5th Australasian Congress on Applied Mechanics, ACAM 2007
10-12 December 2007, Brisbane, Australia

High frequency parameter sensitivity in hydraulically interconnected suspensions

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Abstract: In this paper, the development of a hydraulically interconnected suspension (HIS) system model and its integration into a four degree-of-freedom half-car system is briefly introduced. Frequency response functions are derived in order to simulate the system response to a stochastic road profile. The sprung mass vertical and roll accelerations are considered in the frequency domain up to 1000 Hz. Four key hydraulic system parameters which affect the system's high frequency dynamics but not its low frequency response are identified and investigated. The results indicate that HIS system performance in the high frequency range (50-1000 Hz) can be greatly affected by these hydraulic parameters, while the favourable ride characteristics typical of HIS vehicles are retained.

Keywords: hydraulic, interconnected, suspension, vehicle dynamics.

1 Introduction

Conventional vehicle suspension design involves a trade-off between handling stability and ride comfort. A vehicle with a relatively stiff suspension is likely to possess good handling stability but poor ride comfort, and vice versa. One approach to overcoming this compromise is through the use of hydraulic or mechanical interconnections between the individual wheel stations (spring-damper elements). An *interconnected suspension* system is one in which a displacement at one wheel station can produce forces at other wheel stations [1]. Unlike conventional suspensions, interconnected schemes grant the designer, in theory, complete control over the stiffness and damping of each suspension mode. In practice, the degree to which individual modes can be controlled depends on the method and arrangement of interconnection employed.

In recent experimental studies, vehicles with hydraulically interconnected suspension (HIS) systems displayed significantly improved handling capability compared to their non-interconnected 'equivalents' [2-4]. Meanwhile, a recent theoretical study concluded that, for a half-car model (with 'typical' passenger vehicle parameters) subjected to a stochastic road input, the added roll stiffness achieved with an HIS system resulted in better ride comfort and smaller tyre normal force fluctuations than if the increased roll stiffness had been achieved with a conventional suspension [5]. Very recently, a second theoretical study identified a number of key HIS system parameters and discussed their effects on sprung mass response to road roughness from 0 to 20 Hz [6]. The investigation concluded that, while HIS systems offer potential advantages, "there are a number of hydraulic system parameters which will considerably affect a vehicle's ride performance". One question which remains unanswered, however, relates to the effects that certain hydraulic system parameters might have at frequencies above 20 Hz.

This study, therefore, focuses on the sprung mass response to road roughness in the frequency range from 0 to 1000 Hz. A practically achievable combination of system parameters is selected to deliver desirable low frequency vehicle dynamic characteristics. A number of HIS parameters which significantly affect the vehicle's high-frequency dynamics are then identified and investigated within the constraint that the vehicle's desirable low-frequency dynamics are maintained. Two 'equivalent' vehicles with conventional independent suspensions are also included for comparison.

2 Model description

2.1 System description

The conventional quarter-car approach to ride modelling is inadequate for the study of interconnected suspensions. In order to retain simplicity whilst still accounting for fluid interconnections between wheel stations, a lumped-mass four-degree-of-freedom half-car model is used in this investigation. Numerical simulations of a similar full-car model show that the half-car simplification is capable of capturing the essential dynamics of the system [7]. The half-car, shown in Figure 1, is described by typical passenger vehicle parameters and consists of linear tyre damping and springing, linear conventional suspension springing, and a typical anti-roll HIS system, similar in arrangement to that studied by Liu [8]. The system inputs are the road displacements at both tyre contact 'points' and the system outputs are the vertical displacements of the unsprung masses and the vertical and roll displacements of the sprung mass.

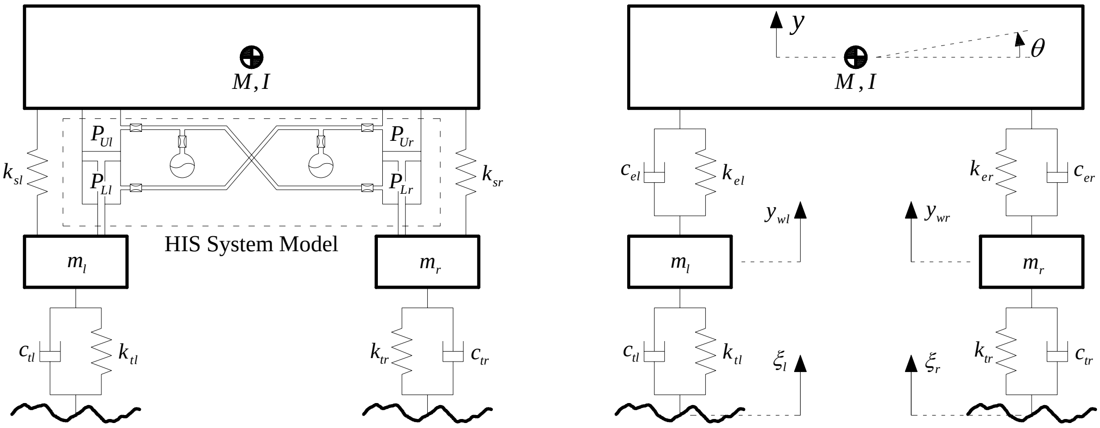

Figure 1: Layout of a half-car with: an HIS (left); a conventional independent suspension (right)

The hydraulic system, shown in Figure 2, consists of: a double-acting cylinder at each wheel-station; hydraulic interconnection between the cylinders; and gas-filled accumulators and dampers, which provide the desired levels of springing and damping. The hydraulic circuits are arranged such that motion in a certain vehicle mode produces a nominal flow distribution which operates particular accumulators and dampers. The arrangement considered here may be described as anti-oppositional [9], meaning that stiffness is added to the vehicle roll mode without significantly affecting the bounce mode.

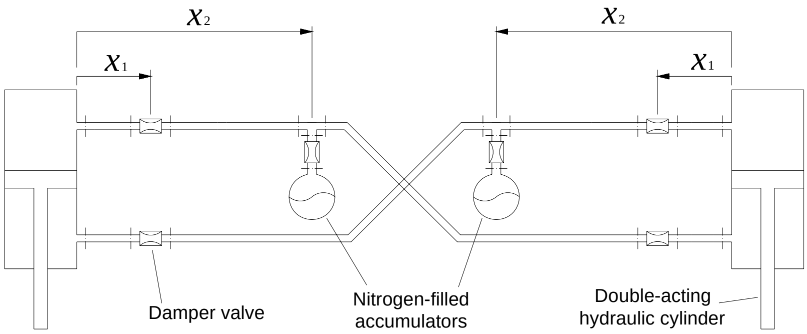

Figure 2: Anti-roll HIS system

In addition to the HIS vehicle, Figure 1 also contains an 'equivalent' conventional independent suspension vehicle. The model is identical to that of the HIS vehicle, except that the suspension springs and double-acting cylinders are replaced with 'equivalent' linear springs and dampers. Both vehicles shown in the figure possess right-left symmetry, although this is not a requirement of the techniques henceforth employed. The equivalent stiffness and damping coefficients are determined by matching the conventional vehicle's bounce and roll modes – determined via free vibration analysis [10] – to those of the HIS vehicle. Two sets of suspension stiffness and damping pairs – termed 'equivalent bounce' and 'equivalent roll' – are thus obtained which give identical modal properties to that of the HIS system [5]. PSD plots of the two equivalent conventional suspension vehicles' responses are displayed against that of the baseline HIS vehicle for comparison.

The aim of this study is to study the dynamic performance of the half-car system up to 1000 Hz with a number of different hydraulic system parameter combinations. A baseline combination of parameters is set, then a few 'key' parameters are adjusted in isolation to indicate their influence on sprung mass response. Clearly, the mechanical system illustrated in Figure 1 is not detailed enough to accurately represent high frequency vehicle dynamics. However, the modelling approach used here is still able to indicate overall trends in parameter effects and the range of frequencies over which these effects occur, even if the actual response values have significant errors.

2.2 Methods of evaluation

Dynamic performance indices are a common tool employed to evaluate and/or optimise suspension system ride performance (0-30 Hz). The most common of these indices is the sprung mass vertical acceleration. Here, since a half-car model is studied, dynamic performance is assessed not only by sprung mass vertical acceleration, but also by sprung mass roll acceleration. Both acceleration responses are considered in the frequency domain in terms of their Power Spectral Densities (PSDs). Although direct human perception of vibration is greatly diminished in frequencies above the ride range, the acceleration PSDs are studied here up to 1000 Hz in recognition of the fact that vibration readily manifests as highly perceptible noise over this frequency range.

Based on vehicle response to a specified road disturbance, the two performance indices for each parameter combination are compared qualitatively via response plots. The key hydraulic system parameters selected for investigation are: the overall hydraulic line length; accumulator location; damper valve location; and hydraulic oil bulk modulus. Simulations suggest that these parameters influence the higher frequency fluid-dominated system modes greatly, while having minimal impact on the lower frequency mechanical-dominated system modes.

2.3 Road surface description

Road profiles in vehicle dynamics simulations are generally treated as either deterministic (bumps, potholes etc.) or stochastic (random road roughness) processes [11]. Here, the latter approach is adopted and the road surface is assumed to be a realisation of a two-dimensional Gaussian homogenous and isotropic random process [12,13]. A single road profile can therefore be conveniently represented by its PSD function. Further information can be found in the relevant literature [13-16]. For our present purposes, a permissible and sufficient form of the direct spectral density is the ubiquitous 'single slope' representation, often expressed in terms of κ , the spatial frequency:

$$S_D^{\xi}(\kappa) = c\left|\kappa\right|^{-2w} \quad (1)$$

The application of isotropy allows us to determine the road displacement spectral density matrix $\mathbf{S}^{\xi}$, which facilitates the calculation of the response spectral density matrix [17]:

$$\mathbf{S}^{R}(\omega) = \mathbf{H}^{*}(s)\mathbf{S}^{\xi}(\omega)\mathbf{H}^{T}(s) \quad (2)$$

where $\mathbf{H}$ is a matrix of the appropriate frequency response functions (FRFs), and * and ^{T} denote the complex conjugate and matrix transpose. The derivation of $\mathbf{H}$ is explained in the next section.

2.4 System equations and vehicle response

The equations of motion for the coupled half-car and hydraulic systems shown in the HIS vehicle in Figure 1 have been derived elsewhere [10] and are in the form:

$$\mathbf{M}\ddot{\mathbf{Y}} + \mathbf{C}\dot{\mathbf{Y}} + \mathbf{K}\mathbf{Y} = \mathbf{F}_{ex} \quad (3)$$

where the system state vector $\mathbf{Y} = \left[y_{wl}, y_{wr}, y, \theta\right]^{T}$. The mass and stiffness matrices ($\mathbf{M}$ and $\mathbf{K}$) are easily derived, but the derivation of the damping matrix $\mathbf{C}$ for the HIS vehicle is more complex and involves impedance modelling of the hydraulic system using the transfer matrix method [10].

The FRFs matrix relating vehicle acceleration response to road displacement is:

$$\mathbf{H}(s) = s^{2}\mathbf{Y}(s)/\boldsymbol{\xi}(s) = s^{2}\left(s^{2}\mathbf{M} + s\mathbf{C} + \mathbf{K}\right)^{-1}\mathbf{F}(s) \quad (4)$$

in which the displacement vector $\boldsymbol{\xi} = \left[\xi_{l}, \xi_{r}, 0, 0\right]^{T}$ and $\mathbf{F}$ is a 4×4 matrix comprising all zero elements except the upper two diagonal terms $F_{11}(s) = F_{22}(s) = sc_{t} + k_{t}$. Upon setting $s = j\omega$, the FRFs describe the system acceleration response to any harmonic road excitation. The FRFs for this system have been published elsewhere and are not repeated here [5].

Equation (4) may be substituted into (2) to obtain the $\mathbf{S}^R$ matrix, the diagonal of which represents the direct spectral densities of the variables in $\ddot{\mathbf{Y}}$, upon which the following results are based.

3 Results and discussion

In this section, vehicle response is considered in the frequency domain up to 1000 Hz. An ‘average’ road type is simulated ($w = 1.25$, $c = 5 \times 10^{-7}\,\mathrm{m}^{0.5}\mathrm{cyc}^{1.5}$ [17]) at 20 m/s vehicle speed, although the speed has little effect since wheelbase filtering does not feature in the roll-plane model [5].

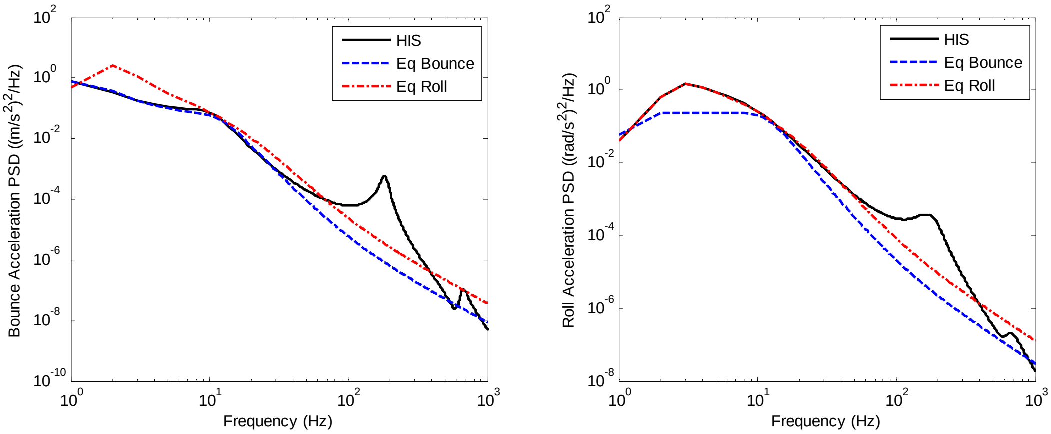

Figure 3: Response plots for baseline HIS, equivalent bounce and equivalent roll vehicles

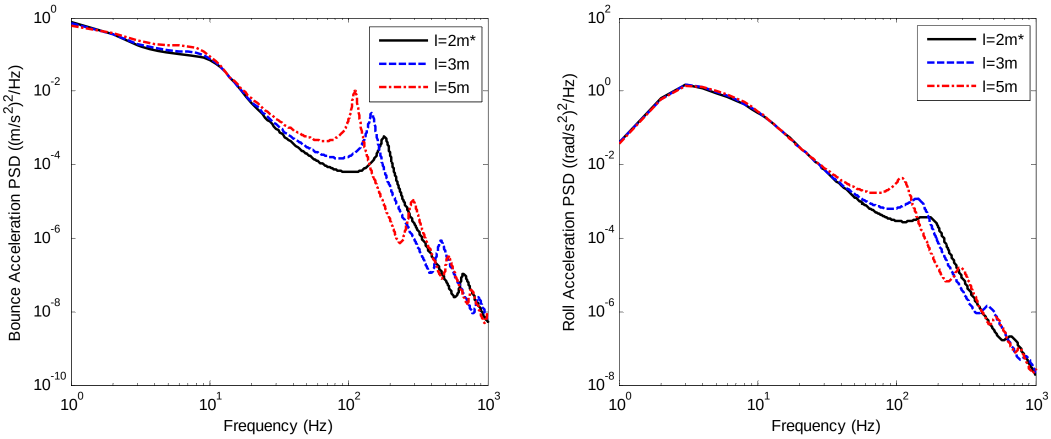

Figure 4: Response plots for HIS vehicle with various hydraulic line lengths (* indicates baseline value)

Figure 3 shows the simulated response plots for the baseline HIS, equivalent bounce and equivalent roll vehicles. As expected, the HIS vehicle response is almost identical to the equivalent vehicles’ in the corresponding modes in the low frequency range. Interestingly, the graphs show a departure in the two curves at about 40 Hz and 60 Hz (bounce and roll, respectively) which indicates the beginning of fluid compressibility effects. In each of the HIS curves, the large peak coincides well with the first fluid-dominated mode of around 180 Hz found with a free vibration analysis.

The effect of overall hydraulic line length (i.e. cylinder-to-cylinder pipe length) is outlined in Figure 4. A longer pipe causes a reduction in the frequency of the fluid-dominated system modes (first mode is reduced to around 110 Hz with a 5m pipe), and this is reflected in the response curves. The curves also display significant troughs between response peaks, meaning that a longer pipe length may be beneficial in certain cases (e.g. to avoid a frequency match with the chassis or vehicle body).

Figure 5 shows the effect of accumulator location (see Figure 2) on dynamic performance. Having the accumulator mid-span (baseline value) gives a maximum value for the frequency of the first fluid-dominated mode, and the peak bounce acceleration is larger than it is when the accumulator is positioned elsewhere. However, the mid-span accumulator position gives a smaller roll acceleration peak, so the determination of an ideal accumulator position would need to take this into account.

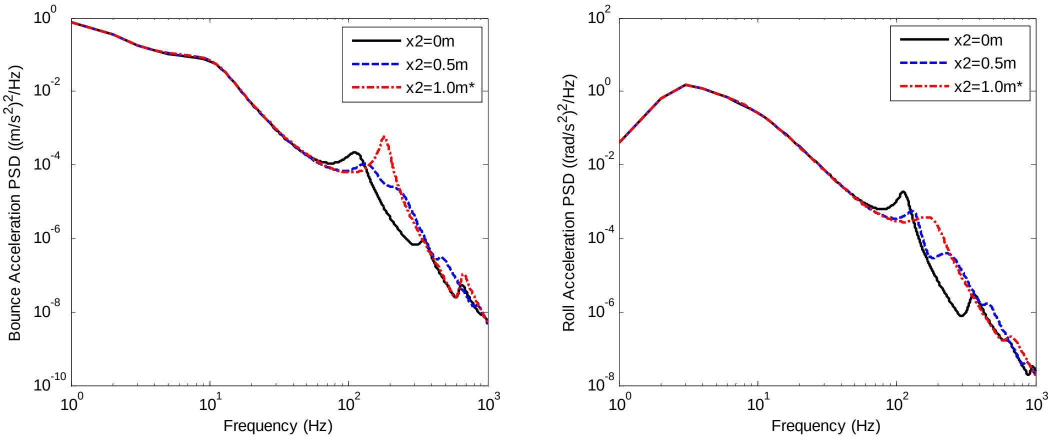

Figure 5: Response plots for HIS vehicle with various accumulator locations

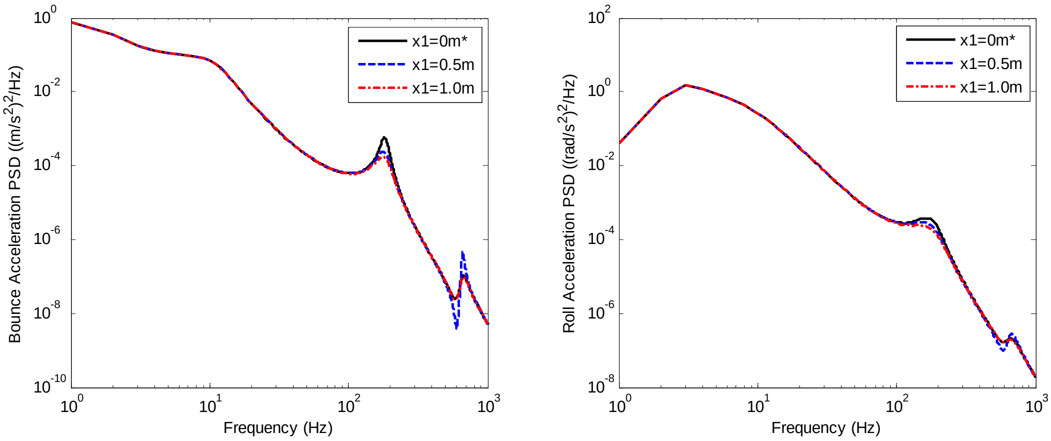

Figure 6: Response plots for HIS vehicle with various damper valve locations

The effect of damper valve location (see Figure 2) is displayed in Figure 6. The curves show that the damper valve position has little impact on the vehicle response in either mode. One potentially important effect valve location does have, however, is on higher frequency modal damping, with the mid-span valve position increasing damping significantly compared to the cylinder valve position (e.g., bounce damping for the 180 Hz peak increases from 7% with $x_1 = 0$ to above 13% with $x_1 = 1$ m).

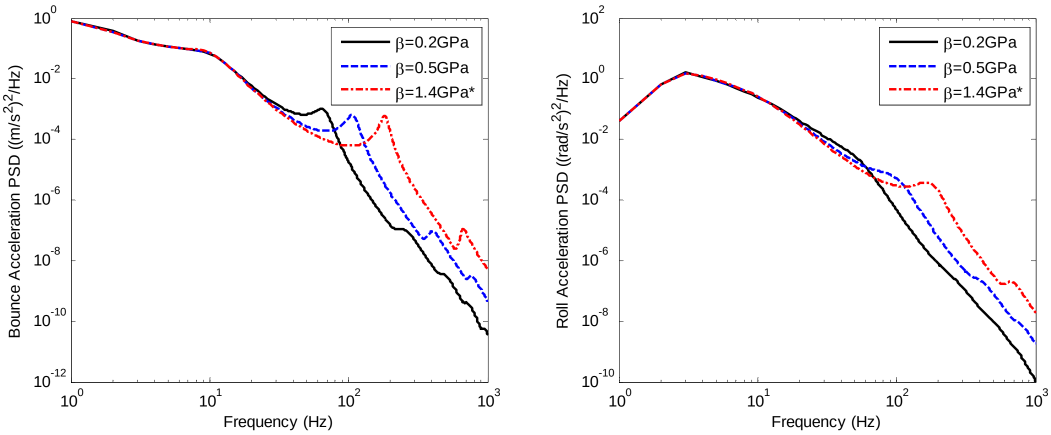

Figure 7: Response plots for HIS vehicle with various hydraulic oil bulk modulus values

Figure 7 shows the effect of hydraulic oil bulk modulus on dynamic performance. The displayed trend is not dissimilar to the effects of line length (Figure 4) in that all the fluid-dominated frequencies are reduced as the bulk modulus is reduced (or the line length is increased). However, the decreased bulk modulus also brings increased damping, which is particularly apparent at higher frequencies, where the response amplitude of the softer system is much lower than the baseline system. To place these results into context, the baseline value of $\beta = 1.4$ GPa corresponds to an 'effective' bulk modulus for an ideal hydraulic oil/steel pipe combination. The effective bulk modulus is affected by factors such as entrained air and pipeline compliance, and in practice is likely to be lower than 1.4 GPa. A β value of 0.2 GPa (which reduces the first fluid-dominated frequency from 180 Hz to 60 Hz) is roughly what one would expect from a hydraulic oil/hose combination in practice [18].

4 Conclusions and recommendations

The results indicate that while hydraulically interconnected suspension (HIS) systems provide a potentially viable method by which to partially overcome the ride/handling compromise, these benefits come at the cost of potential high frequency fluid borne noise issues. Problems are likely to arise when the fluid-dominated modes of the HIS system coincide with structural modes in the vehicle body or chassis. However, this paper shows that there are a number of hydraulic system parameters which will considerably affect a vehicle's high frequency dynamics without adversely affecting the favourable low frequency performance gains achieved with the HIS.

Recommendations for future work include the extension of the study to a full-car system, the use of other interconnection arrangements and model parameters, and the modelling of higher frequency mechanical system dynamics to facilitate a more thorough examination of high frequency phenomena. Experimental validation of the findings is also recommended.

Acknowledgements

Financial support for this research was provided jointly by the Australian Research Council (ARC LP0562440) and the University of Technology, Sydney.

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