EXPERIMENTAL VIBRATION ANALYSIS TO ADAMS INTEGRATION: FLEXIBLE CHASSIS FULL VEHICLE SIMULATION

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ABSTRACT: Multibody models of complete vehicles are often used for the investigation of ride and handling performances. The limits of a standard multibody approach become soon evident when dealing with dynamic phenomena outside the usual frequency range between 0 and 30 Hz. In this case the implementation of deformable components into multibody models is required. This allows to take into account the low frequency dynamic behaviour of the structure (chassis and subframes), but it is bound to the availability of some kind of information on the flexible body properties.
This paper aims to describe a procedure to import experimentally derived flexible properties of structural components into ADAMS models. This method is based on a mixed "rigid+modal" approach. The modal representation allows to synthesize the deformable structure using frequency criteria directly correlated with the scopes of the subsequent dynamic analysis. The deformable body can undergo large rotations and can exchange non linear forces with the other parts of the system. Both the modal data and the rigid body properties are obtained by measuring a wide set of FRFs on the free-free constrained component, performing a modal analysis and transferring the results to ADAMS through suitable interfaces, developed on purpose. The research work has been performed as a part of the E.C. Brite-INVEC project. A full vehicle model with flexible chassis has been developed and validated using the described approach. Detailed results of the experimental and numerical analysis are shown.

1. Introduction

Multibody models of complete vehicles are used for ride and handling evaluations of "virtual" prototypes, or for the solution of dynamic/kinematic problems. Such models are full of several types of non-linearity that heavily affect the vehicle response. In particular, the use of accurate models of non linear components such as shock absorbers, tires, bushing and bumpers leads to simulation results well related to measured data, acquired over rough roads, up to a frequency of 30 Hz [1, 2, 3]. A comprehensive comfort optimization regards the complete range from 0 to 200 Hz. Experimental data show in this range several natural frequencies due to the flexibility of the car body or subframes. Such structures experience large movements and rotations together with structural deflections. The simulation models need to take into account both kinds of phenomena.

2. Brief notes on deformable bodies

The dynamic behaviour of a flexible component can't be properly simulated in terms of rigid body properties and lumped elasticity according to a classical MB approach. Various methods have been proposed to simulate deformable bodies within full non linear MB models. Some of them use the modal approach, but employ constraint modes [4] or are limited to the linear approximation [5]. A superelement formulation [6] is, on the other hand, critical because the condensation procedure produces full mass matrices that a MB code cannot deal with.

2.1. Linear State Equation modelling

For the implementation of experimentally synthesized flexible properties, the "LSE" (Linear State Equation) method, based on a mixed "rigid+modal" approach, has been used. The flexible component is modelled by means of a rigid body (the body reference), a set of ODEs describing the structural dynamic using the modal approach and the matrices transforming back and forth the forces and the displacements between the spatial and the modal domains. The "rigid" equations describe the movements of the body reference according to the usual Lagrange formulation, while the "modal" equations describe the structural deflection referred to the local reference frame. Figure 1 shows the way ADAMS makes use of this approach. The body reference (PART) and the flexible equations (implemented as Linear State Equations) are subjected to the same forces. The part movement and the deformations are summed to produce the effective displacements of the deformable body. Some of these displacements determine the relative deformations of the viscoelastic connections and the corresponding reaction forces.

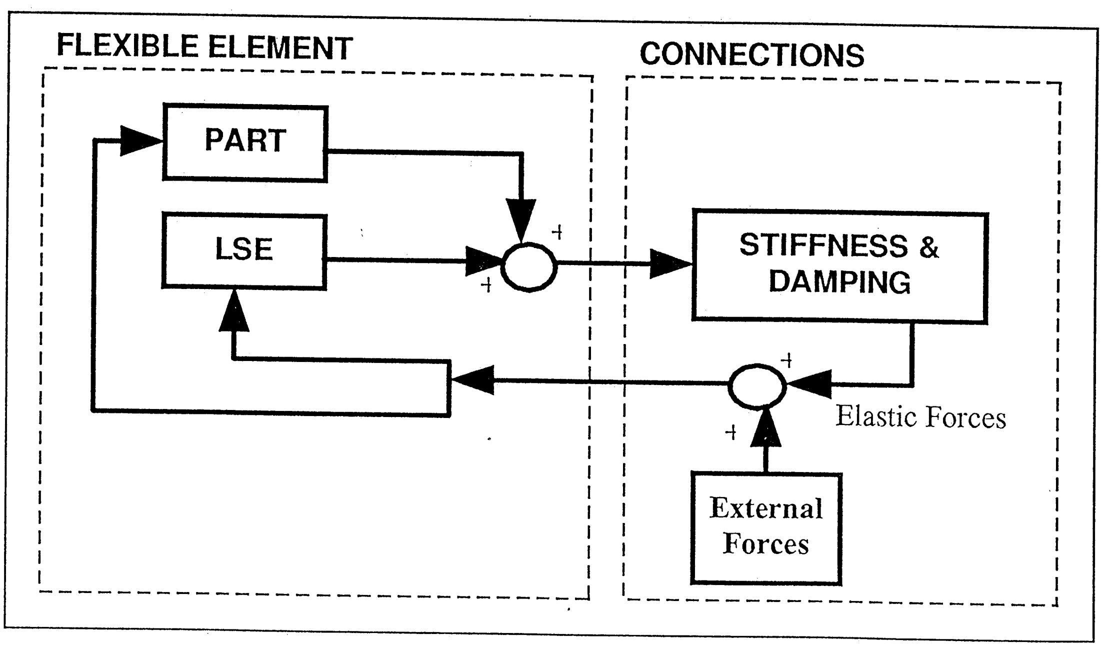


FIG. 1 - ADAMS model of generic flexible component

The equations of motion in the modal domain are written in state space form as follows:

$$\begin{cases} x = Ax + Bu \\ y = Cx + Du \end{cases}$$

where

u = vector of forces and torques acting on the deformable body (inputs)
y = vector of displacements and velocities of deformable body (outputs)
x = vector of modal displacements and velocities of the deformable body

$$A = \begin{Bmatrix} 0 & I \\ -\Omega^2 & -2\xi\Omega \end{Bmatrix}$$

$$B = \begin{Bmatrix} 0 \\ \Phi^T G \end{Bmatrix}$$

$$C = \begin{Bmatrix} \Phi & 0 \\ 0 & \Phi \end{Bmatrix}$$

$$D = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix}$$

I = identity matrix
Ω = diagonal matrix containing the natural frequencies
ξ = modal damping ratio
Φ = matrix of eigenvectors
G = transformation matrix from distributed to nodal forces and torques

The complete development of above formulae and the methodology validation can be found in [7,8].

2.2. Application of LSE approach

The LSE method can be applied according to the following boundary conditions:

a) the flexible body must experience only linear deformations;
b) the internal forces acting on the flexible body must be viscoelastic;
c) the constraint conditions must be "soft" if compared to the local stiffness of the flexible component (inertance of the constrained nodes).

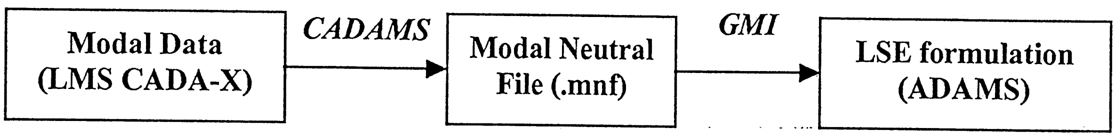
The first restriction is easily met for most applications in vehicle dynamics since the structures normally experience loads not exceeding the linear range.
For what concerns the second condition, the connections between the vehicle components are usually elastic elements (i.e. rubber bushings) with few exceptions that can be described like extremely stiff connections. There are no restrictions on the non-linear characteristics of such viscoelastic joints.

The modal parameters Ω, Φ and ξ can be obtained either by eigenvalue analysis of a FEM model or by experimental modal analysis. The modal basis for the flexible components afterwards analyzed has been synthesized from experimental data coming from a free-free dynamic characterisation. The rigid body properties were estimated using the “massline method” [9].

The modes used in the LSE formulation are a subset of the complete eigendata set of the deformable structure. In this work, for each component, a wide set of FRFs has been acquired and recorded as a standard LMS database. Then the *real* modes selection (zeroes and poles) has been performed to fit properly the measured FRFs of the structure in the frequency range of interest. Generally speaking, the modes with natural frequencies outside the analyzed bandwidth should be taken into account, since they can add significant contribution (residuals) to the FRFs, but at present there is no mean of transferring such residuals into ADAMS models.

2.3. Data transfer procedure

The modal data obtained by LMS CADA-X modal analysis can be translated to ADAMS according to the following procedure:



where

- CADAMS is a program aimed at transferring results (eigenvectors, eigenvalues, modal damping) from experimental vibration analysis within an LMS CADA-X database structure to ADAMS multibody environment. LMS CADAMS translation is based on Modal Neutral File format documentation delivered by MDI Italy.

- GMI (General Multibody Interface) is a new multibody interface allowing the solution of mechanical system through various kind of MB solvers via general "neutral" output file. It is an interface based on general MB concepts, totally independent from the final implementation of the model in a specific solver language.

Both CADAMS and GMI are outcomes of BRITE-INVEC Project

The LSE flexible component properties are resulting from the above defined steps (measurement, modal analysis and translation). The subsequent modal basis is made of: *free-free real modes, without higher frequency residuals*. This can affect the accuracy of the ADAMS flexible model, particularly when dealing with "soft" points of the body, in which static deflection modes have a considerable effect.
The LSE formulation with Normal Modes was chosen instead of a classical Flexible Body approach because:

- experimental data give no information on nodal masses, while the rigid body properties of the whole component can be calculated with modal analysis
- static deflection modes can hardly be measured

3. Flexible Component models

The previously described approach has been applied to the modelization of two components showed in Figure 2 and 3, i.e.:

- a floor panel made of non-conventional material (composite)
- a fully trimmed vehicle body

The floor panel is as a quite simple vehicle structural component and it has been chosen for the methodological validation. The trimmed body is an heavy and complex structure, with high modal damping, that has been firstly analyzed as a separate component and then as a part of a full vehicle model.
Both the components were suspended by means of elastic cords, in order to obtain a free-free system ready for dynamic characterization.

3.1 The panel

The floor panel is 1400 by 1330 mm wide, it has a mass of 26 kg and it is made of composite material. On the surface of the panel 161 uni-axial accelerometers were placed. A random white noise input was applied by means of an electro-dynamic shaker and the corresponding 161 structural FRFs from 0 to 200 Hz were measured and stored in an LMS database structure. A modal analysis was performed starting from this set of acquired functions, to obtain the real modes of the component. Figure 4 shows the first 7 modes of the structure, all lying in the 0-200 Hz frequency range. The presence of a first global bending mode at 29.8 Hz and a well-defined torsion mode at about 44 Hz can be easily seen.

The real modes were transferred to an MNF file by LMS/CADAMS and then translated to an ADAMS model of the single component by means of the GMI interface. The panel flexibility was implemented using the LSE normal modes formulation described in chapter 2
The comparison between the original and calculated FRFs for two different measurement points can be found in figure 5. The shape of the FRFs confirms that the panel behaves almost like a linear dynamic system: the resonance peaks are sharp, poorly damped and clearly connected to the seven modes previously identified. In this case the synthesis was performed using complex modes and taking into account the residuals contribution in order to verify the best achievable correlation between measured and synthesized functions.

3.2 The trimmed body

The analyzed component is a complete chassis of a car, fully trimmed, suspension and engine excluded. The input excitation has been applied to the body by two electro-dynamic shakers. The first one was at the engine suspension right attachment point (lateral direction, Y related to body reference frame), while the other acted vertically, under one of the rear right suspension attachment points. Thirty-seven tri-axial accelerometers were placed all over the structure and a set of 222 FRFs from 0 to 200 Hz were measured and stored. The same procedure used for the composite panel was applied, in order to build up an ADAMS model of the trimmed body, starting from the experimental data acquired.
During the analysis 39 modes were chosen from 0 to 200 Hz. Two of them are shown by figures 6 and 7: a lateral bending of the front chassis at about 24 Hz and a torsion mode involving the full structure at 35.6 Hz.

Inertances and FRFs relative to the excitation points, synthesized by LMS, are compared to experimental curves in figure 8. "Local" inertances are well approximated by synthetic data, while "global" functions don't. The figure 9 shows the equivalent calculated functions obtained with ADAMS/Linear: there is a good correlation between CADA-X syntesized and ADAMS calculated functions.

4. Full vehicle modelling

The fully trimmed chassis has been introduced into the full vehicle model represented in figure 10. The front and rear suspensions were created as standard multibody suspensions using the following component models:

• tyre	Pirelli 4 DOF tyre model
• shock-absorber	Monroe model
• rubber elements	non-linear bushings with static stiffness and damping rates
• links, arms and rods	rigid bodies

The results of a linear analysis on the full vehicle with flexible chassis are compared with those obtained using a standard rigid chassis. Vertical acceleration transmissibilities between the tyre contact patch and some interesting points of the model can be found in figure 11 to 13. Figure 14 shows the correlation between calculated and measured vertical transmissibility to a rear suspension attachment point. Measured data were acquired on a four poster test rig applying an "in phase" random vertical excitation to the four wheels. The contribution of the body deformation to the vertical transmissibility is evident in the 50-200 Hz frequency range.

5. Acknowledgements

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6. Conclusions

This paper described a procedure to import experimentally derived flexible properties of structural components into ADAMS models. This method is based on a mixed "rigid+modal" approach. Both the modal data and the rigid body properties are obtained by measuring a wide set of FRFs on the free-free constrained component, performing a modal analysis and transferring the results to ADAMS through suitable interfaces, developed on purpose.
A full vehicle model with flexible chassis has been developed using the described approach and validated.
The analysis results looks promising, particularly when "stiff" points of the body, in which static deflection modes have little effect, are considered. Further steps have to be taken to:

- improve the FRFs fitting technique by means of real modes
- improve the high frequency dynamic behaviour of suspension component models

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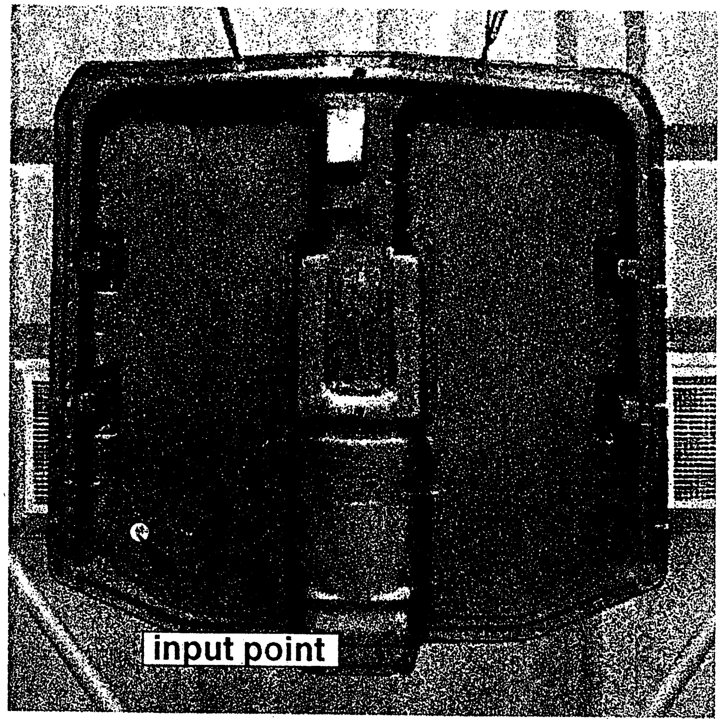

figure 2 – Suspended floor panel

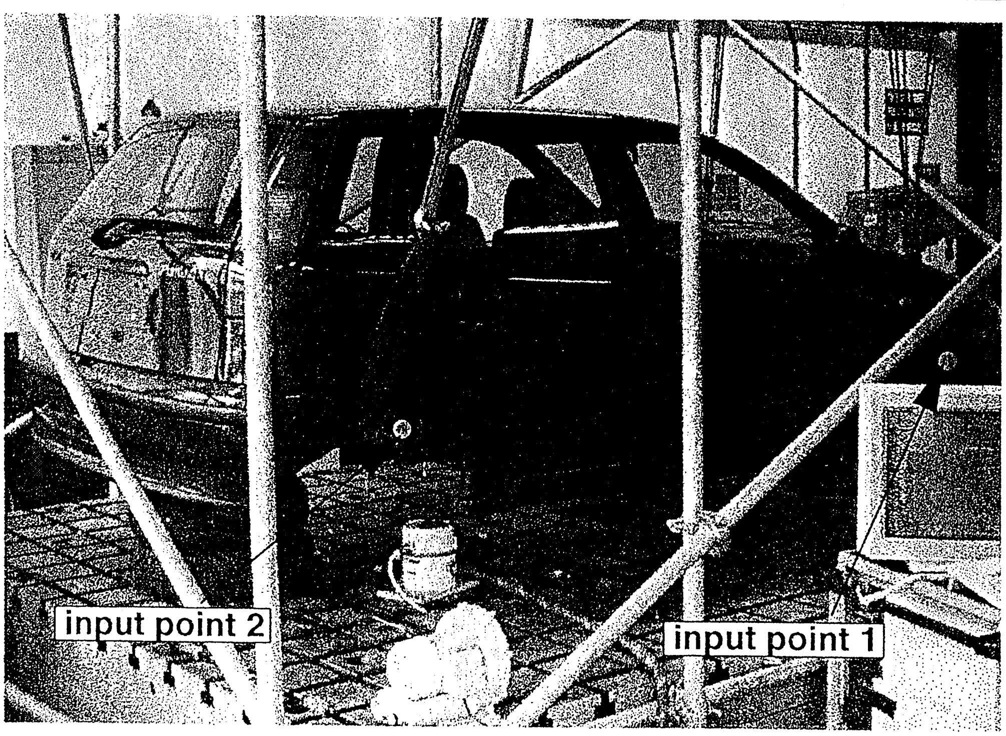

figure 3 – Suspended fully trimmed vehicle body

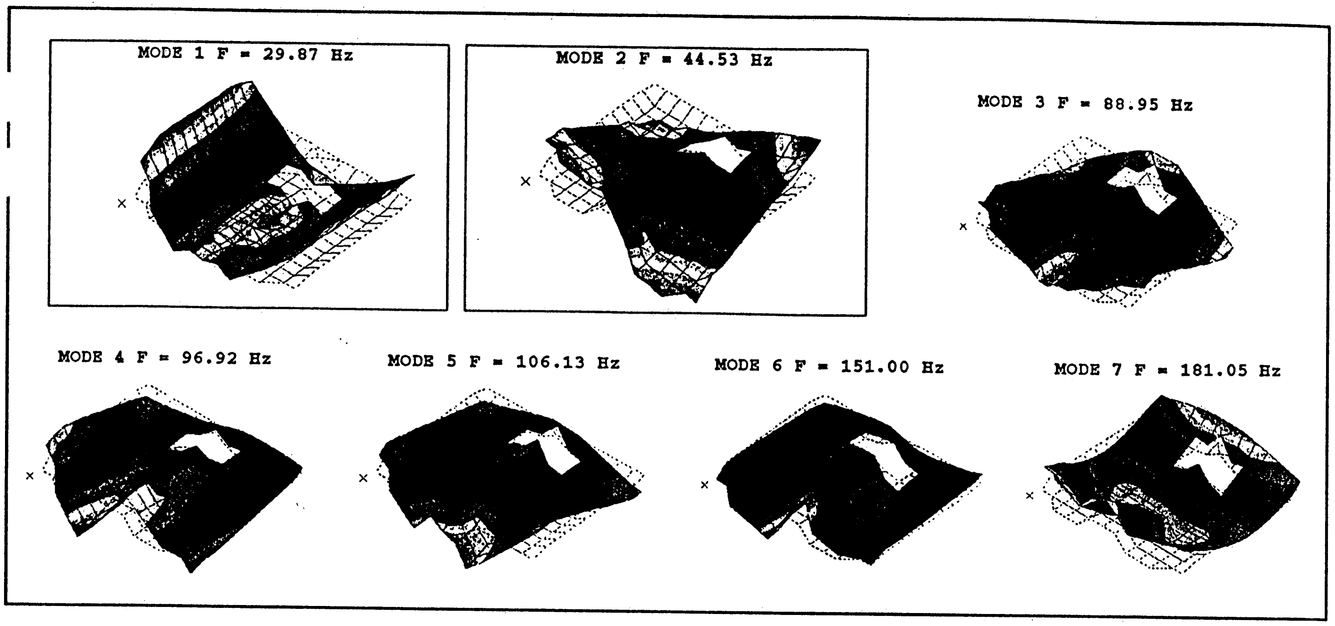

figure 4 – Floor panel modal analysis, deformed mode shapes

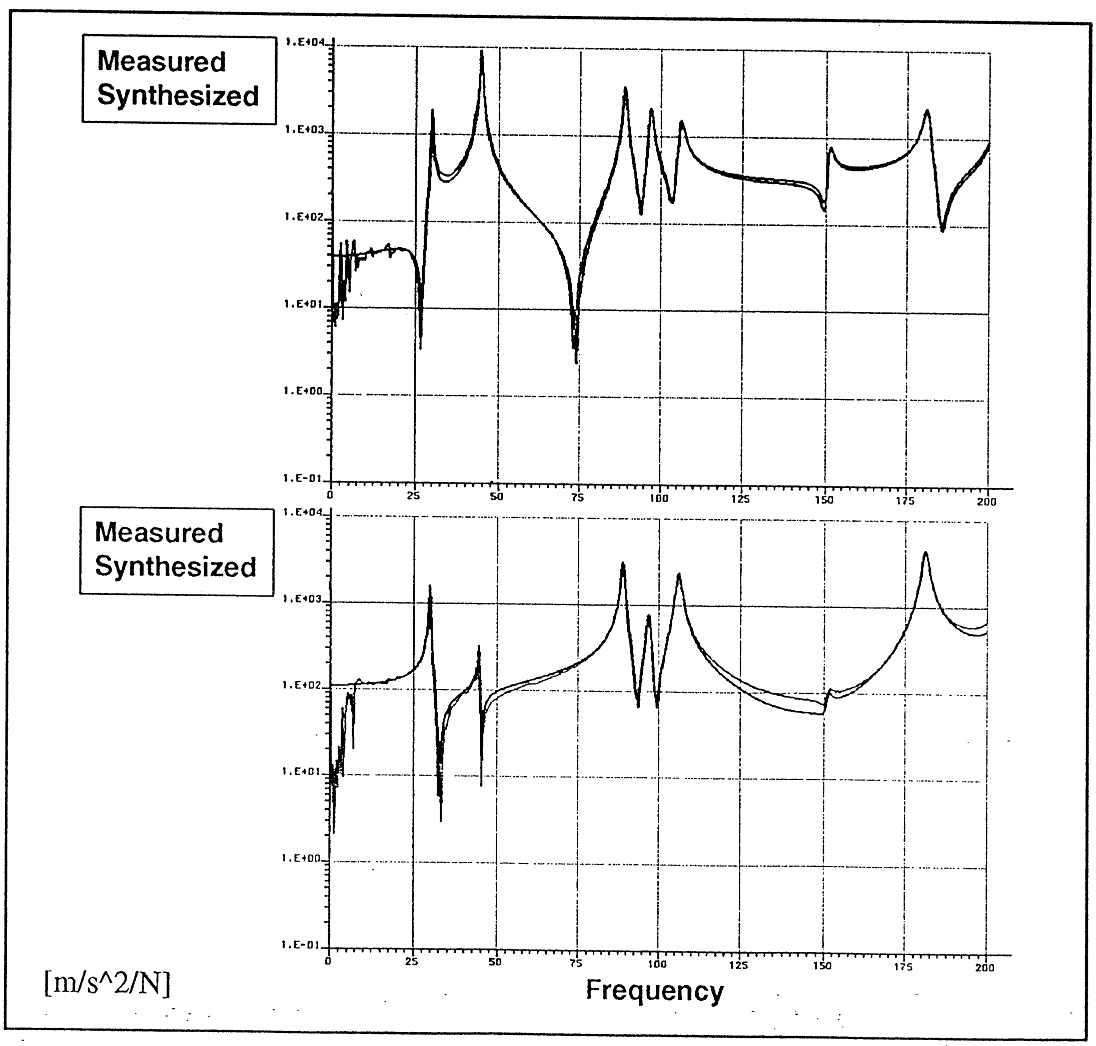

figure 5 – Floor panel modal analysis, synthesized FRFs

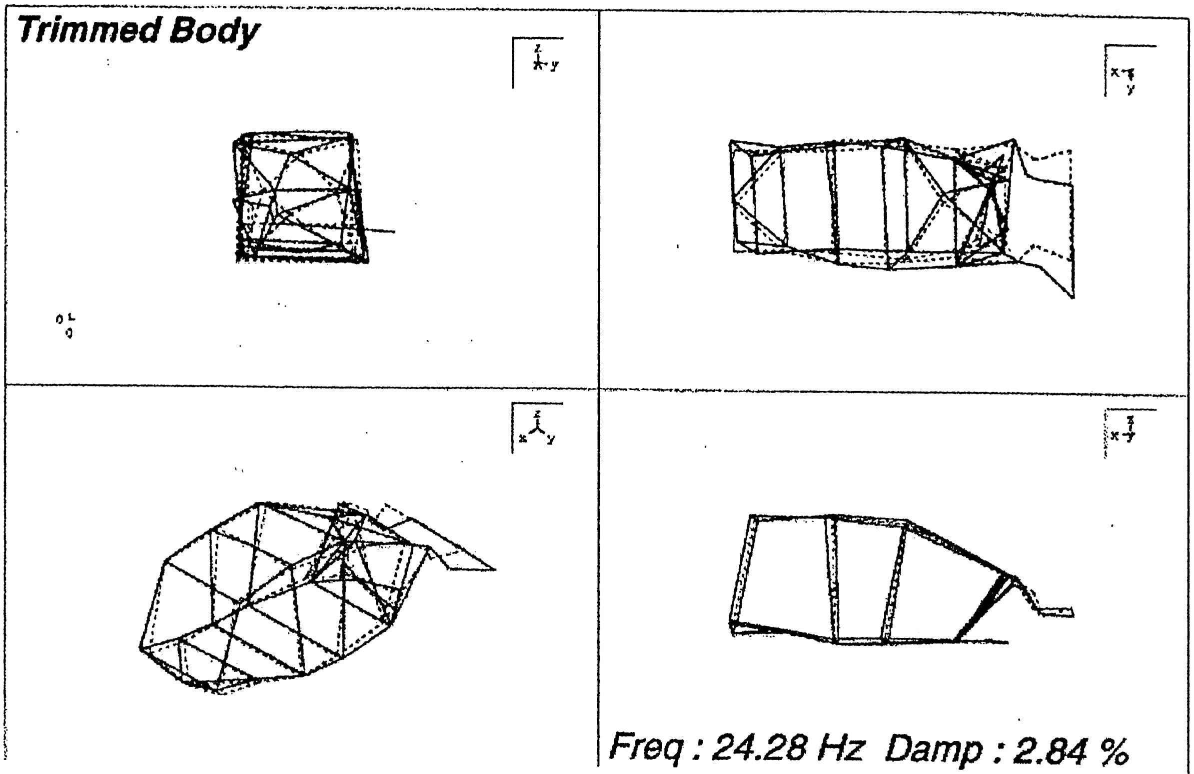

figure 6 – Car body modal analysis, first deformed mode shape

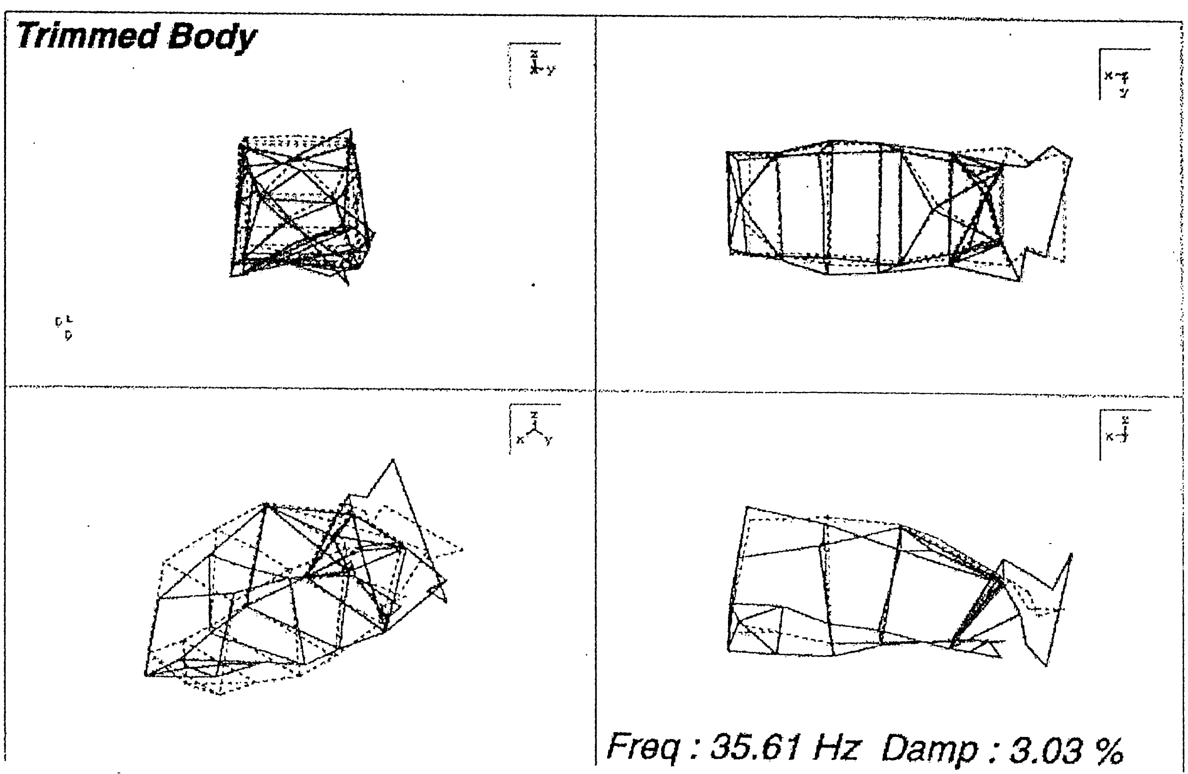

figure 7 – Car body modal analysis, second deformed mode shape

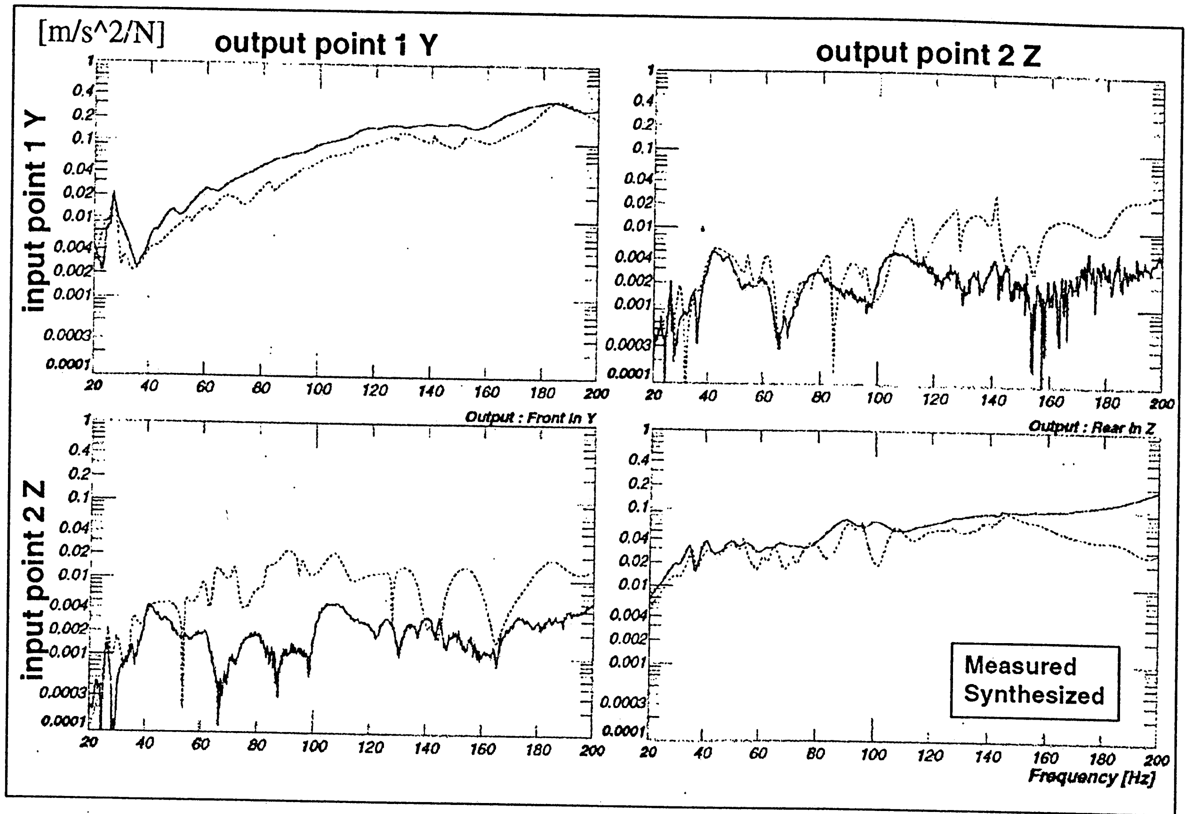

figure 8 – Car body modal analysis, synthesized FRFs, acceleration/force

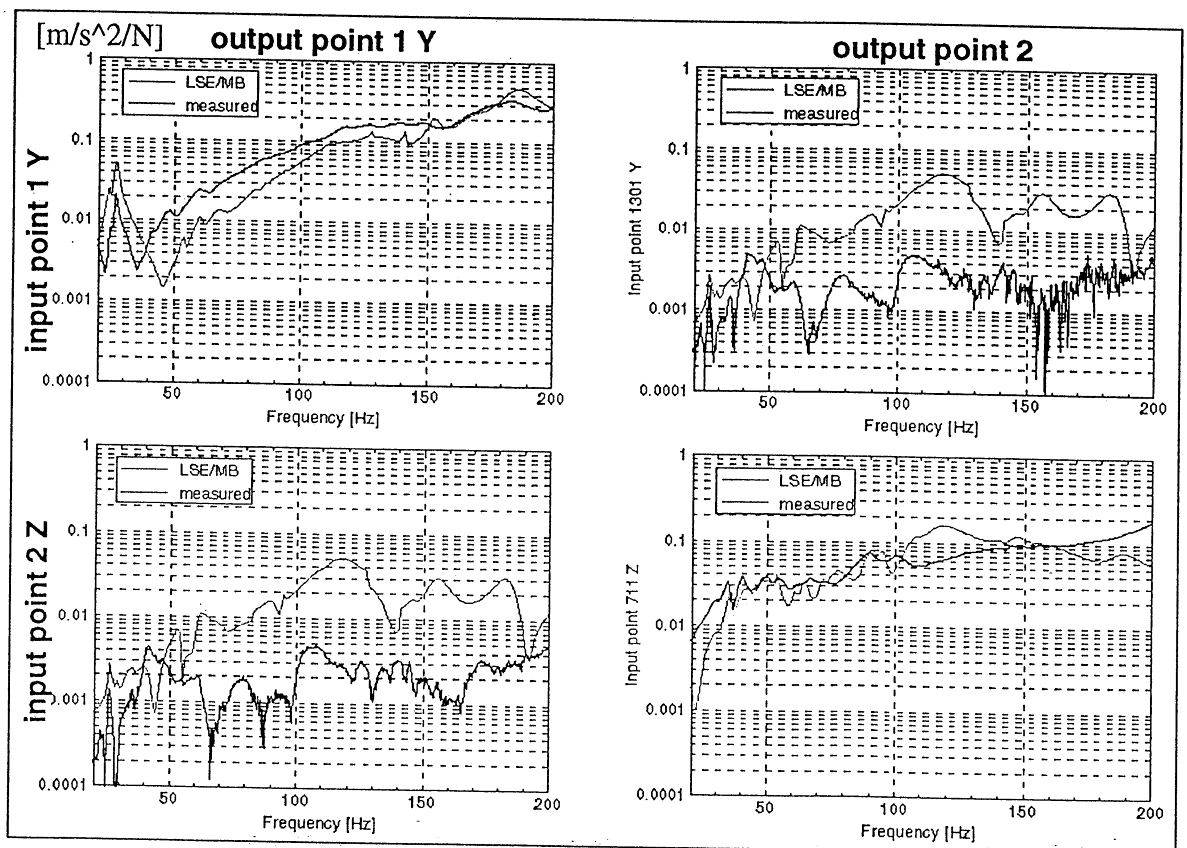

figure 9 – Car body modal analysis, calculated FRFs, acceleration/force

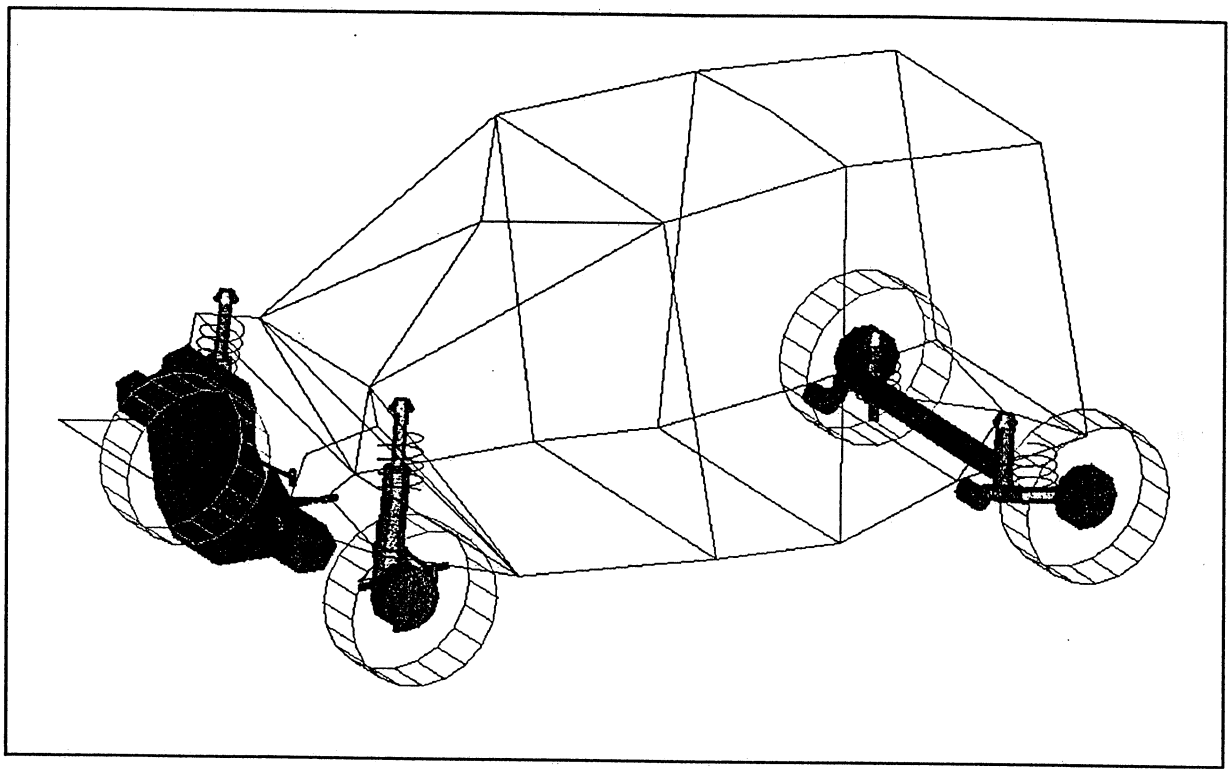

figure 10 – Full vehicle model with LSE flexible chassis

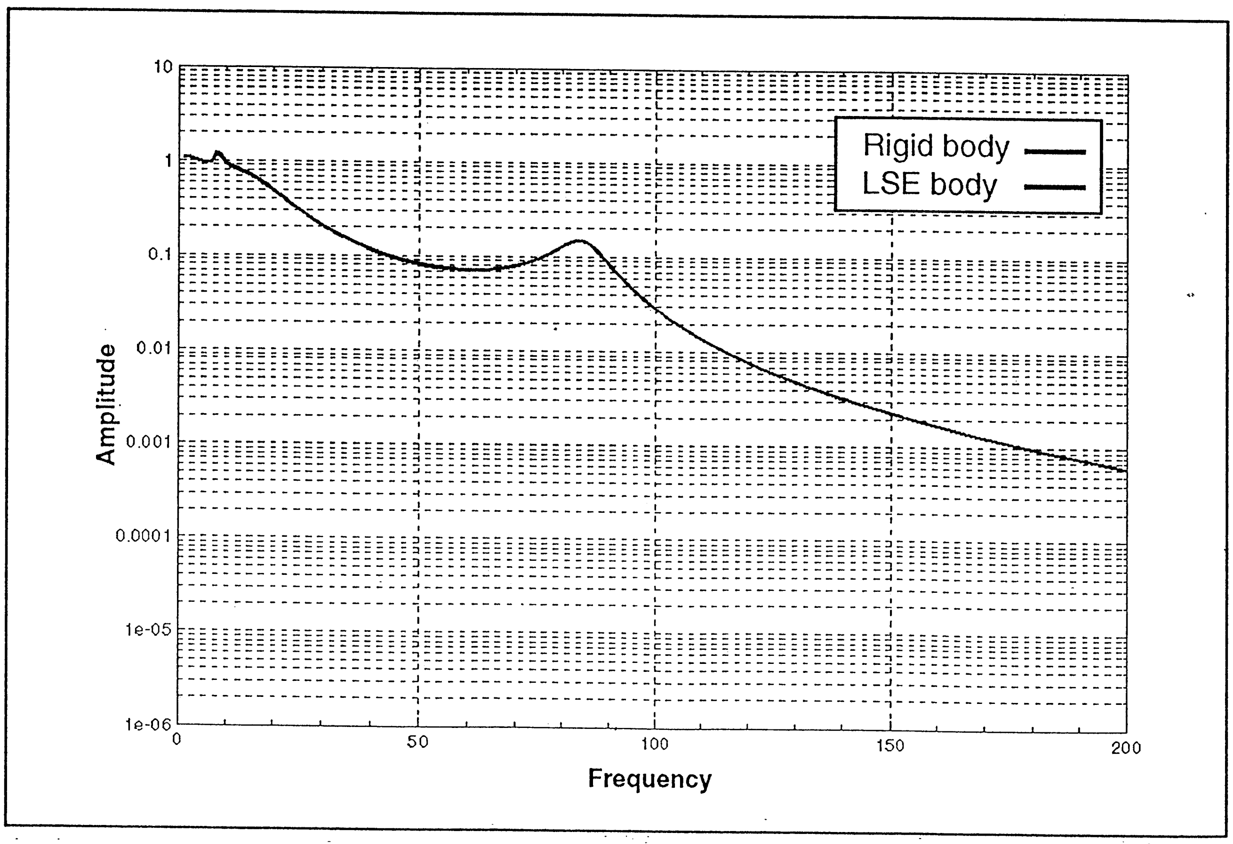

figure 11 – Vertical acceleration transmissibility, Front left wheel hub / ground

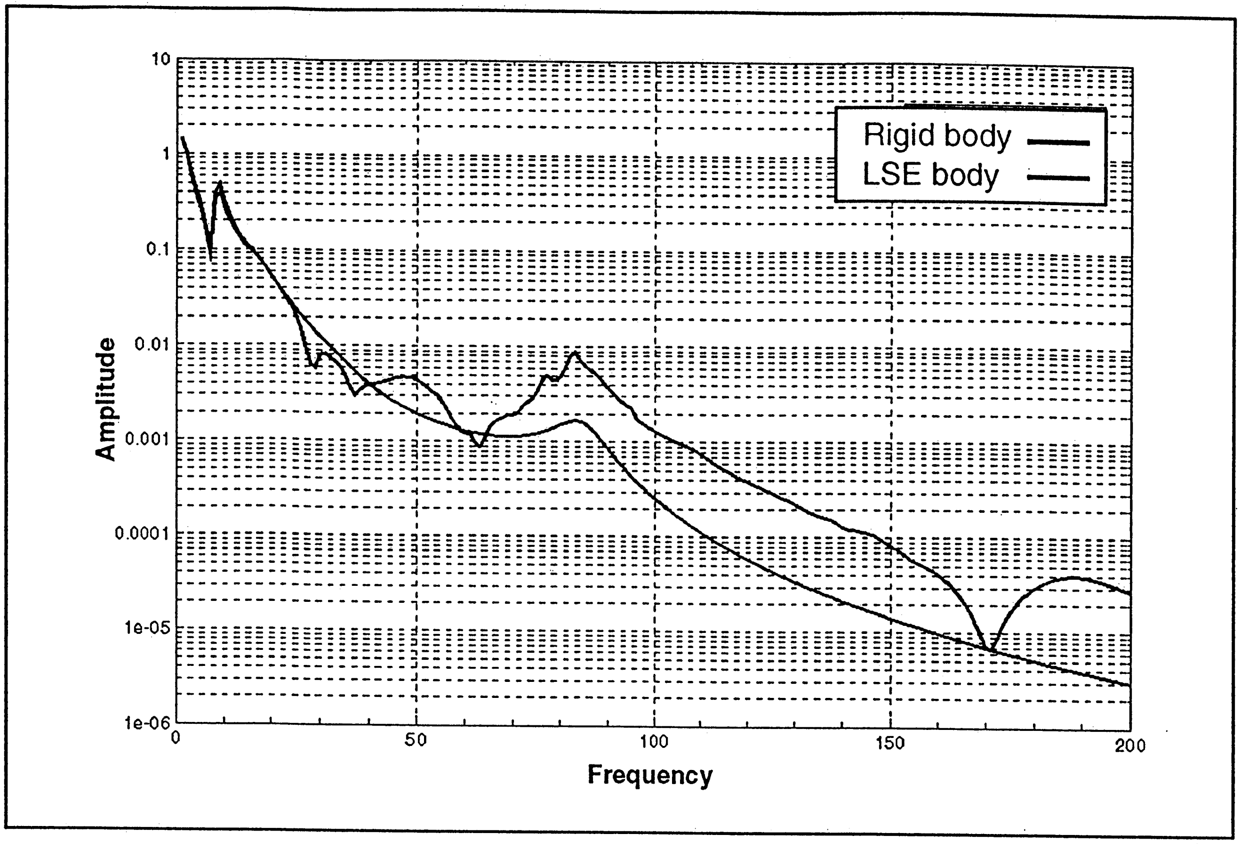

figure 12 – Vertical acceleration transmissibility, Front left dome mount / ground

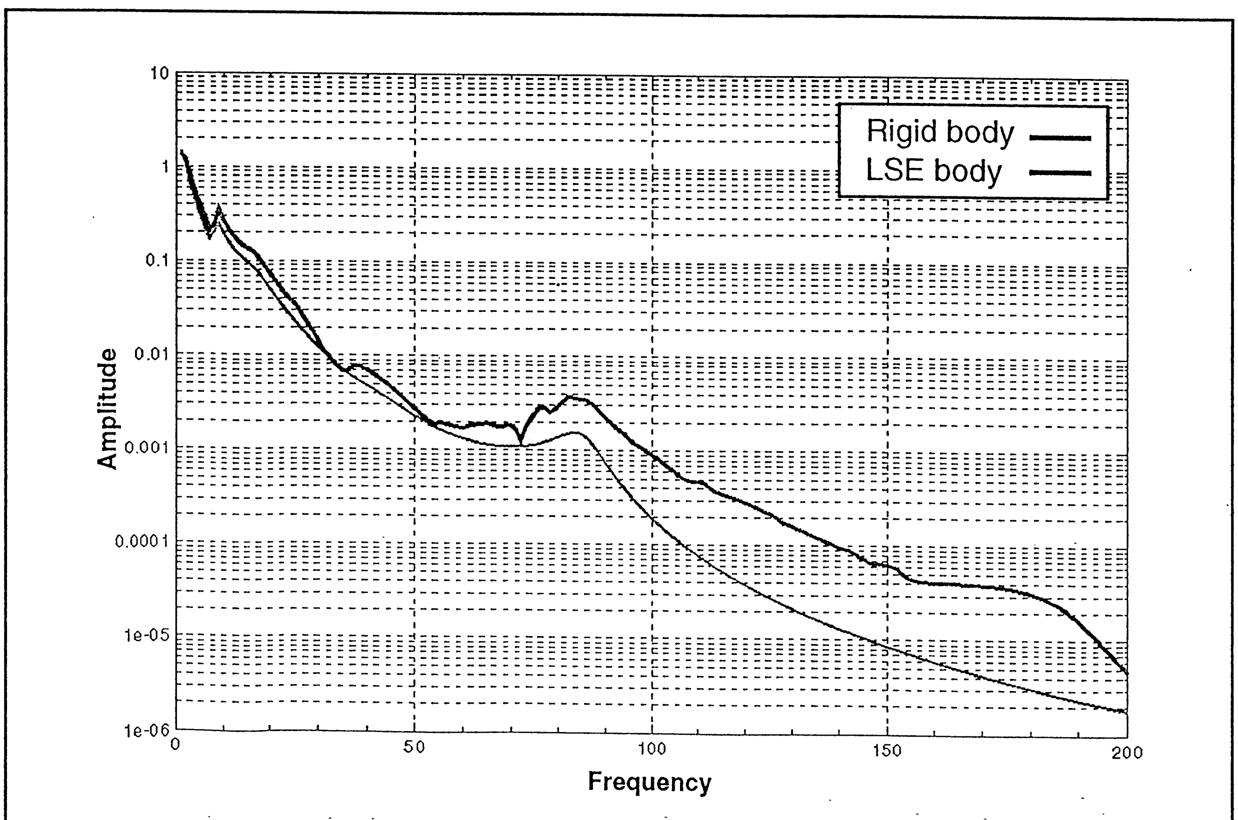

figure 13 – Vertical acceleration transmissibility, Driver seat guide / ground

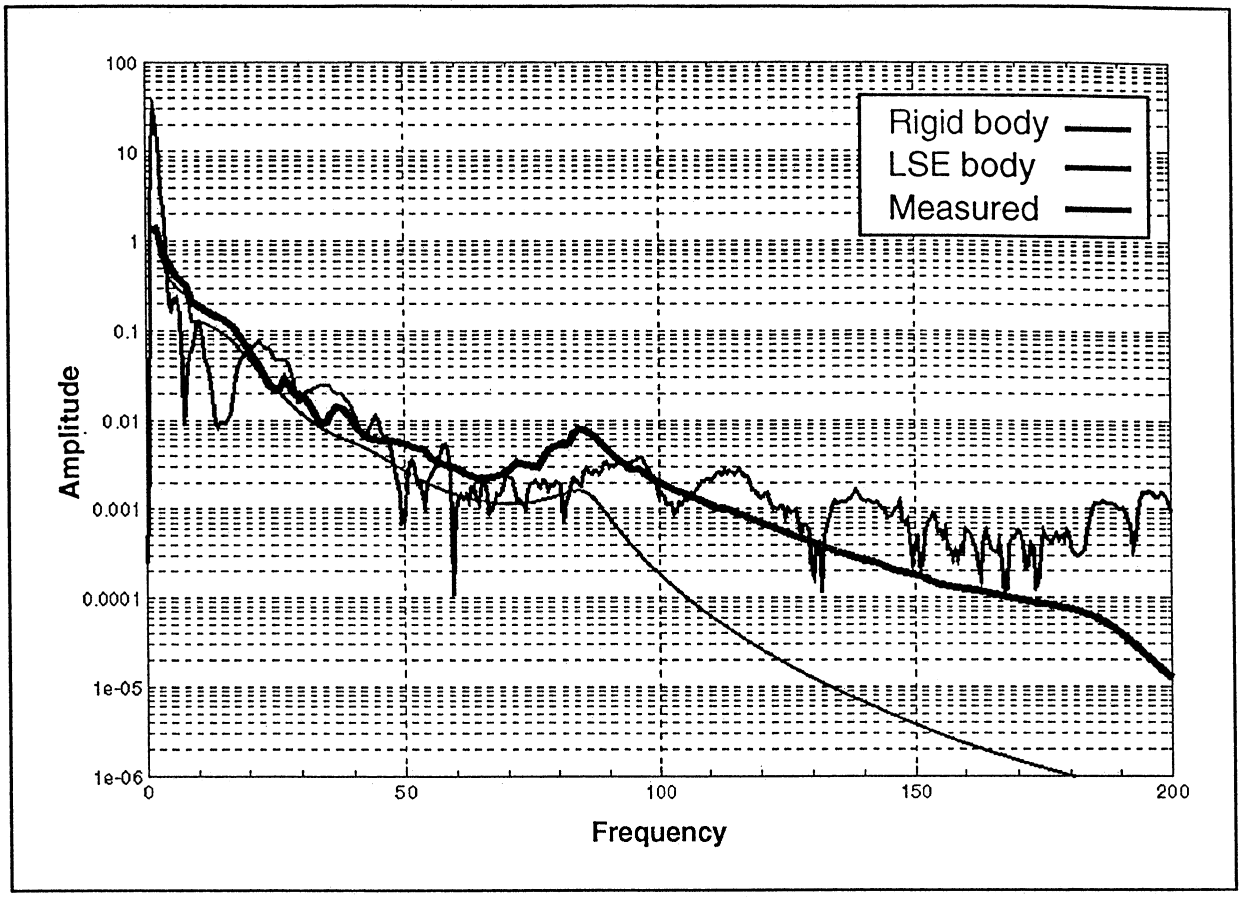

figure 14 – Vertical acceleration transmissibility, Rear suspension att. point / ground