

Prediction of Transient Body Deformation during Handling Maneuvers

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ABSTRACT

A multi-body dynamics model that considers elastic deformation of the body was formulated in order to predict transient body deformation, a factor that affects handling. A comparative analysis with body deformation during handling maneuvers identified using a modal forced response method was conducted, and a good correlation was obtained between vehicle dynamic performance, transient body deformation, and the body modal contribution factor.

1. Introduction

It is well known that body characteristics affect the quality of the dynamics of a vehicle, such as handling performance, ride comfort, and noise and vibration.

In the area of handling performance, examples have been reported of prediction of body deformation during handling maneuvers employing models of body characteristics formulated using a variety of methods in combination with multi-body dynamics models^{(1),(2)}. Examples have also been reported in which the mechanical input to the suspension coupling points on the body framework has been identified from the component forces at the tire contact points and used as the basis for reproduction of body deformation in tests and simulations^{(3),(4)}. There is, however, a paucity of examples of detailed measurement of body deformation during handling maneuvers.

The research discussed in this paper used a modal forced response method to identify the forces acting on the suspension coupling points on the body framework, and rendered body deformation during handling maneuvers visible by inputting these forces into a modal model⁽⁵⁾. The simulation model was a combination of an elastic body model of the body formulated using the residual vector method, and a multi-body dynamics model⁽⁶⁾. This paper will discuss the results for body deformation obtained from the modeling method employed in the project in comparison with measurements of vehicle dynamic performance and body deformation during handling maneuvers.

2. Calculation of Transient Body Deformation during Handling Maneuvers

2.1. Modal Forced Response Method

A modal forced response method was employed as the method of identifying body deformation during handling maneuvers. Strain gauges were attached close to the suspension coupling points on the body framework, and data for body strain during handling maneuvers in the time domain, $\varepsilon(t)$, were measured. Next, the front and rear suspension parts and the powerplant parts were removed from the measurement vehicle to produce a trimmed body, and a strain gauge response matrix, $H(\omega)$, was measured for input to suspension coupling points with n degrees of freedom [Eq. (1)]. The inverse matrix of the force response matrix, $H(\omega)^{-1}$, was found and the inverse fast Fourier transform (FFT) shown in Eq. (2) was performed, enabling determination of the impulse matrix $h(t)$ for the force response of the suspension coupling points at each level of strain. Convolution of this impulse matrix and the measured body strain, $\varepsilon(t)$, enabled the force input to the suspension coupling points during handling maneuvers, $F(t)$, to be identified [Eq. (3)].

Next, accelerometers were attached to the framework, panels, and suspension coupling points of the trimmed body, and electromagnetic shaker was applied to several points on the body to enable the frequency response functions of different areas of the body to be determined. Modal analysis was then employed to obtain the modal shape of the trimmed body, Ψ_r , and the m_r , c_r , and k_r matrices for the system ($r=1,2,3,\dots,m$), and a second-order

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differential equation with the contribution factor of each body mode, $q(t)$, as a variable was formulated [Eqs. (4) and (5)]. The sum of the products of the modal shape and the contribution factors gave the body deformation during handling maneuvers [Eq. (6)].

$$H(\omega) = \varepsilon(\omega)/F(\omega) \quad (1)$$

$$h(t) = iFFT(H(\omega)^{-1}) \quad (2)$$

$$F(t) = h(t) \otimes \varepsilon(t) \quad (3)$$

$$m_r \ddot{q}_r(t) + c_r \dot{q}_r(t) + k_r q_r(t) = f_r(t) \quad (4)$$

$$f_r(t) = \sum \psi_{i,r}^T \cdot F_i(t) \quad (5)$$

$$X_i(t) = \sum \psi_{i,r} \cdot q_r(t) \quad (6)$$

$h(t)$: Impulse matrix for suspension coupling points

$H(\omega)$: Strain gauge response to input at suspension coupling points

$F(t)$: Input to suspension coupling points during handling maneuvers in an actual vehicle

$\varepsilon(t)$: Body strain during handling maneuvers in an actual vehicle

ψ_r : Modal shape ($r=1,2,3,\dots,m$)

$X_i(t)$: Displacement of suspension coupling points during handling maneuvers ($i=1,2,3,\dots,n$)

$q_r(t)$: Body modal contribution factor ($r=1,2,3,\dots,m$)

m_r : Equivalent mass of trimmed body modes ($r=1,2,3,\dots,m$)

c_r : Equivalent damping of trimmed body modes ($r=1,2,3,\dots,m$)

k_r : Equivalent stiffness of trimmed body modes ($r=1,2,3,\dots,m$)

m : Number of trimmed body modes

n : Degree of freedom of suspension coupling points

2.2. Study of Strain Gauge Attachment Points

Strain gauges were attached close to the suspension coupling points on the body side in order to enable identification of input to the suspension. Strain response tests using hammering excitation were conducted on the bench to enable the selection of sites on which sufficient strain would act during handling maneuvers. When a finite element model of the relevant body was available, strain response analyses were conducted to provide a guide for points of attachment of the strain gauges. Figure 1 shows the set-up for a hammering test to measure strain response, with strain gauges attached at several points close to the spring damper coupling points. In this case, response was highest from strain gauge numbers 4 and 6. Figure 2 shows the results of performing the same test using a finite element model. Response was highest in the simulation at the points at which the highest response had

been measured in the test. This is the means by which the attachment points for the strain gauges were determined.

2.3. Screening of Strain Response Matrix

Even when measurements were taken at the strain gauge attachment points that had been carefully selected using the method described in the preceding section, non-coherent results for strain response from multiple suspension coupling points were obtained. Figure 3 shows a strain response matrix with the number of input force locations on the horizontal axis and the number of strain gauge locations on the vertical axis. Figure 4 shows the results obtained when this response matrix was used to identify the input at the coupling points of the spring dampers for the left and right front wheels. Results for the left and right wheels should be identical, but

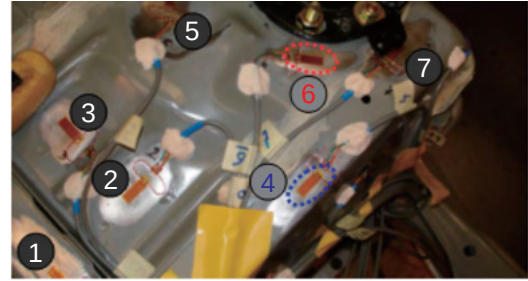


Fig. 1 Pretest for determination of strain gauge setting

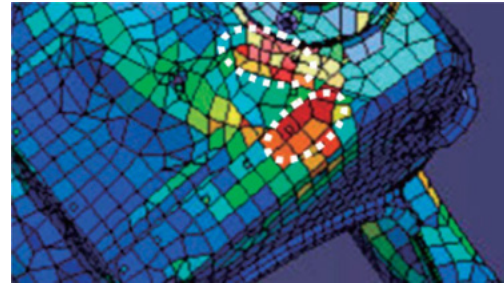


Fig. 2 Result of strain sensitivity simulation

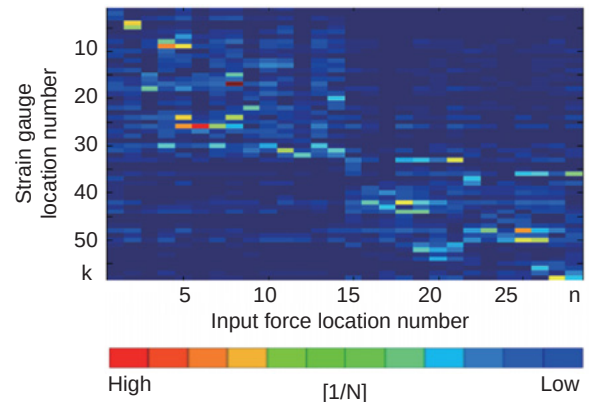


Fig. 3 Strain matrix before cleanup

differences of between two- and four-fold exist in these results.

Threshold values for coherence were therefore set, and any results falling below this threshold were judged as noise and the relevant strain response figures were deleted. Figure 5 shows the results obtained when the resulting matrix was used to identify input. The figures for input at the left and right wheels are almost identical, demonstrating that screening of the strain response matrix is an effective method of increasing the accuracy of identification of suspension input.

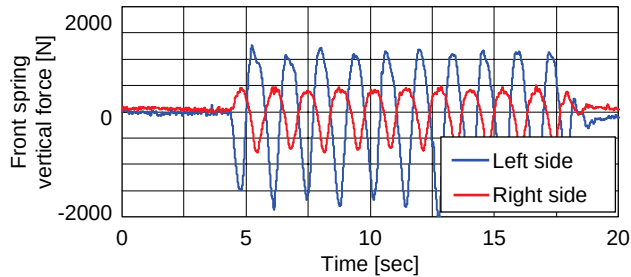


Fig. 4 Result of forced response around front spring damper connecting point before screening

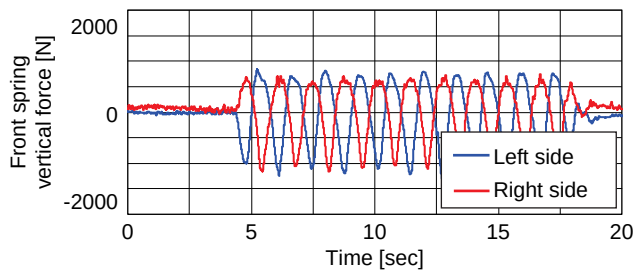


Fig. 5 Result of forced response around front spring damper connecting point after screening

3. Coupled Structural and Multi-body Dynamics Simulation

The project formulated a finite element model of the body system parts and a multi-body dynamics model of the chassis system parts. In order to enable prediction of body deformation during handling maneuvers, a reduced trimmed body model was incorporated in the multi-body dynamics model, and a coupled analysis was conducted.

3.1. Reduction of Body Model

The finite element trimmed body model, employing approximately 700 000 elements, was reduced using the residual vector method to a constrained mode up to 50 Hz and 155 degrees of freedom, including the degree of freedom of external points. This reduced simulation time, to also enable the model to be used in analyses of the

phenomenon of low-frequency noise and vibration. With regard to the total difference in mass between the real vehicle and the simulation model, the missing mass and inertia were incorporated into the trimmed body model and made to conform in a way that would not affect the elastic mode (Fig. 6). This reduction of the body model decreased simulation time to less than one-fifteenth of the original time.

3.2. Method of Formulation of Multi-body Dynamics Model of Chassis System

The front and rear suspension parts and powerplant parts were modeled using rigid elements which included items for coupling point geometry, positions of centers of gravity, mass, and inertia based on design data. The characteristics of the connections between suspension parts, powerplant parts, and body elements were updated with consideration of frequency-dependent and amplitude-dependent characteristics on the basis of a comparison with the results of measurements of the performance of springs, dampers, and rubber bushes in isolation and of an actual vehicle incorporating these components. The mechanical suspension characteristics (wheel rate, wheel center locus inclination brought about by longitudinal force, scuff, roll stiffness, roll steer, and side-force steer) and the frequency characteristic and vibration transmission rate of the powerplant, suspension, and body under forced excitation of the tires in the completed chassis model were

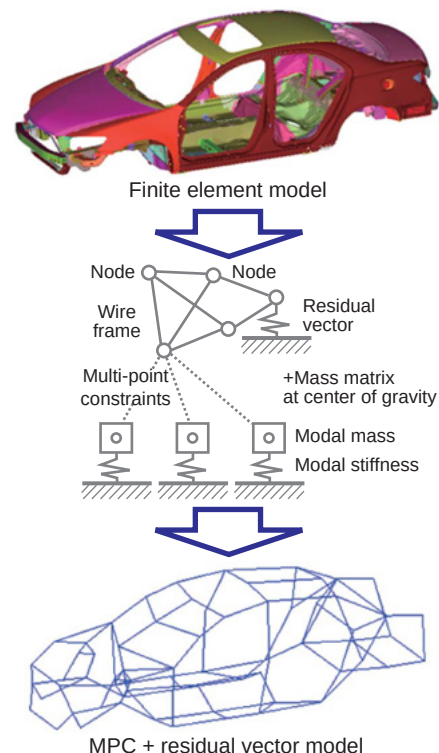


Fig. 6 Method of reduction of body model

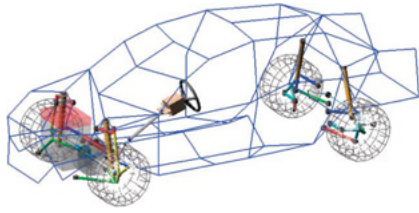


Fig. 7 Multi-body dynamics coupled with structural model

compared with measurement results and verified. Figure 7 shows the completed vehicle model formed by coupling the chassis and body models.

4. Results of Verification of Model

Test results and simulation results for body deformation during handling maneuvers were compared. The driving conditions for the actual vehicle measurements were: after a straight course on a flat road at a fixed vehicle speed of 60 km/h, slalom maneuvers (2.5 sec/cycle) with a steering input such that the peak lateral gravitational acceleration (lateral G) at the center of gravity of the vehicle was 0.4 G. The steering angle and drive force of the test vehicle, the driving conditions, and the measurements taken were input to the simulation.

4.1. Comparison of Dynamic Performance in Actual Vehicle and Simulation

Figure 8 shows a comparison of the results of the slalom maneuver test and the calculation of vehicle dynamic performance in the simulation. A comparison of the yaw rate around the vertical axis through the center of gravity, the lateral G, and the roll angle around the longitudinal axis through the center of gravity shows an error in the time domain of within 4%, indicating that the simulation is able to accurately reproduce the dynamic performance of an actual vehicle.

Figure 9 shows a comparison of results for changes in the lateral force on the suspension coupling points at the front of the front sub-frame from the vehicle test and the simulation. The test results and the input predicted by the simulation are largely identical.

4.2. Results for Body Deformation during Handling Maneuvers

Figure 10 shows the results of calculation of body deformation during handling maneuvers using input to the suspension coupling points calculated from handling data. Body deformation can be seen to change transiently in the time domain.

Figure 11 shows the results of an analysis of the contribution factor of the body modes when lateral G reaches its peak. The modes with the highest contribution factors were torsional mode, in which the entire cabin

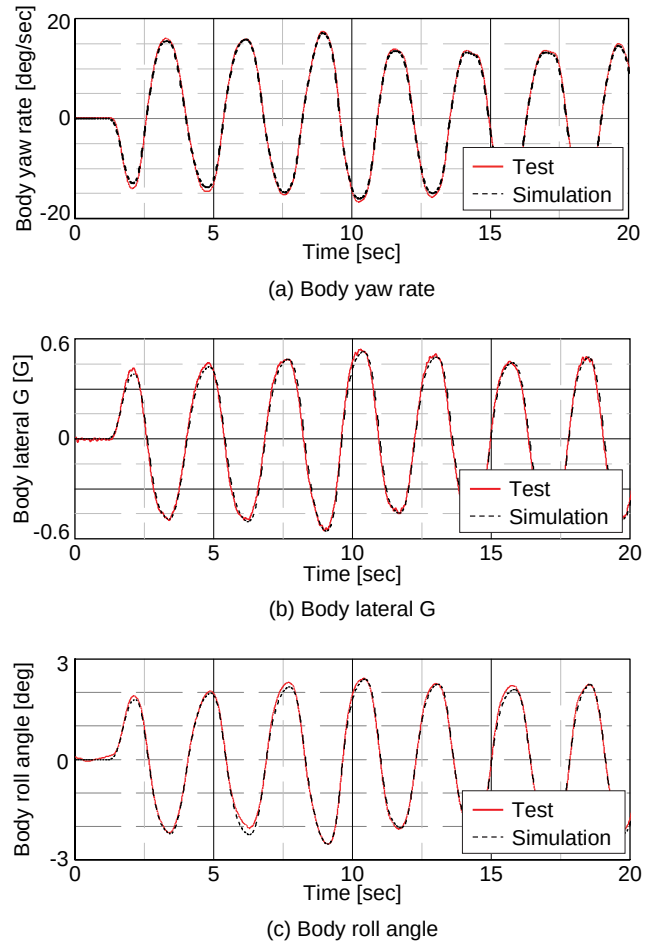


Fig. 8 Vehicle dynamics during slalom maneuver

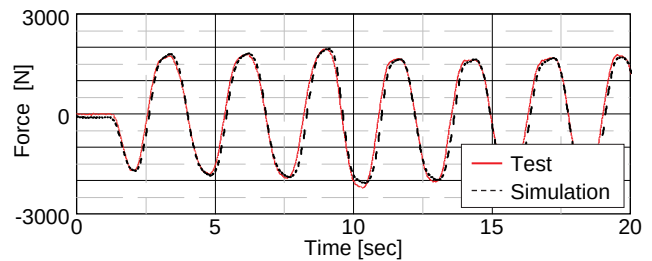


Fig. 9 Front sub-frame lateral force during slalom maneuver

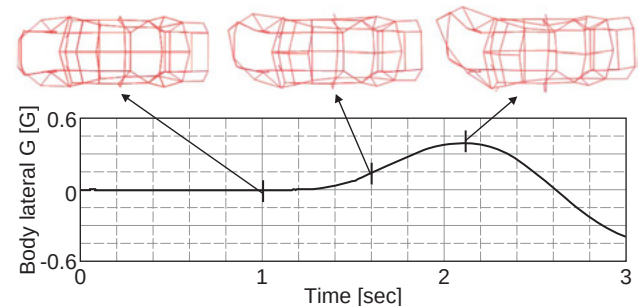


Fig. 10 Test results for body deformation

twists, and front lateral bending mode, in which the front of the frame bends laterally.

4.3. Simulation Results for Body Deformation

The steering angle and drive force from the vehicle test were used as the input conditions for the multi-body dynamics model incorporating the elastic body model, and body deformation was calculated. The results are shown in Fig. 12. A comparison of simulation results and test results for body deformation during the slalom maneuver in each time domain shows that there is a good correlation between results not only for peak values, but also during the transient period.

In addition, the simulation also showed that the torsional mode and the front lateral bending mode displayed higher contribution factors than other modes.

Figure 13 shows a comparison of test and simulation results for the contribution factors of the torsional and front lateral bending modes during handling maneuvers.

The contribution factors of both modes display a transition while changing in the time domain. The test and simulation results also correlate well. This indicates that the applied method was capable of analyzing the modal contributions that affect body deformation in test and simulation results by means of the time axes.

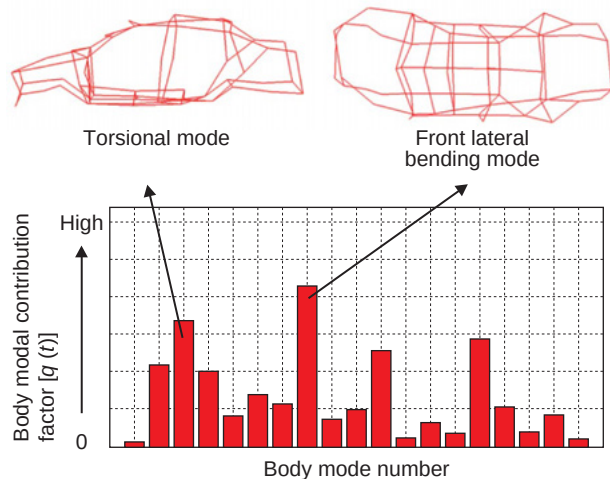


Fig. 11 Test results for body modal contribution

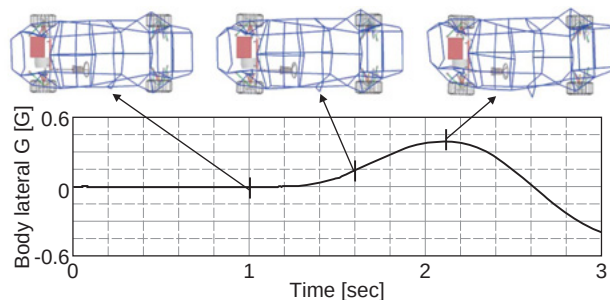
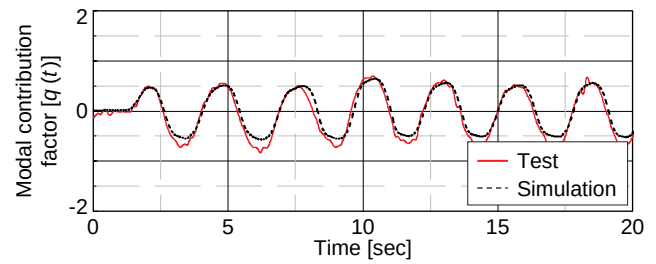
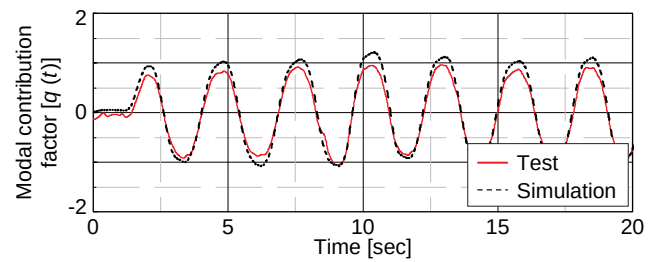


Fig. 12 Simulation results for body deformation



(a) Torsional mode



(b) Front lateral bending mode

Fig. 13 Two time domain body modal contributions

5. Conclusion

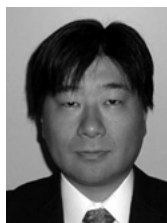
A modal forced response method was used to calculate the input at the suspension coupling points on a body framework, enabling visualization of body deformation during actual handling maneuvers, which had previously represented a challenge. The simulation model combined an elastic body model formulated using the residual vector method, and a multi-body dynamics model. This model was used in a comparison of the modal contribution factors in vehicle dynamic performance and body deformation during slalom maneuvers in test and simulation results, and a good correlation was obtained.

References

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