

Rubber Bushing Model for Vehicle Dynamics Performance Development that Considers Amplitude and Frequency Dependency

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ABSTRACT

In many cars, ride is less comfortable on smooth roads. This is because when the hysteresis in the suspension components rises steeply, the increase of the equivalent spring constant at small amplitude deteriorates the vibration isolation of the suspension. Therefore suspension components should be designed to prevent a steep rises in hysteresis. Investigating the influence of hysteresis, component models, which can reproduce such hysteresis characteristics, should be installed with model parameters in the vehicle model. Using conventional methods, these parameters can be accurately identified if measurement data is provided; however, it is difficult at the earlier phase of vehicle development. Then, if conflicting performances, such as ride and handling, are to be improved, both should be considered concurrently as early in a phase of vehicle development as possible and the design specifications for suspension components should be determined to satisfy both performances. The simulations including 'Multi-Body Simulation' (MBS) should be performed using the same model with the same parameters. Furthermore, they should be done even if measurement data is not available. However, there is no appropriate model for the rubber bushing meeting these demands. Therefore this paper proposes a rubber bushing model and the parameter calculation method that can be used for investigating the vehicle dynamics performances in the early phases of vehicle development accurately and efficiently and specifying the dynamic characteristics of the rubber bushing.

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INTRODUCTION

Figure 1 shows the example measurement result of the vertical vibration transmissibility of the rear suspension installed in a vehicle where the ride is less comfortable on smooth roads. In the example vehicle case, the peak frequency becomes higher when the excitation amplitude becomes small. Figure 2 shows the example measurement results of wheel load to stroke curve (wheel stroke curve), measured with several stroke amplitudes. The equivalent spring constant, which is calculated by the storage energy in one cycle of the wheel stroke curve, increases when the amplitude becomes smaller in the example vehicle [1].

The hysteresis component on the wheel stroke curve is induced by force transfer elements, such as friction and viscous damping, which are involved in the suspension components. Therefore, using the quarter-car model as shown in Figure 3, the vibration transmissibility is changed by the hysteresis component can be understood.

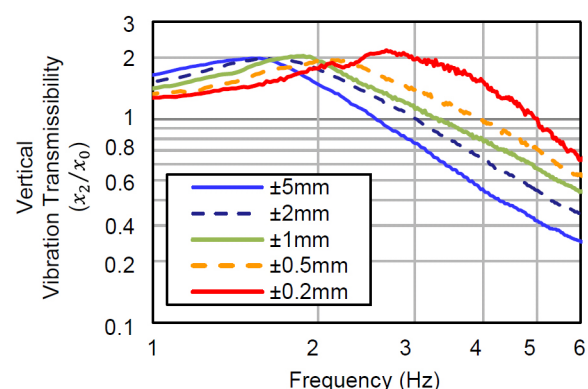


Figure 1. Example measurement results of vertical vibration transmissibility of rear suspension with several excitation amplitudes

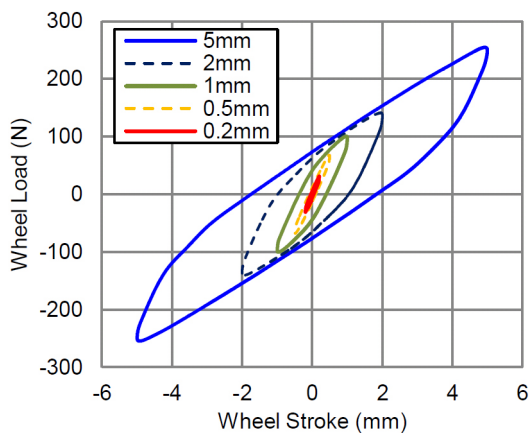


Figure 2. Example measurement results of wheel stroke curves of rear suspension with several stroke amplitudes

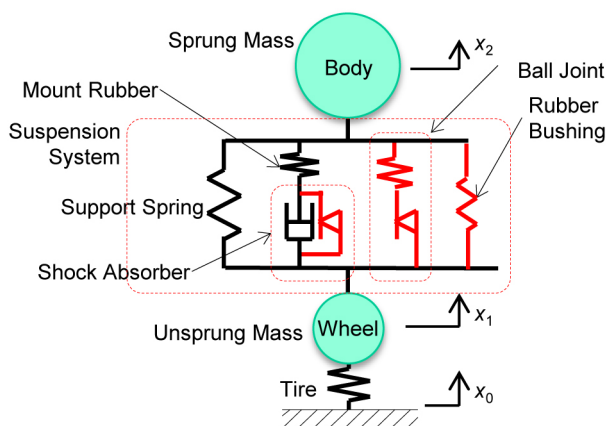


Figure 3. Quarter-car model including hysteresis elements in suspension components

If the exerted force to the shock absorber is smaller than the friction force in the shock absorber, it sticks and the mount rubber behaves like a support spring in parallel. The ball joint also sticks if the exerted force on the ball joint is smaller than its friction, and its spring component behaves like a support spring in parallel, and they increase the equivalent spring constant of the suspension system. Additionally, amplitude-dependent storage stiffness of the mount rubber and the rubber bushing also influence the equivalent spring constant of suspension system.

The support spring is generally much softer than the tire and dominates the vibration transmissibility of the suspension. However, if the equivalent spring constant increases and becomes larger than the tire stiffness, the tire has more influence on the vibration transmissibility. As a result, an increase in the equivalent spring constant of the suspension induces an increase in the body's natural frequency, and ride comfort deteriorates.

The hysteresis should therefore be considered to prevent an increase in the equivalent spring constant at the small amplitude of suspension to improve ride on smooth roads. This can be achieved by adjusting the hysteresis components, such as the shock absorber, ball joints and rubber bushings.

Considering the hysteresis characteristics, the appropriate simulation models should be installed in the vehicle model, and their parameters should be supplied to achieve sufficient accuracy.

In previously proposed modeling methods, dynamic stiffness due to hysteresis, such as storage and loss stiffness, can be modeled accurately using two elements, one for amplitude dependency and the other for frequency dependency, if their hysteresis shape and/or dynamic stiffness are provided [1, 2, 3].

Ride comfort and handling are respective issues of isolation and response in vehicle performance, and sometimes conflict in practical development, so their performances should be considered early in vehicle development concurrently for better efficiency.

The estimation accuracy of handling performance early in vehicle development has been improved by the latest tire characteristics estimation method [4], however ride comfort performance still has some issues with accuracy and efficiency, because actual measurements would be necessary for some suspension components to identify the dynamic characteristics that strongly influence ride comfort.

To improve the estimation accuracy and efficiency of ride comfort during the early phases of vehicle development, the dynamic characteristics of suspension components, such as ball joints, shock absorbers and rubber bushings, should be considered and their design specifications should be determined, even if measurement results are not available. Furthermore, the same component model should be applied to the simulation models for handling performance including MBS using the same parameters.

The hysteresis characteristics of friction involved in shock absorbers and ball joints can be considered using the Masing Model. And their design specifications can be determined using [1, 2].

The static force deformation curve of rubber bushing can be estimated by [5, 6] if detail design shape and rubber material properties are supplied. However, to improve the estimation accuracy and efficiency of the ride during early phases of development, the dynamic characteristics should be considered even if the detailed shape, rubber material properties or measurement results are not available.

If the frequency range near the natural frequency of sprung mass and unsprung mass is considered, the amplitude dependency dominantly influences the ride comfort. Then the dynamic characteristics of rubber bushings can be modeled and the parameters can be calculated by the proposed method [7]. However, to apply the rubber bushing model in general, the frequency dependency should be considered.

This paper therefore proposes a rubber bushing model that fully considers the amplitude and frequency dependency of the dynamic characteristics of the rubber bushing to investigate the ride comfort and other vehicle dynamics performances concurrently during the early phases of vehicle development including MBS using the same parameters.

DESIGN SPECIFICATIONS FOR THE RUBBER BUSHING

The dynamic characteristics of the rubber bushing are generally defined by three values [8]:

- static stiffness K (N/mm)
- dynamic multiplication d (-)
- loss factor L_f (-)

Dynamic multiplication is the ratio of storage stiffness K' (N/mm) and static stiffness K at a specific frequency and amplitude as defined below (1).

$$d = \frac{K'}{K} \quad (1)$$

If static stiffness K and dynamic multiplication d are provided, the specified storage stiffness of the rubber bushing K' can be calculated as follows (2):

$$K' = d K \quad (2)$$

Also, static stiffness can be replaced by the static load-deformation curve when the static characteristics of the rubber bushing show a non-linear change. This is sometimes observed when it has a void in the rubber material part. However, we can use an averaged spring constant near the unloaded position when small amplitude area is considered.

The loss factor L_f is the ratio of loss stiffness K'' and storage stiffness K' at a specific frequency and amplitude as defined below (3).

$$L_f = \frac{K''}{K'} \quad (3)$$

Generally, the dynamic multiplication d is specified at a frequency of 100Hz and amplitude of ± 0.05 mm, and the loss factor is at 15Hz and ± 0.5 mm respectively [8].

DYNAMIC CHARACTERISTICS OF AN ACTUAL RUBBER BUSHING

Actual Measurement Results

The measurement results of storage stiffness K' and loss factor L_f of an actual cylindrical rubber bushing are shown in Figures 4 and 5, respectively, with frequency as the horizontal axis. Figure 4 shows the storage stiffness and Figure 5 shows the loss factor for various excitation amplitudes.

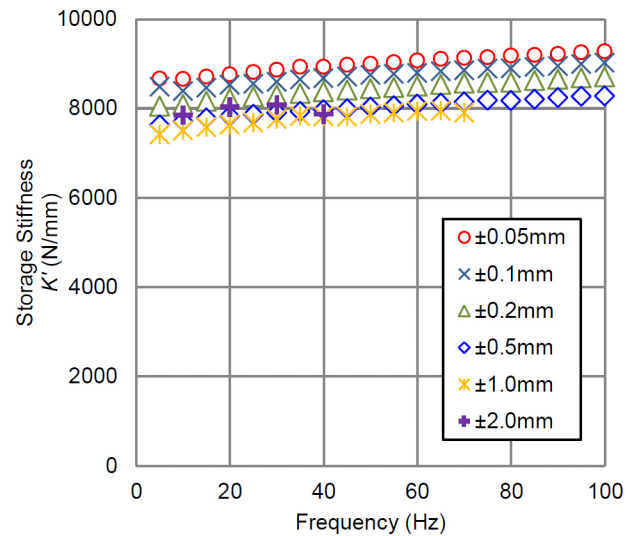


Figure 4. Measured storage stiffness of actual rubber bushing for various excitation amplitudes

In Figure 4, storage stiffness was changed depending on the amplitude and frequency.

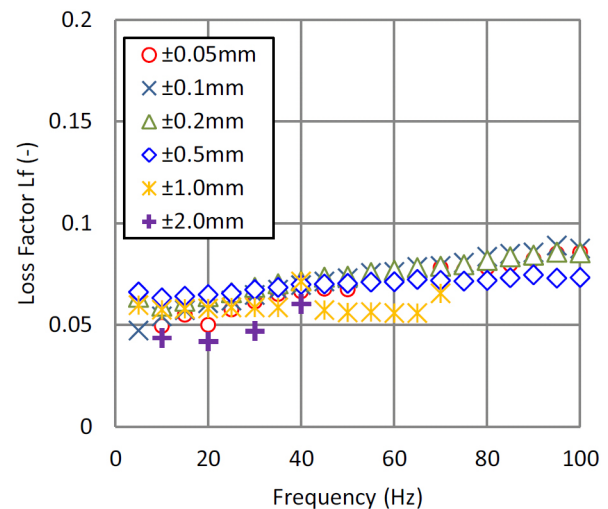


Figure 5. Measured loss factor of actual rubber bushing for various excitation amplitudes

In Figure 5, loss factor showed frequency dependency, but amplitude dependency was smaller than storage stiffness.

Comparison of Amplitude and Frequency Dependency

In Figure 6, the measurement results of storage stiffness shown in Figure 4 are plotted with amplitude as the horizontal axis. The storage stiffness is plotted with several excitation frequencies. In this figure, the storage stiffness was been smallest at amplitude of ± 1 mm and increases when the amplitude becomes smaller.