Studying the Vehicle Response to Various Inputs by Virtual Prototyping.

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1. Abstract

In this paper the influence of interaction of tire with road surface on the car driving subsystem with torsional flexibility is studied. The virtual model of the car is constructed from following subsystems: car body, suspensions, torsional flexible drive line and steering line.
The influence of dynamic loads on the drive line and the overall behaviour of the car moving with diverse velocities and diverse gear ratios is examined.
The usage of the TIRE statement incorporated in ADAMS causes difficulties during the solving of interaction of tire with non-smooth road surface. A new model of contact of tire with road unevenness was designed, in which geometry and GFORCEs are constructed for every element of a road by a set of macros in ADAMS/View.
The properties of new tire model are compared with the Fiala model on standard roads to determine its functionality during simulations.
Simulation with the new designed tire model runs faster and more stable in comparison with to the use of TIRE statement.

2. Introduction

This paper illustrates a particular solution to investigate a middle weight of-road vehicle. There are discussed modelling aspects, properties and behaviour of the vehicle with focus on the drive-line, when the vehicle passes an unevenness such as barrier, bump or waved unevenness.

The modelling and simulation process in the specific environment of technical universities requires more than just a utilisation of standard Vehicle toolbox.
From a methodical point of view, to be an analysis expert in the future, the student should be familiar with the properties and role of each component of the virtual prototype.
A further objective is that the model must enable the designer to prove required modifications. This was the reason for developing macros for automatic creation of every subsystem of a car. Another objective was to improve the behaviour of the simulation process with the use of tire statement.

3. The subsystems of the Model

The model is built from following subsystems:

1) The chassis frame
2) The suspension with stabilisers
3) The axles
4) The engine with gearbox
5) The steering
6) Simple driver

Every subsystem is built with the help of a designed macro. It is possible to build several versions of the car. In the beginning of the modelling process, user simply describes the key geometric points in space and the model is then built up based on these points. By changing the key geometric points, it is possible to prove several geometric designs of suspension, steering or drive line.

The interactions of described subsystems are shown in Figure 1: Scheme diagram.

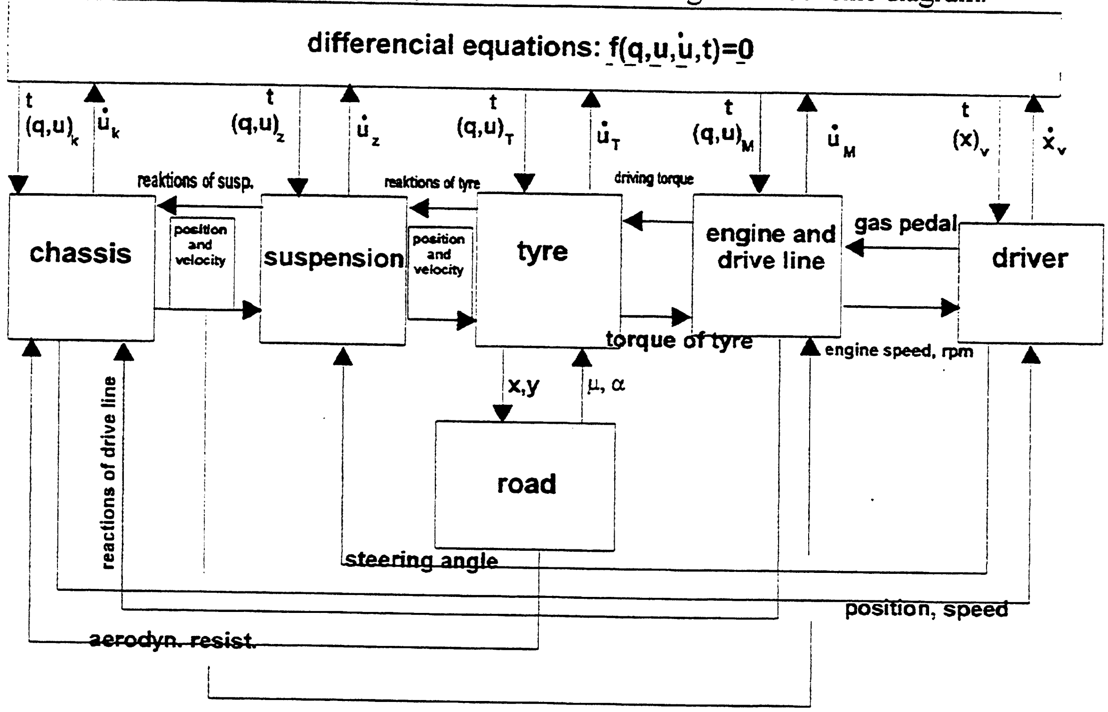


Figure 1: Scheme diagram

3.1. *The chassis frame*

The chassis is represented by a single rigid part, with mass properties calculated from FEM program. After the dynamic analysis solved in ADAMS, the reaction forces were passed to a FEM program for steady-static stress calculation of the chassis.

3.2. *The suspension*

The suspension is modelled as a classic four bar suspension. Connections to the chassis are modelled flexible with ADAMS statement bushing. Other connections are modelled rigid.

The spring-damper element is modelled with non-linear functions for the spring and damper with ADAMS statement SFORCE and functions AKISPL.

The stabilisers for the front and rear axle are geometrically similar, the difference is only in dimensions and stiffness of the torque shaft. The stabiliser is built from a rod connected to lower suspension arm and a flexible shaft connecting the two rods. The shaft is connected to the chassis with two bushings.

3.3. *The axles*

The drive line is modelled with torsional flexible axles. The axles are connected with each other and with chassis by kinematical joints. The torsional flexibility is achieved with the help of SPDP statements. Each axle is discretized in two parts which are connected with a revolute joint and a spring-damper. The differentials are represented with COUPLER statements, because in our experiments there will not be a significant difference between the rotation of axles.

3.4. *The engine with gearbox*

The engine is represented by its fly-wheel and the curve of maximal torque vs. speed. To the fly-wheel is attached a Coulomb friction clutch with non-linear springs. The clutch also works as a dynamical damper.

Two types of gear box are used in the model.

The first is built with the GEAR statement. The shafts are rigidly connected to the chassis. This model was used for first solutions because its more simple to keep the solution stable.

The second model is built with use of GFORCE statements. The model incorporates the parametric stiffness of the teethes couple of gears, the impact angle and the flexible connection of shafts to the chassis. This model is more complex, but also more unstable. Because of its high active frequencies, it also needs more free space for storing the result data. This model was used only for analyses based on results from a model with a simple gear box. Also the computed times were significantly smaller.

The model of the drive line is figured on Figure 2: The drive line.

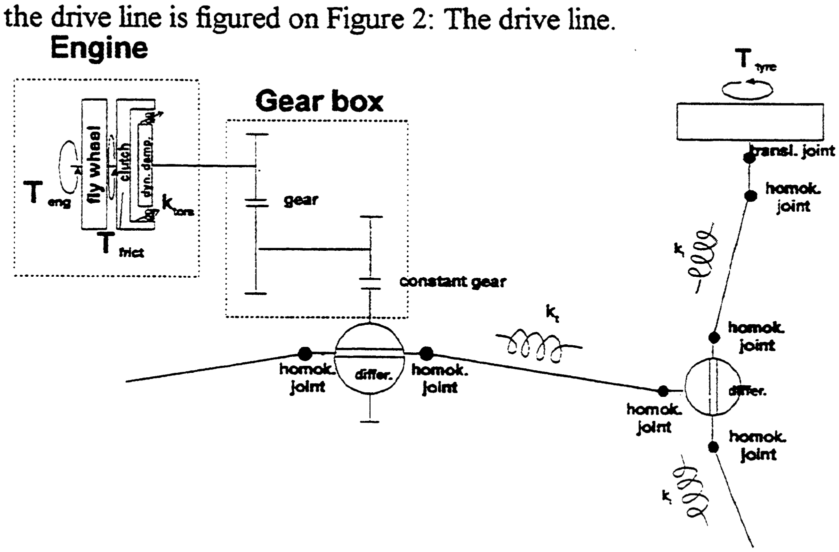


Figure 2: The drive line

3.5. *The steering line*

All parts of the steering line are assumed rigid. The parts are connected with kinematic joints. The non-constant gear ratio is realised with the use of a user function and motion. The steering line is used to control the driving direction of the car.

3.6. *Simple driver*

During the simulations it is needed to catch the behaviour of a human driver. For example a driver holds constant velocity while driving over an obstacle or depresses the clutch. In curves, a driver uses the steering-wheel to drive the curve with a certain velocity or intends to come round an obstacle.

These actions were modelled with user functions which were in fact proportional integrators. It was necessary that the driver holds a certain velocity, accelerates or brakes in a straight line through unevenness, or passes a curve while maintaining velocity.

Inputs were the speed of the engine fly-wheel, speed of the car, direction of the car, required fly-wheel speed during the time and the needed direction of the car. Outputs were position of the gas pedal, position of the clutch pedal, braking torque, steering torque.

3.7. *Overall information*

Degree of freedom		: 71 DOF
Number of parts		: 62 PARTS
Number of connections :	cylindrical	: 4 CYL
	rotational	: 23 REV
	spherical	: 17 SPH
	translational	: 5 TRA
	universal	: 9 UNI
	homokinematik.	: 12 CONVEL
	in line	: 1 INLINE
	couplers	: 3 COUPLER
	gears	: 2 GEAR
Number of motions		: 1 MOT
Number of bushings		: 20 BUSHING (suspension and stabilisers)
Number of linear springs		: 10 SPDP (torsional and axial stiffness of stabilisers)
Number of single forces		: 15 SFORCE (suspension, dampers, bumpers)
		: 1 VFORCE (aerodynamic resistance)
		: 2 GFORCE (gears)
Number of TIRE statements		: 4 FIALA
or with use of designed tire for every unevenness 4 GFORCE.		
User functions		: 2 Variables
Gravitational acceleration		: 1 ACCGRAV

The car is a middle-weight of-road vehicle. From its specification it was necessary to model driving over non-smooth unevenness such as barrier, cyclic unevenness, driving over a bump.

4. Modelling the contact of tire with road surface

The road surface in ADAMS is modeled with triangular elements, where every element is defined by the three co-ordinates of the corner points and two constants describing the friction. The program then determines with which element the tire is in contact and computes general forces for the tire.
Using this method we had negative experience in modeling a strongly non-smooth surface. As an example we can show:

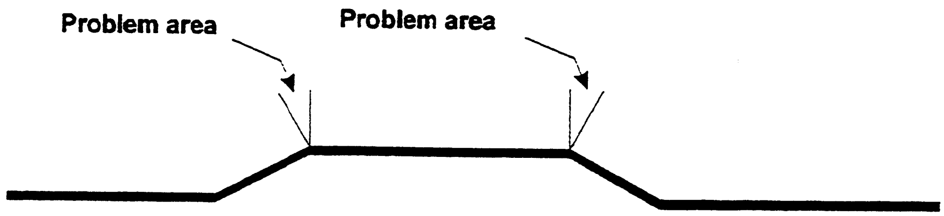


Figure 3: Road surface

Convex angles cause most problems. The program has problems determining with which element the tire is in contact. If the position of the tire is at the edge of the element, the program produces a discontinuity in the solution of the forces. The discontinuity is in most of the cases significant. This behaviour also decreases stability of solution and scales up the computational time.
This was the reason we designed our tire model. The tire model represents the Fiala model implemented to unevenness of types: surface and edge.
The functions are defined within the ADAMS/View using user functions defined with standard ADAMS functions such as IMPACT and STEP.
The function computes with which elements the tire is in contact. The function then determines the deflection and speed of tire deflection. From given constants of friction, lateral, longitudinal and normal stiffness, damping and rolling resistance radius the function computes the appropriate forces. The force and torque determination are shown on the following picture.

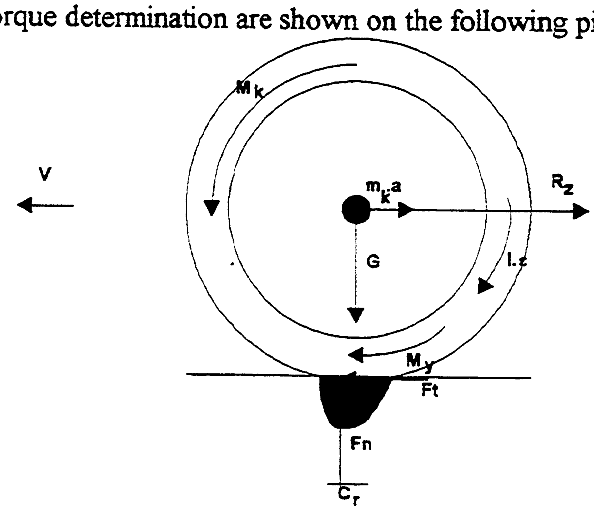


Figure 4: Forces and Torques determination

The advantages of the designed model of tire are: more stable solution during the driving over an unevenness. Stiffness can be non-linear, possibility of observing separately individual forces of the contact, the ground can be moved during the solution, possibility of parametrizing the tire statement.
The disadvantages are: for every element the model uses approximately 4 VARIABLE statements 2 SFORCE and 1 GFORCE statements; the model is not as complex as for example UATIRE.

The problem of driving over an unevenness strongly depends on tire impact angle.

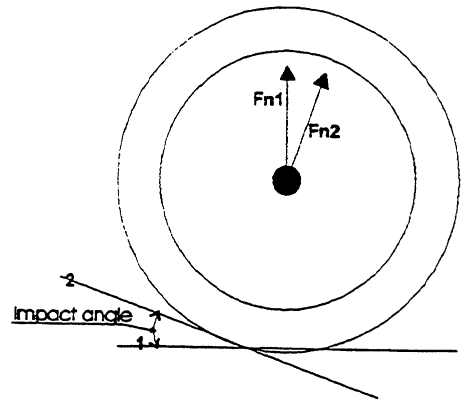


Figure 5: Impact angle

The problem is strongly non-linear and parametric as is shown in the following example:

4.1. Modal analysis of simple suspension

To demonstrate the parametrical behaviour of the problem, modal analyses of a single suspension (with axle and stabiliser) with different impact angles from 0 to 90 degrees were realized. This particular model has 13 DOF.

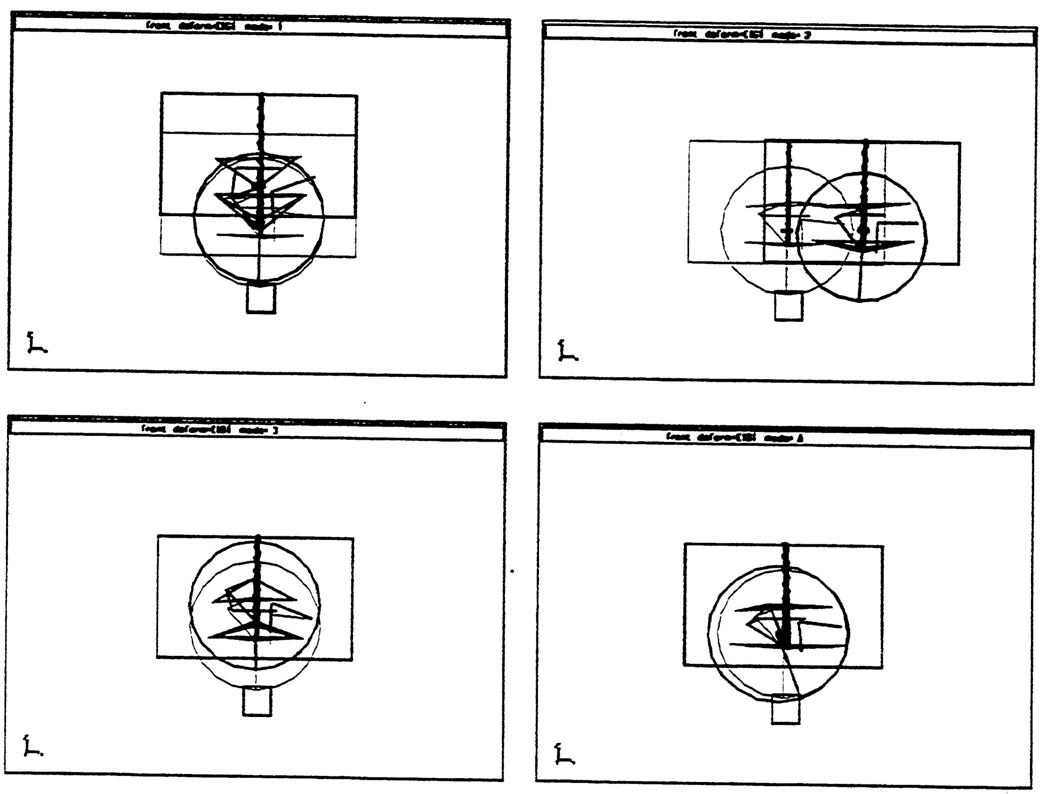
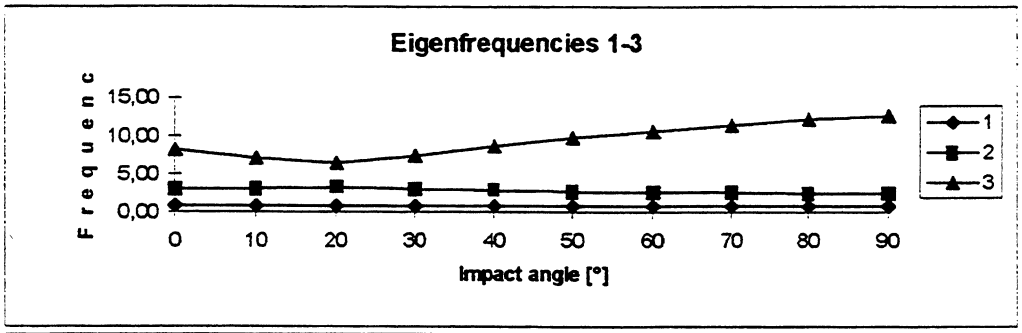


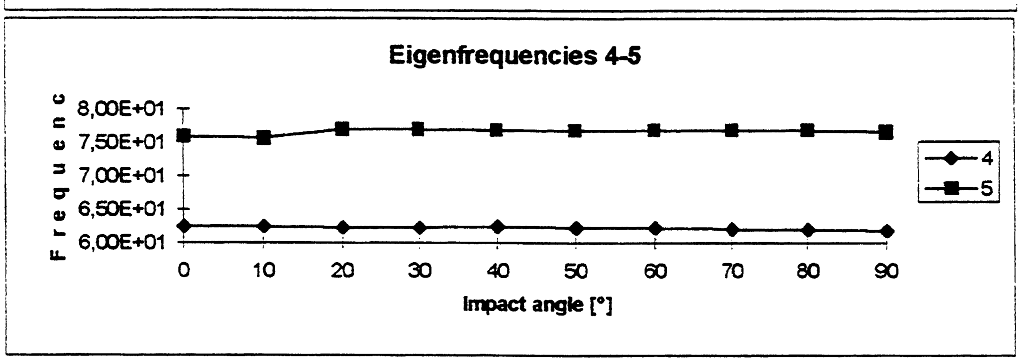
Figure 6: Eigenvectors of suspension

The first eigenvector corresponds to the oscillation of the chassis with respect to the reference frame. The second corresponds to the longitudal oscillation of the whole system including the rotation of the tire with the axle. This eigenvector is influenced by the longitudinal stiffness of the tire. The third eigenvector corresponds to the oscillation of the suspension relative to the chassis. The fifth eigenvector corresponds to the oscillation of the tire on the axle.

Mentioned eigenvectors changes depending on the impact angle:

		Impact angle (in degrees)									
		0	10	20	30	40	50	60	70	80	90
	1	0,90	0,84	0,82	0,74	0,81	0,81	0,80	0,80	0,82	0,84
Eigen-	2	3,01	3,16	3,23	3,00	2,88	2,78	2,69	2,62	2,56	2,52
vector	3	8,10	7,09	6,44	7,40	8,71	9,73	10,64	11,46	12,18	12,61
	4	62,38	62,42	62,29	62,35	62,36	62,33	62,23	62,07	61,91	61,81
	5	75,95	75,74	76,99	76,94	76,85	76,83	76,83	76,84	76,80	76,72





From the charts results that the natural frequencies are changing depending on the impact angle. The changing mass distribution and dominant stiffness cause this change. This is behavior of a parametric non-linear system.

5. Simulation computations

5.1. Simulation to validate the designed tire model

To compare the use of new designed tire model and standard model used in ADAMS we realized two simulations with the same conditions. The modeled unevenness represented a bump 8cm high.

In simulation using the standard tire statement the obstacle is represented by a small surface with impact angle modified so, that tire hits the obstacle in normal direction to the surface.

In simulation using the designed tire the obstacle is represented by an edge. The speed of the vehicle is set to 10 m/s. The driver is set up to produce small changes on gas-pedal depending on the speed of the vehicle. After the impact the torque of the engine decreases because of the front axle having no contact with road (chart a). The first increase of the torque in the moment of impact is also visible. Interesting is also the chart of speed of the vehicle (chart b), where can be seen the decrease of speed. We can also observe the deflection of the front driving axle on chart c. On charts d and e are shown normal forces of tire modeled with standard Tire statement (chart d) and with the designed tire(chart e). It can be seen that the forces are more smooth with the designed model and also the magnitudes are smaller. The excitation of the whole driving system is smaller and the solution is more stable. In areas around the impact both models produce nearly the same results.

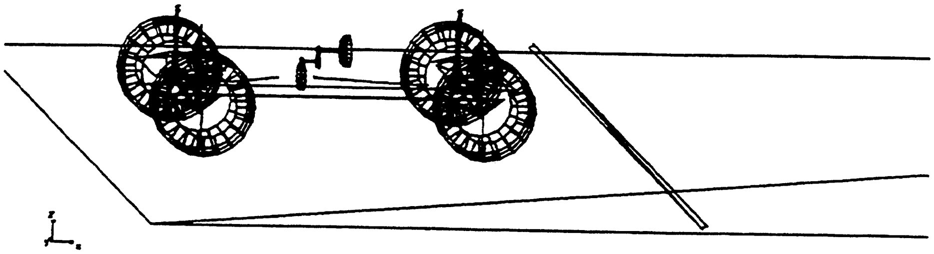
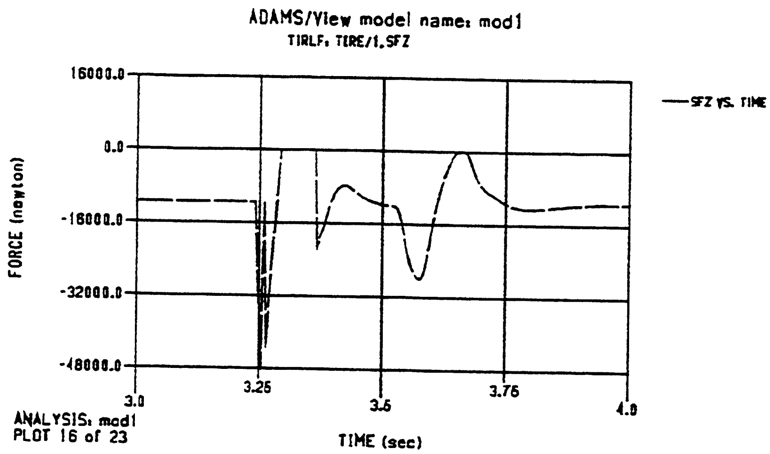


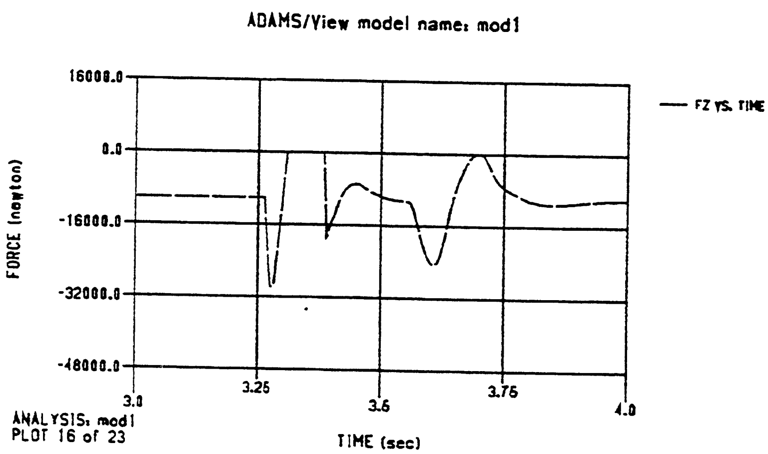
Figure 10: Driving over a bump unevenness

Comparison of the tire models

driving over an unevenness with stand. TIRE

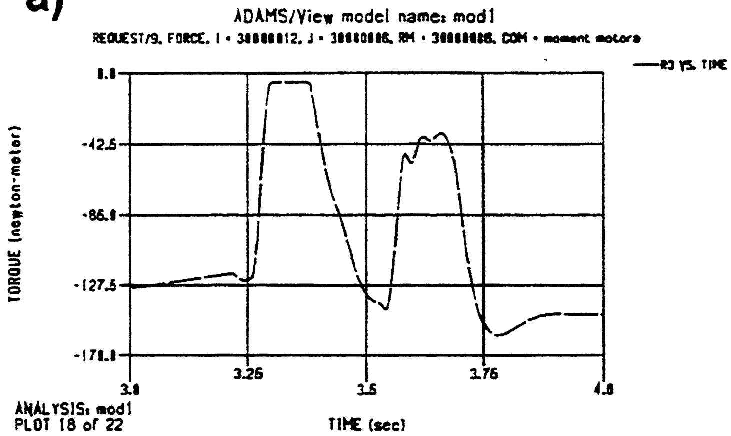


driving over an unevenness with design. tire

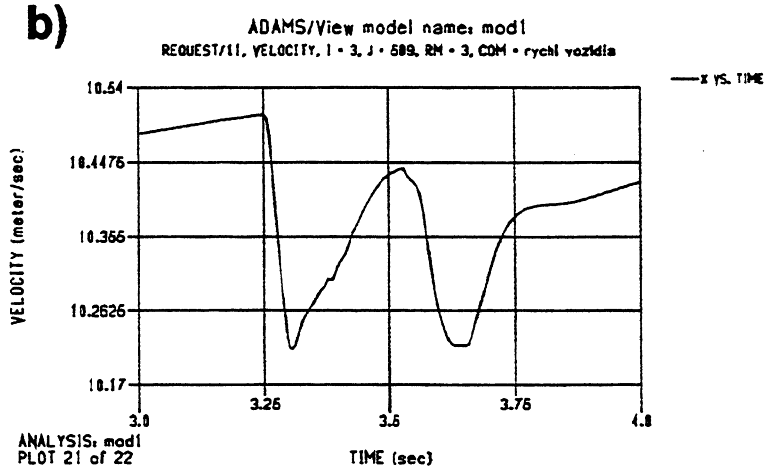


driving over the unevenness with designed statement TIRE

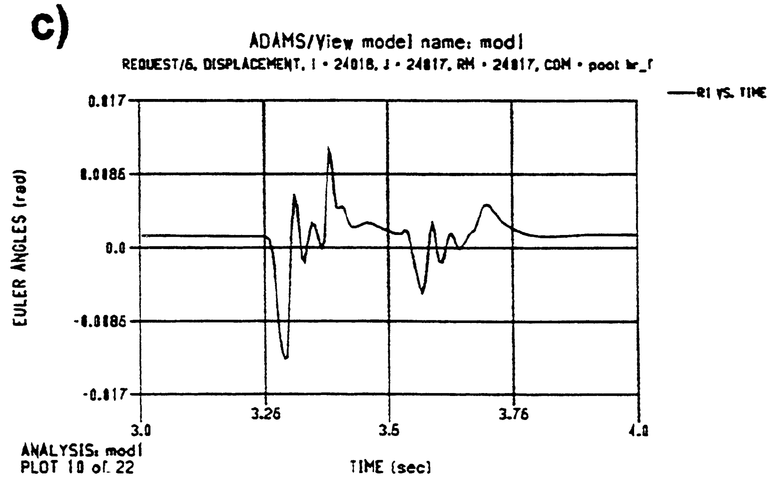
a)



b)



c)



2.2. *Simulation of maneuver driving over an waved unevenness*

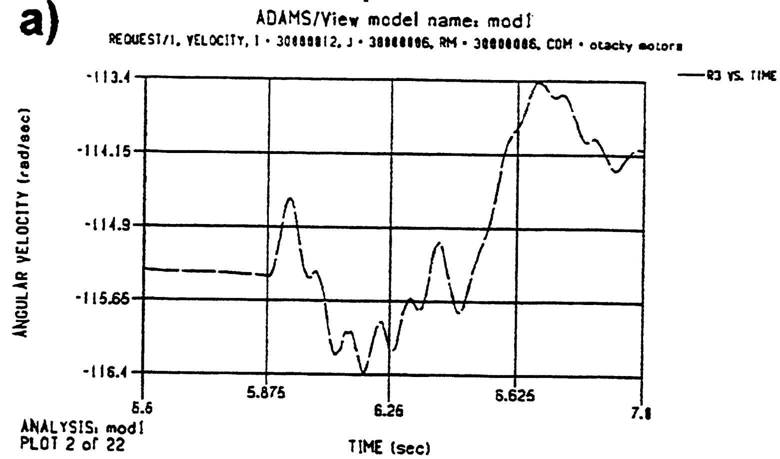
This unevenness excites the driving system approximately harmonically with a frequency depending of the of the vehicle.
The waves are 5cm deep and 50cm long, the total length of the obstacle is 2m.
The simulation is realized with speeds: 2.7 m/s, 5.6 m/s and 10.5 m/s. The driver is set up to hold the velocity of the car with a small change on the gas pedal.
The mean of this simulation is to find the speed of the biggest excitation of the suspension.
Depending of the speed are the tire still in contact with the road or are only hitting the tops of waves, this can be seen on d) chart. This behavior leads to reduce of the speed (chart b), which further strains the drive line (chart c, e).
The angular speed of the fly-wheel is displayed on chart a), the forces on tires are displayed on chart d).
Driving with smaller speed reduces the speed of the car more, with 2.7 m/s is the decrease of speed 12%. The oscillation of the torque on the driving line is about 15% with increase of the average torque by 30%. The normal force on the tires increases by 18%. It is obvious that the dynamic loads on the drive line are bigger than on the suspension.
With increasing speed are the dynamic loads amplified rapidly on the suspension. Because the tires are hitting only the tops of waves (with 10 m/s the normal force on tire oscillates between 0 and 190% of the static load) the average torque decreases by 18%.
On chart f) is displayed the Fast Fourier transformation of the longitudinal force of tire.
From the comparison to solutions with other velocities, results that the biggest amplitudes achieves the car with speed of 5.6 m/s. The frequency is 0.9 1/sec which is near to the first natural frequency.

3. Conclusions

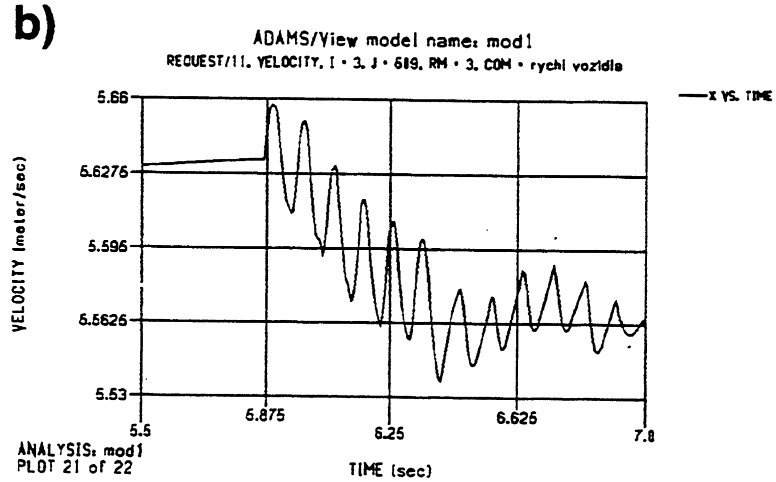
A new tire model was designed and compared with results of model with tire statement used. The models are equivalent in areas of smooth road surfaces. The new model produces a more smooth curve of response to non-smooth inputs.
A simulation of a full vehicle model was realized with focus on drive-line. The results were proved with help of eigenvalues.

WAVES, speed 20 km/hod

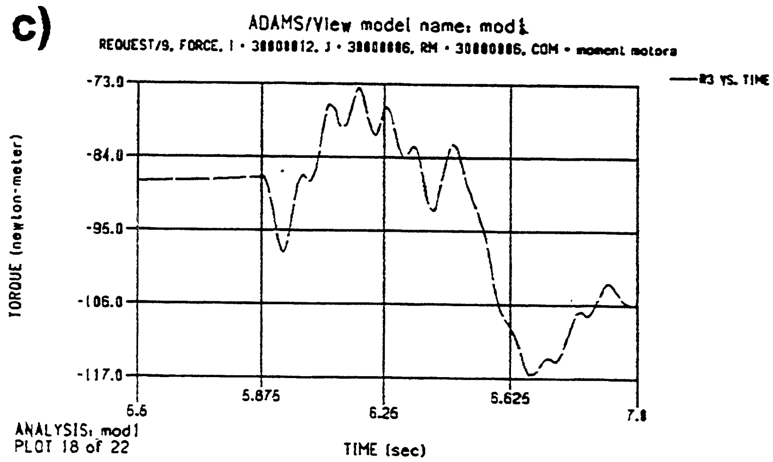
a)



b)

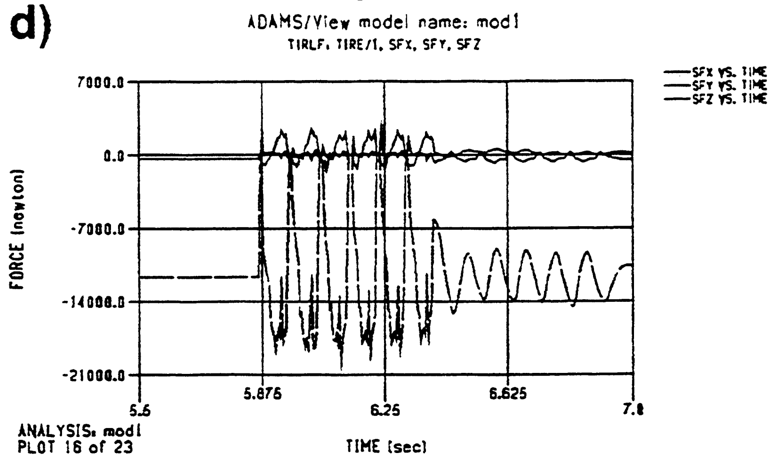


c)

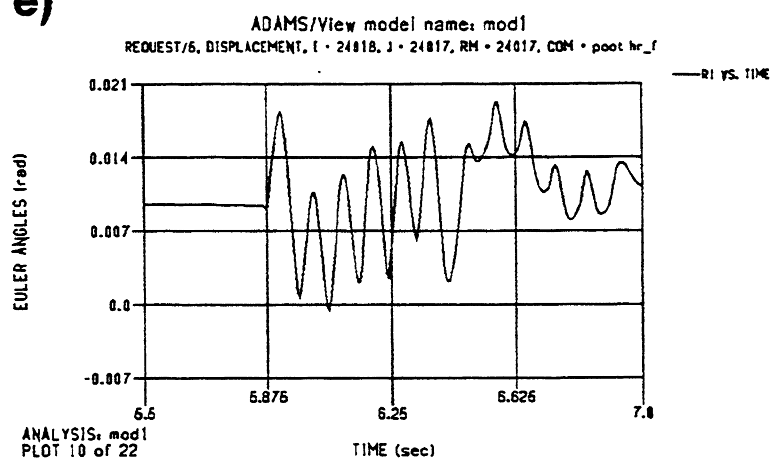


WAVES, speed 20 km/hod

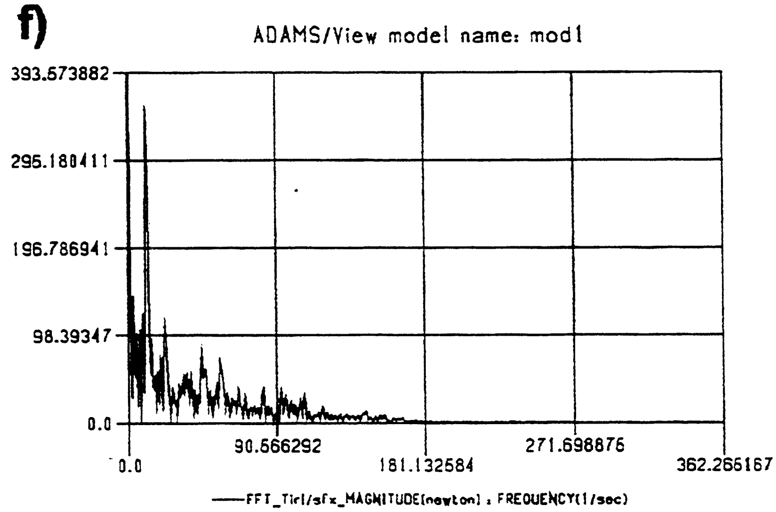
d)



e)



f)



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