

The Influence of Damper Top Mount Characteristics on Vehicle Ride Comfort and Harshness: Parametric Study

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ABSTRACT

The current paper addresses the relationship between the damper top mount characteristics and the ride comfort and harshness of a vehicle. A detailed mathematical damper top mount model which can simulate the vertical force characteristics of damper top mounts is developed and verified with actual tests. The amplitude and frequency dependent parameters of the damper top mount model are extracted from experimental testing of a commercial damper top mount. In order to identify the model parameters, a new procedure based on a two-stage optimization routine using two sets of measurement data for the amplitude and frequency dependent parameters is proposed. The damper top mount model is validated by comparing the measured force of the damper top mount with the simulated force of the proposed model. The developed top mount model is then implemented into a quarter vehicle simulation model for studying the influence of damper top mount characteristics on vehicle ride comfort and harshness. Laboratory tests have been performed with a quarter vehicle test rig in order to verify the vehicle model. A parametric simulation analysis has been performed by varying the characteristics of the damper top mount model. The results of the parametric analysis demonstrate the effect of the damper top mount characteristics on the ride comfort and harshness of a vehicle.

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INTRODUCTION

Damper top mounts are used in the vehicles not only to provide ideal Noise, Vibration and Harshness (NVH) performance but also to improve ride comfort, driving safety, and handling. The ride comfort and harshness can be considered as the vibration response of the vehicle body at different frequency ranges. Vehicle ride comfort can be evaluated with vertical acceleration of the body in the frequency range from 0 to 20 Hz while the harshness can be considered as the body vertical acceleration in the frequency range over 20 Hz until 100 Hz [1]. Another type of the vehicle harshness is the impact harshness (IH) which also affects the subjective impression of ride comfort. This type of harshness involves the vibration responses of the vehicle which are referred to as IH events [2, 3, 4, 5].

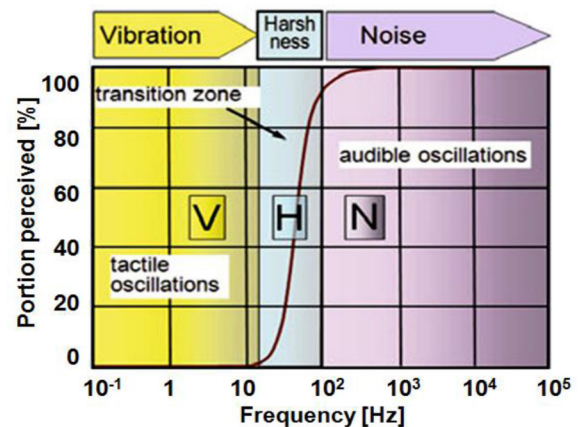


Figure 1. Oscillation frequency ranges associated with the terms noise, vibration, and harshness [1]

While the vehicle is in motion, driver is always in contact with seat track, and some other parts of the vehicle. Therefore, it is expected that once the acceleration responses at the selected locations of these parts are reduced, the vehicle will achieve an improved ride comfort and harshness. Some previous studies have shown that the characteristics of suspension system components have significant effects on ride comfort performance [6, 7]. In addition to these, the influence of the active suspension control laws on the vehicle harshness performance has been analyzed in another research study [8].

It is relatively easy and cost effective to change the compliance of the damper top mounts compared with changing any other suspension component to improve the vehicle ride comfort and harshness [9, 10]. Therefore, a detailed mathematical damper top mount model is required to simulate the actual behaviour of the damper top mounts for studying the impact of top mount characteristics on the vehicle ride and harshness performance. However, only a limited number of research studies about mathematical modeling of the rubber mounts, and especially damper top mounts, are available in the literature.

As generalized studies about the rubber mounts characteristics, [11, 12, 13] proposed a nonlinear dynamic rubber spring model to simulate the behavior of the rubber suspension components used in the road and rail vehicles. The rubber spring model is a one-dimensional model with five parameters, and it is based on a superposition of nonlinear friction, linear elastic, and linear viscous forces. The model parameters are determined from the measured force-displacement hysteresis loops at different harmonic excitation amplitudes and frequencies. Comparisons between the model and the measurement results for two different harmonic excitations pointed out a good agreement. [14, 15] modified the rubber spring model of [12] by adding a fractional calculus element to account for the frequency dependency of the rubber isolator. A better agreement between the measured and the simulated forces is obtained with the proposed model, compared to the well known Kelvin-Voigt rubber model [16].

Recently, [17] improved the rubber spring model of [15] by using a parabolic spring as the elastic element of the rubber mount model. They used the method of [15] for obtaining the rubber mount parameters. However, an additional optimization method is used to find the second parameter of the parabolic spring element. Three isolators having different shapes and different modes of deformation are used for the verification of the model. The simulation results showed a good agreement with the measured counterparts.

After a survey of the previous research studies on the rubber mounts, it can be concluded that, most of the previous work is concentrated on the testing, modeling, and model verification for general rubber vibration isolators. The frequency dependent characteristics of the rubber mounts are considered as linear, and the model parameters are obtained

at different constant frequencies. However, considering a linear behaviour is inadequate for modeling the actual behaviour of the rubber mounts. Furthermore, obtaining the model parameters at different constant frequencies means that, the input frequency must be known for simulating the rubber mount behavior. Therefore, such models are not suitable for the simulation of damper top mounts since it is not possible to know the frequency of the rubber mount input displacement during random road excitations.

In this study, a new damper top mount model based on the restoring force mapping technique is presented. The frequency dependent characteristics of the rubber mount is modeled by using a nonlinear damping element in series with a linear spring. The nonlinear damping element is considered to have a force response which is a function of the excitation velocity and displacement. This damping element together with the linear spring provides frequency dependent characteristics to the rubber mount model. Furthermore, a new procedure is proposed for the identification of the model parameters, which is based on a two-stage optimization routine where two sets of measurement data from the amplitude and frequency dependent tests are used. In order to study the influence of the damper top mount characteristics on the vehicle ride comfort and harshness performance, the proposed damper top mount model is implemented into a quarter vehicle simulation model. Experimental tests have been performed with a physical quarter vehicle test rig for verifying the quarter vehicle simulation model. A parametric study based on the developed damper top mount model has been performed to present the influence of the damper top mount characteristics on the vehicle ride comfort and harshness.

THE DAMPER TOP MOUNT MODEL

MODEL STRUCTURE

The damper top mount model developed in this study employs a nonlinear spring-pot coupled in parallel with a nonlinear smooth friction element and a nonlinear elastic spring. The layout of the model is shown in Figure 2. This structure of the mount model accounts for both amplitude and frequency dependent characteristics of the rubber top mounts. The amplitude dependency of the model is provided by the nonlinear elastic element and the nonlinear smooth coulomb friction element, while the frequency dependency is taken into account by using a linear spring and a nonlinear damper in series.

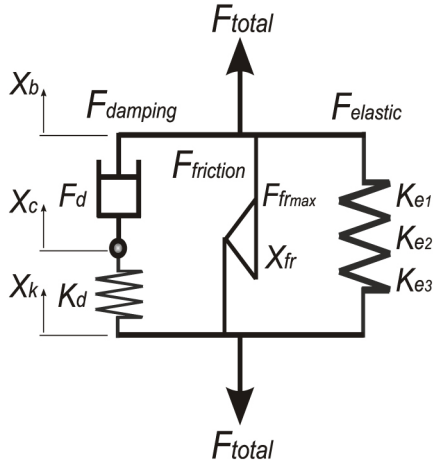


Figure 2. Layout of the proposed IAE damper top mount model

The total restoring force of the mount model is the sum of the forces from each component of the model as follows:

$$F_{total} = F_{elastic} + F_{friction} + F_{damping} \quad (1)$$

where, $F_{elastic}$, $F_{friction}$, and $F_{damping}$ are the elastic, friction, and viscoelastic forces, respectively. The sum of these three forces at a particular time step is based on a superposition assumption which suggests that all forces develop independently and separately. The model is simplified by assuming that the amplitude dependent characteristics is independent of the excitation frequency and the frequency dependent characteristics is independent of the excitation amplitude, as suggested by [11,15] in view of measurement results.

In the proposed IAE (Institute of Automotive Engineering) damper top mount model, the amplitude dependent force response is modeled by using a nonlinear spring in parallel to a nonlinear smooth Coulomb friction element as shown in Figure 2. The nonlinear spring element accounts for the nonlinear elastic characteristics of the rubber mount through a third order polynomial which defines the relationship between the elastic force and the excitation displacement as given with Equation 2.

$$F_{elastic} = K_{e1}(x_b - x_k) + K_{e2}(x_b - x_k)^2 + K_{e3}(x_b - x_k)^3 \quad (2)$$

where, K_{e1} , K_{e2} and K_{e3} are the elastic stiffness parameters defining the nonlinearity of the elastic behavior of the top mount. In the case of a single frequency harmonic excitation, the input displacement can be defined as:

$$(x_b - x_k) = (x_b - x_k)_o \sin \omega t \quad (3)$$

where $(x_b - x_k)_o$, ω and t are the peak amplitude, excitation frequency, and time, respectively. Therefore, the peak elastic force amplitude is given by:

$$F_{e_{max}} = K_{e1}(x_b - x_k)_o + K_{e2}(x_b - x_k)_o^2 + K_{e3}(x_b - x_k)_o^3 \quad (4)$$

The use of a filler, such as carbon black, in the rubber mount construction introduces additional bonds within the material. These bonds exhibit a friction-like behavior. Therefore, modeling of this phenomenon requires a detailed friction model as discussed in [11, 15]. In this study, the friction model proposed by Berg [11] is used for describing the friction force as a function of the top mount displacement. This model actually employs a smoothed Coulomb friction model. Only two model parameters are required for modeling the friction force accurately. The first parameter is the maximum friction force $F_{fr_{max}}$ while the second one is the displacement x_{fr} required to reach half of the maximum friction force. In addition to these parameters, the friction force also depends on a reference state in the top mount force versus displacement curve. The friction force in the damper top mount model develops gradually as a function of the excitation displacement as given in the following equations [11].

$$F_{fr}(x) = \begin{cases} \text{if } x = x_s & F_{fr_s} \\ \text{if } x > x_s & F_{fr_s} + \frac{x - x_s}{x_{fr}(1-\delta) + (x - x_s)} (F_{fr_{max}} - F_{fr_s}) \\ \text{if } x < x_s & F_{fr_s} + \frac{x - x_s}{x_{fr}(1+\delta) - (x - x_s)} (F_{fr_{max}} + F_{fr_s}) \end{cases} \quad (5)$$

$$\delta = F_{fr_s} / F_{fr_{max}} \quad (6)$$

where, $x = (x_b - x_k)$ is the total displacement of the top mount as shown in Figure 2. The parameters x_s and F_{fr_s} represent the reference values of the states x and F_{fr} , and they are updated during each change of the top mount displacement direction. For example, during a sinusoidal excitation of the form $x_o \sin(\omega t)$, the reference displacement x_s will be equal to the maximum displacement amplitudes $\pm x_o$ and the reference force F_{fr_s} is the friction force attained at the peak displacement points. The inclusion of a frictional behaviour introduces an increased stiffness at small displacement amplitudes and results in rate independent hysteresis characteristics. This means that, nonlinearity is introduced into the model due to the friction in addition to the nonlinear elastic behavior of the rubber.

Due to the viscous characteristics of the damper rubber top mounts, a force which depends on the velocity must be included in the mathematical model of the top mount. A viscous element composed of a spring in series with a damper can be used to model viscoelastic forces. Since the force

transmitted through the serial spring-damper element increases with the excitation velocity, a velocity dependent stiffness is obtained. The velocity dependent stiffness represents the frequency dependent part of the total damper top mount dynamic stiffness. The viscous element model is known in continuum mechanics as the Maxwell model, and it is generally composed of a linear spring and a linear damper. However, it is usually not possible to model the damper top mount behaviour accurately by using a linear damper in the Maxwell viscous element. Therefore, in the proposed model, the viscous force has been generated by using a nonlinear viscous damping element in series with a linear spring. In order to represent the frequency dependency of the damper top mount model, the force generated by the viscous element is modeled by using a force-velocity-displacement characteristic map. The force-velocity-displacement map of the viscous element is identified from the measurements together with the amplitude dependent parameters, as described in the following sections. The viscous force generated in the model can be described as given in the following equation:

$$F_{viscous} = f_{damper}((\dot{x}_b - \dot{x}_c), (x_b - x_c)) = K_d(x_c - x_d) \quad (7)$$

Amplitude dependent parameters

The amplitude dependent parameters K_{e1} , K_{e2} , K_{e3} , F_{frmax} , and x_{fr} of the damper top mount model are obtained from the hysteresis loops measured during the low frequency excitations. As shown in [Figure 3](#), the parameter K_{e1} is the large amplitude stiffness, F_{frmax} is the maximum friction force which can be developed during an oscillation, and x_{fr} is the parameter which determines how fast this force develops in relation to the displacement. During the low frequency excitations, the viscous effects are small and therefore can be neglected. Thus, the low frequency tests enable the identification of the amplitude dependent parameters K_{e1} , F_{frmax} and x_{fr} . A low frequency excitation test starts from zero force-zero displacement point. After a few cycles, a steady-state hysteresis curve can be obtained as shown in [Figure 3](#). In this study, the hysteresis curves obtained during the third cycle of the excitations are used. The friction force reaches its steady value as the rubber mount displacement approaches the peak value in both compression and tension deformations. After the direction of deformation changes, the friction force changes smoothly until it attains peak value in the opposite direction. Therefore, the slope of the tangent line to the hysteresis loop is approximately equal to the elastic stiffness parameter K_{e1} . The twice of the maximum friction force F_{frmax} can be considered to be equal to the vertical distance between the two parallel tangent lines of the hysteresis loop. Therefore, the friction displacement parameter x_{fr} can be identified from the distance between the two parallel vertical lines, one of which passes through the

returning point of the hysteresis loop while the other passes through the point where half of the maximum friction force is developed, as shown in [Figure 3](#). The other two elastic parameters K_{e2} and K_{e3} are identified by using a numerical optimization routine, in which the objective is to obtain the best fit between the measured and the simulated hysteresis loops.

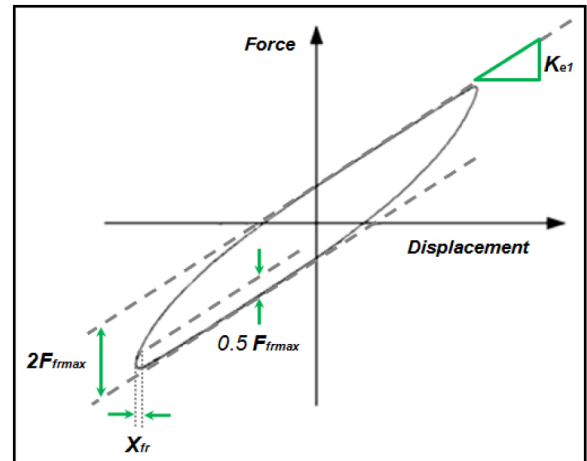


Figure 3. Low frequency hysteresis loop

Frequency dependent characteristics

The frequency dependent characteristics are defined by the stiffness K_d of the spring element and the force-velocity-displacement map of the damping element in the Maxwell model which generates the viscous force. In order to obtain these parameters, a high frequency test must be applied to the rubber mount since the viscous effects can be observed better at high velocities. The commonly used method for obtaining the parameters of the viscous element is the hysteresis loop method [11]. In this method, first the energy loss per cycle during a high frequency excitation is calculated by using the hysteresis loop. Then, the energy loss per cycle obtained during a low frequency excitation at the same amplitude is subtracted from the energy loss per cycle during the high frequency excitation. The result of the subtraction is the energy loss due to the viscous force, which can be used to obtain the parameters of the viscous element. Another method for simulating the viscous force of the rubber mounts is the fractional derivative element method which has been explained in [15, 16, 17]. These studies describe in detail, the use of the numerical estimation directly to obtain the fractional derivative parameters at a constant frequency. The previous methods used in [11, 15, 16, 17] are not adequate for the simulation damper top mounts, since these methods work at a constant and known frequency although it is not possible to obtain the frequencies of the road profile on which the vehicle is traveling. In view of this fact, the non-parametric restoring force technique has been suggested for defining the viscous force behaviour of the damper top mount model [18].

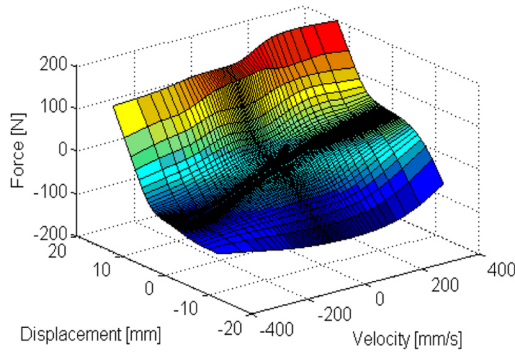


Figure 4. Force-velocity-displacement map of the damping element [18]

As mentioned before, the force that can be transmitted via the serial spring-damper unit increases with the excitation velocity. Therefore, a velocity dependent stiffness is obtained which result in a frequency dependent rubber mount response. Hence, in order to combine the frequency dependency of the rubber mount model with its velocity dependent force characteristics, the force generated by the damping unit is identified as a force-velocity-displacement characteristic map, which is commonly used in the vehicle dynamics simulation analysis. The stiffness parameter K_d and the force-velocity-displacement map of the damping component in the viscous element are obtained by using a numerical estimation process, where the objective is to obtain the best fit between the model behavior and the measurements. The number of nodes in the force-velocity-displacement characteristic map and a parameter which defines the smoothness of the map are used as the optimization variables during the numerical estimation process. The force-velocity-displacement map of the viscous element of the damper top mount model obtained as a result of the optimization process is shown in Figure 4.

IDENTIFICATION OF THE DAMPER TOP MOUNT MODEL PARAMETERS

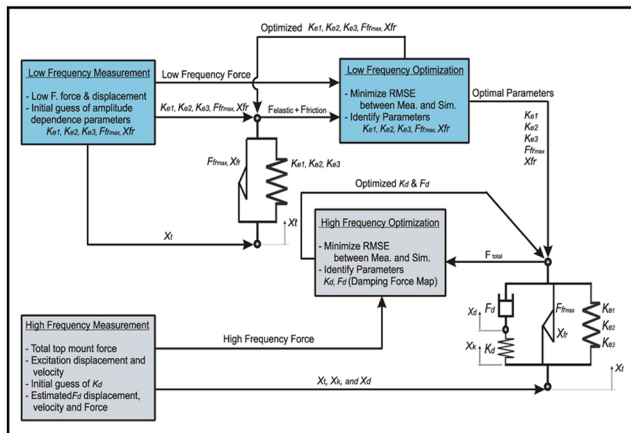


Figure 5. The proposed parameter optimization process

The amplitude and frequency dependent parameters of the damper top mount model are identified by using the measurement results from different tests performed with low and high frequency excitations at different amplitudes. The result of the parameter identification process depends on the tests specifications. For example, the values of F_{frmax} and K_{e1} are affected significantly by the displacement amplitude used in testing. Moreover, it is quite difficult to measure the parameter x_{fr} in practice, since this parameter is naturally small as shown in Figure 3. In view of these observations, it is decided to use a set of optimization routines for increasing the accuracy of the parameter identification for the damper top mount model. The optimization process is designed to optimize the amplitude and frequency dependent parameters of the damper top mount model to obtain the best fit with the measurements according to an objective function. The optimization process is split into two stages; the low and high frequency stages. The objective function for the low frequency stage is considered as the Root Mean Square Error (RMSE) between the measured and simulated damper top mount forces as given with Equation 8. For the high frequency stage, the goal of the optimization is to reduce the error between the measured and simulated dynamic stiffnesses of the damper mount together with the RMSE of the damper top mount force as shown in Equations 9 and Equation 10, respectively. The schematic of the optimization process is shown in Figure 5.

$$RMSE = \sqrt{\frac{1}{N} \sum_{i=1}^N (F_{i,mea} - F_{i,sim})^2} \quad (8)$$

$$RMSE_{DS} = \sqrt{\frac{1}{N_f} \sum_{i=1}^{N_f} (DS_{i,mea} - DS_{i,sim})^2} \quad (9)$$

$$Objective_{H. Frequency} = RMSE + RMSE_{DS} \quad (10)$$

In the above equations, N , $F_{i,mea}$ and $F_{i,sim}$ represent the number of data points in the time domain, and the measured and simulated damper top mount forces, respectively. Furthermore, N_f , $DS_{i,mea}$ and $DS_{i,sim}$ represent the number of frequency points in the frequency domain where dynamic stiffness is evaluated, and the measured and simulated damper top mount dynamic stiffnesses, respectively.

As shown in the optimization schematic, the low frequency measurements are used to obtain the initial guess of the amplitude dependent parameters K_{e1} , F_{frmax} and x_{fr} of the damper top mount model. The damping force is considered to be zero at low frequency excitations. The initial guess for the parameters K_{e2} and K_{e3} are set to zero. The model parameters and the excitation amplitude are provided to the damper top mount sub-model which is composed of the nonlinear elastic and nonlinear friction elements, and the sub-

model force output is calculated. The output force of the sub-model is the sum of the elastic and friction forces only. The low frequency optimization routine uses the output force of the sub-model and the measured force to calculate the RMSE between the measured and simulated low frequency forces, and it optimizes all the amplitude dependent parameters K_{e1} , K_{e2} , K_{e3} , F_{frmax} and x_{fr} to reduce the RMSE. The optimal amplitude dependent parameters are provided to the complete damper top mount model which contains the viscous force element. At the same time, the damper top mount viscous forces are calculated by subtracting the elastic and friction forces which are obtained from the low frequency optimization process from the high frequency measurement results. In addition, the displacement and the velocity of the damping unit of the viscous element is calculated in this stage. The obtained parameters and states, together with an initial guess of K_d , are provided to the damper top mount model and to the high frequency optimization routine. The high frequency optimization routine uses the overall output force of the damper top mount model and the measured high frequency force to calculate the RMSE between the measured and simulated high frequency forces. The frequency dependent parameter K_d , and the number of nodes and smoothness of the force-velocity-displacement map of the damping unit are optimized to reduce the force RMSE and the difference between the simulated and measured dynamic stiffnesses.

DAMPER TOP MOUNT TEST SETUP

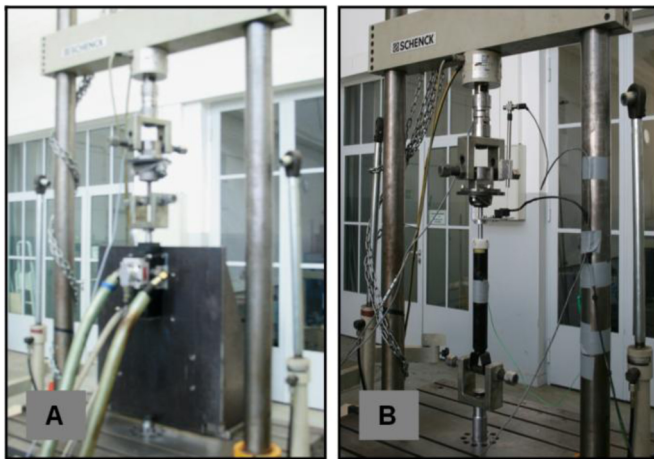


Figure 6. Components test rig setup A. damper top mount, B. damper top mount together with damper

The component test rig used for the damper top mount tests is shown in Figure 6. The hydraulic cylinder of the test rig is fixed to the lower base while the loadcell is connected to the upper plate. The test rig is driven by a hydraulic cylinder equipped with a solenoid valve that can produce harmonic or random excitation signals. The displacement and dynamic force limits of the hydraulic cylinder are ± 25 mm and ± 7 kN, respectively. The excitation displacement is

measured by the displacement sensor of the hydraulic cylinder which is connected to the measurement computer through a 12 bit NI 6062E A/D convertor. A loadcell with a measurement range of ± 10 kN is used to measure the damper top mount force. The force measurement signal produced by the loadcell is supplied to the test computer through an HBM MVD255 amplifier and the A/D convertor. In order to apply tests for the combined damper and damper top mount assembly, a hydraulic cylinder with operating displacement limits ± 50 mm has been used. A linear variable displacement transducer (LVDT) is used to measure the deformation of the damper mount. Also a thermocouple is attached to the damper cylinder wall for observing the damper temperature during the test to avoid the damper overheating.

QUARTER VEHICLE MODEL

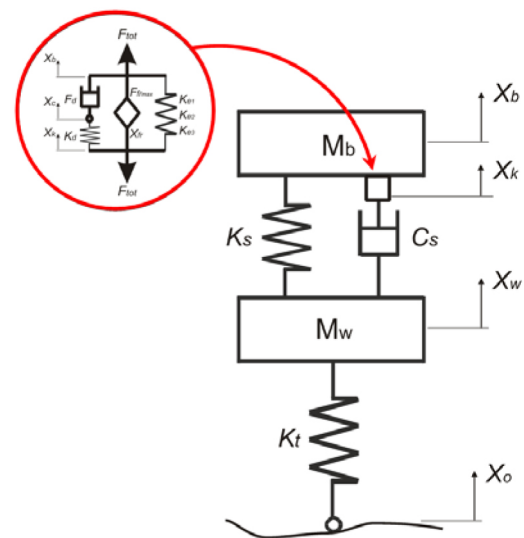


Figure 7. Quarter vehicle model

In this paper, a generic quarter vehicle suspension model with damper top mount has been constructed for the analysis. The vehicle with passive suspension system is modeled as a two degree of freedom model consisting of two masses which represent the body and wheel masses. The damper and top mount weight is neglected. The suspension between body mass and wheel mass is modeled using a linear spring and a viscous damper with rubber top mount as shown in Figure 7. The nonlinear characteristics of the damper and top mount are taken into account. The tyre is modeled as a linear spring and a point contact is assumed between the road and the tyre.

The dynamic equations of motion for the quarter vehicle model are written as follows:

$$M_b \ddot{X}_b = -K_s(X_b - X_w) - C_s(\dot{X}_b - \dot{X}_w) = -K_s(X_b - X_w) - F_{tot} \quad (11)$$

$$\begin{aligned}
 M_w \ddot{X}_w &= -K_t(X_w - X_o) + K_s(X_b - X_w) + C_s(\dot{X}_k - \dot{X}_w) \dots \\
 &= -K_t(X_w - X_o) + K_s(X_b - X_w) + F_{tot}
 \end{aligned}
 \quad (12)$$

In the above equations, M_b and M_w represent the body and wheel masses, K_s and K_t represent the suspension and tyre stiffnesses, C_s stands for the damping coefficient which is a function of the damper velocity, and F_{tot} is the overall damper top mount force. The body, damper top mount, wheel, and road displacements are denoted as X_b , X_k , X_w , and X_o , respectively, and they are given with respect to the inertial frame.

QUARTER VEHICLE TEST RIG



Figure 8. IAE quarter vehicle test rig

Figure 8 shows the quarter vehicle test rig developed by the IAE and used in this study. This hydraulic test rig is a practical realization of the often theoretically considered quarter vehicle, consisting of sprung and unsprung masses including the subframe as well as the suspension components such as spring, damper and rubber mounts. A hydraulic cylinder is used to drive the quarter vehicle test rig and it can reproduce measured road profiles or apply harmonic excitations. The hydraulic cylinder has a displacement range of ± 125 mm and it can provide dynamic forces up to ± 21 kN. In this study, the quarter vehicle test rig is used for the verification of the quarter vehicle model which includes the detailed damper top mount model.

ROAD INPUT SIGNALS

Two types of road inputs were used during experimental and simulation analysis. The first input is a real road profile picked from the road profile measurement database available at the IAE. The details of the road data measurement system can be found in [19]. In Figure 9A, the real road profile that is used for generating the analysis results presented in this manuscript is shown.

The IH events are usually evaluated under specific forward velocities without applying any steering input and therefore they are assumed to be straight-line events. In order

to evaluate IH performance of a vehicle through computer analysis, the mathematical model of the vehicle can be simulated by using a triangle road cleat as the road input. Typical forward speed range for the analysis of IH events is around 60 kph [5]. The triangle cleat used in this study for the analysis of IH event has a height of 30 mm and length of 400 mm length as illustrated in Figure 9B [10].

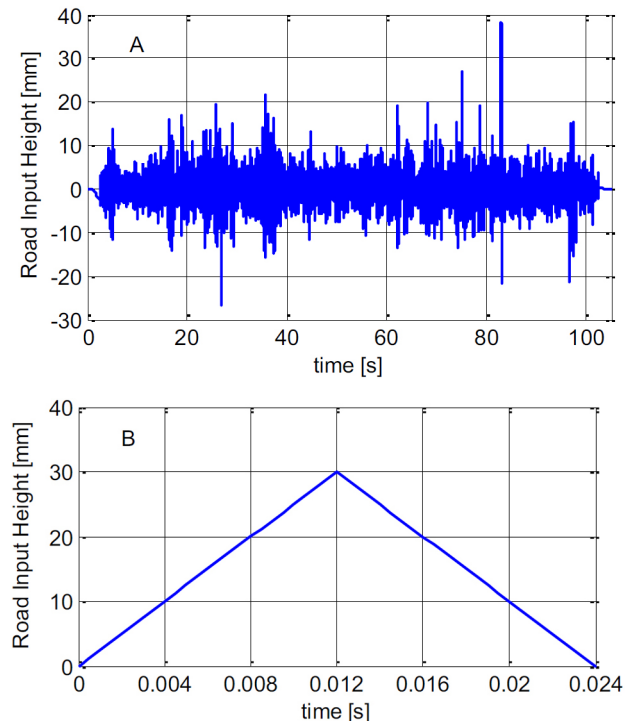


Figure 9. Road inputs A. Random road, B. Triangle cleat

VERIFICATION OF THE DAMPER TOP MOUNT MATHEMATICAL MODEL

The aim of the tests presented in this section is to identify the damper top mount model parameters and to verify the model accuracy. Two types of tests have been performed to obtain the damper top mount model parameters; which are the low and high frequency excitation tests. Both types of tests are performed at various amplitudes. The low frequency harmonic excitation tests are used for the identification of the amplitude dependent parameters of the damper top mount model. The high frequency stochastic excitation tests are used to obtain the restoring force map of the damping component of the viscous element.

The low frequency tests are required for obtaining the amplitude dependent parameters of the damper top mount model, since at low frequencies the viscous effects are very small and can be neglected. The excitation frequency of the low frequency tests performed in this study is selected as 0.1 Hz, and four different amplitudes of 1.0, 1.5, 2 and 3 mm are used.