

The Influence of Road Surface Properties on Vehicle Suspension Parameters Optimized for Ride - Design Trends for Global Markets

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ABSTRACT

Suspension design is influenced by many factors, especially by vehicle dynamics performance in ride, handling and durability. In the global automotive industry it is common to “customize” or tune suspension parameters so that a vehicle is more acceptable to a different customer base and in a different driving environment. This paper seeks to objectively quantify certain aspects of tuning via ride optimization, taking account of market differences in road surface spectral properties and loading conditions. A computationally efficient methodology for suspension optimization is developed using stochastic techniques. A small (B-class) vehicle is chosen for the study and the following main suspension parameters are selected for optimization - spring stiffness, damping rate and vertical tire stiffness. The road is characterized as a stationary random process, using scaling and shaping filters representative of comparable roads in India and the USA. A standard quadratic function is used, including terms for vertical body acceleration, tire load variations and workspace usage. Using the stochastic nature of the ride metrics obtained, it is found possible to further penalize large excursions in the suspension working space. Weighting parameters in the ride metric are selected to match a given design to the US market condition, so that re-optimization assuming conditions relevant to the Indian market immediately provides trend directions for retuning the suspension. It is also shown how assumed increases in payload (passenger numbers etc.) or different driving speeds may be included in the tuning process. Future integration of durability and handling criteria is also discussed. The result is an efficient tuning tool that may be used to support more detailed studies in simulation and track testing, helping to reduce the time to market when suspension tuning is required.

CITATION: Chaturvedi, B., Gordon, T. and Karamihas, S., "The Influence of Road Surface Properties on Vehicle Suspension Parameters Optimized for Ride - Design Trends for Global Markets," *SAE Int. J. Passeng. Cars - Mech. Syst.* 5(1):2012, doi:10.4271/2012-01-0521.

INTRODUCTION

As different markets open up for business, a strategy of designing, producing and selling automotive products at different locations as per the economic feasibility and the demand level is a profitable strategy for a company [1]. While the same product has to be sold in different markets, every market presents different challenges for the designer and the product has to be tuned accordingly [2]. From the point of view of driving dynamics, various factors, like traffic conditions, driving speeds, weather conditions, driving styles and customer expectations vary across markets [3]. However, one of the most important factors to consider while designing a suspension of an automobile is the variation in the road conditions across markets, as it has a direct impact on the ride comfort and the durability performance of a vehicle [4].

Various studies have been conducted which categorize the roads into different classes via the power spectral density (PSD) function [5] which informs about the frequency content present; or profile indices like the International Roughness Index (IRI) [6] and the Roughness Number (RN) which make for a broader characterization of the roads. Although PSD analysis assumes the road surface is represented as a stationary random process, this assumption helps the efficiency of analysis by using various techniques of random vibration theory [7]. It is also common to simplify further by assuming a special form of the PSD, for example to treat the road as being white noise in velocity. In one study [7] the following analytical form was assumed

$$R(\omega) = A\omega^{-n} \quad (1)$$

where ω is the spatial frequency in cycles/m and A is a coefficient which represents the roughness amplitude of the road. The specific case mentioned, white noise in velocity, is then recovered by setting $n = 2$. The response of 2DOF systems to white noise excitation was studied in [8] where the effect of various parameters like the mass ratio, frequency ratio and the damping factors were analyzed. However, if the road is modeled as purely a white noise in velocity, then the variation in road profiles from market to market may not be captured very well. In this paper, to understand the effect of this variation, the road profile is represented as a combination of white noise sources and shaping filters [9] which are lumped together with the dynamics of the plant. This will be explained in detail in the next section.

Traditionally, a suspension is designed to account for durability and packaging requirements, and to achieve the desired performance it is tuned by running the vehicle over “representative” road surfaces of the relevant market and carrying out various iterations with altered component settings; this is an inefficient and time consuming process. Such tuning may be guided using informal design rules based on experience, but the overall process tends to be highly subjective.

It is well known that the requirements of ride comfort, durability performance and handling tend to be conflicting in nature [4, 10]. To improve the ride comfort of a vehicle on rough roads, one may tune the suspension to be as “soft” as possible; however, the vehicle might face problems while maneuvering corners at high speed on smooth roads. Moreover, the durability performance of a “very soft” or a “very stiff” suspension on a rough road may be compromised due to frequent bump stop contact at the suspension stroke limits (soft case) or high body stresses (hard case) [11]. Clearly, there is a need to optimize the performance of the suspension for conflicting performance requirements such as ride comfort, road holding performance and durability performance.

Much research has been carried out in the past to optimize suspension performance by varying parameters like spring stiffness, damping rates and tire stiffness, often in the context of the quarter car model [11,12,13,14]. A multi-objective stochastic optimization has been performed in [12] wherein the performance indices of discomfort, road holding and workspace were optimized by generating Pareto-optimal solutions. Design optimization for random road excitation was performed in [13] using a non linear quarter car model and Monte Carlo simulations. A critical review of various approaches to solve the suspension design problem is reported in [15]. Techniques like: Constraints methods, Linear Quadratic Regulators, Evolutionary Algorithms, Pareto filtering etc have been the focus of many of these studies.

Optimization techniques like LQR have also been applied to active suspensions in order to maximize their potential over passive suspensions [16]. A cost function comprising the vertical acceleration of the sprung mass (ride), suspension

stroke (rattle-space) and tire deflection (handling-related) may be used, with individual weightings for each performance parameter varied across a particular range. For LQR to be applicable, the road input is considered to be a white noise in velocity. Results [16] show how tuning the values of the individual weightings leads to different trade-offs between the various aspects of performance.

None of the above studies has fully considered the effect of varying road conditions in different markets and the effect on suspension optimization. The effect of different roads is studied in [17] by simulating a full 3D nonlinear model of the vehicle over various sections of representative road in a deterministic manner but the process can be time consuming and limited to the particular sections of road selected. In the paper presented here, a computationally efficient optimization methodology is developed, using linear models based on [16]; the aim is to extract the effect of road surface types on the optimized passive suspension parameters such as spring stiffness, damping rate and vertical tire stiffness. The results from the study are then to be validated by simulating vehicle performance over two idealized road surfaces appropriate to two different market regions. Subsequently, the effect of adapting suspension settings to suit the specific region is demonstrated.

ROAD MODELING

It is shown in [7] that the road elevation data may be effectively represented in the two slope form

$$G(\omega) = A(\omega/\omega_0)^{-n_1}, \quad \omega > \omega_0 \quad (2)$$

$$G(\omega) = A(\omega/\omega_0)^{-n_2}, \quad \omega < \omega_0 \quad (3)$$

which is generalized from the form of equation (1); again, ω is spatial frequency and A is the “roughness coefficient”. The two exponents n_1 and n_2 relate to proportional roughness or amplitudes at longer and shorter wavelengths respectively. This road model is considered sufficiently representative and suitable for frequency domain signal analysis. However, it has filtering properties, that are difficult to represent as a causal transfer function, and also is not fully suited to analysis with state space methods.

An alternative model is described in [9]; the PSD of road profile is represented as a combination of three independent white noise sources and shaping filters as follows:

$$G(\omega) = G_a / (2\pi\omega)^4 + G_s / (2\pi\omega)^2 + G_e \quad (4)$$

This allows one to generate the PSD road profile by using up to three white noise sources. The first of the three sources represent a white noise in acceleration, which is integrated twice to determine the displacement profile. Coefficient G_a

represents the contribution of the corresponding long wavelength content present in the road surface; larger values of ' G_a ' represent a more "wavy" nature of the road and this is typically associated with the quality of grading performed while building the road [9]. The second term represents a white noise in velocity which is integrated once to give the displacement profile. The coefficient G_s represents "surface roughness" present in the road; larger values of G_s indicate a rougher road. The third component is white noise in vertical displacement. A PSD with more content in the high frequency region therefore tend to have a larger value of G_e , which may be a result of localized surface defects such as potholes, fault, tar strips etc. [9].

This representation of road surface uses spatial frequency ω , in cycles/m, as independent variable. However, the moving vehicle receives input from undulations in the road as a function of time, which therefore has a dependency on vehicle speed. The spatial frequency (ω , cycles/m) and the temporal frequency (ν , Hz) are related through the speed (V) as follows

$$\nu = V\omega \quad (5)$$

Using this substitution, equation (4) can be re-written

$$G(\nu) = G_{av}/(2\pi\nu)^4 + G_{sv}/(2\pi\nu)^2 + G_{ev} \quad (6)$$

where $G_{av} = V^4 G_a$, $G_{sv} = V^2 G_s$ and $G_{ev} = G_e$.

In case of linear systems, a useful relationship exists between the input and output PSD functions [10]

$$G_{yy} = |H_{xy}|^2 G_{xx} \quad (7)$$

where G_{xx} is the PSD of the input, G_{yy} is the PSD of the output and H_{xy} is the frequency response of the transfer function between the input and the output. Using this, we derive the transfer function of the shaping filters as

$$H_1(s) = \sqrt{G_{av}}/s^2 \quad (8)$$

$$H_2(s) = \sqrt{G_{sv}}/s \quad (9)$$

$$H_3(s) = \sqrt{G_{ev}} \quad (10)$$

where s is the Laplace operator and G_a , G_s and G_e are the coefficients defined above.

SYSTEM MODEL

Since the objective of the research was to come up with a computationally efficient optimization methodology, a linear plant model has been used. The plant has been modeled as a combination of a conventional 2DOF quarter car model and the shaping filters defining the road characteristics.

The quarter car part of the system model consists of a sprung mass (m_s) which comprises the vehicle body, plus mass of occupants and luggage etc. The sprung mass is supported above the unsprung mass (m_{us}) with a suspension system consisting of a spring (spring constant, k_s) and a damper (damping rate, c_s). The damping forces have been assumed to be varying with the piston velocity at a constant rate and are assumed to be equal in magnitude during bump and rebound. The unsprung mass consists of the wheel, tire and the lower suspension links. The tire is assumed to have linear properties for stiffness (k_{us}) and the tire damping (c_{us}).

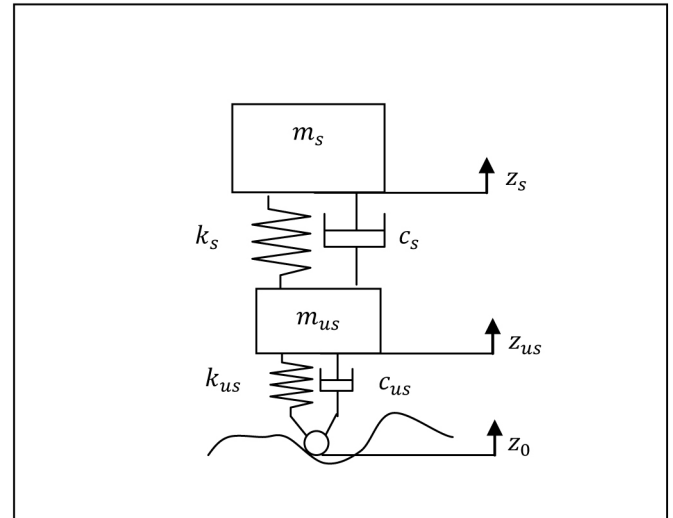


Figure 1. Quarter car model used for analysis

The resulting second order differential equations representing the vertical dynamics of the vehicle are

$$m_s \ddot{z}_s + c_s(\dot{z}_s - \dot{z}_{us}) + k_s(z_s - z_{us}) = 0 \quad (11)$$

$$m_{us} \ddot{z}_{us} + c_s(\dot{z}_{us} - \dot{z}_s) + k_s(z_{us} - z_s) + k_{us}(z_{us} - z_0) + \dots \\ \dots c_{us}(\dot{z}_{us} - \dot{z}_0) = 0 \quad (12)$$

If we normalize the parameters and write these equations in the state space form we obtain