refining the simulation with sophisticated features for confirmation and fine tuning work provided the approximations inherent in this approach are understood.

8.3 Strut Mount Simulation, Design and Implementation

The strut mount simulation and vehicle testing correlate well, giving further confidence in the modelling techniques used. The strut mount itself was also successful in demonstrating secondary ride improvement both objectively and subjectively. Design rules specific to the strut mount concept have been developed and added to the prior knowledge of hydramount technology. The strut mount is reasonably small; for prototype trials it was fitted to a vehicle with minimal disturbance. Further development of the concept, using data gathered during this work has been used to reduce the size further and it can be fitted to an existing vehicle design or new concept with relative ease. The technology used is 'proven' on engine mounting systems.
8.4 Hydro-elastic Suspension

This development is more radical. The simulations performed in this work demonstrate the potential of a strongly frequency dependent damping device, but also reveal problems of roll mode suppression, packaging within the vehicle and sensitivity to variation in operating condition. The roll control and sensitivity could be overcome using adaptive control and additional features, but this would detract from one of the major features of the system, its simplicity. Packaging the suspension within the vehicle remains a problem even when a fully developed design is considered.

8.5 Recommendations for Future Work

Subframe and suspension bushes, suspended seats, semi-active and active mount technology have a great deal to offer in the development of secondary ride characteristics of vehicles. These technologies have been purposely avoided in this work because of ongoing investigation within Avon Vibration Management Systems and other organisations. The main opportunities for building on this work are as follows;

8.5.1 Strut Mount

Work is ongoing in conjunction with various customers to add strut mountings to existing products and new vehicle concepts. Some package reduction has been achieved and further gains may be made. The development of additional features is desirable, particularly switching technology, adaptive control and high frequency noise isolation. These technologies are also well developed for engine mountings, providing a source of experience and knowledge.
8.5.2 Suspension

The concept presented here can be further developed. The additional component to control roll response and reconfiguring the design to improve packaging remain as problems to be overcome. The advantages of frequency dependent damping have been demonstrated; investigation of alternative means of achieving this scheme should be pursued, perhaps a frequency dependent damper, or a new controller concept for semi-active suspension will provide a solution more applicable to a motor vehicle.

8.5.3 Interconnected Systems

Interconnected engine mounting and suspension systems have been a popular topic in the literature in the recent past, but have not yet appeared in production. Researchers claim good results and if effective such a system would be easily fitted to a front wheel drive vehicle. Limited simulation work suggests that these systems are ineffective, which could explain their lack of popularity, however this remains an area of interest which should be investigated further.

8.5.4 Component Testing

The tyre and damper have been identified in this and other work as critical components affecting vehicle ride. The characteristics of these components have been investigated, with some success, leading to improved models for secondary ride analysis. There are still some features of each which have not been investigated. The effect of rolling on the tyre vertical characteristic and the effect of temperature on the damper characteristic. Further work in these areas would produce a more complete picture of their behaviour.
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Fig. 1 'Flat Ride' Response Diagrams

Parallel Sprung Vehicle
front and rear static defns. equal

Flat Ride Concept Vehicle
front static defn. greater than rear

Note the reduced pitch response for the 'Flat Ride' Vehicle.
Fig. 2 Wheelbase Filtering Effects

Response of front and rear of vehicle

Note the effect of interference between front and rear inputs; when the two are in phase, a local maximum occurs, when they are out of phase, a minimum occurs.
Fig. 3 Human Body Response Weightings

For the purposes of ride analysis, the frequency weightings are applied as follows:

- $W_b$ - z - seat, x, y, z - feet,
- $W_c$ - x - back
- $W_d$ - x, y - seat, y, z - back
- $W_e$ - Rx, Ry, Rz - seat
- $W_f$ - z - motion sickness
Fig. 4 Road Profile - Major Road

PSD [m^3/cycle]

Fig. 4 Road Profile - Minor Road #1

PSD [m^3/cycle]

Fig. 4 Road Profile - Minor Road #2

PSD [m^3/cycle]

Application of Hydraulic Engine Mounting Methods to Primary Suspension
**Fig. 5 Road Input Characteristics**
Parallel Track Coherence

**Fig. 6 Road Displacement Measurement**

Displacement (m)
Fig. 7  Effect of Damper and Bush Rate on Transmissibility

Comparison of heavy and light damping  
Comparison of soft and stiff bush for 0.1 m/s input

Body Acceleration (m/s²)

![Graph showing body acceleration for different damping and bush rates.]

Body Acceleration (m/s²)

![Graph showing body acceleration for different damping and bush rates.]

Fig. 8  Effect of Damper and Bush Rate on Tyre Force

Comparison of heavy and light damping  
Comparison of soft and stiff bush for 0.1 m/s input

Dynamic Tyre Force (N)

![Graph showing dynamic tyre force for different damping and bush rates.]

Dynamic Tyre Force (N)

![Graph showing dynamic tyre force for different damping and bush rates.]

Application of Hydraulic Engine Mounting Methods to Primary Suspension
**Fig. 9 - Rising Rate Coil Spring**

As the coil is compressed, successive coils become bound, increasing the rate of the remaining coils.

**Fig. 10 - Rising Rate Leaf Spring**

As the leaf is compressed, the main leaves contact the auxiliaries, increasing the total rate of the spring.

---

**Fig. 11 Air Spring Characteristics**

**Stiffness**

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<th>Stiffness (N/mm)</th>
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**Pressure / Volume Compliance**

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<td>60</td>
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Note constant natural frequency over a wide frequency range for the single convolution bellows.
Fig. 11 Air Spring Characteristics

Natural Frequency of Supported Mass

- twin convolution  + single convolution

Double Convolution Bellows

Single Convolution Bellows

Effective Diameter
Fig. 12 Hydragas Wheel Rates

A comparison of pitch, bounce and roll rates with those of a conventional suspension.

Fig. 12 Section of Hydragas Unit

Application of Hydraulic Engine Mounting Methods to Primary Suspension 122
Fig. 13 Damper Construction & Flow Paths

Base Valve
Damping

Piston Valve
Non Return

Base Valve
Non Return

Piston Valve
Damping

Twin Tube Damper

Fig. 14 Damper Construction & Flow Paths

Gas

Piston Valve
Damping

Mono Tube Damper
Fig. 15 Damper Characteristics
Typical Published Characteristics

Fig. 15 Typical Damper Hysteresis Loop

Fig. 16 Damper Performance
Body Response to Pothole (unfavourable)

Fig. 16 Damper Performance
Body Response to Bump (favourable)
Fig. 17 - Common Engine Mounting Systems

A. 3/4 point system - used on most RWD cars

B. Transverse FWD - mass carriers and roll restrictor(s)

C. Transverse FWD - mass carriers and tie rod(s)

D. Audi FWD with longitudinal engine
Fig. 18 Typical Hydramount Construction

Typical Hydramount characteristics; note greater damping with small amplitude excitation, and high dynamic stiffness.
Fig. 18 Hydramount With Large Flow Channel

Hydramount with large flow channel; note greater loss angle and lower dynamic stiffness.
Fig. 18 Hydramount With Avon Annular Diaphragm

Hydramount with patented annular diaphragm; note lower high frequency stiffness compared with that of competitor parts.
Fig. 19 Active Engine Mount

Active engine mount comprising electromagnetic actuator and seismic mass, used to modulate the passive hydraulic mount positioned below the actuator.
Fig. 20 Switching Hydramount #1

Switching Hydramount #1; the mount has two conditions, a ride mode (air vent blocked) in which the damping rate is high, to improve ride, and an idle mode (air vent open) in which the dynamic stiffness is minimised.

\[ m \pm 0.3 \text{ mm} \]

\[ \text{STATIC RATE 150 N/mm} \]

\[ \pm 0.3 \text{ mm} \]

\[ \text{FREQUENCY (Hz)} \]

Application of Hydraulic Engine Mounting Methods to Primary Suspension 130
Switching Hydramount #2; the mount has two conditions, a ride mode (air vent blocked) in which the damping rate is high, to improve ride, and an idle mode (air vent open) in which the second channel becomes active, producing a minimum stiffness at idle vibration frequency.
The decoupled hydramount characteristic is designed to provide damping for large amplitude excitation, and low dynamic stiffness for small amplitude excitation, due to the snubbed diaphragm.
Fig. 22 Decoupled Hydrabush

The decoupled hydrabush is an enclosed bush development of the decoupled hydramount concept, with similar amplitude dependent properties.
For the simple case of pure side slip or pure longitudinal slip the magic tyre formula may be used. With appropriate experimentally determined parameters the formula expresses side force, aligning moment and longitudinal force in terms of slip angle and longitudinal slip. The general form of the formula, applicable to a single vertical load and camber angle, and the characteristic with coefficients indicated, are shown.

\[
y(x) = D \left( C \arctan(Bx - E(Bx - \arctan(Bx))) \right)
\]

where, \( y(x) = D \left( C \arctan(Bx - E(Bx - \arctan(Bx))) \right) \)
Fig. 24 Hysteresis loops for .01 & 0.1 % Shear

Fig. 24 .01 % at 100 Hz superimposed on 0.1 % at 10 Hz
Fig. 25 Damper With Self-Energising Levelling System

damping valves
high pressure chamber
diaphragm
pump plunger
pump outlet valve
level control port
pump inlet valve
low pressure chamber
piston rod
overload valve

Fig. 26 Karnopp Control Law for Two State Damper

Absolute Body Velocity

Soft

Hard

Relative Velocity (Body to Wheel)

Soft

Hard
Fig. 27 PAWS Passive Suspension Schematic

Fig. 28 Variable Leverage Suspension

Application of Hydraulic Engine Mounting Methods to Primary Suspension
Fig. 29 Slow Active Suspension

Body

Active Suspension
Actuator

Passive
Suspension

Wheel

Tyre

Road Input
Fig. 30  Active Suspension

High Frequency Harshness

Accel. PSD (m/s^2/Hz)

Fig. 31  Active Roll Control System
Fig. 32 Comparison of Seat Performance

Transmissibility

- Rigid Seat
- Conventional Seat
- Suspension Seat
Fig. 33 Single Degree of Freedom Model

Fig. 34 Quarter Vehicle Model

Application of Hydraulic Engine Mounting Methods to Primary Suspension
Fig. 35  Half Vehicle Model

Fig. 36  Full Vehicle Model
Fig. 37 Quarter Vehicle Model With Engine and Passenger

Fig. 38 Vehicle Model Configuration

- Text Editor
- Signal Generator Software
- Vehicle Characteristics & Initial Conditions
- Road Input
- Steering Input
- Solver
- Output File
- Time History
- Signal Processing Routines
- Spreadsheet
- Processed Signals
- Presentation
Fig. 39 Model Internal Function

1. Read Data and Initial Conditions
2. Use position, velocity, stiffness and damping to calculate forces
3. Reduce forces to a single force and moment per mass
4. Calculate response of masses
5. Integrate for velocities and positions
6. Write required variables to output data file
Fig. 40 2 Mass Model Validation - Analytical Solution

Fig. 41 3 Mass Model Validation - Analytical Solution

Application of Hydraulic Engine Mounting Methods to Primary Suspension
Fig. 42 Initial Vehicle Validation Results

Seat Rail Response Vehicle #1
comparison of simulation & experiment

Seat Rail Response Vehicle #2
comparison of simulation & experiment

Seat Rail Response - Vehicle #3
comparison of simulation & experiment
Fig. 43 Validation Investigation
damper tolerance effects

Body Acceleration (RMS g)

Vehicle
Sim. -ve tolerance
Sim. nominal
Sim. +ve tolerance

Frequency (Hz)

Fig. 44 Maxwell Style Models for
Rubber and Tyre Components

Generalised Equation

\[ F = Kx + Cx^n \]

Note: for the usual linear Maxwell model, \( n = 1 \)
Note: Each component was tested at a range of velocities (m/s), as indicated in the legend.
Fig. 48 Tyre Radial Stiffness

Continental CV90 195/65 R 15 V @ Inflation Pressures 2.4, 2.7 & 3.0 Bar

Radial Deflection (mm)

Fig. 49 Typical Cornering Stiffness

Side Force (N)
Fig. 50 Measured Tyre Characteristic vs Dunn & Olatunbosun model

Fig. 50 Measured Tyre Characteristic vs Dunn & Olatunbosun model

Fig. 51 Wheel and Tyre in Test Fixture
Fig. 52 Tyre Stiffness Investigation
variation with preload & excitation freq.

Fig. 53 Tyre Stiffness Investigation
variation with inf. pres. & exc. freq.

Fig. 54 Tyre Stiffness Investigation
variation with exc. ampl. & freq.

Fig. 55 Tyre Loss Angle
variation with exc. ampl. & freq.
Fig. 56 Validation of Enhanced Tyre
Model - Stiffness

Fig. 56 Validation of Enhanced Tyre
Model - Loss Angle

Fig. 57 Front Damper Characteristics
Dynamic Stiffness

Fig. 57 Front Damper Characteristics
Loss Angle

Fig. 58 Rear Damper Characteristics
Dynamic Stiffness

Fig. 58 Rear Damper Characteristics
Loss Angle

Note: Each component was tested at a range of velocities (m/s), as indicated in the legend
**Fig. 59 Damper Model Internal Function**

1. **Read Data and Initial Conditions**
2. **Use position and remaining volume to calculate "valve volume"**
3. **Use "valve volume" and valve stiffness to calculate pressure and hence force**
4. **Use Pressure to calculate valve opening and flow through it**
5. **Integrate flow for remaining volume**
6. **Return data to main model structure**
Fig. 60 Front damper Validation

Dynamic Stiffness

Simulation — Actual  — 0.01
- 0.05  — 0.1

Fig. 61 Rear damper Validation

Dynamic Stiffness

Simulation — Actual  — 0.01
- 0.05  — 0.1

Fig. 60 Front damper Validation

Loss angle

Simulation — Actual  — 0.01
- 0.05  — 0.1

Fig. 61 Rear damper Validation

Loss angle

Simulation — Actual  — 0.01
- 0.05  — 0.1
Fig. 62  Hydramount Characteristics

Contributions of Components and Total Effect

Out of Phase Stiffness (N/m)

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<td>Total Dynamic Stiffness, Kd</td>
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In Phase Stiffness (N/m)
Application of Hydraulic Engine Mounting Methods to Primary Suspension

Fig. 63 Engine Mount Validation

- Simulation
- Actual
- 0.1 mm
- 0.3 mm
- 0.5 mm

Fig. 64 Engine Mount Validation

- Simulation
- Actual
- 0.1 mm
- 0.3 mm
- 0.5 mm

Fig. 65 Engine Mount Validation

- Simulation
- Actual
- 0.1 mm
- 0.3 mm
- 0.5 mm
Fig. 66 Hydramount Model Construction

Fig. 67 Hydramount Schematic With Diaphragm in Channel

Application of Hydraulic Engine Mounting Methods to Primary Suspension 157
Fig. 68 Analysis of Road Input Signal
Time & Freq. Distribution (% of Max.)

Fig. 69 Discrete Event Profiles
Positive 1-Cos Bump

Fig. 69 Discrete Event Profiles
Negative 1-Cos Bump

Fig. 70 Discrete Event Profiles
Positive Step

Fig. 70 Discrete Event Profiles
Negative Step
Fig. 71 Validation Investigation
Vehicle validation model #1

Fig. 72 Vehicle Model Validation
Configuration 1 - seat rail comparison
Fig. 73 Vehicle Model Validation
Configuration 2 - seat rail comparison

- actual - model

Fig. 74 Vehicle Model Validation
Configuration 3 - seat rail comparison

- actual - model
Configuration 1 - chassis comparison

Configuration 1 - wheel comparison

Configuration 2 - wheel comparison

Configuration 3 - wheel comparison
Fig. 81 Hydroelastic Damper Mount
Comparison of Secondary Ride Effects

Frequency (Hz) [0, 5, 10, 15, 20]
Body Response Gain

Conventional Mount  Hydroelastic Mount

Fig. 82 Combined Characteristics
Std. damper / 20 Hz tuned strut mounts

Dynamic stiffness (N/m)
frequency (Hz) [0, 5, 10, 15, 20, 25, 30]

-0.01 -0.02 -0.03 -0.05 -0.07 -0.10

Fig. 83 Hydraulic Strut Mount
comparison of tests & models

Dynamic stiffness (N/m)
frequency (Hz) [0, 5, 10, 15, 20, 25, 30]

Fig. 83 Hydraulic Strut Mount
comparison of tests & models

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Fig. 84 - Strut Mount Prediction
required strut mount dynamic stiffness

Fig. 84 - Strut Mount Prediction
required strut mount loss angle
Fig. 85 Test Vehicle

Fig. 86 Vehicle Instrumentation

Front Strut Tower
Damper Rod
Rear Strut Tower
Front Hub  Seat Rail  Rear of Sill  Rear Hub
Fig. 94 Prototype Strut Mount Section
Fig. 95 Prototype Strut Mount Stiffness

Fig. 95 Prototype Strut Mount Loss Angle
Fig. 96 Front Strut Installation

Fig. 97 Rear Strut Installation

Fig. 98 Rear Rift Bulk Mount
Fig. 99 Strut Mount / Damper Assembly

Fig. 100 Rear Damper Length Adjustment

Fig. 101 Strut Mount / Damper Assembly
Fig. 102 Comparison of rig results
seat rail - mounts fitted front only

Fig. 103 Comparison of Rig Results
Seat Rail - mounts fitted front & rear
Fig. 104 Comparison of road results
seat rail - mounts fitted front only

Fig. 105 Comparison of road results
seat rail - mounts fitted front & rear
Fig. 106 Comparison of road results
Optimised conditions subjective rating
Fig. 107 Subjective Assessment Form

### SUBJECTIVE VEHICLE ASSESSMENT

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#### RATING SYSTEM

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NVR rating

- no trace
- trace
- light
- very light
- method
- requests improvement
- barely acceptable
- borderline
- customer complaint
- poor

At spin by all customers

- average customer
- critical comments
- trained observer
- perceived
Fig. 108 Model prediction comparison
Vibration reduction relative to std.

- model
- rig front
- rig front & rear

% of original vibration

Frequency (Hz)

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Application of Hydraulic Engine Mounting Methods to Primary Suspension

Fig. 109 Suspension Schematic
Configuration 1

Fig. 110 Suspension Schematic
Configuration 2

Secondary channel with diaphragm
Secondary channel with diaphragm in channel
Primary Channel
Bellows
Working Chamber
Top

Diaphragm with vented diaphragm

Secondary channel with vented channel
Application of Hydraulic Engine Mounting Methods to Primary Suspension
Application of Hydraulic Engine Mounting Methods to Primary Suspension
The 'body res.' and 'wheel res.' curves represent the stiffness required to cause a body or wheel mode respectively, over the frequency range shown. Note that the conventional suspension indicates well defined resonance frequencies, however with the hydroelastic system proposed, the system will be close to resonance across a range of frequencies.
Application of Hydraulic Engine Mounting Methods to Primary Suspension
active suspension - A suspension containing one or more actuators governed by a control system and capable of providing or dissipating significant power.

adaptive suspension - A suspension system in which the characteristics can be changed, but only very slowly, for example a system in which the damper characteristic is selected at the beginning of a journey based upon measurements of the vehicle condition.

amplitude decoupled - A particular nonlinear feature, especially of hydraulic mounts, whereby the response to small amplitude excitation is lightly damped and soft, however large amplitude excitation provokes a high level of damping.

bandwidth - The range of frequencies over which a device operates - in the context of vehicle suspension, usually the maximum frequency to which an actuator or system will respond.

boom - An occurrence of sudden noise increase within the vehicle cabin, usually caused by coincidence of engine excitation frequency, body cavity resonance and panel modes.

broadband - Refers to device with a wide bandwidth, ie. device responds to high and low frequencies.

carbon black - Finely divided carbon powder used as a reinforcing filler in the production of rubber. Imparts strength, wear resistance and damping properties.
clipped Karnopp control - A semi-active control technique applicable to multiple state and variable rate dampers, in which the controller attempts to emulate an active suspension. The damping rate is set to provide equal force to the active suspension, except when power input is required, when minimal damping is selected.

contact patch - The area of the tyre tread which is in contact with the road.

cornering stiffness (tyre) - The ratio of side force to slip angle for a tyre, measured within the linear operating range.

dynamic stiffness - The magnitude of the complex stiffness of a component.

dynamic tyre load - The vertical load applied to the tyre in addition to the static load of the vehicle weight.

effective area - A property of the geometry of a hydramount top or similar fluid pumping element. Equal to the area of a piston displacing the same volume of fluid per unit deflection.

effective mass - The mass required for a dynamic absorber to produce a similar damping effect to the hydramount.

elastic stiffness - The component of stiffness which is in phase with the excitation.

electrorheological fluid - A fluid whose viscosity can be controlled by the application of an electrical potential difference.
engine shake - A low frequency (6-12 Hz) vibration problem associated with the vibration of an engine on its mountings due to road input. Becomes most noticeable when a cabin panel, fitting or facia resonates at the same frequency.

frequency domain analysis - Analysis of a component or system in terms of its response to frequency only - efficient method of analysing steady state response of linear or linearised systems.

glass transition temperature - the temperature below which intermolecular movement in an elastomer is suppressed, producing a rapidly increasing modulus and ultimately brittle, glass-like properties.

harshness - High frequency vibration and noise problems encompassing boom, rattles, gear noise etc.

hydrabush - A fluid filled bush type mounting which provides damping tuned to a predetermined frequency by virtue of resonance of a mass within the mount.

hydramount - A fluid filled mounting which provides damping tuned to a predetermined frequency by virtue of resonance of a mass within the mount.

hydropneumatic suspension - A suspension containing both liquid and gaseous media, usually incompressible fluid to transmit forces and compressible gas under pressure to provide a spring action.

idle shake - An engine induced vibration problem, similar to engine shake, but occurring when the engine idles. This is more common with diesel engines or under poor combustion conditions, being driven by the engine half order excitation.
lateral stiffness (tyre) - The stiffness of the tyre along its axis due to deflection of the side walls in that direction.

limit handling - The handling behaviour of the vehicle in the nonlinear region of response, usually high lateral acceleration conditions due to side slip or high speed cornering.

loss angle - The Phase angle between the dynamic stiffness response and the applied deflection excitation.

mobility - Mechanical admittance. The ratio of velocity and force; point mobility is the ratio of velocity and force at a point, cross mobility is the ratio of velocity and force at two different points. Analysis by a mobility method allows prediction of a system response based upon the mobilities of individual components.

passive suspensions - Suspensions containing elements capable of storing and dissipating energy only, and with fixed characteristics.

power spectral density - Power Spectral Density is a means of assessing the frequency content of a signal. The PSD is the square of the signal per unit frequency, and originates from electrical usage, hence the reference to power (electrical power is proportional to voltage squared).

powertrain - In the context of ride analysis, the powertrain is the flexibly mounted 'rigid' mass consisting of the engine, gearbox, associated ancillaries, and in most front wheel drive installations, the final drive unit.

primary ride - Vibration of the vehicle body below 4 Hz as a result of the interaction of road excitation and rigid body motion of the vehicle bodyshell.
scrub (tyre) - Scrub is a property of the suspension system; the lateral movement of the tyre contact patch with change in the vertical deflection of the suspension, due to linkage geometry.

secondary ride - Vibration of the vehicle body in the range 4 - 20 Hz as a result of the body response to road induced rigid body vibration of the powertrain, wheels, subframes etc.

second order transition temperature - The temperature at which the behaviour of an elastomer changes from a brittle, glass-like response to a rubbery behaviour.

semi-active suspension - Suspensions containing elements capable of storing and dissipating energy only, but with variable characteristics under automatic external control.

slip angle - The angle formed between the plane of a wheel, and its direction of travel. A slip angles occur during cornering and side slip conditions, and result in side forces and moments at the contact patch.

slow active suspension - An active suspension in which the actuator has narrow band width, responding to low frequency demand signals only; hence 'slow'.

sky hook damper - A fictional damper installed between the vehicle body and a fixed, horizontal reference plane. The effect of this imaginary damper can be partially simulated with active suspension.
static deflection - The deflection of a component under static conditions. In the case of vehicle suspension systems, this parameter relates stiffness to supported mass (weight) and hence indicates natural frequency.

static stiffness - The stiffness of a component measured under static conditions - the gradient of a load - deflection curve

strut mount - An elastomer mounting fitted between the suspension strut and body.

suspension ratio - The ratio of suspension spring load to vertical wheel load.

triaboelastic - Description of a material exhibiting a combination of frictional damping and elastic spring properties.

vehicle ride - Vehicle ride is the response of the sprung body containing passengers to excitation of the vehicle by road roughness.

viscoelastic - Description of a material exhibiting a combination of viscous damping and elastic spring properties.

wave number - Wave number is the spatial equivalent of frequency, i.e. cycles per metre. This is commonly used for road characterisation since it is independent of vehicle speed.

wheelbase filtering - The wheelbase filter effect arises from the time shift between excitation of the front and rear wheels of a vehicle. For any given speed, certain discrete frequencies result in only bounce excitation, and others only pitch.
### Appendix B - Vehicle Parameters

#### Masses (Kg)

<table>
<thead>
<tr>
<th>Component</th>
<th>Mass (Kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Body</td>
<td>856</td>
</tr>
<tr>
<td>Powertrain</td>
<td>297.4</td>
</tr>
<tr>
<td>Front Wheel</td>
<td>44.5</td>
</tr>
<tr>
<td>Rear Wheel</td>
<td>37.5</td>
</tr>
<tr>
<td>Subframe</td>
<td>14</td>
</tr>
</tbody>
</table>

#### Inertia Properties (kg m²)

<table>
<thead>
<tr>
<th>Component</th>
<th>Roll</th>
<th>Pitch</th>
<th>Yaw</th>
</tr>
</thead>
<tbody>
<tr>
<td>Body</td>
<td>299</td>
<td>1210</td>
<td>1350</td>
</tr>
<tr>
<td>Powertrain</td>
<td>11</td>
<td>11.75</td>
<td>26</td>
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<tr>
<td>Front Wheel</td>
<td>3</td>
<td>5.2</td>
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</tr>
<tr>
<td>Rear Wheel</td>
<td>3</td>
<td>4.8</td>
<td>3</td>
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<tr>
<td>Subframe</td>
<td>1.3</td>
<td>0.5</td>
<td>1.8</td>
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</table>

#### Centre of Gravity Positions (m)

<table>
<thead>
<tr>
<th>Component</th>
<th>X</th>
<th>Y</th>
<th>Z</th>
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<tr>
<td>Body</td>
<td>0</td>
<td>0</td>
<td>-0.589</td>
</tr>
<tr>
<td>Powertrain</td>
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<td>0.004</td>
<td>-0.438</td>
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<tr>
<td>L/H Front Wheel</td>
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<td>-0.7614</td>
<td>-0.310</td>
</tr>
<tr>
<td>R/H Front Wheel</td>
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<td>0.7314</td>
<td>-0.310</td>
</tr>
<tr>
<td>L/H Rear Wheel</td>
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<td>-0.310</td>
</tr>
<tr>
<td>R/H Rear Wheel</td>
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</tr>
<tr>
<td>Subframe</td>
<td>1.1894</td>
<td>-0.017</td>
<td>-0.410</td>
</tr>
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</table>
### Mounting Coordinates

<table>
<thead>
<tr>
<th>Component</th>
<th>X</th>
<th>Y</th>
<th>Z</th>
</tr>
</thead>
<tbody>
<tr>
<td>L/H Engine Mount</td>
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<td>-0.2873</td>
<td>-0.6158</td>
</tr>
<tr>
<td>R/H Engine Mount</td>
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<td>0.2908</td>
<td>-0.6158</td>
</tr>
<tr>
<td>L/H Gearbox Mount</td>
<td>1.13</td>
<td>-0.4215</td>
<td>-0.2203</td>
</tr>
<tr>
<td>R/H Gearbox Mount</td>
<td>1.13</td>
<td>0.3915</td>
<td>-0.2203</td>
</tr>
<tr>
<td>L/H/F Subframe Mount</td>
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<td>-0.4215</td>
<td>-0.19925</td>
</tr>
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<td>R/H/F Subframe Mount</td>
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<td>0.3915</td>
<td>-0.19925</td>
</tr>
<tr>
<td>L/H/R Subframe Mount</td>
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</tr>
<tr>
<td>R/H/R Subframe Mount</td>
<td>0.962</td>
<td>0.295</td>
<td>-0.2275</td>
</tr>
</tbody>
</table>

**Tyre Stiffness** 340000 N/m

**Front Suspension**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stiffness</td>
<td>29000 N/m</td>
</tr>
<tr>
<td>Damper Bush Stiffness</td>
<td>250000 N/m</td>
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</tbody>
</table>

**Damper Parameters**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seal Stiffness</td>
<td>50000 N/m</td>
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<tr>
<td>Seal Friction</td>
<td>60 N</td>
</tr>
<tr>
<td>Piston Area</td>
<td>1200e-6 m&lt;sup&gt;2&lt;/sup&gt;</td>
</tr>
<tr>
<td>Fluid Density</td>
<td>7000 Kg/m&lt;sup&gt;3&lt;/sup&gt;</td>
</tr>
<tr>
<td>Suspension Ratio</td>
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</tr>
<tr>
<td><strong>Compression Valve</strong></td>
<td></td>
</tr>
<tr>
<td>----------------------</td>
<td>--</td>
</tr>
<tr>
<td><strong>Radius</strong></td>
<td>0.03 m</td>
</tr>
<tr>
<td><strong>Volumetric Spring Stiffness</strong></td>
<td>3.9e11 N/m⁵</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Extension Valve</strong></th>
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</tr>
</thead>
<tbody>
<tr>
<td><strong>Radius</strong></td>
<td>0.06 m</td>
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<tr>
<td><strong>Volumetric Spring Stiffness</strong></td>
<td>11e11 N/m⁵</td>
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</table>

Simulating Damper Characteristic (see fig. 60)

<table>
<thead>
<tr>
<th><strong>Rear Suspension</strong></th>
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</thead>
<tbody>
<tr>
<td><strong>Stiffness</strong></td>
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<tr>
<td><strong>Damper Bush Stiffness</strong></td>
<td>250000 N/m</td>
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</table>

<table>
<thead>
<tr>
<th><strong>Damper Parameters</strong></th>
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<tbody>
<tr>
<td><strong>Seal Stiffness</strong></td>
<td>50000 N/m</td>
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<tr>
<td><strong>Seal Friction</strong></td>
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<td><strong>Piston Area</strong></td>
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<td><strong>Fluid Density</strong></td>
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</table>
Compression Valve

Radius 0.03 m
Volumetric Spring Stiffness 3.9e11 N/m$^5$

Extension Valve

Radius 0.06 m
Volumetric Spring Stiffness 11e11 N/m$^5$

Simulating Damper Characteristic (see fig. 61)

Subframe Mount Vertical Stiffnesses (N/m)

Front 150000 Rear 700000

Front Engine Mounting

Vertical Stiffness 180000 N/m

Working Chamber Volumetric Stiffness 1.8 e10 N/m$^5$
Working Chamber Effective Area 3750e-6 m$^2$
Fluid Density 1000 kg/m$^3$
Channel Length 0.295 m
Channel Area 100e-6 m$^2$
### Rear Engine Mounting

<table>
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<td>Working Chamber Volumetric Stiffness</td>
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<tr>
<td>Fluid Density</td>
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<td>Channel Area</td>
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### Appendix C - Strut Mount Characteristics

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<td>Static</td>
<td>Dynamic</td>
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<td>Test</td>
<td>Stiffness (N/mm)</td>
<td>Natural Freq theta max (deg.)</td>
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<tr>
<td></td>
<td>Sd »</td>
<td>(Hz)</td>
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<tr>
<td>Test</td>
<td>Front Static</td>
<td>Dynamic</td>
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<td>-------------</td>
<td>---------</td>
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<td>3000</td>
</tr>
<tr>
<td>29</td>
<td>1000</td>
<td>3000</td>
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</tbody>
</table>
Hydroelastic Mountings for Vehicle Vibration Control:
  Modelling the Nonlinear Characteristics
Hydroelastic Mountings for Vehicle Vibration Control: 
Modelling the Nonlinear Characteristics

Martin A. Shaw
Avon Vibration Management Systems Limited / University of Bath

Jos. Darling
University of Bath

Abstract

Hydroelastic mountings are becoming increasingly commonplace in the suspension of power trains in passenger cars. They exhibit a performance advantage with respect to conventional elastomer mounts in terms of idle vibration isolation and control of engine motion, while offering significant cost savings compared with active mounting systems. This paper documents the construction and basic operating principles of typical passive hydroelastic mounts which utilise elastomeric materials and hydraulic elements. A component modelling technique is presented, with emphasis on the nonlinear features of the hydraulic components. A comparison is made between the model predictions and tests conducted with prototype parts.

1.0 Introduction

"In an increasingly competitive market the pursuit of cars with low fuel consumption and manufacturing costs has resulted in a trend to lower weight construction. This has increased the difficulties in achieving acceptable levels of noise and vibration at a time when consumers are expecting the standards to be continually improved." (Ref. 1)

A typical engine mounting system consists of the powertrain, supported by three or four mounts that must address the following:

i) Reaction of static and low frequency dynamic forces, namely the weight, lateral and longitudinal inertial loadings, and drive torque. This is essential to restrain the power unit within the vehicle and provide tractive effort.

ii) Control of powertrain resonances, especially vertical, pitch and roll modes. These modes are excited by random road and periodic wheel inputs to the body, and are perceptible as engine 'shake'.

iii) Isolation of high frequency vibration from engine imbalance and firing forces. This high frequency excitation can cause both noise and vibration within the passenger compartment. If a cavity resonance coincides with the excitation a 'boom' condition occurs, producing a marked increase in cabin noise.

The recent trends in passenger car design serve to make an acceptable compromise more difficult to achieve with elastomer mountings. Increase in powertrain mass, as encountered with diesel engines or front wheel drive vehicles and increase in drive torque, again arising from front wheel drive or from more powerful engines, tend to require mountings with higher static stiffness. Control of powertrain resonances, requiring damping forces, dictate the use of high hysteresis rubber, which stiffens remarkably with reduction in amplitude and increase in the frequency of vibration. Acceptable isolation of high frequency vibration, however, requires low dynamic stiffness.

Hydramounts offer a fundamental improvement over conventional elastomer mountings. For a given static stiffness, a far greater level of damping is possible, over a limited frequency range, than with elastomer properties alone and with a smaller 'penalty' in terms of the high frequency stiffness. Thus, the conflict between static stiffness, damping and high frequency stiffness is reduced.
2.0 Mount Construction and Main Features

A typical hydramount comprises five significant features (fig. 1); A main elastomer spring which supports the static load, displaces fluid when deflected and provides some volumetric compliance. A bellows that, together with the main spring, contains the fluid within the mount. A plate that divides the mount into a working chamber and a compensation chamber. An inertia track or channel, often incorporated in the divider plate, that joins the working and compensation chambers. Most hydromounts also include a diaphragm or decoupler to introduce a non linear modification to the volumetric compliance of the main spring.

3.0 Mount Operation

The principle of operation is similar to that of a helmholtz resonator or tuned mass absorber, using the hydraulic components to amplify the effect. The frequency to which the mount is tuned is governed by the combined volumetric compliance of the working chamber, the dimensions of the channel and the fluid density. Relative movement between the engine and body fixings of the mount cause the main spring to deflect, modifying the volume of the working chamber. At low frequency the displaced fluid passes through the channel to or from the compensation chamber. The velocity of fluid in the channel is much greater than that measured across the mount because of the large ratio between the effective area of the main spring and that of the channel. As the excitation frequency approaches the tuned frequency, the mass of fluid in the channel begins to resonate on the spring provided by the volumetric compliance, causing pressure fluctuations in the working chamber that are phase shifted with respect to the excitation. These pressure fluctuations act on a large area inside the main spring, producing a damping force. When the excitation frequency is increased beyond the resonance, the movement of fluid in the channel diminishes, and displaced fluid is absorbed by the volumetric compliance, creating an in phase pressure variation and contributing to the dynamic stiffness.

The diaphragm or decoupler element is used to modify the volumetric compliance of the main spring. It may be non linear because of its geometry or increasing contact with snubbing surfaces as it is forced to move. A rising rate characteristic is used to provide more stiffness when the excitation or response amplitudes are large, thus producing damping for excitation in the tuned frequency region, and lower stiffness for small amplitudes associated with high frequencies.

4.0 Characteristics and Formulae

The hydramount operation results in a characteristic as shown in fig. 2a,2b, in which a static stiffness and a dynamic stiffness can be clearly identified. In the region between these points, the stiffness dips below the static value, then peaks before falling back to the dynamic stiffness. Simultaneously, the loss angle (phase angle between deflection and force) rises to a maximum consistent with the mid point of the frequency rise and tuned frequency. Bosworth and Fursdon (ref. 2) derive equations for the dynamic stiffness, the tuned frequency and effective mass as follows;

\[ K_d = K_s + K_2 \]

\[ K_2 = A c^2 \cdot K_p \]

where

- \( K_d \) = High frequency dynamic stiffness
- \( K_s \) = Static stiffness
- \( K_2 \) = Increase in stiffness
- \( A_c \) = Effective piston area of the main spring; fluid volume displaced per unit linear deflection
- \( K_p \) = Puffing stiffness; the volumetric compliance of the working chamber defined as the change in pressure per unit change in volume
\[ W_n = \sqrt{\frac{K_p \cdot A_c}{\rho \cdot L_c}} \]

where

- \( W_n \) = Resonant frequency (rads/sec)
- \( A_c \) = Cross sectional area of channel
- \( \rho \) = Density of fluid
- \( L_c \) = Length of channel

\[ M_e = \rho \cdot A_c \cdot L_c \left( \frac{A_e}{A_c} \right)^2 \]

where

- \( M_e \) = Mass of equivalent dynamic absorber

Fig. 3 is based upon the simplified linear model proposed by Seto et. al. (ref. 3) and illustrates the action of each part of the mount and the total effect. Note that the initial action of the channel motion is equal and opposite to that of the top, balancing the volume of the working chamber and resulting in static stiffness only. Then at the resonant frequency, the channel movement is a maximum, and is producing damping forces. Finally, the channel movement diminishes, leaving only the elastomer spring and in phase top pumping action, which produces the dynamic stiffness.

When a mount with non-linear characteristics is considered, the value of \( K_p \) becomes amplitude dependant and consequently the resonant frequency, \( W_n \), stiffness rise, \( K_2 \), and damping effect also vary widely with amplitude (fig. 4a, 4b).

5.0 Component Modelling Technique

The model presented operates in the time domain, enabling non-linear and transient analyses of the mounting, and thus avoiding the limitations of linearized modelling methods (ref. 3, 4).

To allow modelling of a wide range of mount and mounting system types, the hydramount sub components were modelled separately and combined to produce a complete hydramount. This modelling technique enables a single modelling system to be used for simple hydramounts as demonstrated here, mountings with multiple channels, interconnected systems and so on. The methodology is similar to that used in the Bath FP modelling environment (ref. 5), in that the sub components can be combined in any configuration that ensures compatible inputs and outputs between linked models (fig. 5). For the hydramount, separate models are produced for the main spring, working chamber, channel and compensation chambers. In all cases, the fluid is considered incompressible. This is justified by the low pressures at which the unit operates (1 Bar) and the low compressibility of the operating fluid in comparison with the gaseous element.

6.0 Models and Nonlinear Features

The main spring model is an idealised spring with non-linear stiffness, achieved by interpolating between points on a load vs deflection curve.

\[ F_{spring} = f(x) \]

where

- \( F_{spring} \) = spring force
- \( f(x) \) = value of spring force from interpolation of load vs deflection points
- \( x \) = deflection of mount
The working chamber is modelled as a piston with an effective area equal to that of the main spring, an additional volume term to provide the connection to channel elements, and a volumetric compliance, accounting for the combined main spring, diaphragm and decoupler elements. This compliance is non linear and is achieved by interpolating between points on a characteristic curve in a similar way to the non linear stiffness.

\[
P_{\text{chamber}} = K_p(x \cdot A_e - V_{ole}) \quad \quad F_{\text{fluid}} = P_{\text{chamber}} \cdot A_e
\]

where

- \( P_{\text{chamber}} \) = Pressure in working chamber
- \( V_{ole} \) = Additional volume term due to fluid transfer into the channel(s)
- \( F_{\text{fluid}} \) = Force due to fluid pressure acting on the inside of the main spring
- \( K_p(\text{vol}) \) = Value of chamber pressure from interpolation of pressure vs volume points

In addition, a check is performed to limit the minimum value of \( P_{\text{chamber}} \) to a user defined value. This feature simulates cavitation in the working chamber.

The channel model includes inertial mass, viscous and hydrodynamic loss effects. The model requires values for the pressure applied at each end of the channel. As the fluid is considered incompressible, it is treated as a solid 'slug' within the channel (Ref. 6). The viscous loss is calculated for laminar flow; \( \text{Re} < 2300 \).

\[
P_{\text{visc}} = 2L_c \mu \rho \frac{V_{\text{chan}}}{(\text{Rhyd}^2)} \quad \quad P_{\text{phyd}} = \rho / 2 \cdot \sum C_d \cdot V_{\text{chan}}^2
\]

\[
P_{\text{net}} = P_1 - P_2 - P_{\text{visc}} - P_{\text{phyd}} \quad \quad P_{\text{acc}} = \frac{P_{\text{net}}}{\rho \cdot L_c}
\]

where

- \( P_{\text{visc}} \) = Pressure loss due to viscosity and channel cross section characteristics
- \( \mu \) = Viscosity of fluid
- \( V_{\text{chan}} \) = Velocity of fluid in channel
- \( \text{Rhyd} \) = Hydraulic radius of channel (area / perimeter)
- \( P_{\text{phyd}} \) = Pressure loss due to hydrodynamic effects (entry, exit, flow path)
- \( C_d \) = Discharge coefficients
- \( P_{\text{net}} \) = Net pressure differential acting on the channel
- \( P_1, P_2 \) = Pressures acting on opposite ends of the channel
- \( P_{\text{acc}} \) = Acceleration of the fluid slug in the channel

The compensation chamber model operates in the same way as the working chamber model. The only difference is that the 'x' term, relating to mount deflection is always zero, hence the model simplifies to;

\[
P_{\text{chamber}} = K_p(V_{ole})
\]

where

- \( P_{\text{chamber}} \) = Pressure in compensation chamber
- \( V_{ole} \) = Additional volume term due to fluid transfer from the channel(s)
7.0 Model Prediction and Prototype Comparison

Figs. 6-8 show comparisons of model predictions and the corresponding mount characteristics.

Mount #1 (fig. 6a,b) has a linear diaphragm and main spring, and the channel is relatively large, resulting in minimal resistance to flow. This produces a 'peaky' mount with large loss angle and little change in the resonant frequency with amplitude. The high frequency stiffness varies only slightly with amplitude.

Mount #2 (fig. 7a,b) also has a linear diaphragm and main spring, but contains a smaller, more restrictive channel. As a result, the mount becomes more strongly amplitude dependant, since the larger amplitude resonance is less well developed. This produces a mount with smaller loss angles and some reduction in frequency with increased amplitude. In addition, the rubber compounds used in this mount have more hysteresis, producing a greater increase in high frequency stiffness with reducing amplitude.

Mount #3 (fig. 8a,b) has a linear main spring, but the diaphragm is replaced by a decoupler. This produces a mount that is strongly amplitude dependant, producing little damping for small amplitude excitation, but greater loss angle at increasing frequency as the amplitude is increased. The action of the decoupler also serves to reduce the high frequency stiffness of the mount with reducing amplitude, resulting in a mount with better noise performance.

8.0 Discussion

The use of a hydraulic means to amplify the effect of a small dynamic absorber gives the hydramount its primary advantage over conventional mountings and those incorporating traditional tuned masses, a few grams of fluid provide a similar effect to several kilograms applied at the mounting points. In addition, non linear features offer further advantages in tuning mountings for a particular application. Modelling the hydramount as separate component models is intended to allow a wider range of mount configurations to be represented with a single modelling system. This can be used as a product development tool, and in addition, the modular nature of the model lends itself to incorporation in a vehicle model, and thus predictions of the effect of hydramounts on the ride characteristics of a complete car. The inclusion of non linearities is essential to accurately model hydramounts now in common use on passenger cars. The variable stiffness and damping characteristics are put to good use in vehicle applications where well known specific frequencies must be isolated from the cabin. Likewise, the same techniques could be used in mounts for high pressure hydraulic machinery where pump and motor pumping frequencies must be attenuated. In this way some of the structure borne vibration associated with pump and motor systems could be reduced.

9.0 Conclusions

1) Accurate reproduction of a range of common hydramount characteristics incorporating the non linear features can be achieved with the proposed model.

2) The correlation demonstrated can probably be further improved by incorporating the effects of rubber hysteresis into the model.

Acknowledgement

The author wishes to thank the following:
The directors of Avon VMS Limited for permission to publish the findings of this study.
References

1) "Engine Mounting Performance All Rubber, Hydraulically Damped and 'Semi' Active Hydraulically Damped Compared", J P West

2) "A Frequency Domain Representation of Hydramounts For Ride Prediction", R. Bosworth & P M T Fursdon.


6) "The Hydraulic Trainer", A Schmitt, Mannesmann Rexroth GMBH

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**Fig. 1 Typical Hydramount Construction**

![Diagram of a typical hydramount construction](image_url)
Fig. 2a  Typical Hydramount Dynamic Stiffness

Fig. 2b  Typical Hydramount Loss Angle

Fig. 3  Hydramount Characteristics
Contributions of Components and Total Effect

\[
\text{Out of Phase Stiffness (N/m)}
\]

\[
\text{In Phase Stiffness (N/m)}
\]
Fig. 4a Hydramount Model Validation
Actual Dynamic Stiffness, Mount #3

Fig. 4b Hydramount Model Validation
Actual Loss Angle, Mount #3

Fig. 5 Hydramount Model Construction
Appendix D - Publications

Development of Hydroelastic Strut mountings for Vehicle Secondary Ride Enhancement: From Concept to Prototype Evaluation
Development of Hydroelastic Strut Mountings for Vehicle Secondary Ride Enhancement: From Concept to Prototype Evaluation

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Synopsis

Hydroelastic mounts are becoming commonplace in powertrain mounting systems, but their characteristics make them suitable for solving other vibration problems in vehicles. This paper describes the development of a hydroelastic mount that improves secondary ride when used as a strut mount on suspensions with telescopic dampers. The performance limitations of conventional bushes are discussed, leading to an explanation of the rationale behind the hydroelastic strut mount. A ride model is described, including hydroelastic mounts, with ride predictions for suspensions using conventional and hydroelastic bushes. Finally, the results of these trials are presented, demonstrating an improvement in the ride quality of the vehicle.

1 Introduction

1.1 The conventional suspension system

A conventional suspension system consists of a spring, usually a steel coil or leaf, and a damper unit in which a fluid is forced through small orifices to provide a damping action. The design and construction of dampers vary depending upon the application, and their characteristics can be tuned by the manufacturer, however all dampers exhibit an increasing dynamic stiffness with frequency. In order to produce acceptable noise and harshness performance, it is common practice to fit elastomer bushes in series with the damper unit, thus limiting the maximum stiffness attained by the suspension. These bushes progressively reduce the effectiveness of the damper with increasing frequency of excitation.

Figure 1 shows the effect that varying the stiffness of the damper bushes has on the ride performance of a vehicle. Stiff bushes which force the damper unit to deflect are advantageous for primary ride performance, maximising the effect of the damper and controlling the body resonance, but increase the transmission of vibration over the rest of the
frequency range including secondary ride and noise regions, producing higher levels of vibration and noise. Soft bushes tend to deflect themselves, in preference to the damper unit and especially at high frequency, providing improved isolation at the expense of primary ride control. This compromise is an inherent limitation of the conventional suspension.

In addition to the problem of finding suitable damper bush rates, the ride engineer must also consider the damper characteristic and its effect (ref. 1). The critical damping required for the two major modes of vibration, body bounce and wheel hop are, for a typical vehicle, 2300Ns/m and 4200Ns/m respectively. As a first approximation, a road profile may be considered to be a constant velocity spectrum, thus for a particular road, the damper operates at a similar point on its force - velocity characteristic for all frequencies of excitation. Under these conditions it is impossible to damp both modes equally; either the wheelhop mode will be poorly damped, or the body mode excessively damped, or both. The effect of the damper bushes, by reducing the effect of the damper with increasing frequency, makes this shortcoming worse by further reducing the damping for the wheelhop mode.

1.2 The requirements for an enhanced system

What becomes apparent from figure 1 is that the ideal suspension would be relatively soft, with heavy damping around the resonant frequencies only, thus providing control of the body and wheel modes, but isolating the higher frequencies and the important secondary ride region between the main modes, in the frequency range 6 - 12 Hz. With the conventional system the first of these requirements, control of the body mode, can be achieved satisfactorily, but increasing the damping to suppress the wheel hop mode introduces a trade-off between isolation and control of the wheel.

2 The Hydramount

2.1 A Typical Configuration

The hydramount is a hydroelastic device which is currently used predominantly for powertrain mounting applications. A mount, shown in figure 2, normally consists of a main spring that contributes the static stiffness and contains the working fluid, a valve plate, which together with the main spring forms a working chamber, and a bellows which contains fluid within a compensation chamber. In addition, the valve plate contains a diaphragm, used to modify the volumetric compliance of the working chamber, and a channel connecting the working chamber to the compensation chamber.

2.2 Operation of the hydramount

In operation, deflection of the main spring causes a change in the volume of the working chamber. At low frequency, fluid passes through the channel to compensate for the difference, and the stiffness of the mount remains low. As the frequency of the excitation is increased, the mass of fluid in the channel and the volumetric compliance of the working chamber form a resonant system, generating large pressure fluctuations in the working chamber as the resonance frequency is approached. The pressure generated acts upon the inside of the main spring, producing a damping force. At frequencies above resonance fluid motion through the channel subsides, the change in the working chamber volume is accommodated by the volumetric compliance and an increased dynamic stiffness is observed.
2.3 Hydramount characteristics

Figures 3a and 3b show a typical hydramount characteristic. The static stiffness in this case is 200 N/mm, this value rising to a maximum, then settling to a high frequency stiffness around 600 N/mm as the excitation frequency is increased. Note that the damping, measured as a loss angle, can be more than 90 degrees and is confined to a relatively small frequency band, in this case centred on 10 Hz. This combination of features is exactly that which was identified as desirable when considering figure 1; a controlled level of damping in a narrow, tuneable frequency range, and an upper limit on the stiffness achieved at high frequency. The characteristics are also amplitude rather than velocity dependent, offering further scope for tuning the system.

3 Vehicle modelling

A vehicle model was produced to investigate the replacement of the conventional damper bushes with a hydroelastic strut mounting. A quarter vehicle model was used, incorporating wheel and body only. The effect of powertrain motion was omitted to improve the clarity of the prediction, but it was also felt that the strut mount would be equally applicable to front and rear suspensions, and so the powertrain would not necessarily be present in all installations.

The wheel and body were modelled as point masses, the tyre as a lightly damped spring, and the main suspension as a constant rate spring. The damper was modelled as an idealised viscous damper with a nonlinear force - velocity profile, incorporating a bump / rebound bias and knee points. The damper bush was modelled as a lightly damped spring, the strut mount as a hydramount unit. The hydramount model was a simplified version of Avon's hydramount prediction software, incorporating the main non linear geometrical and fluid flow effects. Further details of hydramount modelling techniques are given in refs. 2 and 3. A time stepping routine was used with a Newton - Cotes integration scheme to model the response in the time domain, with a Fourier analysis to generate the frequency response plots.

A range of strut mount characteristics were investigated and the performance compared to determine an optimum mount condition. The intention was to preserve the primary ride and noise performance, and to maximise the improvement in the control of the wheel hop, thus reducing secondary ride vibration. Figure 4 compares the model prediction for the original condition with elastomer bushes with that of the hydroelastic strut mount equipped vehicle. The model predicts a 25% reduction in the wheel hop vibration transmitted to the body.

4 Vehicle trials

4.1 Prototype manufacture

The next stage of development was to manufacture prototype mounts for vehicle trials. The mounts were designed to fit an existing vehicle with minimal modifications, and initially to be fitted to the front suspension only. A range of mounts were produced with varying stiffness and damping characteristics, nominally the optimum condition predicted by the model with positive and negative variations.
4.2 Benchmarking and Initial Tests

Benchmark measurements were made with the standard vehicle for sinusoidal sweep rig tests and objective and subjective road tests. The front hub, damper rod, strut tower, floor and seat rail were instrumented for the initial trials, with the rear hub and rear corner of the sill being added during the investigation.

The first tests with hydroelastic mounts were performed exclusively on a four poster rig using constant velocity sinusoidal sweep inputs to gain an insight into the response of the vehicle with the strut mounts. These tests confirmed the model predictions to be reasonably accurate.

4.3 Optimisation - front suspension

The rig tests continued in parallel with objective and subjective road tests to optimise the system. For the road tests, a specific section of road was used, and tests were carried out at a range of speeds to assess the effect of wheel base filtering, due to relative phasing of the inputs to the front and rear wheels. Care was taken to ensure minimal change in the fuel load of the vehicle. The results of these investigations were promising, but also a little disappointing because the impact of the hydroelastic mounting was reduced. The effect of the changes to the strut mount diminished as the measurement location moved away from the front of the vehicle. This is not surprising, since there is a progressively increasing contribution from the standard rear suspension and a reduction in the contribution of the front suspension. As a consequence, the benefit as measured at the strut tower was considerable, but at the seat the effect was much smaller and difficult to detect subjectively.

4.4 Optimisation - front and rear suspension

In order to demonstrate the full potential for ride improvement, more prototypes were produced and fitted to the rear suspension in addition to the front. This configuration was also optimised through rig and road testing, and using the knowledge gained from the work on the front of the vehicle. The same testing procedures were followed, with additional accelerometers fitted to the rear of the vehicle.

5 Test results

5.1 Rig tests

Figure 5 shows a summary of the seat rail responses of the vehicle, measured on the 4 poster rig. Responses are shown for the standard benchmark condition, for a vehicle fitted with the optimised front hydroelastic strut mounts only and for a vehicle with optimised mounts front and rear. The front only condition shows some improvement across the whole secondary ride region, from 3 to 30 Hz, the greatest effect occurring around 17 Hz. This is consistent with the tuning of the mounting used and the model prediction. It was found that none of the hydroelastic mountings tested made the secondary ride performance worse than that of the standard vehicle, however the static stiffness of the bush had an effect on the primary ride, and the dynamic stiffness influenced the high frequency response, as predicted in figure 1.
The result for mounts fitted front and rear shows further gains in secondary ride performance, since the improvement is now present front and rear; the standard ride performance of the rear suspension is not diluting the improvements made at the front. A reduction in the vibration of one third is achieved across a broad range of frequencies from 15 to 25 Hz. The effect on primary ride is not so good, however, exhibiting a worsening of the body bounce mode. In fact this is not due to the damper bush, but can be explained by the modifications made to install hydroelastic mounts in the rear suspension. It was necessary to remove the rear bump stops to fit the mountings, and their absence allowed greater travel of the rear suspension.

5.2 Objective road tests

Figure 6 shows the road test results for the same optimised mounting conditions as figure 5. Once again seat rail response is used as a measure of the suspension performance. The benefit of the mounts is less marked in these tests, but an improvement over the benchmark condition can still be seen for both hydroelastic mounting configurations. The effects are limited to the secondary ride region, above 15 Hz.

The main differences between the rig and road tests are the tyre condition, longitudinal input and input content. For the rig tests, the tyre is not rolling; the effect of sidewall stiffness and filtering due to the contact area are changed relative to the normal operating conditions on the road. The 4 poster rig applies vertical excitation only, there is no facility for replicating the longitudinal inputs experienced on the road. Finally, the rig test utilises a sinusoidal input at a single frequency and the response is allowed to settle, whilst the road excitation is almost exclusively random, producing predominantly transient response from the vehicle. With these considerations, the ability to identify similar trends in both rig and road tests is encouraging.

5.3 Subjective road tests

Subjective assessment of vehicle ride is considered to be an important technique for quantifying vehicle ride performance. The acid test for any proposed ride-improving product is that its effect be noticed by passengers in the vehicle. To this end a group of ride engineers made subjective assessments of each set of strut mounts fitted to the vehicle. Figure 7 shows a summary of the average ratings awarded to the same optimised configurations as presented in figures 5 and 6. The assessment panel were asked to rate the vehicle performance against a standard 1 to 10 scale for noise, primary and secondary ride and general vibration. The noise assessment was intended to ensure that the effects of high frequency stiffness were considered. This revealed a similar small increase in the noise experienced by passengers for both front only and front and rear installations. The primary and secondary ride characteristics were judged to have been progressively improved by the addition of front and rear hydroelastic mountings, and the general vibration, used to assess rattles, trim vibration etc. was also considered to be better.

The overall effect, then, was perceived to be improvement in the primary and secondary ride of the vehicle and a small reduction in the general vibration.
5.4 Model prediction comparison

Figure 8 shows a comparison of the model prediction and actual performance improvements measured in the vehicle tests. The improvement is presented as the vibration measured with hydroelastic mounts as a percentage of the vibration in the standard condition. The model prediction shows similar performance below 12 Hz, with a reduction in vibration between 12 and 20 Hz. This trend tallies with the results for the rig tests, the performance of the front only condition is very similar and the front and rear installation shows considerable improvement. This suggests that the mounts have been more effective than expected, though this may be attributable in part to the damping produced by the prototype being greater than predicted by the model. There is also some discrepancy around 10 Hz, since the model does not include the powertrain effects.

6 Conclusions

It is known that the conventional suspension system exhibits design conflicts which result in a compromise in the primary and secondary ride performance of passenger cars. However it is shown that it is possible to partially resolve these conflicts, increasing the wheel hop damping, by using a hydroelastic strut mounting in place of the conventional elastomer bush.

This development is capable of improving the vehicle ride when fitted to the front of a vehicle only, but maximum benefit is obtained with mountings fitted to all the wheel stations. The front and rear of the vehicle interact at the frequencies associated with secondary ride, hence the mountings for front and rear suspensions must be tuned as a system for maximum benefit.

The improvement is measurable both objectively and subjectively, the improvement achieved in this study amounting to approximately half a point for primary ride and one point in secondary ride, measured on a subjective scale of 1 to 10. This is a significant gain in performance.

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References


Fig. 1 - Effect of damper bush rate on transmissibility

Comparison of soft and stiff damping bush for 0.1 m/s input

Fig. 2 Typical Hydramount Construction
Fig. 3a Hydramount characteristic
Typical dynamic stiffness

Fig. 3b Hydramount characteristic
Typical loss angle

Fig. 4 Hydroelastic damper mount model
Comparison of secondary ride effects
Fig. 5 Comparison of rig results
Optimised conditions - seat rail

Fig. 6 Comparison of road results
Optimised conditions - seat rail
Fig. 7 Comparison of road results
Optimised conditions subjective rating

Fig. 8 Model prediction comparison
Vibration reduction relative to std.