

Non-linear CAE Technology to Balance Handling and Ride Comfort

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ABSTRACT

Honda's in-house vehicle dynamics simulation tool has been enhanced to balance ride comfort and handling performance more effectively. To deal with the characteristics that affect ride comfort, ride comfort tire model was newly introduced and its parameter identification method was developed for better reproduction of vibration transmission characteristics from road to tire. In order to consider suspension vibration transmissibility from tire to vehicle body, simulation models for rubber bushings, ball joints, and dampers were newly developed that can consider amplitude and frequency dependence. Their parameter calculation methods were also developed. In addition, to enhance the reproducibility of a vibration originated from engine, a hydraulic engine mount model that can reproduce amplitude and frequency dependence as well as its parameter identification method were developed. By incorporating the above technologies into our vehicle dynamics simulation tool, it became possible to study design more detailed specifications to balance handling and ride comfort performance at early stages of vehicle development.

1. Introduction

Contemporary vehicle development is expected to create attractive products that balance driving pleasure and comfort as well as good fuel efficiency and safety. In addition, enhancement of the product development process to support rapidly changing and diversifying customer demand is a common issue for vehicle manufacturers.

Under these circumstances, CAE is increasingly important to enhance the efficiency and quality of vehicle development. CAE is used in the early phases of vehicle development to study design concepts that balance opposing performances such as fuel efficiency and dynamic performance. CAE is also widely applied in later development stages, for example to predict vehicle performance and enhance specifications in the stage where vehicle and subsystem characteristics are reflected in the drawings. Multibody simulation (MBS) is commonly used to evaluate vehicle dynamics performance areas during this process.

MBS is a nonlinear numerical simulation method that calculates the dynamic behavior of mechanical systems comprising multiple parts. MBS solvers formulate equations of motion from inertia properties of bodies and kinematic constraints or force elements between parts, and solve these equations numerically to simulate the complex behavior of

mechanical system⁽¹⁾.

Honda began employing MBS in the vehicle dynamics performance area from the latter half of the 1980s. Honda initially applied MBS to evaluate suspension kinematics, for example prediction of alignment characteristics during the suspension stroke. Thereafter, use was expanded to various analyses, such as vehicle handling simulations and flexible multibody dynamics simulations⁽²⁾. In the area of handling performance, MBS is used to study the characteristics of parts that make up subsystems such as the suspension to meet subsystem requirements as determined by the design concept. Design employing MBS to predict performance feasibility in parallel with parts layout is now standard practice.

Handling and stability as aspects of handling performance are mainly studied by analyzing the motion of rigid bodies up to 5 Hz. In contrast, the study of ride comfort, which has a trade-off relationship with handling performance, deals with vibration phenomena up to around 50 Hz, making the expression of nonlinear parts a key factor. The nonlinearity of parts such as the rubber bushings, dampers, and tires and changes in the vibration transmission characteristics due to the amplitude and frequency of vibration have a large impact on the transient response when running on an uneven road. Because this calls for detailed modeling of these parts when studying

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ride comfort, it is not a simple matter to consider the balance between handling and ride comfort performance in the product planning phase, and ride comfort has generally been studied using prototype vehicles in the later stages of development.

However, as mentioned above, ride comfort performance should essentially be designed in parallel with handling performance. This is because enhancement of both types of performance calls for a good balance between responsiveness and isolation characteristics, and to achieve this balance it is necessary to appropriately set subsystem requirements and design the specifications of subsystem parts before determining the overall vehicle layout.

Against this background, the authors expanded the functions of an MBS design tool conventionally used to predict handling performance by adding model elements related to ride comfort. This made it possible to study handling and ride comfort performance in parallel for the same parts specifications in the initial phase of vehicle development.

2. Development Goals

Ride comfort refers to the phenomenon where input from the road is transferred to the vehicle body and steering system via the tires and suspension and perceived as vibration by vehicle occupants. Figure 1 shows typical phenomena that influence ride comfort.

For example, on an undulating road, sprung-mass vibration of 1 to 2 Hz in frequency occurs with relatively large amplitude of several mm to 10 mm or more. Drivers perceive this as a float feel, so the balance between front and rear natural frequency is designed to achieve rapid convergence of sprung-mass vibration.

Even when running on a smooth-surfaced road, unevenness of the road surface may generate suspension strokes with amplitude of 1 mm or less. Under these conditions the nonlinear characteristics of parts such as the dampers, ball joints, and rubber bushings result in a

boulevard jerk feeling with vibration of approximately 3 to 8 Hz, which is an issue for ride comfort.

In addition, rotational shimmy of the steering system is a form of vibration where imbalance between the tires and wheels generates periodic longitudinal input to the suspension. This input frequency is proportional to the running speed, and occurs in the range of approximately 10 to 20 Hz, which is near the longitudinal natural frequency of the unsprung mass.

As this indicates, ride comfort phenomena occur over a wide range of amplitudes and frequencies. To enhance performance in this area, it is therefore necessary to design characteristics that contribute to ride comfort in the target frequency and amplitude ranges, using models able to accurately reproduce parts with nonlinear characteristics.

Furthermore, to perform these studies in the initial phase of development, it is desirable to quickly build models from parts specifications, predict handling and ride comfort performance using the same model, and study parts specifications that balance the respective performance demands.

Therefore, the following three goals were set to address the above issues.

- (1) Realization of the ability to express the nonlinearity of parts within the range of amplitudes and frequencies relevant to ride comfort phenomena.
- (2) Realization of functionality that enables designers to study performance for typical ride comfort phenomena with simple operations.
- (3) Realization of accuracy sufficient for studying parts specifications.

3. Model Expression related to Ride Comfort Characteristics

Figure 2 shows the main elements comprising the dynamic stiffness of the suspension in the vibration transmission paths from the tires to the body.

Vehicle motion is excited by input from the tires, so

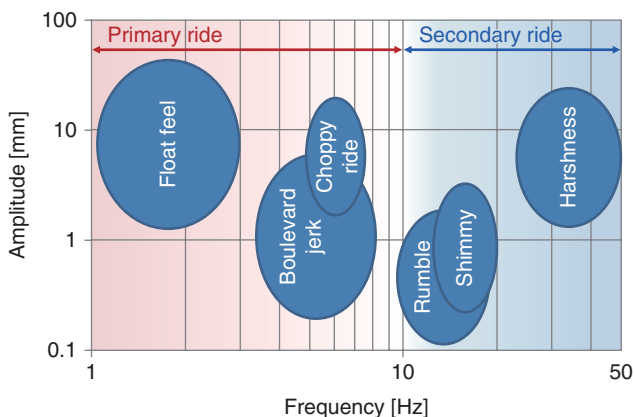


Fig. 1 Typical ride comfort feeling

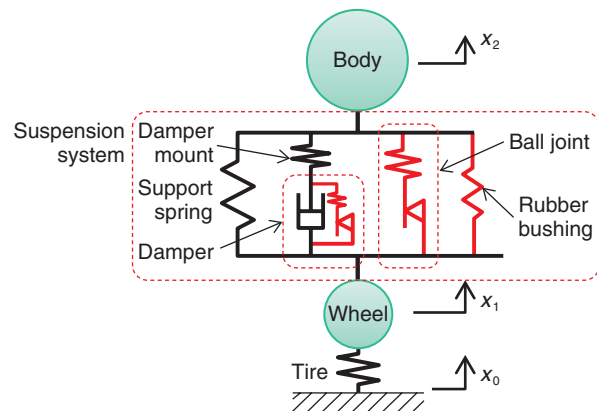


Fig. 2 Quarter-car model including hysteresis elements in suspension components

the reproduction accuracy of the forces generated on the tires has a large influence on the predictive accuracy of simulations for both handling and ride comfort performance.

In addition, the forces generated on the tires move the vehicle body via the suspension arms, springs, and dampers, but it is known that the friction of the joints and the stiffness of the dampers and rubber bushings in the transmission paths fluctuate greatly depending on the amplitude and frequency of the input. Furthermore, the characteristics of the engine-mount bushings that support the engine also greatly affect ride comfort and NVH. Incorporation of these characteristics models into the MBS design tool would make it possible to study ride comfort and NVH characteristics, which have a trade-off relationship with handling, together with handling in the initial phase of development, thus enabling creation of design proposals that achieve a good balance of conflicting performances. This chapter gives an overview of this modeling method.

3.1. Ride Comfort Tire Model

Due to increasing demand for better fuel efficiency, rolling resistance is being reduced by reducing the thickness of tire structures and using rubber compounds with low hysteresis loss. As a result, characteristics related to handling and ride comfort performance are changing greatly. Examples of computer simulation tire models (hereafter, tire models) that can reflect these changing tire characteristics include RMOD-K or FTire models that use small tread segments to reproduce both detailed tire contact patch behavior and whole tire structural responses, and FEM tire models that reproduce details down to the internal structure of tires⁽³⁾⁻⁽⁶⁾.

Honda adopted the theory of Magic Formula (MF)⁽⁷⁾ published in 1987 to study handling performance when beginning to apply MBS to chassis development. Since then, in-house tools have been used to create an MF tire model from measured characteristics, and the tire model has been enhanced while examining performance in the nonlinear region where lateral and longitudinal forces are combined, such as BIT (Braking In Turn).

In addition, Honda quickly focused on the 1st-order lag characteristics equation (1) of the lateral force transmission due to lateral deflection of the tire, theoretically clarified its influence on both vehicle dynamics performance and the driver-vehicle system⁽⁸⁾⁻⁽¹⁰⁾, and incorporated it into the tire model.

In vehicle development it is increasingly important to simultaneously consider the nonlinear characteristics and frequency dependency of the bushings, the stiffness of the body, and other characteristics in the initial phase of development when deciding the suspension member layout and the body framework, in order to find the best trade-off solution between ride comfort and handling performance^{(2), (11), (12)}. As a result, it was necessary to expand the functions of Honda's tire model to handle even higher-frequency vibration issues.

$$\tilde{F}_{y0}(t) = \int_0^t F_{y0}(\tau) \frac{1}{T_i} e^{-\frac{t-\tau}{T_i}} d\tau \quad \text{Magic Formula with 1st order lag} \quad (1)$$

where

$$\left\{ \begin{array}{l} T_i(\alpha|F_z) = \frac{\frac{\partial F_{y0}}{\partial \alpha_y}|_{\alpha_y=0}}{K_y V} \quad \text{Time constant} \\ F_{y0}(t) = D_y \sin \{ C_y \arctan \{ B_y \alpha_y - E_y [B_y \alpha_y - \arctan(B_y \alpha_y)] \} \} \end{array} \right.$$

When focusing on reproducibility of tire force input when running on a typical road with a frequency range up to approximately 50 Hz and undulation amplitude up to approximately 10 mm, the MF-Swift model⁽¹³⁾ that simulates behaviors of elastically supported tire tread rings excited by road undulation offers good calculation speed and is highly compatible with handling performance. Honda is therefore working to standardize this MF-Swift tire model and its application scheme in development.

The MF-Swift model is accurate enough to reliably reproduce measured cornering characteristics such as longitudinal force, lateral force, and SAT, as shown in Fig. 3. With regard to ride comfort characteristics, the model can accurately reproduce the enveloping characteristics of tires, that is to say the behavior of the tread surface to envelope road surface unevenness at low speeds, as shown in Fig. 4.

In addition, the reproduction of transient tire forces when the tires roll over a cleat in the road also shows good approximation of the peak values, frequency, and convergence of vibration (Fig. 5).

As tire measurements for the identification of tire models need large-scale, specialized engineering facilities, Honda had mainly relied on tire manufacturers and/or external organizations for these measurements. However, a tire tester was newly developed and introduced to measure tire characteristics that are not affected by test

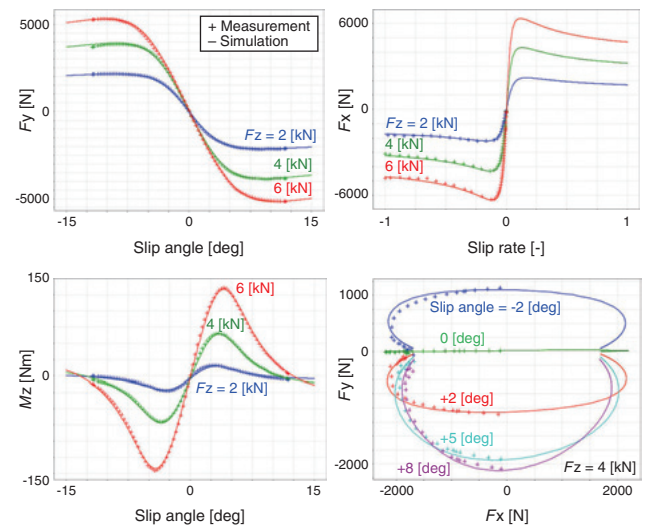


Fig. 3 Example of force and moment characteristics of tire 175/65R15, IP = 220 [kPa], measurement and its simulation with MF-Swift

facilities or conditions. This enabled the formulation of models using the test results and accurate comparison of tire characteristics⁽¹⁴⁾. The use of MF-Swift models helps to accurately predict input forces from tires, which is important for the study of ride comfort phenomena, and enables theoretical examination of the trade-off between handling and ride comfort.

3.2. Suspension Friction Model

When addressing vibration phenomena, measures are put into effect from low frequencies that have a broad influence. An effective approach for addressing the low-frequency ride comfort phenomena of boulevard jerk and rumble is to reduce the friction included in the ball joints and dampers

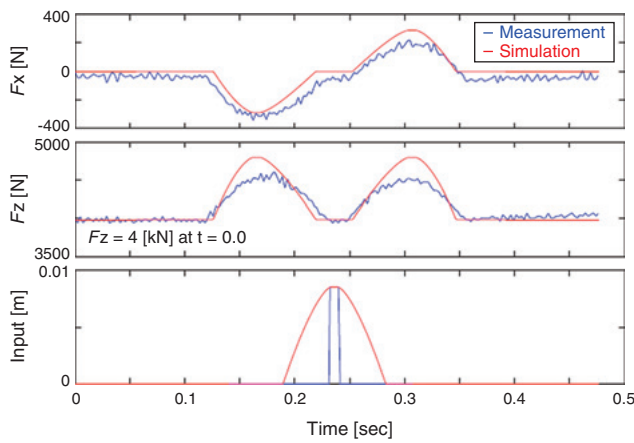


Fig. 4 Example of enveloping behavior of tire 175/65R15 with $F_z = 4$ [kN], $IP = 220$ [kPa], $V = 0.27$ [km/h], obstacle of 8.5×8.5 [mm], measurement and its simulation with MF-Swift

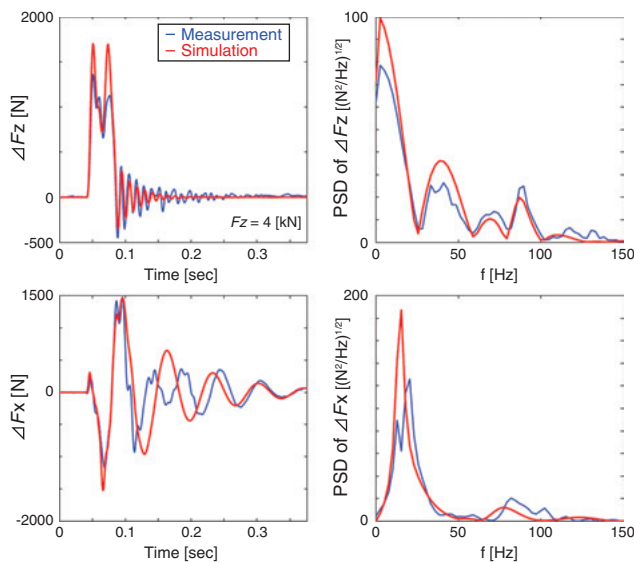


Fig. 5 Example of 2D (15 × 15 [mm]) 90 [deg] cleat response of tire 175/65R15 $V = 20$ [km/h]

that contributes to these phenomena during the suspension stroke, in order to suppress the increase in the natural frequency of the suspension in the small amplitude range⁽¹⁵⁾.

3.3. Modeling of Damper Force

Reducing the damping force is an effective means of addressing the boulevard jerk and rumble phenomena produced by the dampers, but this also reduces the handling performance, so it is not easy to achieve a good balance.

However, if the damping force at small velocity area that contributes to ride comfort and the damping force needed to suppress vibration for handling behavior could be adjusted separately, this would enable studies to achieve a better balance between the two demands.

Therefore, the force produced by the dampers was divided into a component originating in friction and one originating in the oil as shown in Fig. 2, and the nonlinear series spring elements in the dampers were taken into consideration. Use of this model made it possible to reproduce the damper force, which changes according to the frequency and amplitude, as shown in Fig. 6, and to separately design the damper characteristics in the range of frequencies and amplitudes associated with handling and ride comfort phenomena.

3.4. Rubber Bushing Model

The rubber bushings are the structural elements that exhibit the next-greatest degree of nonlinearity after the tires and dampers. The spring characteristics of the damper mount rubber and the compliance bushings in particular contribute greatly to the handling characteristics, and the amplitude dependency and frequency dependency of these characteristics also contribute to ride comfort performance.

Figure 7 shows a schematic of the applied model⁽¹⁶⁾. This model consists of three nonlinear elements: a displacement-dependent spring element in parallel mainly used to study handling performance, and amplitude- and frequency-dependent elements mainly used to study ride comfort performance.

This model is compatible with MBS, and can be used to analytically study dynamic stiffness together with time-

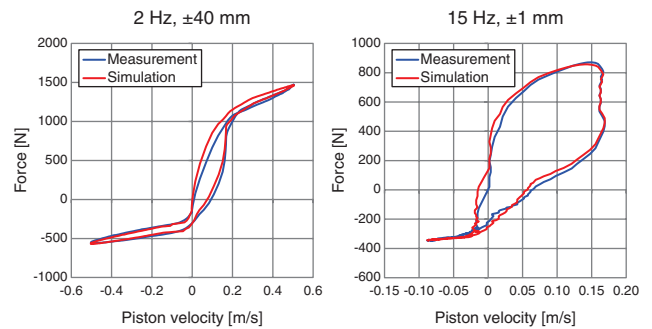


Fig. 6 Example of damper force characteristics with different input amplitude and frequency

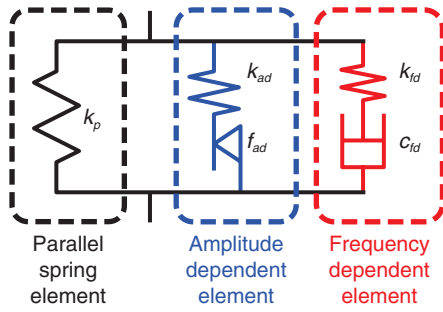


Fig. 7 Rubber bushing model for vehicle dynamics performance design

series simulations that take a certain amount of calculation time, enabling more efficient study of performance.

When the friction force f_{ad} and the series spring constant k_{ad} are given, the storage stiffness $K_{ad}'(a)$ and loss stiffness $K_{ad}''(a)$ of the amplitude-dependent element can be determined from the following equations where a denotes vibration amplitude.

$$K_{ad}'(a) = \frac{f_{ad}}{a} \tanh \frac{a}{a_{ad}} \quad (2)$$

Table 1 Parameters for rubber bushing model

Static stiffness K	7500 N/mm
Dynamic multiplication d	1.24 (± 0.05 mm, 100 Hz)
Loss factor L_f	0.05 (± 0.5 mm, 15 Hz)
Freq. dependence ratio r_{fd}	690 N/mm/100 Hz

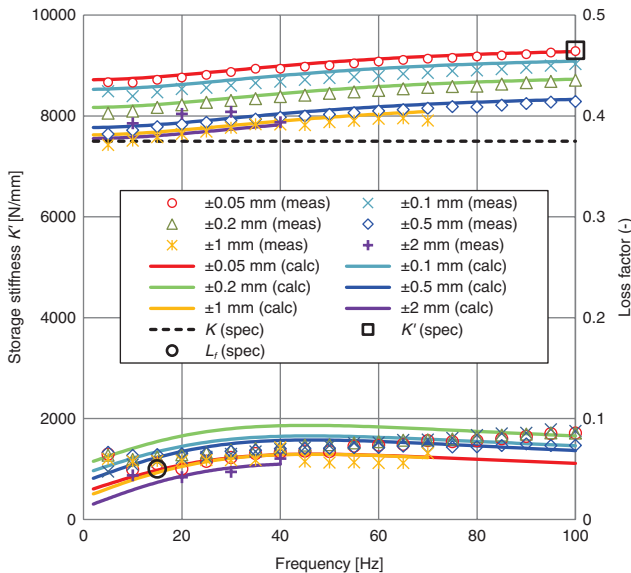


Fig. 8 Comparison between measurement and calculation results for dynamic stiffness of rubber bushing

$$K_{ad}''(a) = \frac{4f_{ad}}{\pi a^2} \left(a - a_{ad} \tanh \frac{a}{a_{ad}} \right) \quad (3)$$

Likewise, when the viscous damping factor c_{fd} and the spring constant k_{fd} are given, the storage stiffness $K_{fd}'(\omega)$ and loss stiffness $K_{fd}''(\omega)$ of the frequency-dependent element can be determined from the following equations where ω denotes vibration frequency.

$$K_{fd}'(\omega) = \text{Re} [G(j\omega)] = \frac{\frac{c_{fd}^2}{k_{fd}} \omega^2}{\left(\frac{c_{fd}}{k_{fd}} \right)^2 \omega^2 + 1} \quad (4)$$

$$K_{fd}''(\omega) = \text{Im} [G(j\omega)] = \frac{c_{fd} \omega}{\left(\frac{c_{fd}}{k_{fd}} \right)^2 \omega^2 + 1} \quad (5)$$

By calculating the given parameters for Eq. (2) to Eq. (5) from the minimum parameters that express the dynamic characteristics of the part (Table 1)⁽¹⁶⁾, the dynamic stiffness of the rubber bushing can be reproduced and applied to study performance as shown in Fig. 8.

3.5. Hydraulic Mount Model

The use of hydraulic mounts to enhance ride comfort and quietness has become the main solution for the mounting of automotive power plants. The basic structure and principle of operation of these mounts are described below (Fig. 9). The interior of the main rubber of a hydraulic mount is filled with fluid, and closed off with a diaphragm. A separator is used to form two fluid chambers, with an orifice running between them. This orifice is called a shake orifice, and the damping characteristics can be determined by the shape of the main rubber and the inner diameter and length of the orifice. When the main rubber is deformed by a load, fluid passes through the orifice, producing damping effects at a specific frequency due to the frequency response of the fluid mass within the orifice. This frequency is normally set near the resonant frequency of the power plant.

The resonance of the shake orifice produces damping effects, and at the same time the internal pressure of the mount rises, which increases the dynamic stiffness with small amplitude and would increase the amplitude

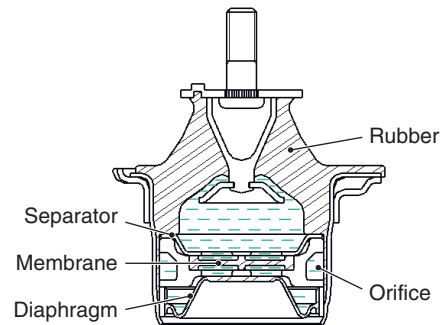


Fig. 9 Basic structure of hydraulic engine mount

dependency of the dynamic characteristics. To address this issue, low dynamic multiplication materials are generally applied for the main rubber, and a liquid pressure-absorbing mechanism such as a membrane is also often used.

To accurately model these hydraulic mount characteristics, the actual phenomena are reproduced in the simulation by using a combination of multiple-degree-of-freedom models as shown in Fig. 10.

Here, F_1 , F_2 , and F_3 are respectively defined to reproduce the static characteristics of the main rubber, the frequency dependency of the main rubber, and the expansion stiffness and amplitude dependency produced by the hydraulic effect. The total of these forces is the transmitted force relative to the mount displacement ($X - X_g$).

The above process makes it possible to reproduce the complex characteristics of hydraulic mounts in simulations (Fig. 11). This increases the predictive

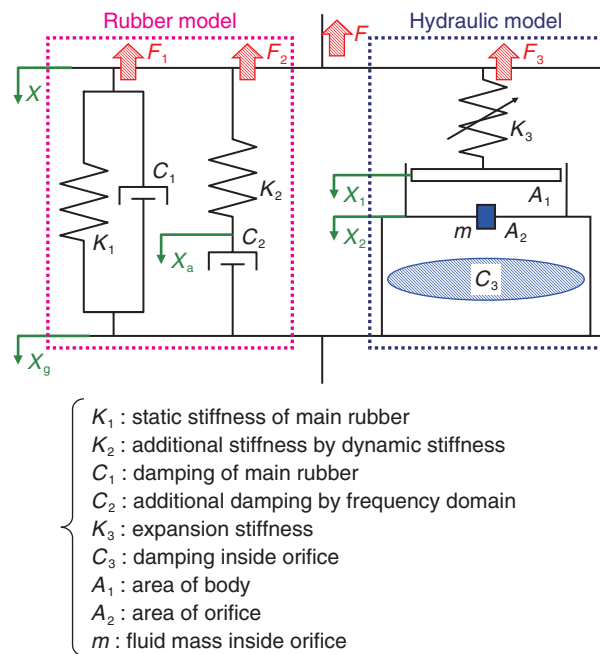


Fig. 10 Hydraulic engine mounting model for MBS

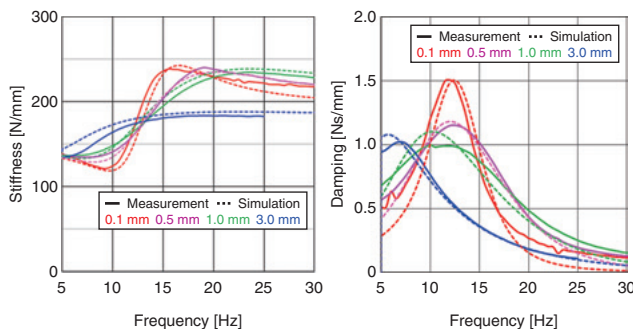


Fig. 11 Characteristics of hydraulic engine mounting model

accuracy of simulations under conditions of non-steady-state displacement that approximate actual operating conditions in the initial design phase, and enables setting of the optimum engine mount positions and sizes, which are challenging to change from the intermediate design phase onward (Fig. 11).

4. Dynamic Performance Design Tool

Studies using MBS that take into account the model elements discussed in the previous chapter are needed to design specifications of parts that balance handling and ride comfort.

However, it is not easy for designers to create models with the degree of detail needed to perform studies, run simulations, and analyze the results. Therefore, dedicated software was created that enables designers to easily perform this series of operations.

Figure 12 shows an example of screens used to define bushing characteristics in the MBS tool for designers. When the designer defines parts specifications such as static stiffness and dynamic stiffness, those characteristics are automatically applied to the simulation model.

In addition, Fig. 13 shows an example of an excitation test screen. The designer simply inputs the test conditions

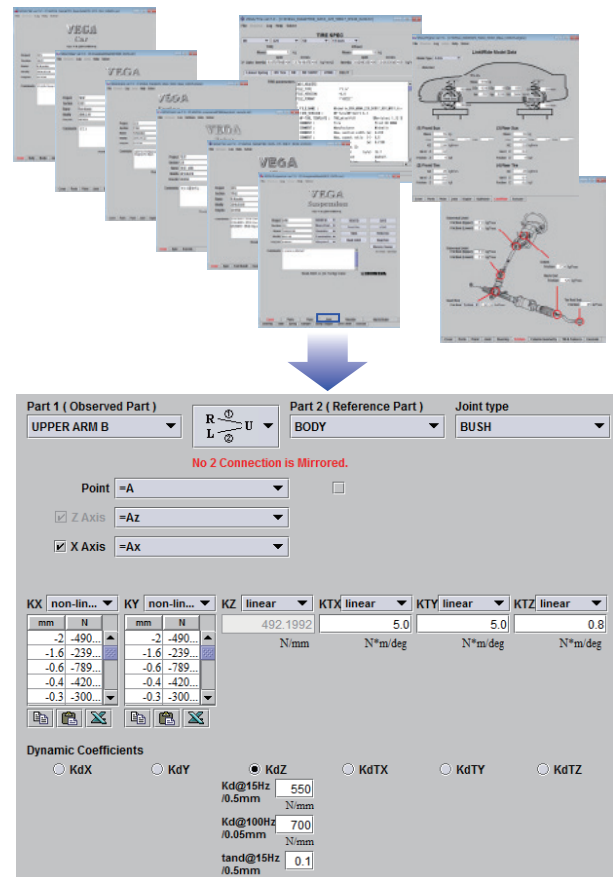


Fig. 12 Joint definition panel of MBS tool for designer

such as the subsystem assembly conditions, the excitation points, and the excitation waveform, and the software automatically generates a simulation model based on the model definitions and test conditions (Fig. 14), and runs the simulation.

Analysis screens are provided with menus offering simulations of typical vehicle tests and bench tests. The designer can run the necessary simulations and study parts specifications that achieve a balance between handling and ride comfort.

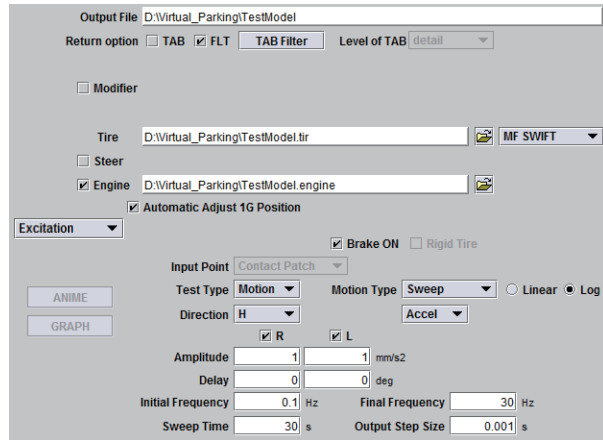


Fig. 13 Chassis excitation test panel in MBS tool

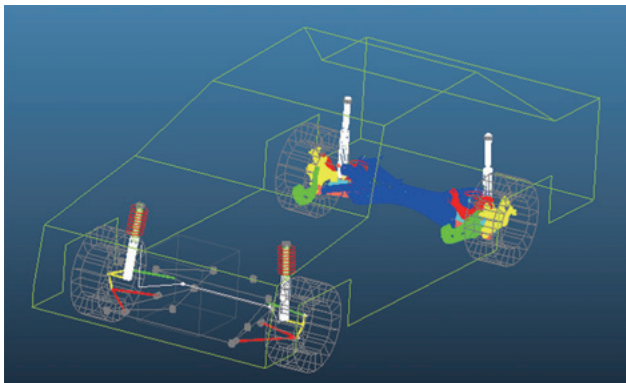


Fig. 14 MBS full vehicle model

5. Case Studies

5.1. Balancing Yaw and Roll Phase Delay and Flat Ride Performance

The phase delay of the vehicle yaw rate, lateral acceleration, and roll angle in response to steering are important characteristics that affect the feeling of responsiveness of steering operations. When steering operations produce a cornering force on the tires, first yaw motion is generated, followed by lateral motion and roll

motion. When the phase difference between this series of motions is large, the driver loses a sense of responsiveness and nimbleness in steering operations.

For example, correction of the roll phase delay relative to yaw motion is achieved by optimally designing specifications such as the roll center height in the initial phase of development, and then optimizing specifications such as the roll stiffness in later phases of development to enhance the ride feel. However, the roll center height is subject to layout constraints such as the minimum height above ground, and if the roll stiffness is adjusted by means of the suspension springs, ride comfort performance in the float feel region is affected. In the actual development process it is necessary to determine parts specifications in consideration of these opposing performance demands.

Figure 15 shows an example of evaluation results for yaw and roll phase delay and vertical vibration transmission characteristics in a midsize vehicle, with the front suspension specifications as the design variables.

Looking first at the spring rate, increasing this parameter reduces the phase delay, but also increases the natural frequency of the sprung mass, which reduces the ride comfort performance. Next, modification of the stabilizer stiffness and the roll center height makes it possible to reduce the phase delay with little change in the ride comfort performance, but in practice it is necessary to consider durability and layout constraints. By understanding the contribution of the characteristics of each part and studying their combinations in this manner, the proposed specifications can be determined as shown in Fig. 15.

Other parameters that affect ride comfort performance, such as the dynamic multiplication of the suspension bushings, can also be made the subject of design focus, and specifications from suspension kinematics to dynamic stiffness can be examined theoretically while evaluating handling and ride comfort performance.

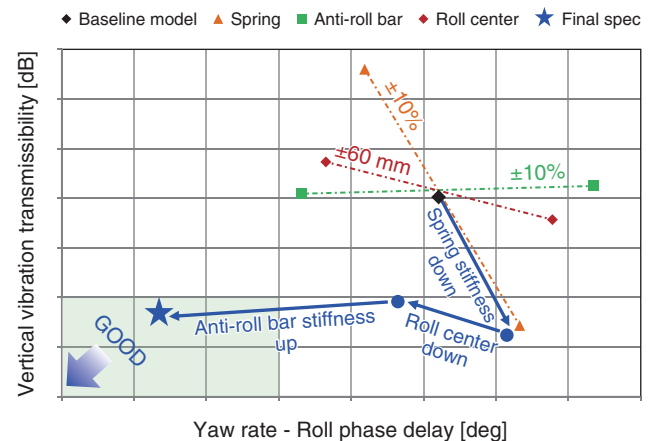


Fig. 15 Example of parameter study in which spring, anti-roll bar and roll center height are design parameters

5.2. Balancing Ride Comfort and Idling Vibration

Idling vibration is the phenomenon where periodic excitation force corresponding to the engine idling speed is generated around the engine roll axis and transmitted to the vehicle body via the engine mounts and other parts.

Therefore, the basic principle for addressing this issue is to separate the frequency of the excitation force and the resonant natural frequency of engine mount system. However, lower idling speeds are demanded from the perspective of fuel efficiency, so if the mount stiffness is reduced to help separate the natural frequency from the excitation force frequency, ride comfort performance declines in the primary region.

For this reason, when addressing idling vibration, the level of vibration is reduced by the rotational elastic axis of the engine mount system closer to the engine center of gravity in order to align the rotational elastic axis and the torque roll axis.

In addition, with regard to ride comfort relative to vertical input from the road, it is known that the level of sprung-mass vibration can be reduced by adjusting the distribution and positioning of mount stiffness to couple the vertical mode and pitch mode⁽¹⁷⁾.

Figure 16 shows an example of evaluation of the vertical vibration transmission properties of the vehicle body when the tire contact patches of a compact car are excited in the vertical direction. Test and simulation results were compared for two specifications: the baseline specification and the specification where the engine side mounts use hydraulic mounts with the characteristics shown in Fig. 17. In Fig. 17, K_d represents storage stiffness and K_l represents loss stiffness. These spring constants change according to the frequency, and are plotted for each amplitude. By positioning the damping peak of the hydraulic mounts close to the resonant frequency of the power plant, the vertical vibration level of the vehicle body can be reduced while at the same time realizing the low spring characteristics needed to secure idling vibration performance.

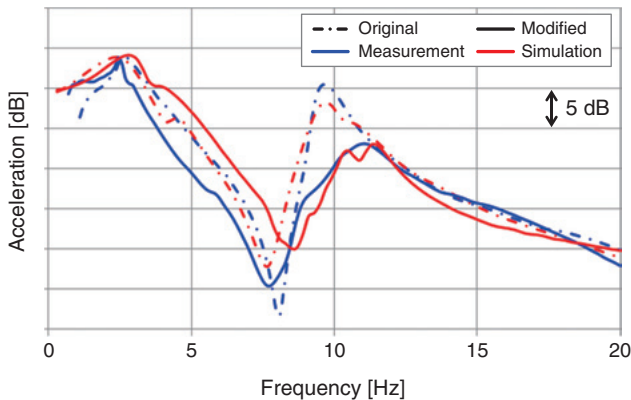


Fig. 16 Example of body vertical acceleration response in four-poster test

This demonstrates that the use of parts models enabling study of multiple functions makes it possible to accurately predict the performance of the complete vehicle in the initial phase of vehicle development, and to quantitatively evaluate the effects of using parts.

5.3. Harshness

Harshness is the phenomenon where transient vibration and noise are produced in the vehicle body when the vehicle passes over differences in level such as road seams or bridge joints. The backward inclination angle of the wheel center trajectory ϕ , the damper inclination angle θ , and the longitudinal stiffness of the suspension during the vertical suspension stroke are known to contribute to harshness performance as shown in Fig. 18^{(18), (19)}. When designing the suspension, changes in the virtual arm angle and the damper inclination angle need to take into account factors such as the positioning of the driver's heel point and the hip points of the rear-seat passengers, the balance with lowering of the floor, and changes in the vehicle posture

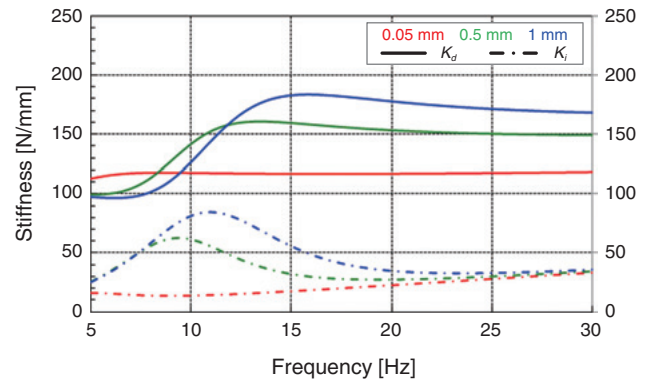


Fig. 17 Example of dynamic characteristics of hydromount bush model

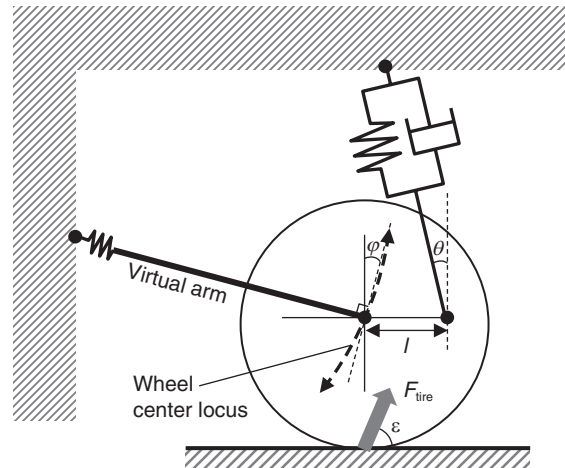


Fig. 18 Side view of conceptual suspension model with virtual arm and spring-damper

when accelerating and braking. Regarding longitudinal stiffness of the suspension, in order to achieve a balance with requirements related to handling performance, in many cases harshness performance is secured by reducing the dynamic multiplication or increasing the damping of the compliance bushings.

As an example, Fig. 19 compares the driving test and simulation results the rear suspension of a midsize vehicle. There is good correspondence between the calculated values and measured values, indicating that the developed calculation model makes it possible to judge the viability of, and performance prospects offered by, design proposals in consideration of the dynamic characteristics of the parts.

Figure 20 shows the results of a simulation study of the relationship between body input force and the design variables (damper inclination angle θ and virtual arm angle φ), based on the verified simulation model. The total body input force is reduced when the virtual arm is positioned so that the wheel center trajectory has a backward inclination and the damper inclination angle θ is negative (forward inclination).

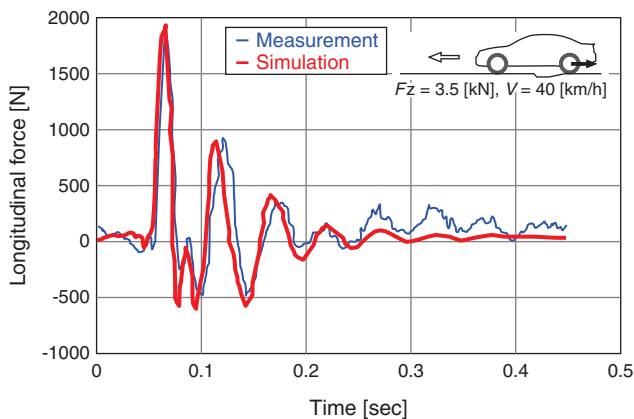


Fig. 19 Example of body longitudinal force ($V = 40$ km/h)

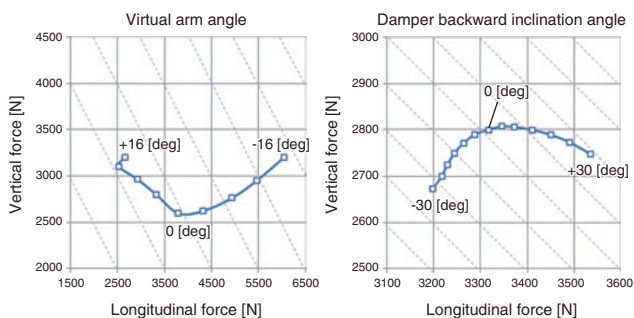


Fig. 20 Relationship between body input force, virtual arm angle, and damper inclination angle

6. Conclusion

This paper discussed the addition of model elements related to ride comfort to a design tool conventionally used to study handling performance in order to conduct studies aimed at achieving a balance between handling and ride comfort performance.

- (1) Specifications used in the models of bushings, dampers, and tires, which are parts that affect ride comfort performance, can be applied as is to study ride comfort performance in parallel with handling performance.
- (2) The developed tool enables simplified study of the rubber bushing, suspension damper, and engine mount characteristics needed to study ride comfort.
- (3) Comparison of the results of performance predictions using the developed tool with the results obtained in actual vehicles demonstrated that the developed tool possesses sufficient accuracy for initial studies, and can be used to study specifications that balance handling and ride comfort performance down to the level of detail of individual part specifications.

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