Vibration Analysis of a Passenger Vehicle with a Flexible Car Body

車體의 彈性을 고려한 乘用車의 振動 解析

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초 록

자동차가 고르지 않은 도로를 주행할시 차체의 각 부분에 일어나는 진동을 고려하기 위해서는 차체 각 부분의 탄성효과를 고려하여야 만족할 만한 결과를 얻을 수 있다. 이 논문에서는 범용 유한요소해석 프로그램과 범용 기구해석 프로그램을 이용하여 자동차의 주행시 일어나는 차체내의 진동에 대하여 연구하였다. 예제로는 승용차를 이용하였고 차체의 유한요소해석을 한 후 그 결과를 범용 기구해석 프로그램에 이용하여 기구해석에 가장 많이 쓰이는 강체 모델의 결과와 탄성을 고려한 모델의 결과를 비교하였다.

1. Introduction

If elastic deformation is negligible compared to the overall movement, then the concept of a rigid body is acceptable for many instances. However, if flexibility of bodies are crucial in determining motion of the system, the rigid body assumption may not be acceptable. Lightweight systems such as aircraft and automotive structures, robot manipulators, and articulated space structures are a few examples where flexibility of bodies should be considered. If rigid body assumptions are made for all components of those systems, the analysis result may be unacceptable due to the resulting position inaccuracies. Also system failure due to fatigue can not be predicted because elastic deformation analysis is not performed.

To account for the flexibility effect of the system, one can solve seperately for nonlinear gross rigid-body motion and elastic deformation. Small elastic deformation is assumed to have negligible effects on gross body motion on this approach. Rigid body analysis is performed to obtain gross motion, then the resultant forces are used as external forces into a structural analysis problem. Total motion of the system, including elastic deformation, can be obtained by superposing elastic deformation on gross body motion. However, this approach has a serious efficiency problem due to the large number of degrees-of-freedom in the model. Also large errors may

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be introduced because the coupling effect of rigid and deformation coordinates is not considered.

In this paper, a method which couples rigid and deformation coordinates is presented. A general purpose FEM program is used to obtain necessary information for the flexible bodies. A general purpose mechanical system analysis code, DADS[1], is used to model a combined rigid and flexible body system. A passenger car is modeled as an example problem. The modeling and analysis procedure is explained and the benefit of the flexible body model is demonstrated.

2. DADS flexible body code

The DADS flexible body program uses the modal synthesis method[2] to combine the small elastic motion with the large displacement rigid body motion. Component mode synthesis[3] is an approach allowing the representation of a complex mechanical system by a reduced number of degrees-of-freedom.

Many complex mechanical systems consist of several structures and a finite element model of those systems may have a very large number of degrees-of-freedom. In many cases, components are connected to adjacent bodies through kinematic joints or spring-damper-actuators which may produce large concentrated forces and local deformations. A large number of mode shapes are required to capture these effects if only normal modes are used. Thus, it is desirable to have a method which provides a reasonable representation of the displacements with a small number of mode shapes.

The motion of each component is represented by a set of component modes[3]. Component modes may include dynamic modes and static modes which are introduced to include local deformations which are caused by joint reaction forces or other external forces.

The component mode synthesis method of structural dynamics is applied in the DADS code by introducing dynamic normal modes and static correction modes on the component level to account for constraints induced deformations. Generally, the static modes enforce geometric compatability at a preselected set of points due to interaction between any two adjacent components.

The modal synthesis method from structural dynamics is also applied to dynamic analysis of flexible systems. However, flexible body dynamics is significantly different from conventional linear structural dynamics. First, small elastic deformation is coupld with the geometrically non-linear global motion of the system. Second, large joint reaction forces and non-linear spring-damper-actuator forces appear between components. Thus, a carefully chosen combination of dynamic normal modes and static attachment modes[4] are employed in the DADS code, unlike structural dynamics which uses only a few of the lowest dynamic modes.

3. Interface between FEM codes and DADS

In the current version of the DADS code, an efficient dynamic analysis method for mechanical systems consisting of rigid bodies and flexible components is employed. The component mode synthesis method enables the DADS code to solve large-scale FEM models efficiently. Information from FEM analysis is necessary for each flexible body to form the equations of motion[1]. The detailed procedure for interfacing an existing FEM code and the DADS code is as follows:

(1) Set up a finite element model of a flexible body component of the

mechanical system. Determine preferable boundary conditions for the components, after considering internal/external forces, kinematic constraints, and non-structural masses.

(2) Find the eigensolution by solving the eigenvalue problem for the flexible component.

(3) Extract the following data from the finite element data base using the DADS intermediate processor.
   a. global stiffness matrix
   b. global mass matrix in diagonal matrix form
   c. geometric information
   d. desired number of normal modes including rigid body mode shapes if they exist

(4) Determine the number of static correction modes to be used in the flexible body analysis. Calculate the inertia relief load vector [4] using the DADS inertia relief load vector calculation program and run the finite element analysis program to get the static attachment mode shapes.

(5) Calculate static correction modes (residual inertia relief attachment modes)[4] and prepare the DADS input data for a flexible body using the DADS intermediate processor. The resultant file contains the following information.
   a. nodal coordinates
   b. modal mass and stiffness matrices
   c. global mass matrix in diagonal matrix form
   d. mode shapes

(6) Create a rigid body model of the mechanical system using the DADS preprocessor and include the file for a flexible body created by the DADS intermediate processor to make the model with rigid and flexible bodies.

(7) Run the DADS program to get the desired result.

4. Example: Vibration analysis of a passenger car

4.1 Description of the rigid body model

a. Front suspension system

Eleven rigid bodies are defined for the front suspension system. Left and right knuckles, left and right lower control arms, left and right shock rods, left and right stabilizer bars, left and right tie rods and the rack are defined as rigid bodies. The front suspension system is shown in Figure 1.

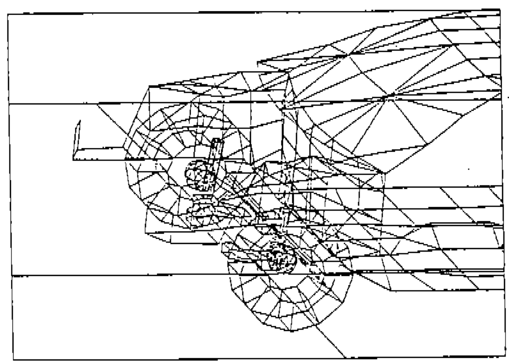

Fig. 1 The Front Suspension System

Various joints and force elements are used to connect parts. Two cylindrical joints are used to connect left and right knuckles and shock rods. The rack and the car body are linked with a translational joint to transfer the steering effect to the tie rod. Left and right lower control arms and knuckles are connected with 2 spherical joints, right and left tie rods and knuckles are connected with 2 spherical joints, and 2 universal joints are used to connect left and right tie rods with the rack.

Other than kinematic joints, 10 bushing elements, 2 bump stops, 2 shock absorbers, 2 springs, and a rotational spring for a stabilizer bar is used. Each lower control arm has two bushings, at the front and rear, which are attached to the car body. Each shock rod is connected to the car body with a bushing. The stabilizer bar is attached to lower control arms and a car body with bushing elements. Bump stops, shock absorbers, springs are attached in between shock rods and knuckles.

b. Rear suspension system

Twelve rigid bodies are defined for the rear suspension system. Four lateral links, two each at left and right sides, left and right trailing arms, left and right shock rods, left and right stabilizer bars and left and right knuckles are defined as rigid bodies. The rear suspension system is shown in Figure 2. Unlike the front suspension system, only two joints are used in the rear suspension system. Two cylindrical joints are defined in between left and right shock rods and knuckles. All other bodies except shock rods are connected with bushings, bump stops, shock absorbers, and a stabilizer bar.

Two bushing elements are defined to attach left and right front lateral links and knuckles, 2 bushing elements in between rear lateral links and knuckles, 2 bushing elements to connect trailing arms and knuckles, 2 bushing elements to attach trailing arms and the car body, 2 bushing elements to attach shock rods to the car body, and a stabilizer bar is attached to knuckles and the car body with 4 bushing elements. Springs, shock absorbers, bump stops are attached in between shock rods and knuckles and a rotational spring is attached in between left and right stabilizer bar.

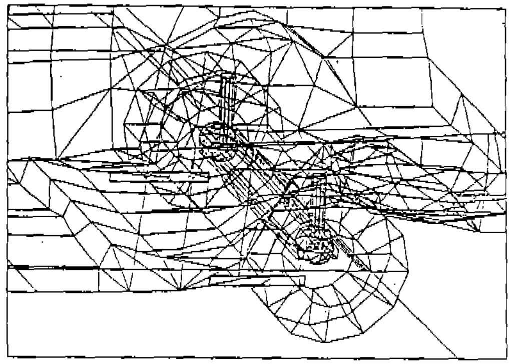

Fig. 2 The Rear Suspension System

c. Modeling of the transverse leaf spring and stabilizer bars.

A composite leaf spring is used in the rear suspension system. Since the behavior is a little bit different than the normal spring, the leaf spring in the rear suspension is investigated in depth to determine the most effective modeling technique. The vertical and lateral deflection curves of the nonlinear finite

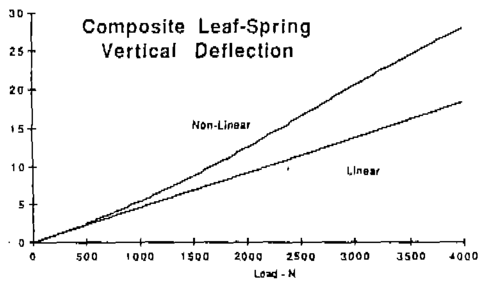

Fig. 3 The Vertical Deflection Curve of the Leaf Spring

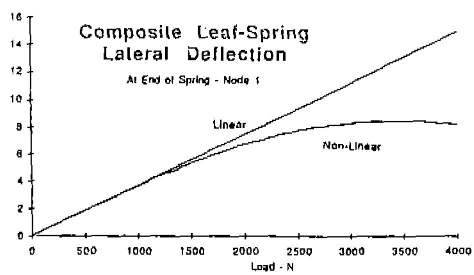

Fig. 4 Composite Leaf-Spring Lateral Deflection

element analysis model are shown in Figures 3 and 4, respectively. This leaf spring can be modelled by dividing the leaf spring into 4 to 6 seqments and represent each seqment as a flexible beam element.

Another simplified, but fairly reliable, method to account for this non-linear effect is, with non-linear translational spring elements in vertical and lateral directions between the knuckles and the body. This technique is used in the model rather than using a flexible beam element. The force- deflection curve of the beam used in the model is obtained from nonlinear finite element analysis of the spring as in Table 1.

Table 1 Force-deflection data of a leaf spring

Non-linear Deflections of Composite Leaf-Spring – Symmetric Loading on Nodes 1 and 27

Deflections at end of spring-Node 1			Node 9-Support	Node 14-CL	Defl. of Nodes
Load(N)	Defl(cm)-Uz	Defl(cm)-Ux	Defl(cm)-Ux	Defl(cm)-Uz	1+14 (Uz)
0	0	0	0	0	0
400	1.949	1.491	0.07788	–0.3703	1.5787
800	4.18	2.969	0.1531	–0.7731	3.4069
1200	6.698	4.385	0.2235	–1.207	5.491
1600	9.484	5.674	0.2863	–1.668	7.816
2000	12.47	6.772	0.3389	–2.15	10.34
2400	15.65	7.622	0.379	–2.643	13.007
2800	18.86	8.189	0.4055	–3.135	15.725
3200	22.02	8.469	0.4182	–3.618	18.402
3600	25.06	8.481	0.4181	–4.082	20.978
4000	27.93	8.263	0.4069	–4.522	23.408

Linear Deflections of Composite Leaf-Spring at End of Spring - Node 1

Load(N)	Defl(cm)-Uz	Defl(cm)-Ux
0	0	0
4000	18.4	14.96

To represent the front and rear stabilizer bars, each stabilizer bar is cut into two segments and connected with a revolute joint. A rotational spring is attached between two bars to calculate the torsional force from the stabilizer bar.

d. The coordinate system, the road profile and the tire model

The coordinate system of the model is defined as: positive x axis is toward the driver's side lateral direction, positive y axis is in the rearward longitudinal direction, and positive z axis is in the upward vertical direction. The model is driven over the bump located at the driver's side with a speed of 25m/sec.

The road used during the simulation is a flat surface with a single sinusoidal bump. The bump has the shape of a half sine curve. The length of the bump is 150cm and the height is 5cm.

The tire force element is used to model a vehicle tire interaction with a road profile. Six components of forces are calculated; vertical, lateral, longitudinal direction forces and three rotational forces at each axis. Forces are calculated in the tire/ground interface plane and then applied to the body where the tire is attached in the model. The tire vertical force-deflection curve used in the model is shown is Figure 5. To calculate lateral force and aligning torque, usually testing data

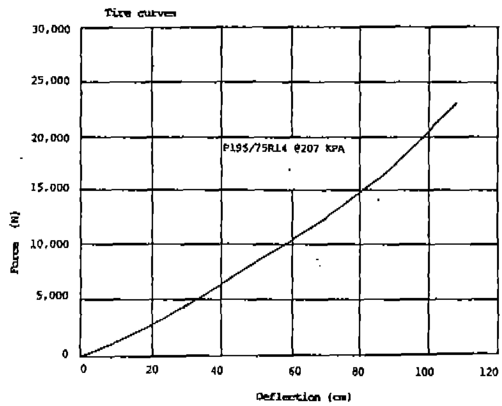

Fig. 5 Tire Vertical Force-Defelection Curve

is used. But in this example, lateral cornering stiffness of the tire is assumed as 200 N/degree and cubic polynomial approximation is used to fit the curve.

4.2 DADS flexible body model

To make a DADS flexible body model, NASTRAN[5] program is used. First, a FEM model of 707 grid points were established and normal vibration analysis was performed with a free-free boundary condition. From normal vibration analysis, 20 normal modes are extracted including 6 rigid body modes. The extracted modes are shown in Table 2.

After a carful study of the result, it has been found that the frequency of the first torsion mode is 24.92 Hz and the frequency of the first bending mode is 28.1 Hz.

To simplify the model, the first torsion mode and the first bending mode are used for the DADS flexible body analysis. The DADS intermediate processor is used to read the geometry, mass matrix, stiffness matrix, and mode shapes from NASTRAN output files. The final output file of the DADS/NASTRAN intermediate processor is the DADS3D input file which contains geometric information, modal coordinates, modal velocities, modal

Table 2 Extracted Eigenvalues

MODE NO.	EIGENVALUE	CYCLES	GENERALIZED MASS	GENERALIZED STIFFNESS
7	5,927966E+02	3.875301E+00	2.167284E+00	1.284759E+03
8	1.587293E+03	6.340867E+00	2.354872E+00	3.737871E+03
9	2.395805E+03	7.790151E+00	1.389106E+00	3.328029E+03
10	3.052094E+03	8.792636E+00	1.826786E+00	5.575521E+03
11	3.346279E+03	9.206640E+00	1.881359E+00	6.295521E+03
12	4.524586E+03	1.070556E+01	1.478666E+00	6.690350E+03
13	8.779172E+03	1.491238E+01	1.022904E+00	8.980250E+04
14	2.452063E+04	2.492217E+01	1.013272E+00	2.484606E+04
15	3.116724E+04	2.809760E+01	2.063670E+00	6.431891E+04

mass matrix and modal stiffness matrix of the flexible body.

The file prepared by DADS intermediate processor is merged with the prepared rigid body model by the DADS preprocessor.

5. Discussion of the Results

A total of 4 runs were made for this study.

(1) 5 second rigid body settling run.
(2) 2 second bump run of a rigid body model.
(3) 5 second flexible body settling run.
(4) 2 second bump run of a flexible body model with two normal modes.

5.1 Dynamic settling run

A dynamic settling run is preformed for 5 seconds to obtain the equilibrium position and check the stability of the model. As indicated in Figure 6, the model reaches an equilibrium position and the acceleration of each component at the end of the run are very small as shown in Figure 7. From Figure 6 it can be seen that the flexible body model reaches an equilibrium position faster than the rigid body model. Further, very interesting results are shown in Table 3. Loads from the suspension to the car body differ as much as 4.65% between the two settling runs.

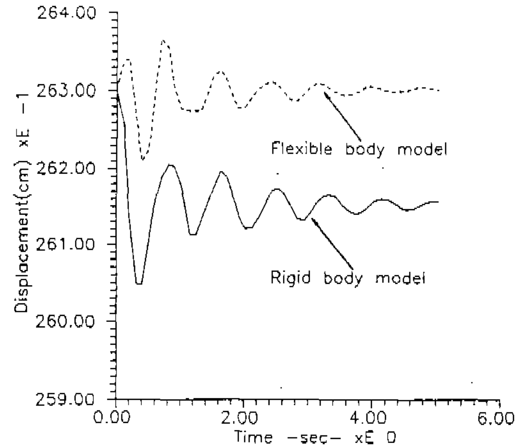

Fig. 6 Front Left Knuckle Displacement

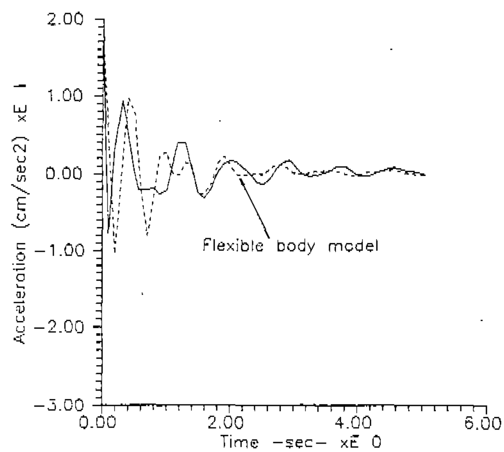

Fig. 7 Front Left Knuckle Vertical Acceleration

Table 3 Equilibrium loads to the body from the suspension

Location	Rigid body	Flexible body	Difference
Front left spring to body	3942(N)	3844	−2.6%
Front right spring to body	3477	3514	1.0%
Rear beam to left body	2314	2279	−1.6%
Rear beam to right body	2045	2145	4.6%
Front springs to body (left & right)	7419	7368	−0.7%
Rear beam to body (left & right)	4359	4424	1.5%

Loads are more evenly distributed in the flexible body case than the rigid body case. It can be concluded that the equilibrium position of the flexible body model gives more physically meaningful results for the following reason: In rigid body analysis, the body is represented as one big mass concentrated at the center of the body. This implies that if a concentrated mass causes significant localized effect to the vehicle body, the flexible body model will give more accurate results than the rigid body model. The difference between the two models is a function of both the mass and the relative stiffness of the flexible frame compared to the suspension components. Modern unibody chassis construction has increased the importance of using a flexible model due to its greater flexibility.

5.2 Comparisons between rigid and flexible body analysis

Bump runs of a rigid body model and a flexible body model with two normal modes are done for 2 seconds. The CPU time on the APOLLO 4000 workstation for those runs are 2 hours for the rigid body model and 2.4 hours for the flexible body model with 2 normal modes. If the magnitudes of the modal coordinates are small and the frequencies used in flexible body model are not much higher than the frequencies of the rigid body model, then the computational time of the flexible body model is not much more than that of the rigid body.

The additional CPU time for flexible body analysis is used to calculate additional terms in the mass matrix and right hand side terms in the equations of motion. The total degrees of freedom in the model are 175 for the rigid body model and 177 for the flexible body model, when two modes are used for the flexible body analysis.

Table 4 shows the peak load difference between the two models. Generally, loads to a car body from flexible body analysis are much different than loads generated by rigid body analysis. Figure 8 shows vertical forces for the bushing element from the front shock to the car body during the two second simulation.

Figure 9 shows the driver's seat vertical acceleration. The peak accelerations are almost the same for the rigid body model and the flexible body model as shown in Figure 9.

Table 4 Peak loads between rigid and flexible bodies

Bushing element force	Rigid body	Flexible body	Difference
Front left shock to body bushing force (at t=0.32)	392(N)	286(N)	−37%
Front right shock to body bushing force (at t=0.34)	97	53	−83%
Left rear lower control arm to body (at t=0.32)	902	744	−21%
Rear stabilizer bar to left body (at t=0.50)	1157	1170	1%
Front stabilizer bar to left body (at t=0.32)	2009	1794	−12%

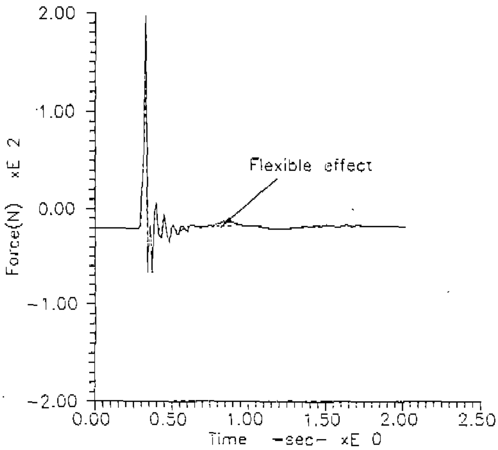

Fig. 8 Front Left Shock to Body Bushing Vertical Force

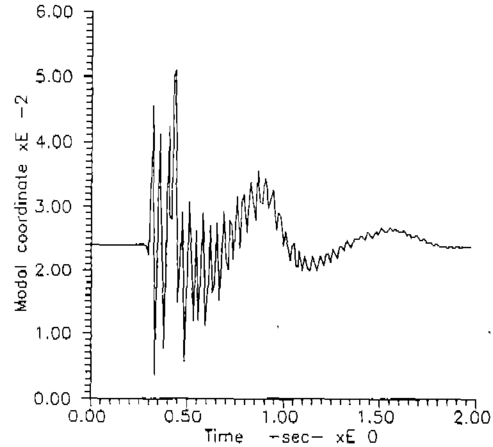

Fig. 11 Body First Bending Mode Modal Coordinate

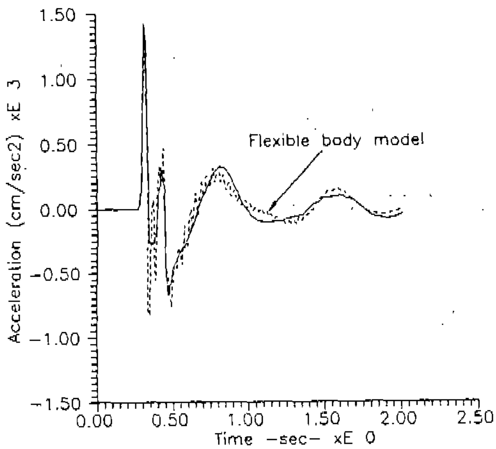

Fig. 9 Driver's Seat Vertical Acceleration

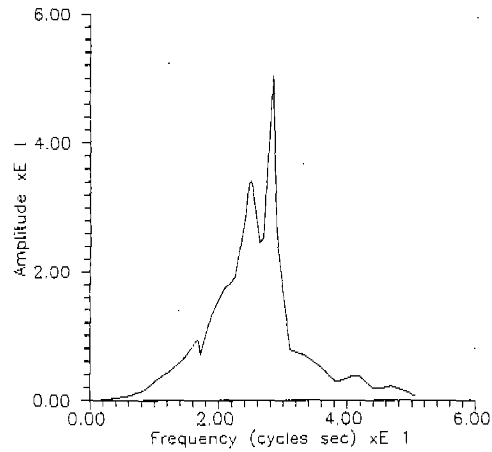

Fig. 12 FFT of Seat Vertical Acceleration

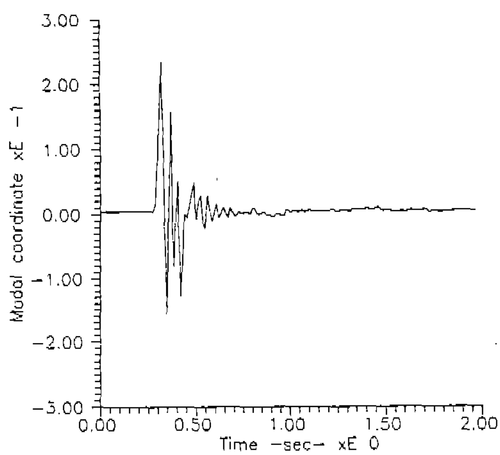

Fig. 10 Body First Torsion Mode Modal Coordinate

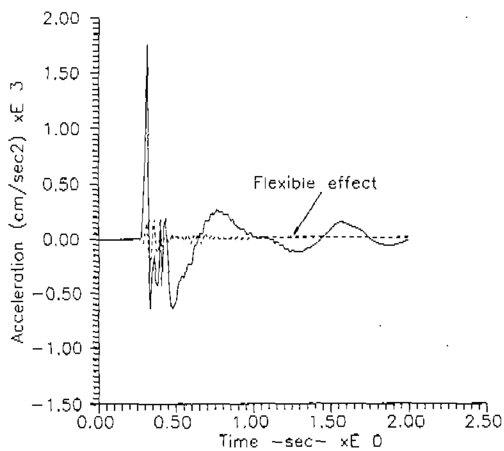

Fig. 13 Steering Column Vertical Acceleration of the Flexible Body Model

However, after the bump, the magnitude and frequency of the accelerations are different.

5.3 Bump run of the flexible body model

The flexible body model with 2 normal modes was run for 2 seconds. Since the bump is on the driver's side, the first torsion mode should give more contribution to the flexible body effect than the bending mode. As seen in Figures 10 and 11, the first torsion mode has larger magnitude than the first bending mode.

If the contribution of each mode at a particular node is desired then the FFT(Fast Fourior Transformation), which is available in the DADS post processor, can be used to measure the sensitivity of each mode at a particular node. An example of the FFT for the driver's seat vertical acceleration is shown in Figure 12. Since the modes used in the analysis are 24.92 Hz and 28.1 Hz, Figure 12 indicates that the driver's seat vertical acceleration is sensitive to 24.92 Hz and 28.1 Hz, and also indicates that the torsion mode has more effect than the bending mode.

Figure 13 shows a plot of steering column vertical acceleration. The solid line indicates the total acceleration of the node and the dotted line is the local acceleration due to flexibility of the body. The difference between the total and local acceleration is the effect of the rigid body motion.

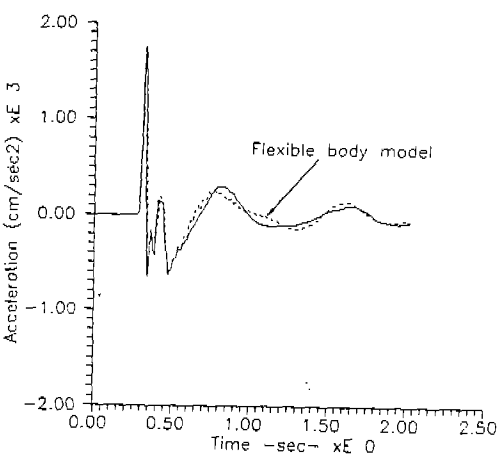

Fig. 14 Steering Column Vertical Acceleration of the Rigid and Flexible Body Model

Figure 14 shows steering column vertical acceleration for the rigid body model and the flexible body model. It can be noticed that the force from the suspension components are smaller in the flexible body analysis. However, due to the flexibility effect of the body, the acceleration of points on the flexible body are generally higher than the rigid body model.

6. Conclusions

The advantages of the flexible body model compared to the rigid body model can be summarized as,

1) Static loads from the flexible body model are more precise than the rigid body model due to the distributed mass.
2) The load history is more accurate than that given by rigid body analysis.
3) The vibration effect of the flexible body can be monitored for ride analysis while the total vehicle travels the prescribed road.

In this study, only two normal modes, the first torsion mode and the first bending mode, are used. By using more modes and correlating the analysis results with test results, a flexible body model for ride analysis can be verified. That model can be used to monitor the vibration behavior at various points in the car body. To improve ride quality, some parameters in the vehicle, such as bushing stiffnesses, can be altered to see the effect due to that change.

As seen in this study, there is a significant difference in the system response when chassis flexibility is included. Even though the loads from the chassis components are lower, the acceleration are higher in the car body because of the body flexibility when compared to the corresponding rigid body analysis.

As an alternative to the use of finite element based flexible body data, testing results could be used. Natural frequencies and mode shapes based on physical tests could be used in the same manner as the analytical results. The differencies between the two approaches needs to be examined further.

The use of active/semi-active suspension systems depends on accurate information about the behavior of the system components. As seen in this paper, there is a significant difference in the system response when the chassis flexibility is included. It is recommended that camparisons of rigid and flexible body models, including controls, be made. This is especially important since both amplitude and frequency of the system response are altered by flexible bodies. This directly impacts the control system behavior.

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