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ANALYSIS OF AUTOMOTIVE RIDE COMFORT CONSIDERING VEHICLE BODY FLEXIBILITY

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Abstract

In most researches on ride comfort analysis of passenger vehicles, the flexibility of the vehicle body has not been considered as an important factor because the resonance frequencies of the vehicle body related to pitching, yawing, and rolling motions are approximately below 10 Hz while the resonance frequencies of the vehicle body related to the flexibility are approximately above 20 Hz. However, in this paper, we found that modification of the local flexibility (or local stiffness) of the 4 parts on which shock absorbers are mounted in the vehicle body has some influence the level of ride comfort. A one-dimensional, theoretical model is devised to qualitatively examine the effect of the change in the local stiffness of the vehicle body on the ride comfort. Based on the results obtained from the analysis of the one-dimensional model, multi-body dynamic analysis considering the flexibility of the vehicle body is performed using MSC/ADAMS and MSC/NASTRAN. More concretely speaking, natural frequencies and mode shapes computed by MSC/NASTRAN are used as input data for multi-body dynamic analysis in MSC/ADAMS. It is confirmed that the ride comfort can be improved by appropriately changing the local stiffness of the vehicle body through several simulations using MSC/ADAMS. The simulation results agree well with the experiment results.

INTRODUCTION

In general, vehicle dynamics analyses involve the use of the rigid body model of a vehicle body and chassis. However, it has been reveled in recent researches that analyses using the rigid body model do not offer accurate or reliable results. In the paper, we were able to determine how much the flexibility of a vehicle body effects the dynamic characteristics of the vehicle body. For this, a finite element model of the vehicle body, which was made by the finite element analysis program (MSC/ NASTRAN), was used for a Multi-Body Dynamics (MBD) analysis using MSC/ADAMS. The main objective of this research is to improve the dynamic characteristics of the vehicle body to achieve enhanced vehicle comfort. Although a good vehicle comfort has been achieved through a full design modification of the vehicle body, this work consumes too much time and is very costly. Therefore, in this paper, a method of obtaining a better vehicle comfort by means of changing the local stiffness of the vehicle body is introduced [1, 2].

In MBD analysis using a finite element of a vehicle body, the input point inertance (IPI) and transfer point inertance (TPI) analysis were carried out at locations where the vehicle body and the chassis are connected. The results of these analyses are used to develop an improved plan for the vibration transfer reduction and for performing the verification analysis. FEM model, which is the same as an actual automotive and composed of a vehicle body, used Body In White (B.I.W.) and Trimmed model, and the model was compared with the test results obtained through the modal analysis.

In this research, we performed the modal analysis by performing a simulation on the effects on the vehicle ride comfort about the local stiffness of the vehicle body. A study on the stiffness definition method and the design modification for vehicle ride comfort evaluation were performed on the vehicle mounting point through the test along with analytical method.

ANALYSIS AND COMPOSITION OF SIMPLE THEORITICAL MODEL

The vehicle dynamic characteristic analysis that uses a flexible body is different from that using a rigid body through the conventional method because of the flexibility and the vibration mode effects. However, the complicated theoretical flexible model as a full vehicle model is difficult to solve as an analysis method, the vehicle dynamic characteristic can be estimated as a composition of the simple theoretical model. As a result, it is possible to use to search for an access method about the full vehicle analysis.

Free Vibration Analysis

A simple model is shown in Figure 1 where a vehicle body is assumed as a flexible beam. For the thickness variation of the chassis mounting parts, the Heaviside Function such as that shown in Figure 1, was used [3]. The governing equations based on Euler -Bernoulli Beam (EBB) theory can be represented as Eqs. (1).

$$\frac{d^2}{dx^2}\left[EI(x)\frac{d^2Y(x)}{dx^2}\right]-\omega^2 m(x)Y(x) = 0 \tag{1}$$

$$\begin{aligned} EI(x) &= EI_u + EI_{Fa}\left[H(x-x_1)-H(x-x_2)\right] + EI_{Ra}\left[H(x-x_3)-H(x-x_4)\right] \\ m(x) &= m_u + m_{Fa}\left[H(x-x_1)-H(x-x_2)\right] + m_{Ra}\left[H(x-x_3)-H(x-x_4)\right] \end{aligned} \tag{2}$$

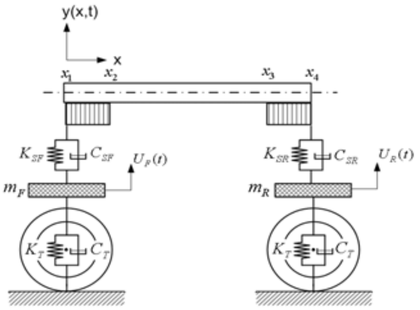

Figure 1 - Simplified model for flexible beam

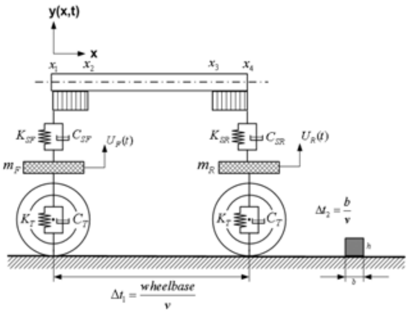

Figure 2 - Simplified model for forced response

Flexible beam simplified model which has altering stiffness and mass can be written as Eqs. (2). Substituting Eqs. (2) into Eqs. (1), in can be written as Eqs. (3).

$$
\begin{aligned}
& Y^{(4)}(x) - \omega^2 \frac{m_u}{EI_u} Y(x) + \left[\frac{EI_{Fa}}{EI_u} Y^{(4)}(x) - \omega^2 \frac{m_{Fa}}{EI_u} Y(x) \right] (H_1 - H_2) \\
& + \left[\frac{EI_{Ra}}{EI_u} Y^{(4)}(x) - \omega^2 \frac{m_{Ra}}{EI_u} Y(x) \right] (H_3 - H_4) + 2\frac{EI_{Fa}}{EI_u} (\delta_1 - \delta_2) Y^{(3)}(x) \\
& + 2\frac{EI_{Ra}}{EI_u} (\delta_3 - \delta_4) Y^{(3)}(x) + \frac{EI_{Fa}}{EI_u} (\delta_1^{(1)} - \delta_2^{(1)}) Y^{(2)}(x) \\
& + \frac{EI_{Ra}}{EI_u} (\delta_3^{(1)} - \delta_4^{(1)}) Y^{(2)}(x) = 0 \; (H_i = H(x - x_i), \quad \delta_i = \delta(x - x_i), \quad \delta_i = \frac{dH_i}{dx}, \quad \delta_i^n = \frac{d^n \delta_i}{dx^n})
\end{aligned}
\tag{3}
$$

$$
[\mathbf{SM}]\{\mathbf{Y}\} = \begin{bmatrix} AE_{ki} & AF_{ki} & AG_{ki} & AH_{ki} \\ BE_{ki} & BF_{ki} & BG_{ki} & BH_{ki} \\ CE_{ki} & CF_{ki} & CG_{ki} & CH_{ki} \\ DE_{ki} & DF_{ki} & DG_{ki} & DH_{ki} \end{bmatrix} \begin{Bmatrix} Y^{(2)}(x_{Fi}) \\ Y^{(2)}(x_{Ri}) \\ Y(x_{Fi}) \\ Y(x_{Ri}) \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{Bmatrix} \tag{4}
$$

By using continuous conditions on the shift section of the material, Laplace Transform(LT) can be performed through Eqs. (3). If the shift section of the material is wide, the shift section can be integrated through a Gauss Integration. Gauss Integration Parts are applicable in continuous condition, and uses partial fractions. It applies boundary condition to both ends of the beam. Thus, Equation Of Motion (EOM) can be given as Eqs. (4). In order for the solution in Eqs. (4) to exist, the matrix formula of System Matrix (SM) should be zero. Then, the natural frequency can be obtained from β , the natural mode can be obtained by using the natural vector [4]. Table 1 shows

Table 1 - Natural frequency of the uniform beam

Mode	Theoretical (Hz)	FEM (Hz)	Difference (%)
1	1.13	1.14	0.88
2	1.98	1.99	0.51
3	9.47	9.47	0
4	10.27	10.28	0.10
5	24.16	24.15	0.04
6	66.3	66.06	0.36
7	129.92	128.75	0.90
8	214.74	211.28	1.61

the results of FEM and theoretical analysis which is partially the uniform beam of the unchangeable material. Since the results of the theoretical analysis are the same as FEM results, the theoretical analysis can be considered a reasonable analysis tool.

Forced Vibration Analysis

When a simple model vehicle as shown in Figure 2 is acting on displacement vibration, the response can be obtained using Forced Vibration Analysis (FVA) [5]. The governing Equation to solve forced vibration response can be represented as Eqs. (5).

$$\frac{\partial^2}{\partial x^2} EI(x) \frac{\partial^2 y(x,t)}{\partial x^2} + m(x) \frac{\partial^2 y(x,t)}{\partial t^2} = f(x,t) \tag{5}$$

$$y(x,t) = \sum_{r=1}^{\infty} Y_r(x)\eta_r(t) \tag{6}$$

$$\begin{aligned} &\int_0^L m(x)Y_r(x)Y_s(x)dx = \delta_{rs} \quad r, s = 1, 2, .. \\ &\int_0^L Y_s(x)\left[\frac{d^2}{dx^2} EI(x) \frac{d^2 Y_r(x)}{dx^2}\right] dx = \omega_r^2 \delta_{rs} \end{aligned} \tag{7}$$

$$\begin{aligned} &\ddot{\eta}_r(t) + \omega_r^2 \eta_r(t) = N_r(t) \quad , r = 1, 2, 3, ... \\ &N_r(t) = \int_0^L Y_r(x) f(x,t)dx \quad , r = 1, 2, 3, ... \end{aligned} \tag{8}$$

$$\begin{aligned} &m_F \ddot{u}_F(t) + (C_{sF} + C_T)\dot{u}_F(t) + (K_{sF} + K_T)u_F(t) \\ &= C_T \dot{z}_F(t) + K_T z_F(t) + C_{sF} \sum_{s=1}^{N} Y_s(0)\dot{\eta}_s(t) + K_{sF} \sum_{s=1}^{N} Y_s(0)\eta_s(t) \\ &m_R \ddot{u}_R(t) + (C_{sR} + C_T)\dot{u}_R(t) + (K_{sR} + K_T)u_R(t) \\ &= C_T \dot{z}_R(t) + K_T z_R(t) + C_{sR} \sum_{s=1}^{N} Y_s(L)\dot{\eta}_s(t) + K_{sR} \sum_{s=1}^{N} Y_s(L)\eta_s(t) \end{aligned} \tag{9}$$

$$RMSI(t) = \sqrt{\frac{\int_{t-\Delta T/2}^{t+\Delta T/2} |y(x,t)|^2}{\Delta T}} \tag{10}$$

The displacement $y(x,t)$ is assumed in Eqs. (6). Substituting Eqs. (6) into Eqs. (5), multiplying $Y_s(x)$ by both sides, integrating with length, and then using orthogonal condition of mass, natural mode can be expressed respectively in Eqs. (8). As a result, EOM of unsprung mass can be expressed in Eqs. (9).

In this research, the theoretical analysis was performed at a vehicle speed of 40 km/h. When the width and the height of impulse input shape was 0.01m, the acceleration at the driver (2.2m) and VIP (2.85m) seat position was measured. To verify the acceleration response about vibration, the Impulsive Root Mean Squares (RMSI) in Eqs. (10) was used. Where $x(t)$ is time history of the transient signal, and ΔT (time

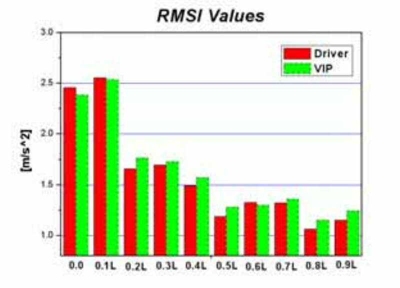

Figure 3 - RMSI Mean Value

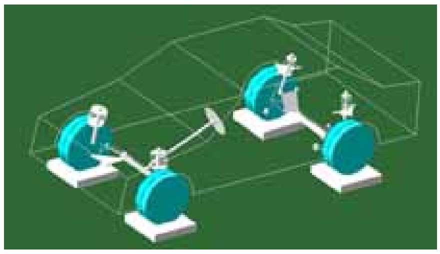

Figure 4 - Quasi-static analysis model

interval) is 0.04. The mean value of RMSI is the minimum value at 0.8 L position, and the mean value of RMSI is the maximum value at 0.1 L position. From this result, the frequency response variation can be verified by local stiffness variation, and rear suspension for the stiffness of chassis mounting parts may turn out to be more sensitive than front suspension.

COMPOSITION OF THE VEHICLE ANALYSIS MODEL

Quasi – Static Analysis

The analysis of vehicle dynamic through a flexible kinematic & compliance analysis method can be further divided into Kinematic Analysis, Quasi-Static Analysis (QSA), and Dynamic Analysis [6]. QSA is to maintain static equilibrium condition at each step can be analyzed spring, bush, and flexible beam as an analysis method of reaction force and movement of the system when vertical force and side force acting on the system. Thus, it is called Compliance Analysis. QSA result was compared with Suspension Parameter Measurement Device (SPMD) result acquired through the vehicle test. Therefore, MBD model can be composed through the input and modify of the vehicle suspension data which are hard points and moments of inertia of the steering system, spring, damper, bush, and tire property. QSA model is shown in Figure 4.

Caster change (deg) is usually set up 0 ~ 3 (deg). Test and simulation results are shown in Figure 5 where a 0.7 ~ 0.8 (deg) change can be seen. Camber change (deg) is usually set up 0 ~ 2.5 (deg). Test and simulation results are shown in Figure 6 where a 1.8 (deg) change can be seen.

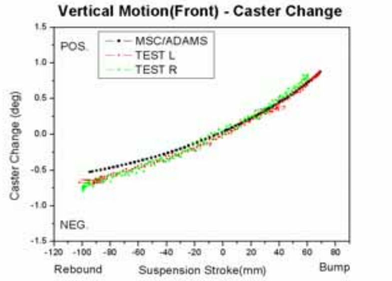

Figure 5 - Caster change

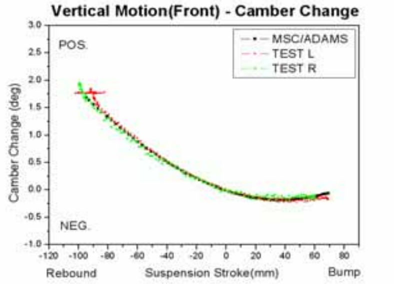

Figure 6 - Camber change

Composition of Flexible Model

Using B.I.W. vehicle body, a finite element analysis was performed by MSC /NASTRAN. This model consists of about 500,000 nodes and elements. To adjust the mass of a full vehicle using the trimmed vehicle body, Engine/TM modeling was performed. Lumped mass was used as an Engine. Engine was supported at 4 points, i.e., two in the front and two are the rear of which are located at the center member and the others (right and left) at the vehicle body.

Finite element models are composed of mixed vehicle body using rigid body element 2 (RBE2) and spring element (CELAS2). The normal mode analysis result of the test and MSC/NASTRAN [7] after natural vibration analysis can be verified by Modal Neutral File (MNF) result of MSC/ADAMS. It was used $1^{st} \sim 7^{th}$ modes among the results of a vehicle simulation [8].

Frequency Response Function

To change Young's Modulus according to vehicle body stiffness, it is the process for selecting stiffness variation parts through IPI and TPI analysis. IPI is called Point Inertance Functions (PIF). It is excited at the same position and direction using unit force, and measured at the same position, and it can be represented the characteristic variation of the local stiffness as the ratio of the frequency function. In other words, it is FRF between the same measuring point and exciting point. TPI is called Transfer Inertance Functions (TIF). It is excited at the same position and direction using unit force, and measured the difference position, and it can be represented the characteristic variation of the global stiffness as the ratio of the frequency function. In other words, it is FRF between the different measuring point and exciting point.

Figure 7 shows the measuring point when TPI analysis was performed. They are the measured points which have the effect on the ride comfort of driver such as floor, front/rear seat track, and steering wheel. The amplitude of node 208017 and 208018 is relatively higher than other nodes. Therefore, these nodes are relatively weaker than other nodes because of rear chassis mounting parts.

Figure 8 represents the measuring point of the front seat track when TPI analysis is performed. From the result, the response of node 208017, 208018 related with the ride comfort is relatively higher than other nodes. Because the stiffness of the rear mounting parts set up relatively stronger than front one, it could be better effect related with the ride comfort. Natural frequency of the flexible model through the thickness variation of

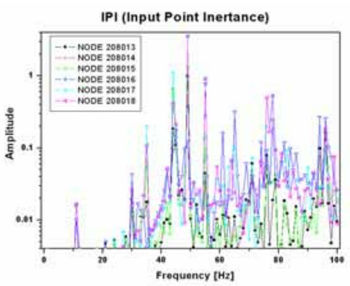

Figure 7 - Results of IPI

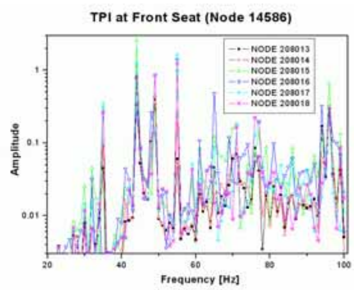

Figure 8 - Results of TPI at front seat track

the chassis mounting parts elements was reduced from 4th mode to 7th mode.

FULL VEHICLE SIMULATION

Vehicle Simulation

In this research, the result is computed using MBD simulation program MSC/ADAMS [9], and Ftire which is durable tire about bar input. Figure 9 shows vehicle body and suspension shape when two passengers get on the car. Figure 10 and Figure 11 show the acceleration value compared with the test value on the knuckle in frequency. They agree well with the test value.

Figure 9 - Flexible body vehicle model

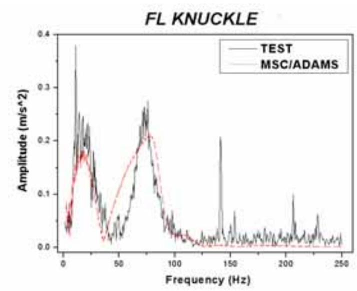

Figure 10 - Accelerations at front left knuckle

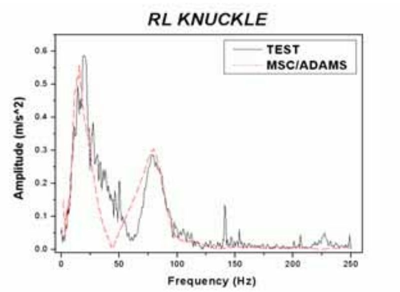

Figure 11 - Accelerations at rear left knuckle

Comparison of the Ride Comfort Analysis

It was compared with the ride comfort through the vehicle simulation after stiffness variation of the chassis mounting parts using the FRF result of the vehicle body. Figure 12 and Figure 13 show respective RMSI values about X axis and Z axis. STIFF 1 was

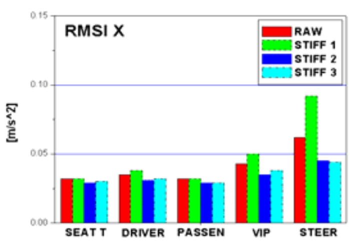

Figure 12 - RMSI Values of X direction

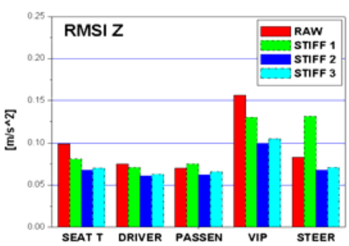

Figure 13 - RMSI Values of Z direction

expected worse ride comfort than original stiffness. STIFF 2 which is optimal stiffness target value is expected the best ride comfort value.

SUMMARY

In this paper, a study on the scheme for ride comfort improvement of carrying out the integrated numerical analysis of the suspension system and the vehicle body which is modeled flexible using MSC/NASTRAN finite element analysis and MBD analysis program MSC/ADAMS. It is presented simplified vehicle model of the vehicle considered flexible modeling characteristics of the vehicle body and predicted the vibration characteristics of the vehicle with the variation of various design parameters about bar input using this model. It was utilized concretely the design modification of the vehicle which is used MBD simulation program MSC/ADAMS based on the results. The feasibility about suspension system modeling of the vehicle verified and compared the test results with kinematic & compliance analysis results of the target vehicle using MSC/ADAMS. IPI & TPI function was acquired using the local stiffness variation of the vehicle about bar input through FEM and MBD integrating analysis and can be selected a strut mounting part as an optimal position. Parameter study of flexible body through the thickness variation of the strut mounting parts was conducted. Optimal stiffness target value at the strut mounting part for the best ride comfort value of the driver and VIP position was presented. Through this analysis process, it was found out the effect on the strut mounting about ride comfort and presented concretely the design modification scheme for the improvement of the ride comfort value.

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