DYNAMICS OF LARGE SCALE VEHICLE MODELS USING ADAMS/FLEX

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ABSTRACT
A general methodology is first presented for capturing dynamics of large order vehicle models in a systematic and computationally efficient way. The equations of motion of a simplified vehicle model are set up by applying classical finite element techniques and they may include nonlinearities in the suspension and the tires. The first step is to reduce the dimensions of the original system substantially by applying a component mode synthesis approach. This allows the application of appropriate numerical methodologies for predicting response spectra of nonlinear models to periodic road excitation. Results obtained by direct integration of the equations of motion are also presented for transient road excitation. In all cases, the accuracy and validity of the applied methodology is verified by comparison with results obtained for the original models. Finally, similar results are obtained by employing ADAMS/Flex, for a more involved vehicle model. These results represent the outcome of current efforts to develop a fully automated procedure of using ADAMS/Flex, based on the car model and the road profile alone.

1. INTRODUCTION
Research in the area of vehicle dynamics has progressively become more systematic and intensive since the early sixties. As a result, the fundamental concepts are now well documented in classical books and review papers (e.g., Ellis, 1969; Gillespie, 1992). These research efforts were significantly assisted by the simultaneous rapid improvements in computer technology and numerical analysis methods. On the other hand, there was an equally great progress in the area of dynamical systems analysis. In particular, new analytical, numerical and experimental methods have been developed, permitting a more global study and explanation of the behavior of dynamical systems with nonlinear characteristics (e.g., Nayfeh and Balachandran, 1995). Much of the new knowledge that has been accumulated in the area of nonlinear dynamical systems has already been transferred and applied to many practical applications (e.g., Natsiavas, 1993; Ben Mrad et al. 1994; Verros et al., 1999). This transfer is currently assisted by a large number of special software packages, which help in applying quite complicated and theoretical methodologies in an automatic way. However, there are still certain limitations in applying these methodologies to high order systems.

The main objective of the present work is to develop and apply systematic methodologies for an efficient study of large order vehicle models. This is necessary to explain and improve the dynamic behavior of contemporary vehicles. Usually, satisfaction of current design requirements demands the examination of nonlinear vehicle models with a quite large number of degrees of freedom. This poses serious difficulties in the successful execution of existing numerical codes. On the other hand, in a typical vehicle structure the major nonlinearities appear in the suspension and the tires. This permits the application of suitable methods, causing a substantial reduction in the number of degrees of freedom of the system. In the present study, this is achieved by employing an appropriate component mode synthesis methodology, which is commonly applied in structural dynamics (Craig, 1981). This reduction, in turn, makes possible the subsequent application of efficient numerical methodologies for determining the dynamic response of the reduced system.

The organisation of this paper is as follows. First, a simplified vehicle model is presented in the following section, together with the basic steps of a component mode synthesis methodology. Next, response to periodic road excitation is examined in section 3, by employing suitable methodologies leading to direct determination of steady state motions. Vehicle response to transient road excitation is also determined. The accuracy and effectiveness of the methodology is established by comparison of results obtained for reduced and the corresponding complete dynamical models. Finally, a more involved vehicle model is examined and similar results are obtained by employing ADAMS/Flex.

2. SIMPLIFIED VEHICLE MODEL - COMPONENT MODE SYNTHESIS

The most important ideas and steps of the present work are illustrated by examining the mechanical model shown in Fig. 1. It presents a half-car model, where the vehicle body is modeled as a beam, while the wheel and the suspension are represented by appropriate discrete elements. First, the equations of motion for the vehicle body are derived by the finite element method in the following form

$$\hat{M}_B \ddot{\underline{x}}_B + \hat{C}_B \dot{\underline{x}}_B + \hat{K}_B \underline{x}_B = \hat{\underline{f}}_B(t)\,. \tag{1}$$

For a typical vehicle model, the order of the equations of motion (1) is quite large. However, for a given level of forcing frequencies it is possible to reduce significantly the number of degrees of freedom, without sacrificing the accuracy in the numerical results, by applying standard component mode synthesis methods (Craig, 1981). Namely, through a coordinate transformation with form

$$\underline{x}_B = \Psi \underline{q}_B\,, \tag{2}$$

the original set of equations (1) can be replaced by a considerably smaller set of equations, expressed in terms of the new generalized coordinates $\underline{q}_B$.

The Ritz transformation (2) includes a contribution from a set of free-interface normal modes, whose selection is based on the frequency content of the excitation. Moreover, the effect of the deleted normal modes is taken approximately into account by including residual flexibility modes. The number of these modes is equal to the number of the boundary degrees of freedom of the component, which are those subjected to external loads or connecting the vehicle body to the suspension. Application of (2) into the original set of equations (1) yields the smaller set

$$M_B \ddot{\underline{q}}_B + C_B \dot{\underline{q}}_B + K_B \underline{q}_B = \underline{f}_B(t)\,, \tag{3}$$

where

$$M_B = \Psi^T \hat{M}_B \Psi\,, \quad C_B = \Psi^T \hat{C}_B \Psi\,, \quad K_B = \Psi^T \hat{K}_B \Psi \quad \text{and} \quad \underline{f}_B = \Psi^T \hat{\underline{f}}_B\,.$$

Similar sets of equations of motion are then obtained for each of the suspension/wheel components. Taking into account all of their degrees of freedom and ignoring the nonlinear terms, temporarily, these equations are first written in the form

$$M_A \ddot{\underline{q}}_A + C_A \dot{\underline{q}}_A + K_A \underline{q}_A = \underline{f}_A(t)\,, \tag{4}$$

Then, the equations of motion of the composite system are derived in the classical form

$$M \ddot{\underline{q}} + C \dot{\underline{q}} + K \underline{q} = \underline{f}(t)\,. \tag{5}$$

This is accomplished by considering appropriate energy quantities. For instance, for the total potential energy of the system it is true that

$$V = V_A + V_B \qquad \tfrac{1}{2}\underline{q}^T K \underline{q} = \tfrac{1}{2}\underline{q}_A^T K_A \underline{q}_A + \tfrac{1}{2}\underline{q}_B^T K_B \underline{q}_B\,,$$

which eventually determines the stiffness matrix of the composite system. In a similar fashion, consideration of the kinetic energy and the virtual work determines the mass matrix and the excitation vector of the composite system, respectively.

In the final stage, the system nonlinearities are appropriately added in (5). This step can be performed easily, since the nonlinearities appear in the suspension and the wheels only and all the degrees of freedom involved have explicitly been included in the displacement vector $\underline{q}$. For instance, the contribution of the suspension dampers may be expressed in the bilinear form (Wallaschek, 1990), while the tire stiffness may be modeled as Duffing type spring. Therefore, the equations of motion can eventually be put in the form

$$M \ddot{\underline{x}} + C \dot{\underline{x}} + K \underline{x} + \underline{g}(\underline{x}, \dot{\underline{x}}) = \underline{f}(t)\,, \tag{6}$$

where the vector $\underline{g}(\underline{x}, \dot{\underline{x}})$ includes the nonlinear terms, while the vector $\underline{f}(t)$ represents the external forcing, resulting from road irregularities.

Apart from increasing the computational efficiency and speed, the reduction of the system dimensions makes amenable the application of several numerical integration techniques, which are applicable and efficient for low order dynamical systems. When the effect of the system nonlinearities is negligible, its dynamic response can be determined by applying standard modal analysis or direct integration procedures. Also, when only the dual-rate damping nonlinearities are of importance, exact response of the system to periodic excitation can be obtained by applying appropriate procedures developed for piecewise linear systems (Natsiavas, 1993). For the general nonlinear case, the response can be determined by direct integration of the equations of motion. However, when the excitation is periodic, suitable methodologies can be applied which capture directly periodic steady state responses and their stability properties (Nayfeh and Balachandran, 1995).

3. RESPONSE SPECTRA

Typically, evaluation of the vehicle ride performance is based on examination of frequency spectra for three types of response quantities, within the range of 0-25Hz (Gillespie, 1992). Namely, the maximum absolute acceleration at specified locations (related to passenger comfort and isolation), the suspension travel (related to wheel space dimensions) and the wheel traction (related to vehicle handling and safety). Next, numerical results are presented for vehicle models subjected to harmonic road excitation.

The original beam model consists of 24 dof and its first 8 (undamped) natural frequencies are depicted in the first column of Table 1. The corresponding reduced model is chosen to have 8 dof, including 2 dof for the wheels and 6 dof for the vehicle body (2 rigid body modes, 2 residual flexibility modes and 2 free-interface normal modes), so that it is accurate up to excitation frequencies of about 30Hz. The natural frequencies of the reduced model are shown in the second column of Table 1. Finally, the third column of Table 1 includes the natural frequencies of the corresponding model with rigid body.

Figure 2 presents typical response diagrams obtained for a road with harmonic profile and different values of the horizontal velocity v_0. It shows the absolute acceleration $\ddot{w}_A$ at the connection point A between the vehicle body and the rear suspension. The damping coefficients c_f and c_r are selected so that both suspensions posses a damping ratio of $\zeta = 0.3$. The thick/thin continuous curves correspond to response of the complete/reduced beam model, respectively. The results obtained for the reduced model virtually coincide with the results of the original model, within the specified frequency range of interest. Likewise, the broken lines represent response obtained by simply truncating the complete model to its 6 lowest frequency modes, while the dotted lines represent response of the corresponding model with rigid vehicle body. These results demonstrate the appearance of significant numerical errors throughout the frequency range examined when the residual flexibility effects are not taken into account. In addition, complete omission of the vehicle body flexibility effects results in even larger errors, especially for frequencies above 15Hz.

Next, Fig. 3 shows response spectra obtained for suspensions with damping ratio $\zeta_- = 0.1$ and $\zeta_+ = 0.3$, in jounce and in rebound, respectively. The steady state response of the reduced model is again quite similar to the response of the complete model, within most of the frequency range of interest. Also, in accordance with results obtained for similar but single degree of freedom vehicle models (Verros et al., 1999), all the periodic solutions presented are harmonic and stable. Finally, the response diagrams obtained for tires with Duffing type restoring forces in the vertical direction are very similar, indicating a small effect of the tire nonlinearity considered on the vehicle response. One important factor leading to this behavior is the relatively large amount of the vehicle suspension damping, permitting only small tire deformations, which are not capable of activating the nonlinearity effects in a significant way.

4. TRANSIENT ROAD EXCITATION

This section presents numerical results obtained for transient excitation. First, the beam vehicle models are assumed to pass over a road with an isolated irregularity of the form $s(z) = s_0 \sin(2\pi z/\lambda)$, within the interval $0 \le z < 10\lambda$. The road wavelength is chosen so that for a horizontal velocity $v_0 = 150 km/h$ the resulting forcing frequency is $\omega = 20 Hz$. In all cases, the response is determined by direct integration of the equations of motion.

First, Fig. 4 shows the absolute acceleration at point A, for a linear model with $\zeta = 0.3$. Clearly, the agreement between the response of the complete model (thick line) and the reduced model (thin line) is in both cases satisfactory. The main deviations are observed at the beginning of the intervals following the entrance and the exit of the vehicle from the road irregularity. This is justified, since at these instances the system starts two distinct vibration intervals and the effect of all the modes is initially important, with the higher frequency modes being damped out more quickly.

Figure 5 presents similar results for the case corresponding to the beam model, with bilinear suspension dampers ($\zeta_- = 0.1$, $\zeta_+ = 0.3$). Here, some difference is observed throughout the time interval where the vehicle passes over the road irregularity. To some extent at least, this may be attributed to the continuous switch of the damping coefficient of the system, which is also indicated at the top of the same figure.

The folllowing set of numerical results was obtained for a more involved vehicle model, whose body is modeled by appropriate finite elements, as shown in Fig. 6. The complete model was assembled and run by using ADAMS/Flex, in cooperation with the NASTRAN code. First, Fig. 7 presents the vertical force developed on the hub of the front left wheel for one run of the vehicle on a specific straight road, with a constant horizontal velocity. The continuous line indicates results obtained by assuming that the vehicle body is rigid, while the broken line was obtained by taking into account the flexibility of the vehicle body. For comparison purposes, it is mentioned that the total CPU time needed to run the model with the flexible body is increased by an order of magnitude. However, this time is drastically smaller than the time necessary to run the flexible model, without performing the coordinate reduction. These results reinforce the efforts aimed at developing an automated procedure of using ADAMS/Flex, based on the car model and the road profile alone.

The final set of results was also obtained for the car model shown in Fig. 6, by following a slalom path. More specifically, the continuous/broken line of Fig. 8 indicates the desired/followed path of the vehicle, respectively. In addition, Fig. 9 presents the vertical force on the front left wheel hub obtained for the vehicle model with a rigid body (continuous line) and the model with a flexible body (broken line).

5. SUMMARY AND EXTENSIONS

The first part of this work presented a systematic methodology for determining the ride response of large order vehicle models in a computationally efficient way. The basic idea was to start the solution process by first applying an appropriate component synthesis method in order to eliminate some of the degrees of freedom, so that the reduced model is accurate up to a specified level of forcing frequencies. For nonlinear vehicle models, this methodology exploits the fact that the most important nonlinearities appear in the suspension substructures. Apart from the substatntial direct computational savings, the reduction in the model dimensions made possible the application of modern powerful numerical codes capturing steady state response and stability of nonlinear dynamical systems in a direct way.

The accuracy and effectiveness of the new methodology was illustrated by presenting numerical results for several vehicle models. First, frequency spectra of several response quantities related to vehicle ride performance were constructed for steady state motions resulting from periodic road excitation. Then, transient response was also investigated for several typical combinations of the system parameters. Finally, similar results were obtained for a more involved vehicle model, by employing ADAMS/Flex. These results showed that application of such methodologies leads to a substantial decrease of the computing time, without sacrificing the accuracy of the results.

A great advantage of the present methodolody may be realised by extending its capability to dealing with hybrid models, consisting of components modeled by analytical means and components modeled with experimental data. Similar advantages are expected in problems of structural and acoustic interaction. Finally, the advantages of reducing the dimensions of the original model will also be beneficial in other important areas, such as diagnostics and vehicle control.

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Table 1: Natural frequencies of vehicle model with beam body

ω_n [Hz]	Complete model	Reduced model	Rigid model
1	0.89	0.89	0.89
2	1.27	1.27	1.27
3	10.17	10.17	10.27
4	10.27	10.27	10.36
5	10.42	10.42	
6	28.15	28.15	
7	55.21	62.70	
8	91.41	140.75	

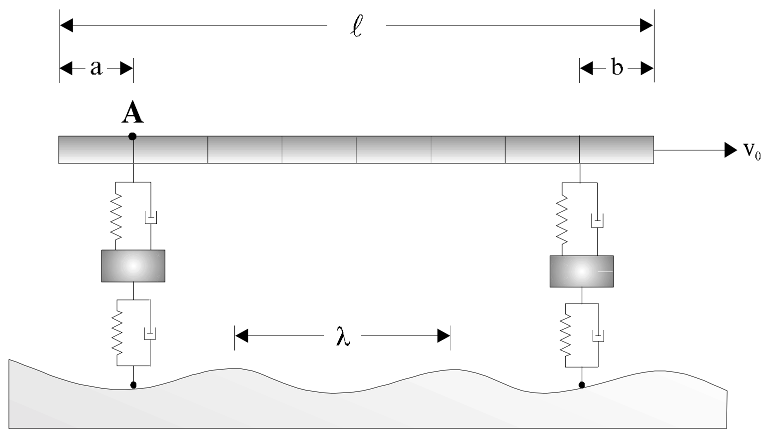

Figure 1. Simplified vehicle model.

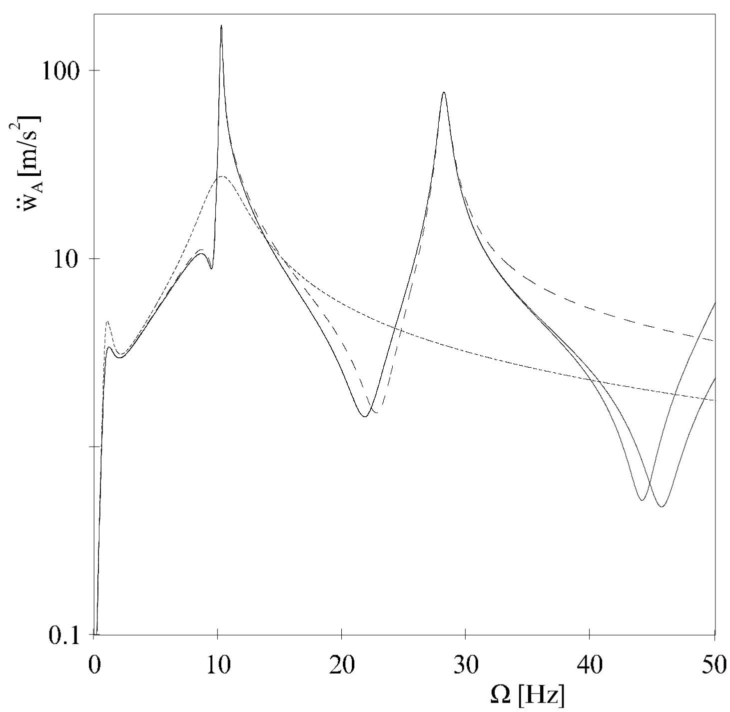

Figure 2. Response diagram of linear beam vehicle models with $\zeta = 0.3$.

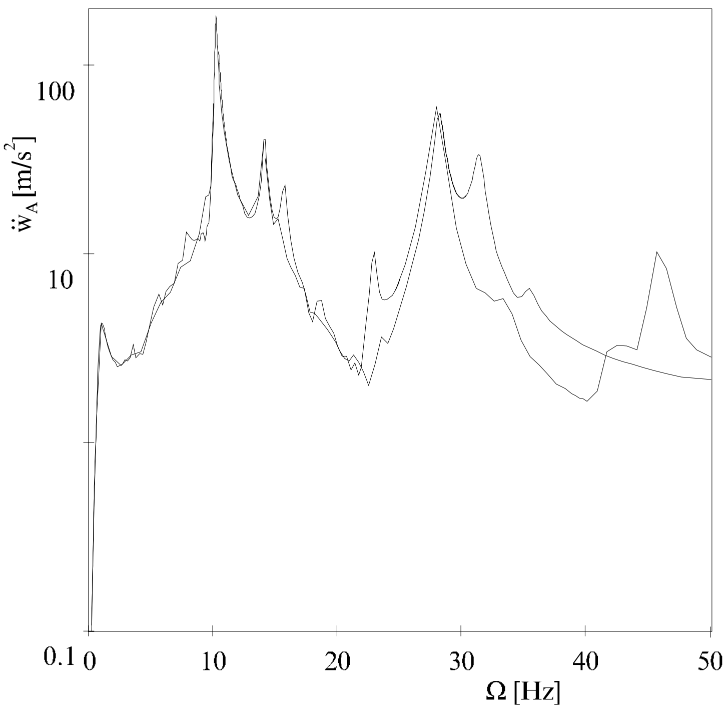

Figure 3. Response diagram of vehicle models with bilinear suspension dampers.

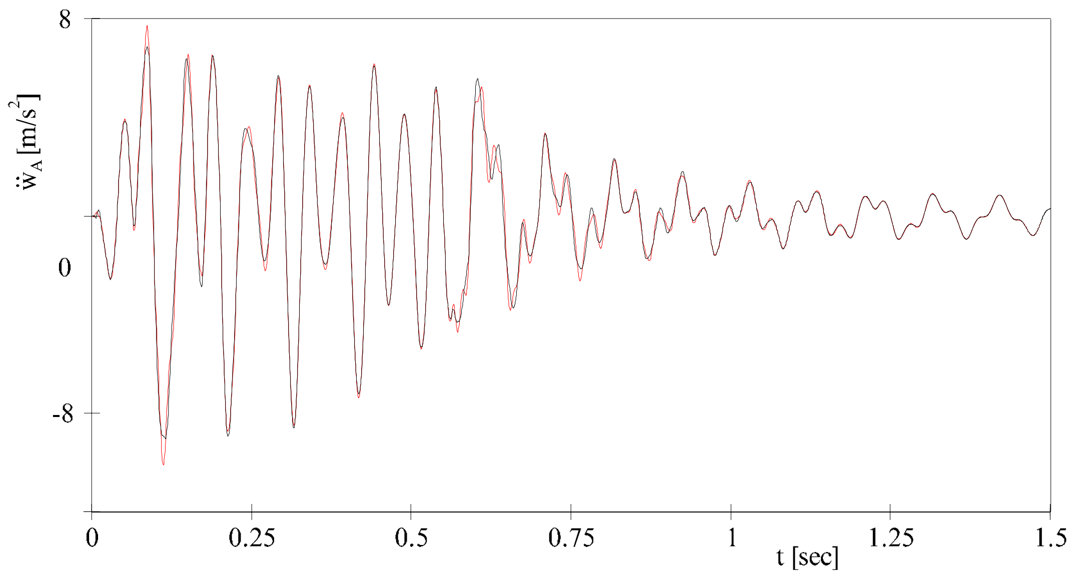

Figure 4. Response history of absolute acceleration at $\omega = 20Hz$ for linear beam model.

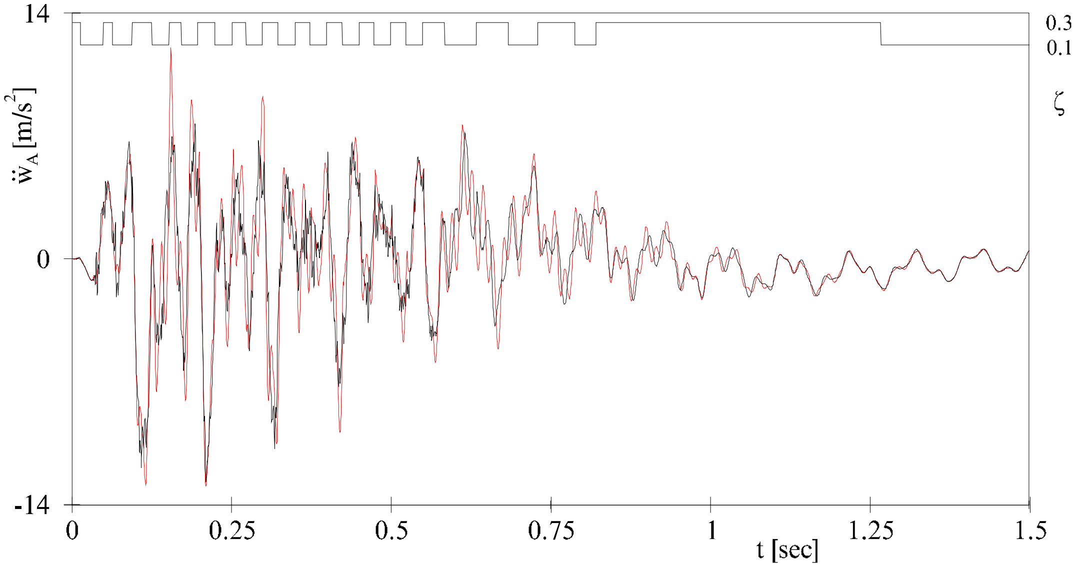

Figure 5. Response history at $\omega = 20Hz$ for a vehicle model with dual-rate dampers.

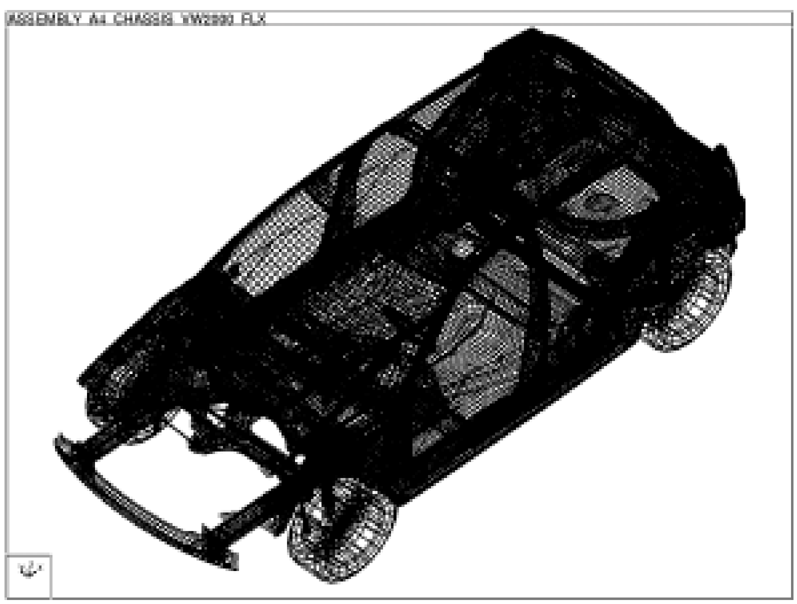

Figure 6. Finite element model of vehicle body.

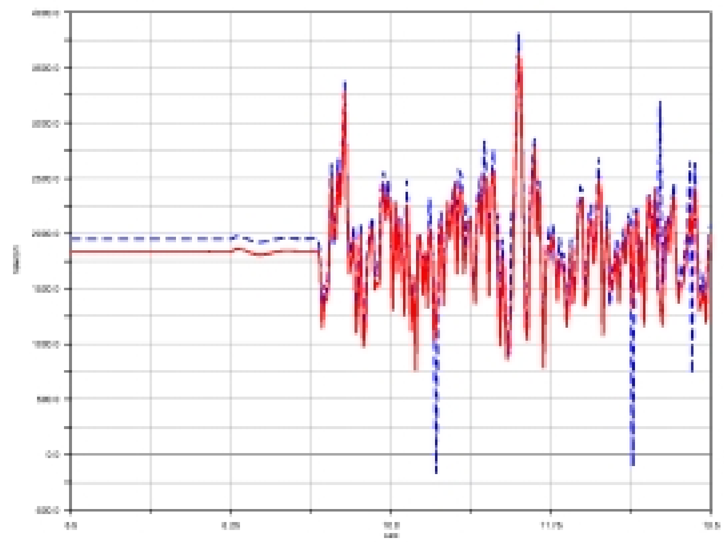

Figure 7. Force developed on the front left wheel (straight path).

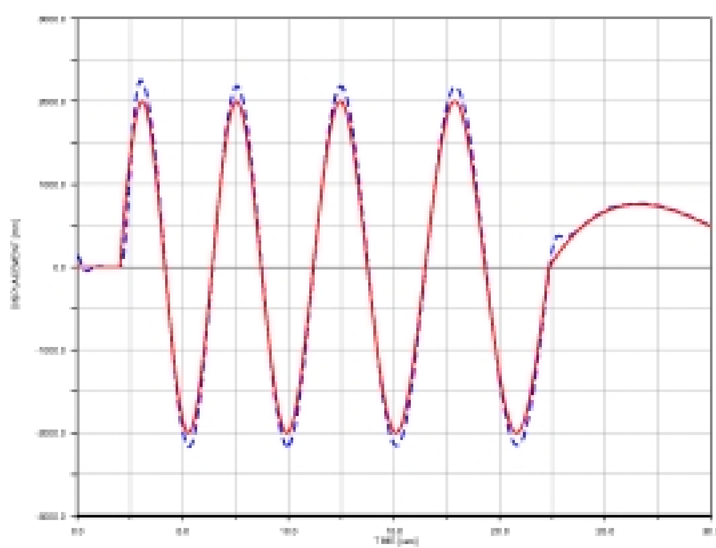

Figure 8. Desired/ Followed vehicle trajectory.

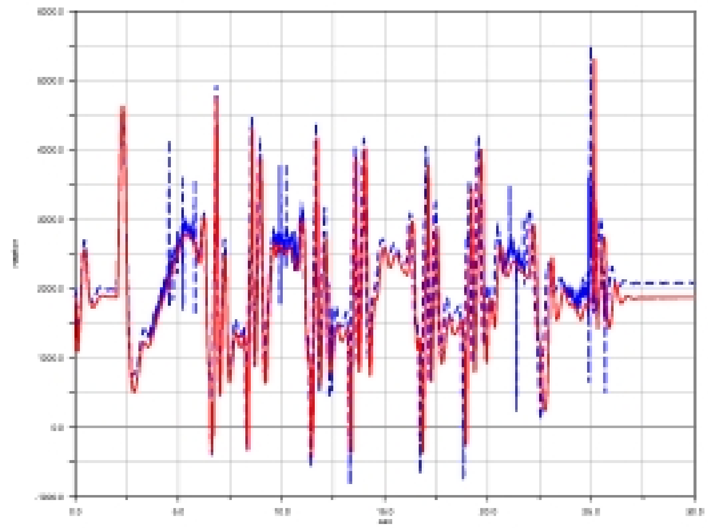

Figure 9. Force developed on the front left wheel (curved path).