# A Study of Vibrational Behavior of a Medium Sized Truck Considering Frame Flexibility with the Use of ADAMS

**Dr. A. Costa Neto**
Universidade de Sao Paulo - Sao Carlos

**L. C. Ferraro**
**V. L. Veissid**
**C. A. M. Freitas**
Mercedes Benz do Brasil

**M. A. Argentino**
**R. R. Ripoli**
**O. T. Perseguim**
debis humaita IT Services Latin America

## ABSTRACT

The present work describes the modeling and analysis processes of a medium sized truck manufactured in Brazil with regard to vibration and comfort behavior. The vehicle model includes Hotchkiss suspensions front and rear with shackle and with a double stage with bump stops at the rear. It is also included frame flexibility in ADAMS in an approximate manner based on a Finite Element Analysis of the frame. Nonlinear shock absorber curves are also represented for the vehicle and cab suspensions. Viscoelastic bushings for cab and powertrain suspension are also included. Random track profile is generated as input and vehicle comfort is described in terms of the ISO 2631-85 Standard. The effect on vehicle comfort of changing a design parameter can be predicted in the model and verified experimentally. Also, vehicle vibrational behavior is improved with the decoupling of some subsystem motions from certain vehicle modes.

## INTRODUCTION

Present trend in motor truck design involves the reduction of costs and the increase in transportation efficiency. The pursuit of both these objectives, results in lighter trucks, which uses less material and carries less dead weight. Also very important to this respect is driver comfort that has to operate safely and comfortably for many hours. One part of the truck strongly influenced by these guidelines of weight and cost reduction is the frame. The consequence of a lighter frame is a vehicle that has structural resonance within the range of typical rigid body vibrations of the truck subsystems. The likelihood of coupling of structural and rigid body motions in a given resonance is high. Therefore, realistic truck models to predict vehicle vibrational behavior must describe adequately these effects.

## VEHICLE MODELING

A truck has a very large number of natural frequencies in the range of concern to vehicle ride. Typically it may be found around 20 frequencies and modes in this range. In the early stages of vehicle design the use of simplified models is helpful in pointing out dominant behavior obtained. For real vehicle analysis it helps distinguish frequencies and modes of the truck and/or its subsystems. Such models may be easily developed and calibrated in ADAMS, due to its small number of parameters. Later, however, in order to have a good representation of the dynamic behavior of the whole truck, comprehensive models have to be developed.

PRELIMINARY MODELS: For the reasons pointed before, in the early stages of design, and for the analysis of very complex systems it is strongly recommended the use of simplified models. These models allow the perception of basic characteristics of the system. They can also be used to quickly interact and achieve near desired values for vehicle parameters.

Plane Model: A plane preliminary model of the truck was developed in ADAMS as it can be seen in figure 1 illustrated below.

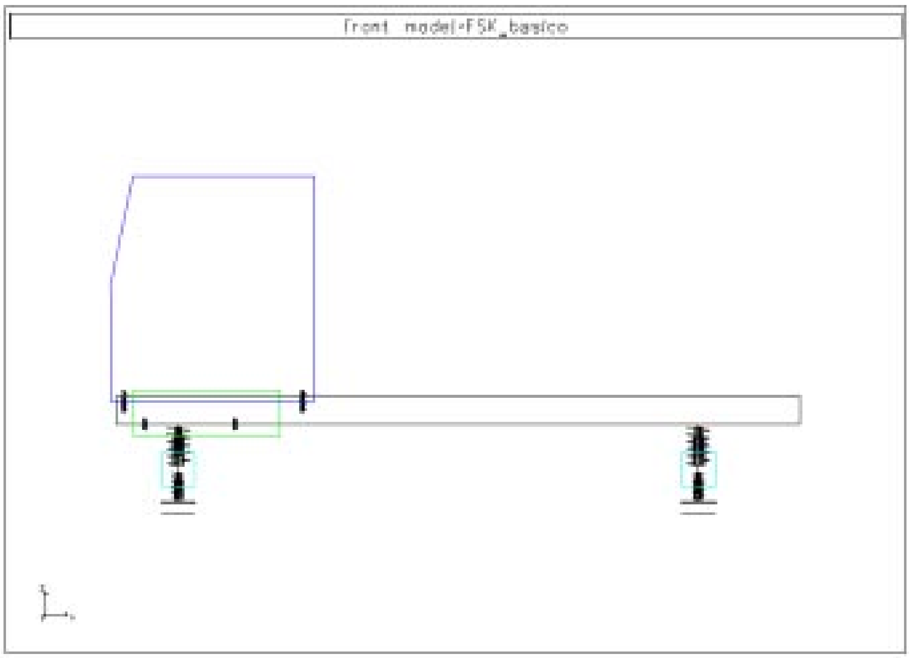


Figure 1 -- Plane model in ADAMS

This model has 8 degrees of freedom and it is made of 5 bodies. It was used to validate pitch plane rigid body modes of the truck.

It was performed a modal analysis of this model and the frequencies and modes described in table 1 resulted.

| Mode description | Frequency [Hz] |
|---|---|
| front axle | 7.5 |
| rear axle | 8.0 |
| front end bounce (FEB) | 1.8 |
| rear end bounce (REB) | 2.1 |
| engine FEB | 11.7 |
| engine REB | 6.3 |
| cab FEB | 12.5 |
| cab REB | 6.0 |

Table 1 -- Frequencies and modes of simple vehicle model

Powertrain Model - It was developed a separate model for the powertrain, suspended on its mounts and including the driveshaft. It was supposed to be supported on a rigid base. This model can be seen in figure 2 shown below.

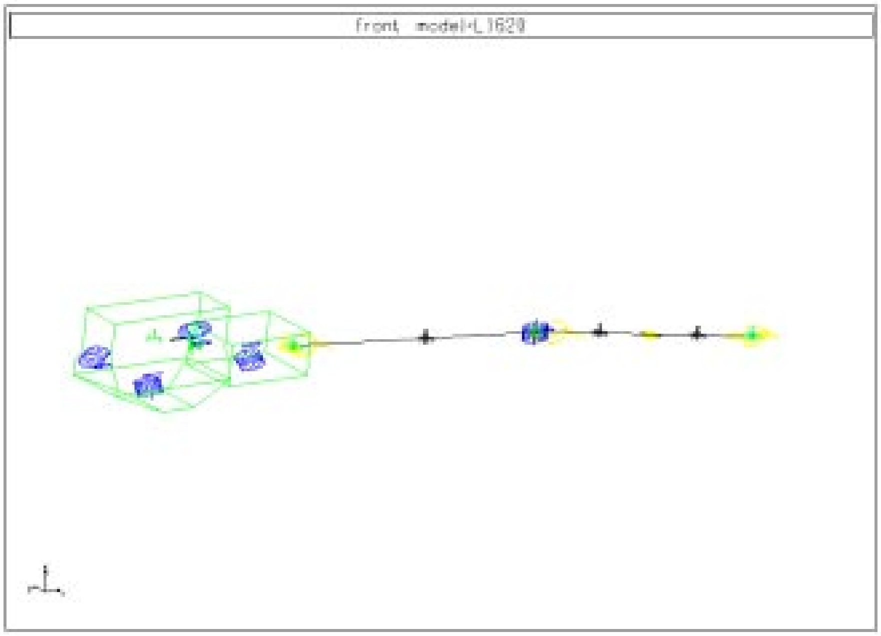

Figure 2 -- Powertrain and driveshaft model

It was also performed a modal analysis of this subsystem and the following main frequencies and modes described in table 2 resulted.

| Mode description | Frequency [Hz] |
|---|---|
| longitudinal | 5.7 |
| pitch REB | 6.3 |
| lateral and upper roll | 7.3 |
| yaw | 7.8 |
| FEB | 10.7 |
| lower roll | 17.3 |

Table 2 -- Powertrain frequencies and modes

Cab Models - The cab was modeled in two different ways. First it was considered the cab and the hood to be rigidly attached, this situation is illustrated in figure 3.

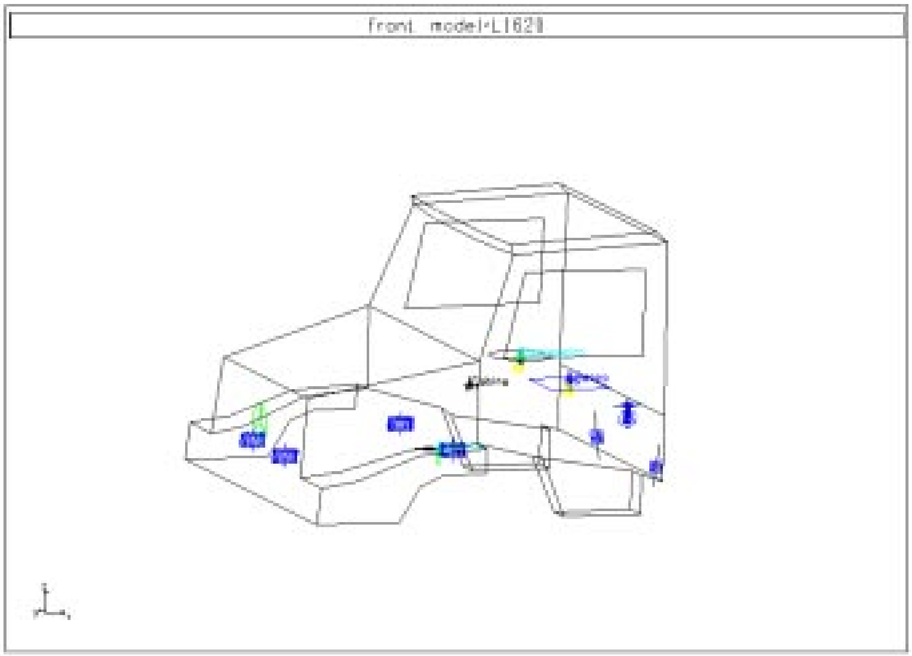


Figure 3 -- One body cab model

Afterwards, they were modeled as separate bodies, connected by bushing elements. The reason for this approach was tests carried out with the vehicle without the hood. Figure 4 shows this model.

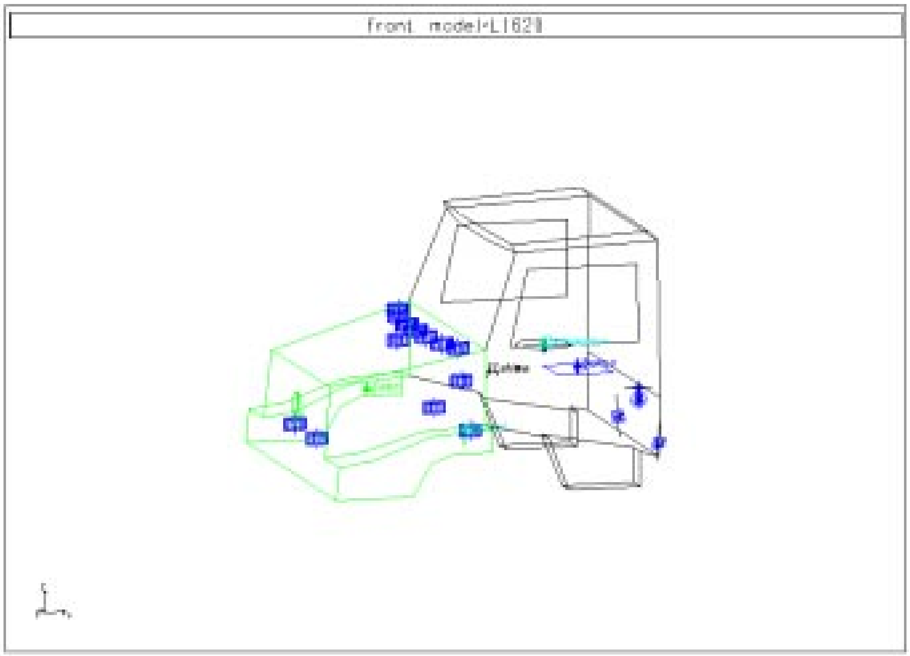


Figure 4 -- Cab and hood separate model

The results of a modal analysis in ADAMS gave the results shown in table 3 for the main frequencies and modes for both models of the cab and hood.

| One body cab model | |
|---|---|
| Mode description | Frequency [Hz] |
| cab roll | 5.8 |
| cab pitch | 6.3 |
| cab yaw | 7.8 |
| longitudinal | 9.9 |
| lateral and roll | 12.6 |
| FEB | 19.8 |
| Cab and hood separate model | |
| Mode description | Frequency [Hz] |
| cab pitch REB | 3.5 |
| hood and cab roll | 5.8 |
| cab lateral and yaw | 7.5 |
| cab longitudinal | 9.8 |
| cab FEB, hood REB | 11.8 |
| hood yaw | 11.9 |
| cab lateral and yaw | 12.8 |
| hood longitudinal | 15.9 |
| hood lateral | 16.3 |
| hood roll | 18.0 |
| hood REB | 21.9 |
| hood FEB | 30.4 |

Table 3 -- Cab and hood frequencies and modes

These models will be used later on to generate the full model in ADAMS.

FINITE ELEMENT MODEL - A full vehicle model with great detail of the frame and with the rigid bodies represented in an approximate manner was developed in NASTRAN. The model has 30,000 degrees of freedom, and it can be seen in figure 5 pictured below.

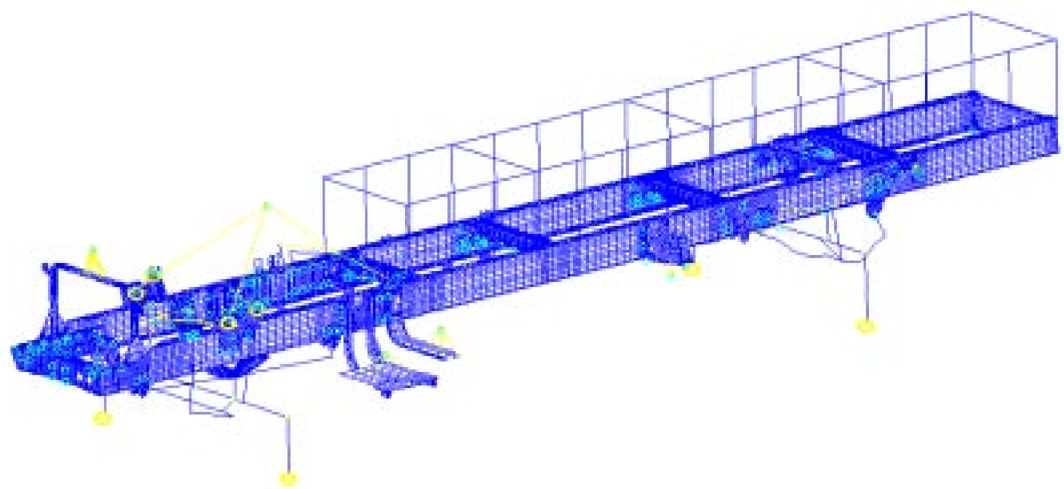

Figure 5 -- NASTRAN model

A modal analysis of the FE model was carried out. The mode illustrated in figure 6 below with a frequency of 6.0 Hz was considered to represent the most prone to cause problems. The large number of subsystem motions involved is the reason.

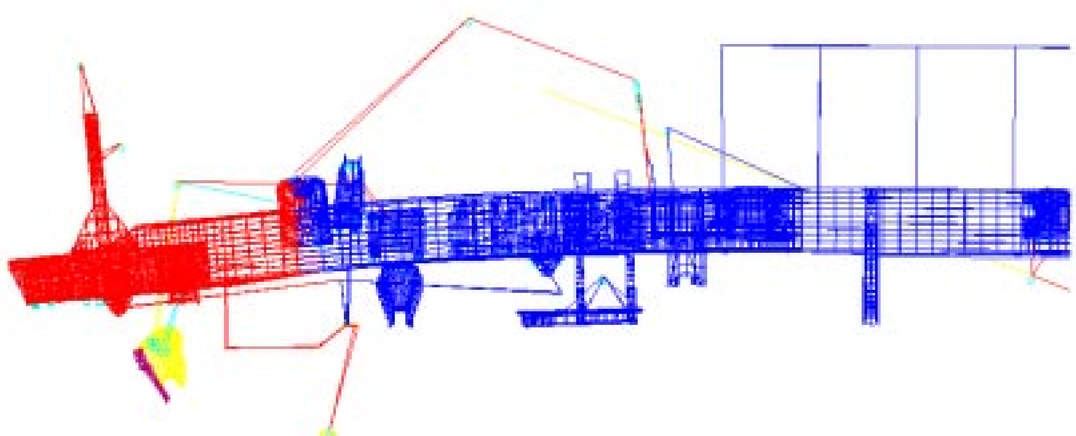

Figure 6 -- First bending mode of the frame

Frame Model - From the Finite Element modal analysis of the vehicle, in NASTRAN, a frame model in ADAMS was developed using the beam element. It can be seen in figure 7, pictured below.

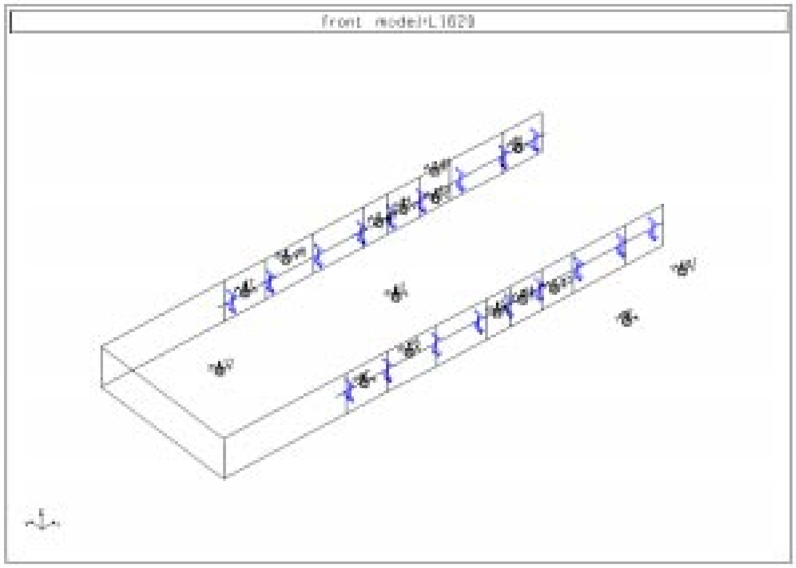


Figure 7 -- Frame model in ADAMS

FULL ADAMS MODEL - A full ADAMS model was built using the previous analyses. The powertrain, cab and frame subsystem models were used. The Hotchkiss suspension template of the ADAMS/Vehicle module was also used. A special topology was developed to represent the tire and road contacts. Figure 8 illustrates the full vehicle model developed in ADAMS.

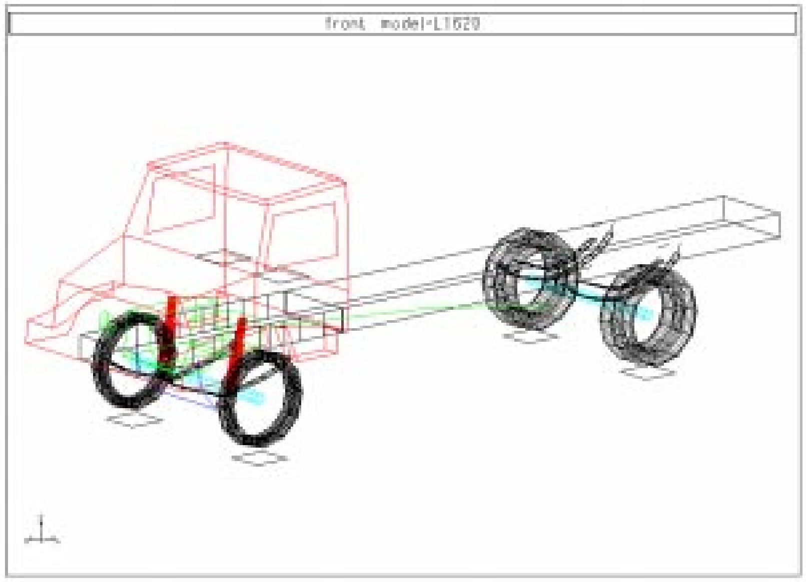


Figure 8 -- Full ADAMS model

The model comprises 145 bodies and 634 degrees of freedom for the truck. This model also considers the second stage of the rear suspension. The main reason for this large number of parts and degrees of freedom is the leaf springs of the suspensions, which are modeled as beam elements in ADAMS. This fact makes full vehicle model simulation rather slow, even with a powerful machine.
One factor which strongly affects flat leaf spring behavior is internal friction. The effect on suspension characteristic is an increase in equivalent vertical stiffness when the suspension is subject to small displacements. Various ways of modeling such behavior have been described in the literature [2]. In our case, when the vehicle was driving over the ramp or on a bad quality paved road, suspension travel was large enough to minimize this effect. In the case of cab longitudinal vibration, an equivalent stiffness was adopted by changing the E Modulus of the leaves. Later models have already incorporated hysteresis effects.

**MODEL VALIDATION**

For the purpose of model validation, time history, as well as frequency domain spectral analysis were used. Two types of inputs were considered. Transient terminated ramp and random road profile.

TERMINATED RAMP INPUT - Model validation using terminated ramp input was performed in time and frequency domain. Ramp description is according to figure 9 shown below. Vehicle speed, used to transverse the ramps, was 15 km/h.

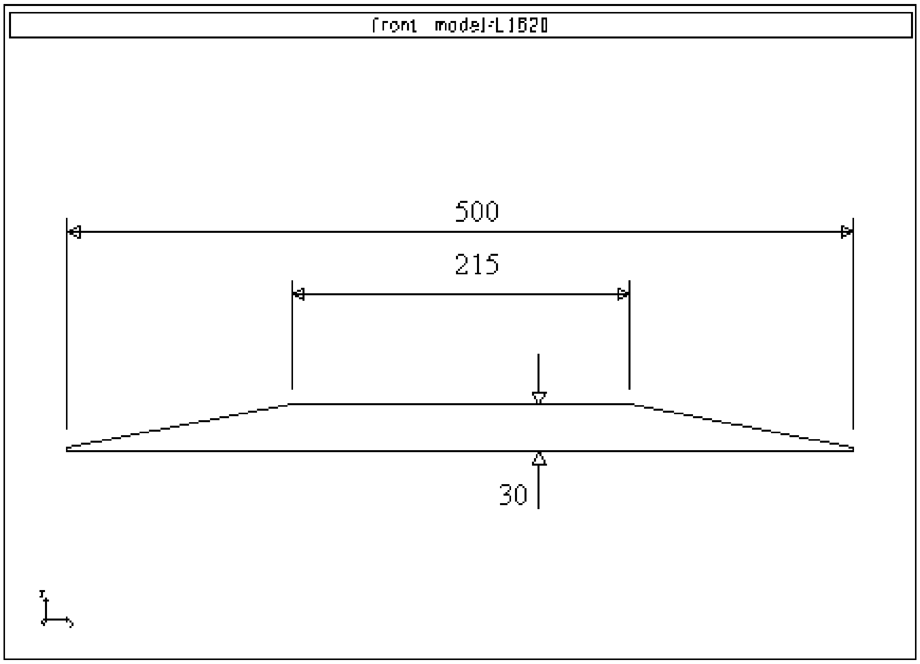


Figure 9 -- Elevation of the ramps

Two kinds of tests were carried out: Parallel and out-of-phase left and right ramps. These two cases are shown in figure 10 below.

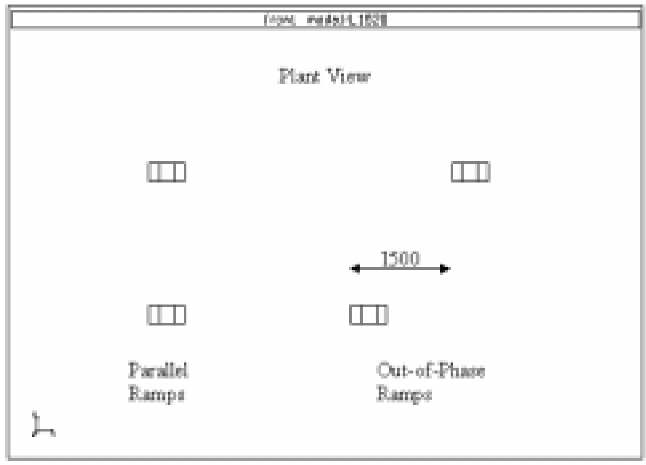


Figure 10 -- Plant view of ramp arrangements

Time histories - Parallel ramps were used to excite pitch plane modes and out-of-phase arrangement to excite roll modes. Quantities used to validate the model were vehicle vertical accelerations. It was measured front and rear, left and right accelerations in the axles and the frame, left and right accelerations in the cab, and at the driver's seat in all three directions. Figure 11 below shows some of these results for the parallel ramps. The red line corresponds to test results and the black one to the ADAMS full model response for the unloaded vehicle.

{ EMBED Word.Picture.8 }

Figure 11 -- Time histories of some vertical accelerations of the truck

It can be observed good agreement between experimental results and simulation. One of the main sources of discrepancies, however, is the nonlinear hysteretic behavior of the leaf springs when subjected to small displacements, which they are subject to before and after the ramps.

Frequency analysis - Performing a spectral analysis on these results, together with the previous time histories, one can obtain the value of frequencies and modes of the truck. In the ADAMS model an eigenanalysis is carried out in order to validate the model in the frequency domain and to better understand the complex modes that result for the full vehicle. Table 4 describes the main frequencies and modes of the full vehicle model. From this analysis it can be observed coupling of motions of various subsystems at certain modes.

| Mode description | Frequency [Hz] |
|---|---|
| front chassis | 1.7 |
| rear chassis | 2.0 |
| cab pitch and chassis pitch on tires | 3.4 |
| cab roll | 5.2 |
| frame bending and engine pitch | 6.3 |
| frame bending and engine yaw | 6.8 |
| front axle | 7.5 |
| rear axle | 8.0 |

Table 4 - Frequencies and modes of full vehicle model

Spectral analysis of the previous time histories are illustrated in figure 12 below and they confirm ADAMS modal analysis results.

**{** EMBED Word.Picture.8 **}**
Figure 12 – Frequency spectra of some vehicle vertical accelerations

## COMFORT STUDY

After being validated, vehicle model can be used to analyze the effect of some design parameters on vehicle comfort based on the ISO Standard. In order to carry out this study it is necessary to generate a random road profile.

RANDOM ROAD GENERATION - For this purpose, a random profile was generated using SIMULINK, a simulation toolbox from MATLAB. Figure 13 illustrates the block diagram used to generate such road.

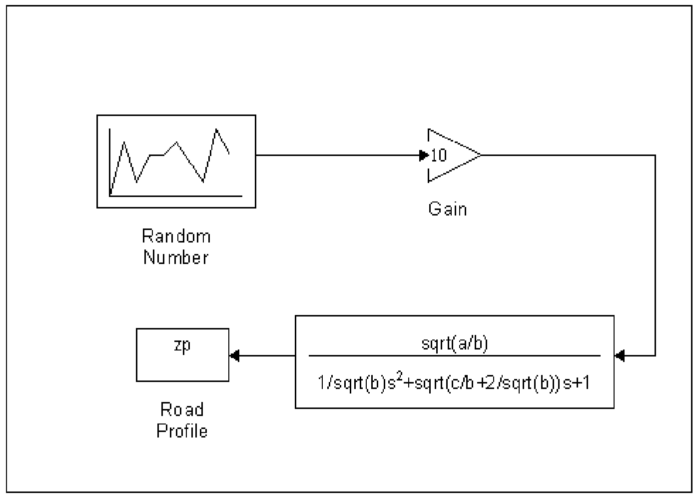


Figure 13 – Block diagram for road profile generator

Data from the literature and real road tests were used to adjust road model parameters. Time histories of ADAMS model and road responses measurements can be seen in figure 14 below. The black line represents real vehicle response and the red, ADAMS model output.

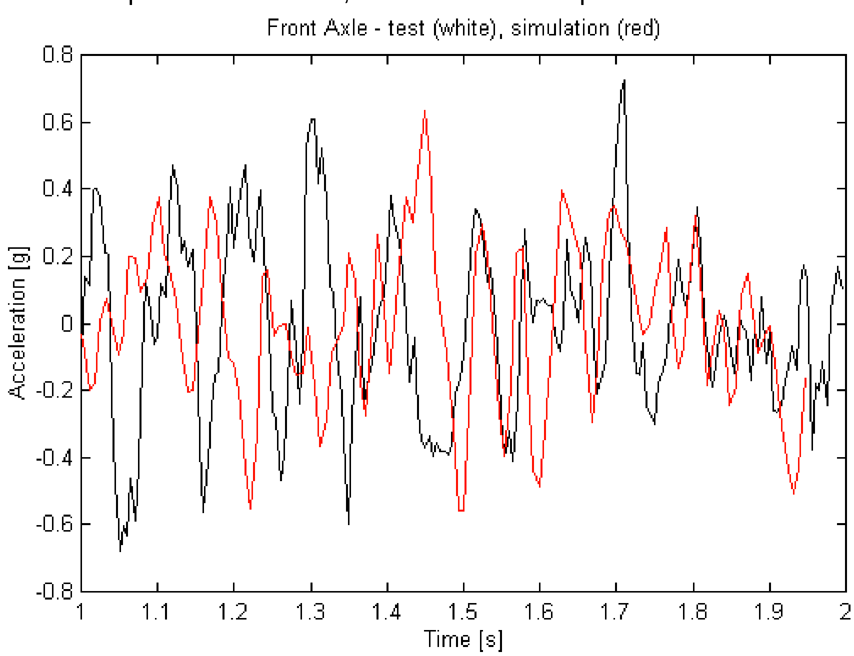


Figure 14 -- Time histories of front axle random responses

From the previous figure it can be observed good agreement between both responses and therefore vehicle and road models can be used to study the effect of a design parameter on vehicle ride comfort.

DESIGN PARAMETER VARIATION - For illustrative purpose only, it was chosen to study the effect of the rear spring of cab suspension. It was chosen to simulate two springs rates that Mercedes Benz already has in stock and fit the present vehicle. One is 60 N/mm (green) and the other 210 N/mm (black). Following the guidelines of the ISO Standard it was obtained the graphs shown in figure 15 for the vertical direction and for the fatigue decreased proficiency boundary. This figure, as well as all the calculation necessary to reduce the simulation data to the ISO format were carried out using m-files in MATLAB.

{ EMBED Word.Picture.8 }

Figure 15 -- Comparison of comfort level according to ISO 2631-85 for two rear springs of cab suspensions

It can be observed an increase in the time tolerance for the driver in the vertical direction, near the critical 4-8 Hz band, for the softer spring, considering the criteria adopted.

**CAB LONGITUDINAL VIBRATION**

During vehicle testing it was observed a certain amount of longitudinal cab vibration at the driver and passenger level. This situation occurred when the vehicle was loaded and driving at a speed of 73 km/h. The situation happened more markedly on good paved roads. The main component of this vibration occurred at a frequency of 6.3 Hz. Figure 16 illustrates time histories and power spectral densities of longitudinal roof acceleration.

{ EMBED Word.Picture.8 }
Figure 16 -- Longitudinal roof acceleration

Further calculations indicated this to be the rotating frequency of the wheels. This characteristic had already been observed in the literature before [1]. Its source was traced to wheel run-out errors. They excite certain vehicle modes which present low impedance and has motions of various subsystems coupled

The literature indicates that wheel run-out can be adequately described by a periodic signal with three significant components. It is given by the following expression, and it is illustrated in figure 17.

{EMBED Equation.2}

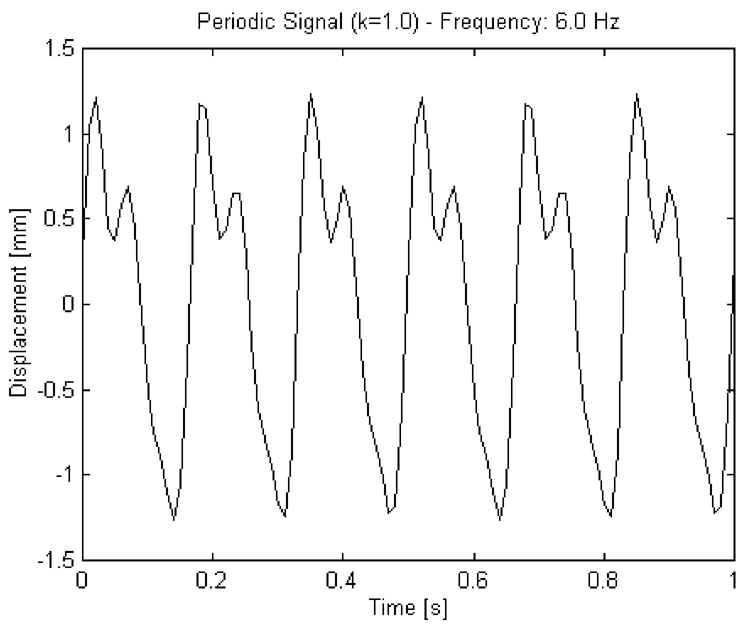


Figure 17 – Wheel run-out

Initially, it was thought that cab pitch, powertrain front end bounce and frame bending motions were coupled in the full vehicle system mode. Tests carried out later with the removal of the hood, indicated that cab was carried by its support during frame bending, as a consequence of the higher frequencies of cab longitudinal and cab FEB when compared with the first frame bending frequency. The results of a modal analysis of the full vehicle model developed in ADAMS clearly indicated the above mentioned behavior. Figure 18 illustrates the superposed modal animation obtained in ADAMS.

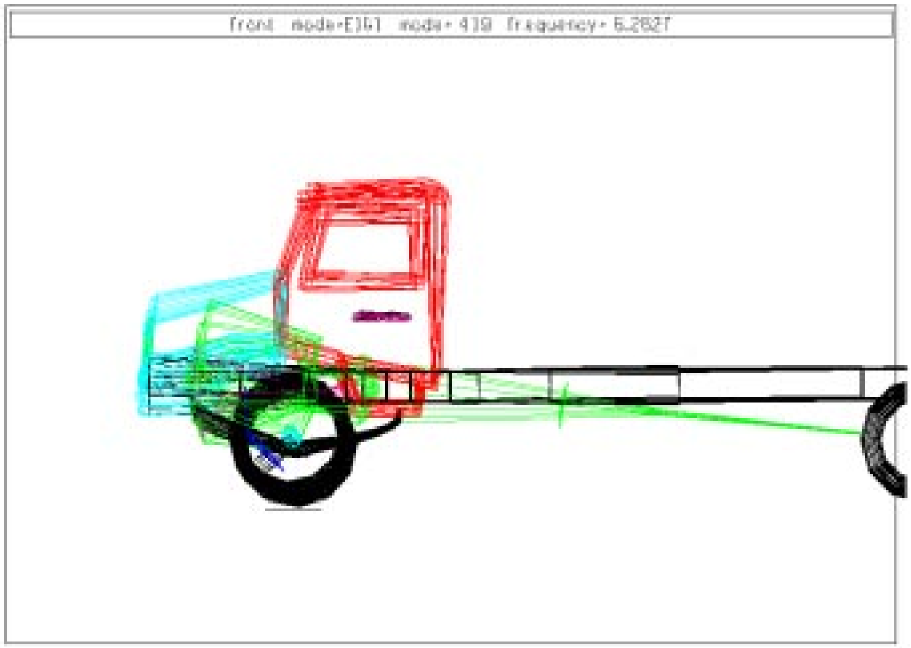

Figure 18 – Full vehicle mode with cab longitudinal vibration

Further measurements of other quantities confirmed the diagnosis. Figure 19 below illustrates vertical accelerations of front cross-bar of the frame. Figure 20 shows vertical acceleration of front end of the engine.

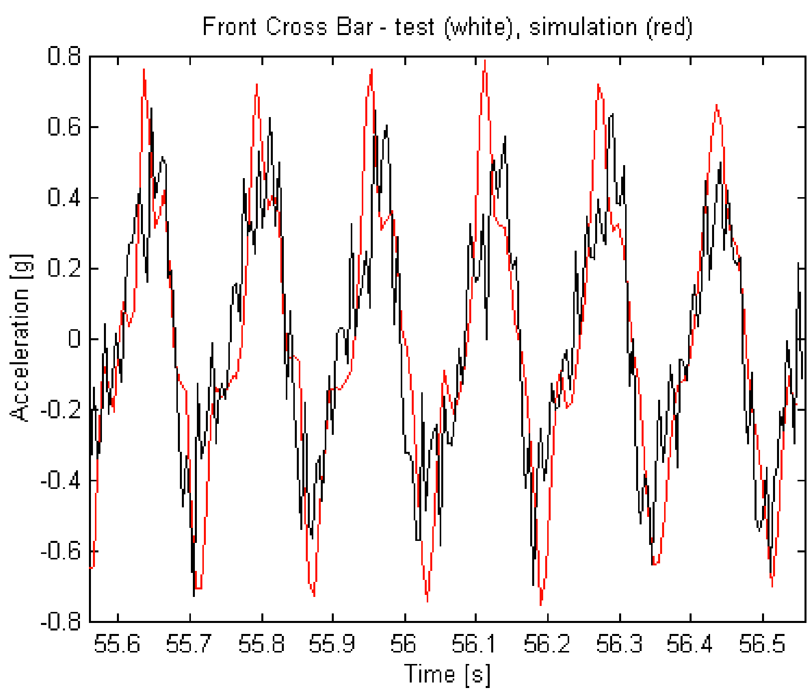


Figure 19 – Vertical acceleration - front crossbar test and simulation

{ EMBED Word.Picture.8 }
Figure 20 - Vertical acceleration - front engine test and simulation

Some possible directions for design changes to decrease the vibration level were: frame stiffening, softening of cab suspension and modification of the powertrain mounting system.

Experience and analysis with the model indicated the little effectiveness of frame stiffening due to its global bending characteristic. The softening of cab suspension at levels which would put its higher frequency below the 6.3 Hz were impractical given the present cab suspension concept. However, some tests were carried out, which indicated the solution. Figure 21 shows the modal analysis with a soft cab suspension. There is little cab movement when compared with the frame. The only feasible alternative for modification was to change the engine mounts. A retune of its stiffness were performed and new values for powertrain frequencies were obtained. These new values are given in table 5 shown below.

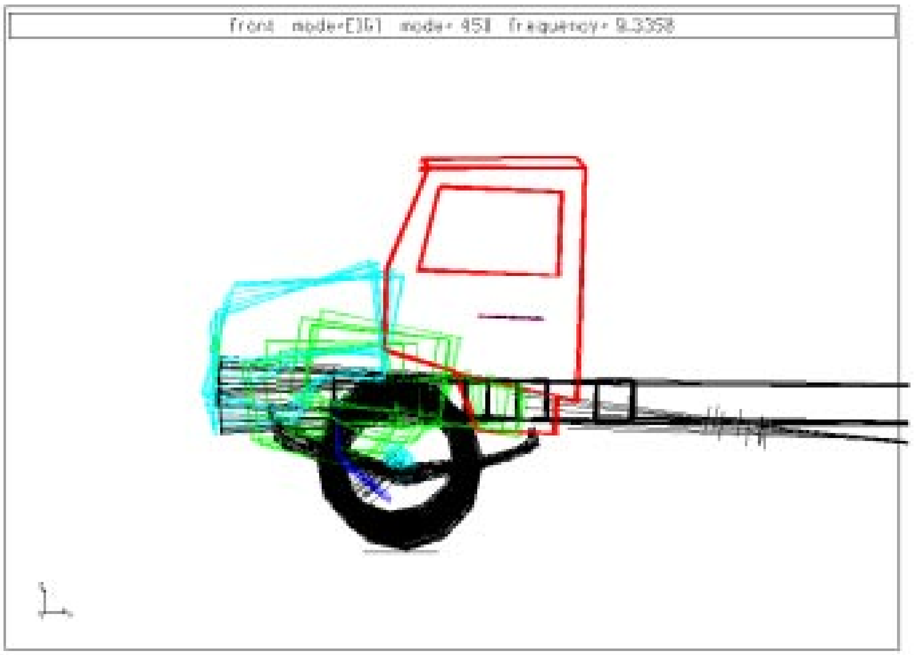

Figure 21 - Full vehicle mode with frame bending vibration - soft cab mounts

| Mode description | Frequency [Hz] |
|---|---|
| pitch | 4.5 |
| yaw | 4.8 |
| longitudinal and yaw | 5.3 |
| lateral and lower roll | 6.5 |
| vertical | 8.6 |
| roll | 15.9 |

Table 5 – Powertrain frequencies with new mounts

These values decoupled powertrain motion from frame bending at that frequency and the level of vibration was significantly reduced. Modal analysis in ADAMS of the full model with the new engine mounts can be seen in figure 22, illustrated below.

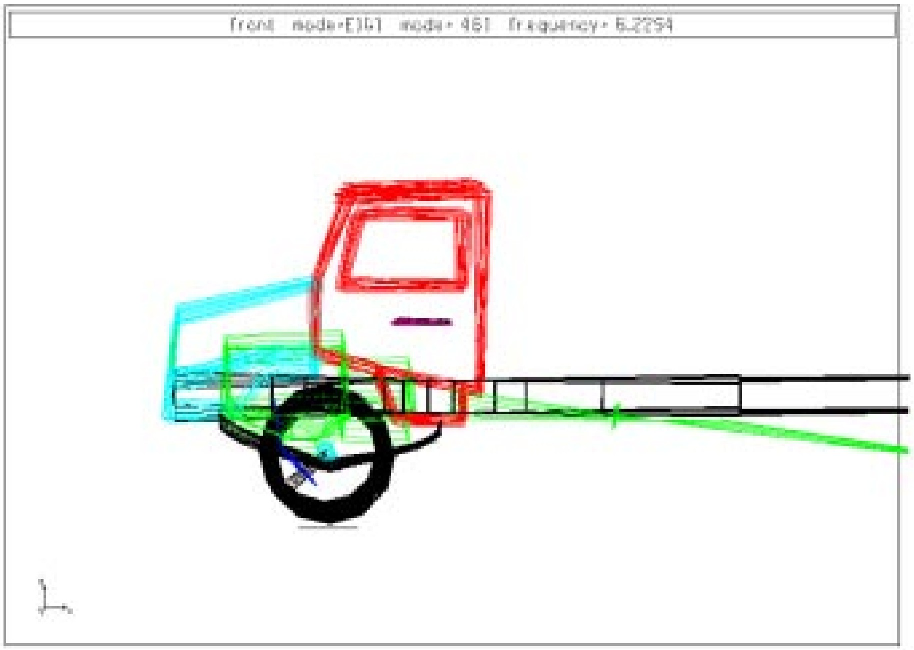

Figure 22 – Full vehicle mode with cab longitudinal vibration and  new engine mounts

The proposed mount values were implemented in the truck and tests confirmed simulation results. Test results for the previous mentioned quantities are illustrated in figure 23, shown below.

**{** EMBED Word.Picture.8 **}**
Figure 23 – Vehicle test results with new engine mounts

**CONCLUSIONS**

The present work described the use of multibody system simulation in ADAMS for the study of the vibrational behavior of a medium sized Brazilian truck. Preliminary models were developed which greatly helped understand vehicle behavior in the early stages of the study. Full vehicle model was validated using simplified deterministic ramp inputs. Road data was used to generate a virtual road profile for comfort studies. Vehicle comfort was improved through the use of vehicle simulation and verified subjectively by test drivers. Vehicle model was used to decreased cab longitudinal vibration through redesign of engine mount system. Full vehicle model was very important to describe and understand such a complex phenomenon involved in vehicle vibration.

**ACKNOWLEGEMENTS**

The authors wish to thank Mercedes-Benz do Brasil for authorization for publishing this paper.

**REFERENCES**

[1] - ‘Ride Comfort Aspects of Heavy Truck Design’ - J.I.Ribarits, J.Aurell, E.Andersen - AB Volvo Div.
[2] - ‘Heavy Truck Ride’ - Thomas D. Gillespie
[3] - ‘Engine Mount System Focusing to Reduce Vehicle Shake’ - A.R.Solomo - GM Truck Group
[4] - ‘Rahmenfestigkeit HPN-M96 L1620/5170’ - Berechnung Nutzfahrzeuge - 12.09.95
[5] - ‘A Heavy Truck Cab Suspension for Improved Ride’ - Albert W.Foster - Ford Motor Co.

**ABOUT THE AUTHORS**

Dr. Alvaro Costa Neto
B.Sc. in Mechanical Engineering, 1981, USP Sao Carlos
M.Sc. in Mechanical Engineering, 1986, USP Sao Carlos
Ph.D. in Engineering, 1991, University of Warwick, UK.
Professor at USP -São Carlos, System Dynamics, Multibody Systems, Vehicle Dynamics
Member of SAE Brasil, ABCM, IAVSD.
Odilon Terreri Perseguim
B.Sc. in Mechanical Engineering, 1996, USP Sao Carlos
Present work: Dynamic analysis of mechanical systems - debis humaitá IT Services Latin America
Member of SAE Brasil.

| | |
|---|---|
| Filename: | pap_brasil.doc |
| Directory: | \\Davinci\UserConf |
| Template: | C:\Program Files\Microsoft Office\Templates\Normal.dot |
| Title: | Paper ######## |
| Subject: | |
| Author: | Humaitá Informática e Tecnologia |
| Keywords: | |
| Comments: | |
| Creation Date: | 06/04/98 2:08 PM |
| Change Number: | 4 |
| Last Saved On: | 06/25/98 2:50 PM |
| Last Saved By: | Marilyn Lee |
| Total Editing Time: | 22 Minutes |
| Last Printed On: | 08/06/98 3:38 PM |
| As of Last Complete Printing | |
| Number of Pages: | 20 |
| Number of Words: | 2,615 (approx.) |
| Number of Characters: | 14,910 (approx.) |