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Three dimensional dynamic model for a quick simulation of vehicle collisions

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ABSTRACT

The aim of this work is to develop a simulation tool able to foresee the dynamic behaviour of a vehicle involved in an emergency manoeuvre. This tool is suitable to reproduce the real dynamic of an accident and so could be exploited both for road safety analyses and both for accident reconstruction.
In order to perform the dynamic analysis we have developed a detailed three dimensional handling model able to estimate interactions between tyres and road; furthermore an interaction model is required to simulate the interaction between vehicles and/ or between vehicle and ground, taking into account high speed plastic strain. The latter interaction was evaluated according a three dimensional simplified model, called “collision points model”, based on the assumption that bounding surface of the car, meshed with a polyhedron, is connected to the rigid body representing the car, by means of a damper and a spring in series. After contact detection, interaction loads are computed imposing equilibrium and compatibility conditions.
Overall interaction loads together with external forces due to weight are used to integrate equations of motion for each rigid body on a explicit time step basis.
Correctness of handling model was checked simulating a standard test and obtaining a very good agreement between numerical and experimental results. Interaction capability were widely tested starting with simple applications consisting in a bouncing sphere and a spinning top. Capability to reproduce accident dynamic was also tested on the basis of measured data.

INTRODUCTION

Vehicles collision simulation can be addressed to accident reconstruction or to road safety design. In both these fields are required a model to simulate the dynamic of vehicles, i.e. a handling model, and a tool able to estimate the behavior of the interaction between vehicles and between vehicle and ground. At present the investigation of crash dynamic is conducted buy means of multi-body analysis with impulse interaction between vehicles or FEM methods.

A wide range of commercial codes are devoted to the numerical simulation of rapid transient events. Pamcrash, Dyna3D, Dytran exploit explicit FEM analysis to evaluate the dynamic behavior of complex structures subjected to large strain and strain rate with multiple surface contact. An accurate crash reconstruction is achieved, although a great computational effort is required. For this reason such powerful technology is difficultly applicable for the simulation of the complete accident (typically 10 seconds) while is the standard tool for crash structural design. The use of a three dimensional handling model, comprehensive of the tire-ground interaction, joined to an high speed plastic strain simulator (based on the model of the “collision point”) permits to obtain a good reconstruction of the collision dynamic in a quick time.

MATHEMATICAL MODEL

INTERACTION BETWEEN TIRES AND ROAD

The interactions between the vehicle and the ground, due to the tires forces, are modeled using longitudinal forces and lateral forces, acting independently on each tire. The forces can be calculated according to the following expressions:

$$X = -fZ$$

$$X_f = -m_x Z$$

$$Y = -C\alpha\left[1+\left(\frac{3Y_{\max}}{C\alpha_{\max}}-2\right)\cdot\frac{|\alpha|}{\alpha_{\max}}-\left(\frac{2Y_{\max}}{C\alpha_{\max}}-1\right)\cdot\frac{\alpha^2}{\alpha_{\max}^2}\right]$$

if $0 \le |\alpha| \le \alpha_{\max}$

$$Y = -Y_{\max}\cdot\frac{\alpha}{|\alpha|} \qquad (1)$$

if $|\alpha_i| \ge \alpha_{\max}$

where X is the longitudinal force, f is the friction coefficient during non-braked motion, X_f is the longitudinal force during a braking maneuver, Y is the lateral force, α is the slip angle, and Y_{max} is the maximum lateral force corresponding to α_{max} angle (the empiric formulation is due to [1]). It is also modeled the interaction between longitudinal forces and lateral forces. Defining X_0 and Y_0 as the maximum values for longitudinal and lateral forces, X and Y must agree with the condition

$$\left(\frac{X}{X_0}\right)^2 + \left(\frac{Y}{Y_0}\right)^2 = 1$$

INTERACTION BETWEEN VEHICLES AND BETWEEN VEHICLE AND GROUND

The model used to simulate the interaction between vehicles and vehicle and ground is called “collision point” model [2]. The model consists in a system of a spring and a damper, connected in series. This is a simple schematization to take into account the relation existing between load and strain rate during dynamic plastic deformation. Elastic behavior is represented by the spring while yielding limit is modeled introducing a threshold value to activate the damper. In figure 1 a typical deceleration curve for a mass connected to the spring damper series.

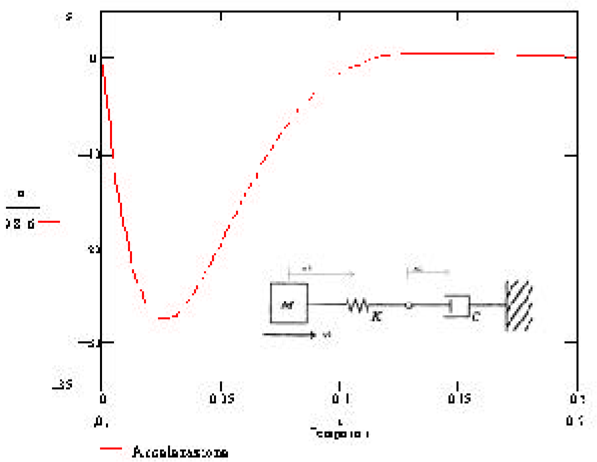

Figure 1. Normalized acceleration of a spring damper mass in series model.

In this study the surface of vehicle and ground are meshed with triangular (three nodes) elements. Using three-system spring-damper in series (one for each direction x, y, z) for each node of the mesh is possible to simulate the high-speed plastic strain during the contact. It is assumed that the mass is always concentrated in the center of gravity of the vehicle i.e. that the dynamic behavior of the vehicle is characterized by a rigid body and the inertial properties remain unaffected by the motion of the surfaces during the strain. The values of model constants are identified using acceleration measured in real car crash tests [2]. In order to simulate the interaction between tires and ground a simplified system of spring and damper in parallel is used. In only two constant (the spring stiffness and the damping value) are summarized the characteristic of both tire and suspension. To simulate the contact between two vehicles the following procedure is given. For hypothesis the contact happens between a triangle (of the meshed surface) and a node of the other surface. First is determined the offset, i.e. the value of the penetration of the node into the triangle. After the stiffness of the triangle in the contact point is evaluated and used into an equivalent two collision point model (fig.2).

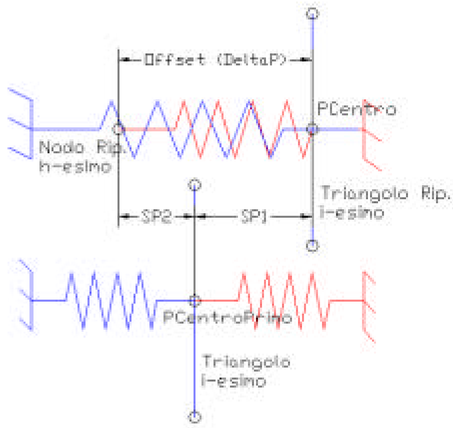

Figure 2. Equivalent model to simulate the node-triangle contact.

Using the equilibrium and congruence equations it is possible to determine the forces and the displacements of the nodes

$$F_n = -K_{1,EQ} \cdot Sp_1 = -K_2 \cdot Sp_2$$

$$Sp_1 + Sp_2 = \Delta P$$

After is possible to evaluate the plastic strain and the new configuration for the mesh nodes.

EQUATIONS OF MOTION

Once the forces of interaction during the contact are known, is possible to determine the motion of the vehicles. In this work the equations are written in the inertial reference system, and the forces defined into local reference system, such as tire forces, are evaluated using appropriate rotation matrix.

$$\frac{d(M\underline{\underline{v}})}{dt} = \sum \underline{F}$$

Where M is the mass and F_x, F_y, F_z are the forces applied. Furthermore defining

$$\underline{H} = I\underline{w}$$

where *I* is the inertial tensor and $\underline{w}$ the rotation vector, the moments equations assume the simple form:

$$\frac{d\underline{H}}{dt} = \sum \underline{M}$$

where ΣM is the sum of the moment of the forces applied.

NUMERICAL IMPLEMENTATION

The numeric integration was carried out with a constant time step scheme. The equations are written in an explicit form as follows:

$$[M]\ddot{X} = f(X, \dot{X}, \sum F)$$

where ΣF is the sum of the forces on the vehicle. Using the initial condition X_0 ,X_1 it is possible to perform the first step:

$$X_1 = X_0 + \dot{X}_1 \cdot \Delta t\,.$$

During the standard calculation cycle the external forces are evaluated at old values:

$$\sum F = f(X_1, \dot{X}_1)$$

A first order approximation for velocity can be calculated:

$$\dot{X}_2 = \dot{X}_1 + \Delta t \cdot \left([M]^{-1} f(X_1, \dot{X}_1, \Sigma F)\right)$$

So a first order approximation for position can be calculated:

$$X_2 = X_1 + \Delta t \cdot \dot{X}_2$$

that is the output required. Variable swapping is then imposed for the next step:

$$\dot{X}_1 = \dot{X}_2$$

$$X_1 = X_2$$

VALIDATIONS

VALIDATION OF THE RIGID BODY MOTION EQUATION

In order to verify the accuracy of the solver of the rigid body motion equation, a spinning top frictionless motion has been simulated. The spinning top, leaned to the ground on a single damperless collision point, has been set to move with a constant angular speed and two different values of inclination angle. The output evaluated with the numeric code exposed in par. 2.3 has been compared with the output of the same model obtained with the commercial multi-body code Working Model 3D[6]. In figure 3 are shown the results. The time-histories of the coordinates (x, y, z) at the contact point obtained with proposed numeric code and Working Model 3D.

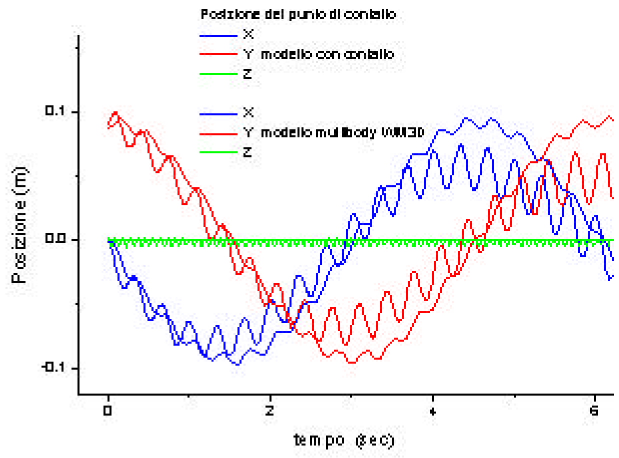

Figure 3. Comparison between spinning top simulation using numeric code and WM3D.

Friction effects and compliance of proposed model produce a quite different evolution, although the first transient shows a good agreement. Interaction capability are further highlighted by this application, actually, after some revolutions the spinning top loses stability and falls on the ground; we can observe an inversion in angular speed and a rolling of the cone.

VALIDATION OF THE ELASTIC SURFACE INTERACTION MODEL

This validation test consists in a simulation of the motion of an elastic bouncing ball. The ball, while is moving bouncing on a horizontal ground, bumps against a vertical wall along a direction that forms an angle of 45 degree with the normal of wall surface.

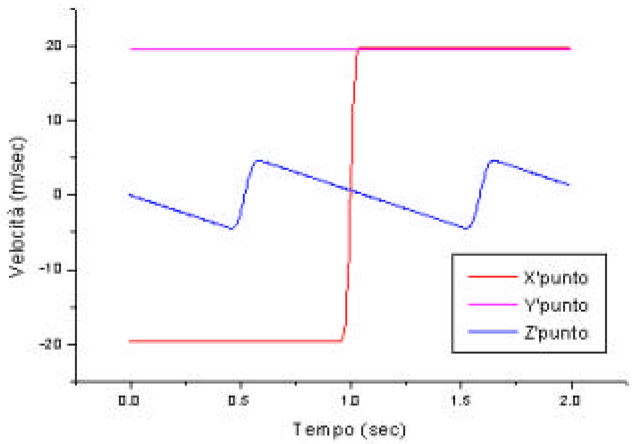

Figure 4. Time-history of speed of center of gravity of a bouncing ball.

The figure 4 and 5 show how speed and height of the center of gravity, and direction angle time-histories, agree with the hypothesis of conservation energy and momentum, valid in presence of pure elastic strain.

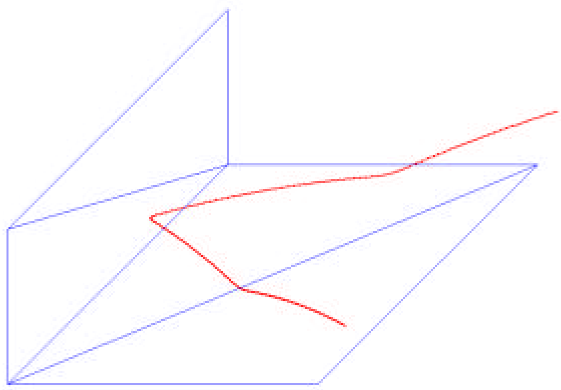

Figure 5. 3D view of the trajectory of the center of gravity of a bouncing ball.

VALIDATION OF ELASTIC-PLASTIC STRAIN MODEL

The validation of the elastic-plastic strain model was performed comparing the closed form solution of the mass damper spring in series model with the results of the simulation of a crash between a pyramidal deformable body and a rigid wall (fig.6).

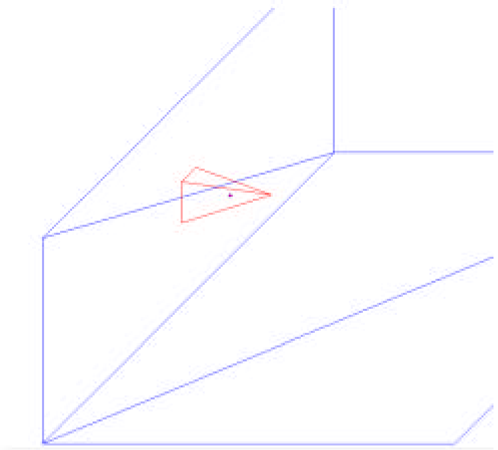

Figure 6. Representation of the crash simulation used to validate elastic-plastic model.

In figure 7 and 8 are show the comparison between the closed form solution and the model output in terms of mass acceleration and strain speed, showing the right implementation of the model.

VALIDATION OF THE 3D HANDLING MODEL

In order to perform a fine simulation of vehicle motion, a handling model has been developed. The model features a six DOF car body, four independent single DOF suspension and tire models, following equations described in paragraph 2.1. To validate the model, a typical maneuver [3] of double change of line has been simulated.

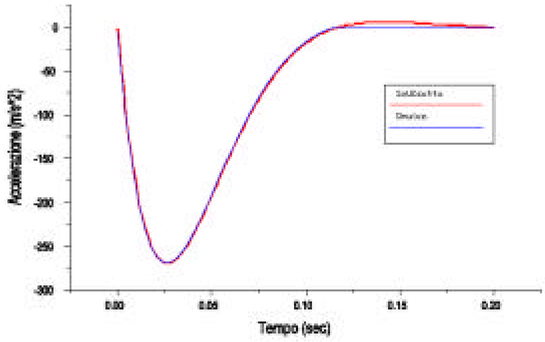

Figure 7. Mass acceleration comparison between well-known solution and model output.

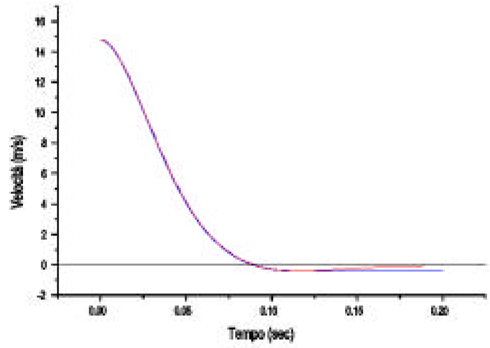

Figure 8. Strain speed comparison between well-known solution and model output.

The output obtained has been compared with the measures acquired during a on-road car test executed by CRF[1] on Alfa Romeo 156. In figure 9, 10, 11 and 12 are shown the comparisons between road test measures and simulation output in terms of yaw speed, rolling speed, lateral acceleration and attitude angle.

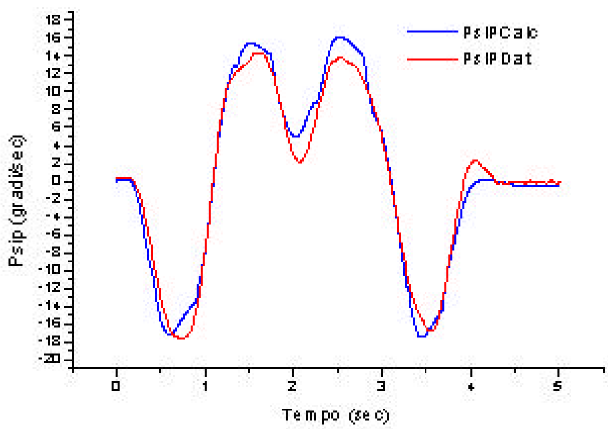

Figure 9. Yaw speed time-histories.

[1] Fiat Research Center, Orbassano (TO).

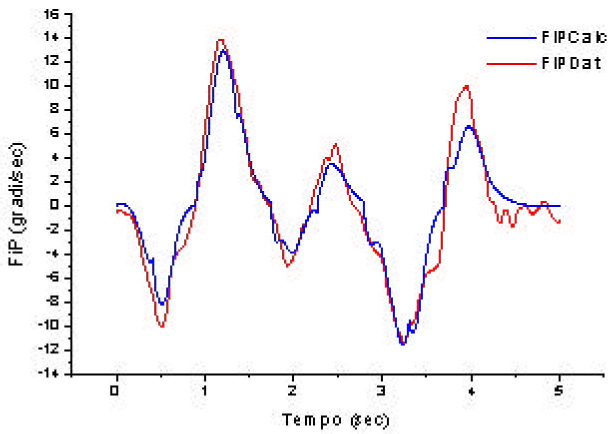

Figure 10. Rolling speed time-histories.

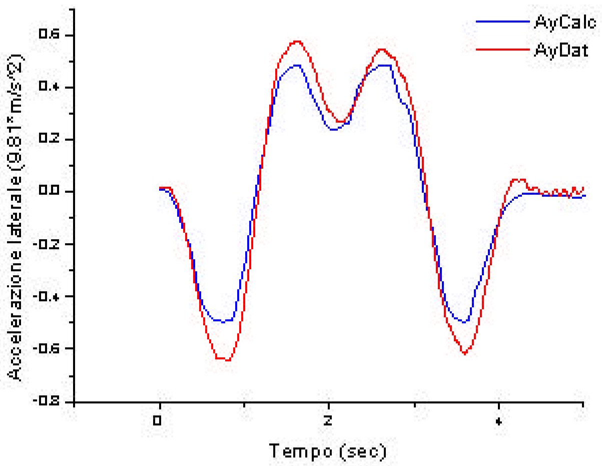

Figure 11. Lateral acceleration time-histories.

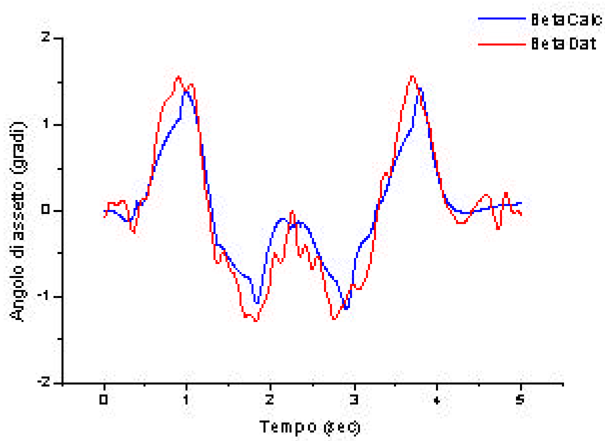

Figure 12. Attitude angle time-histories.

The figures show that the simulation model has a good agreement with all the road test measures.

THE ACCIDENT SIMULATION

After the validation of all the single effects, proposed model was run to simulate an accident between two vehicles, and between a vehicle and an obstacle, after a generic road maneuver. First, the constants for the collision point model was chosen to met experimental deceleration data measured in a crash against fixed barrier according to the procedure described in [2]. The values calculated (stiffness and damping) must be calibrated for the surface mesh used, i.e. to the density of collision point per unit of volume. Then, surface data for vehicles and ground was prepared by an external FEM preprocessor FEMAP 7.0 and written in ASCII format; further, geometrical information regarding car and tires are written in the input file, together with the input time histories of steering angle, throttle and brake. Various applications of the model implemented are shown in the following tests.

FULL FRONTAL BARRIER COLLISION

In this test a vehicle (mass=1065 kg, density of collision point σ=88 nodes/m^3) crash against a rigid fixed barrier while is moving with constant speed (the test is conducted with two different value of speed: 40.2 and 48.2 km/h). Stiffness and damping value for the collisions point are K=49999 N/m and C=869 Ns/m. The acceleration diagram (fig.13) agrees with experimental and calculated values taken from [2] (fig.14).

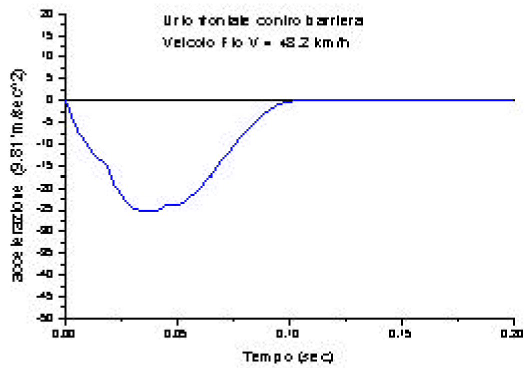

Figure 13. Acceleration during full frontal barrier collision (48.2 km/h).

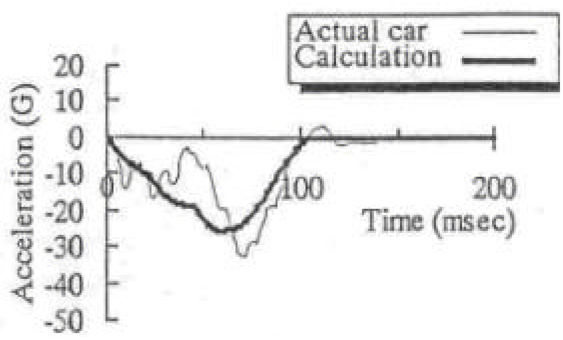

Figure 14. Acceleration during full frontal barrier collision (experimental and calculated in [2]).

Another comparison with experimental values can be made between the maximum residual strain present on the vehicle after crash. The figure 15 shows the permanent displacement for each node of the contact surface.

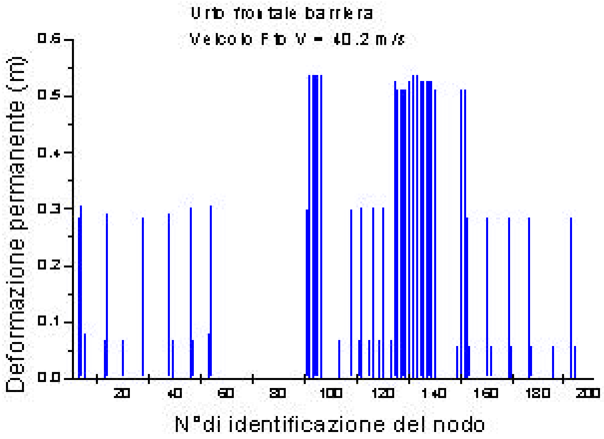

Figure 15. Permanent strain versus identifying node number.

The maximum value obtained by the simulator (0.547 m) agrees with the maximum value (0.590 m) of the experimental test. In figure 16 is shown the evolution of simulated crash, represented by a set of frame (time interval = 0.04 sec.).

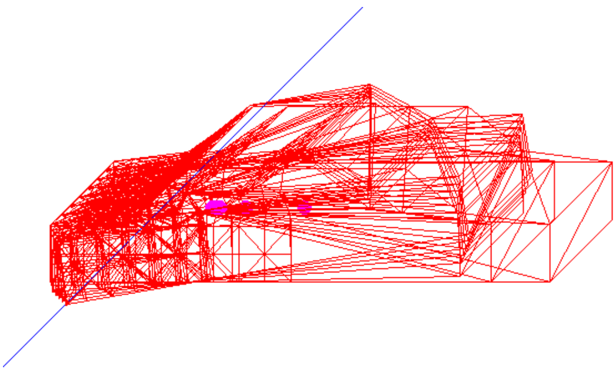

Figure 16. Simulated crash representation.

OVERLAP 50% OFFSET COLLISION

This final application consists in a simulation of two vehicles frontal crash. The first vehicle (A), in uniform rectilinear motion (speed = 94.8 km/h), collides with a standstill vehicle (B). The interaction interests the 50% of the frontal surface of the vehicle. According with the experimental data available, the model is set with the following characteristics:

Vehicle A: number of nodes = 98, mass M=505 kg, Collision Point stiffness K=108482, C.P. damping constant C=1886

Vehicle B: number of nodes = 78, mass M=820 kg, C.P. stiffness K=221316, C.P. damping constant C=3062.

Figures 17 and 18 show the agreement between acceleration diagrams (simulated and experimental).

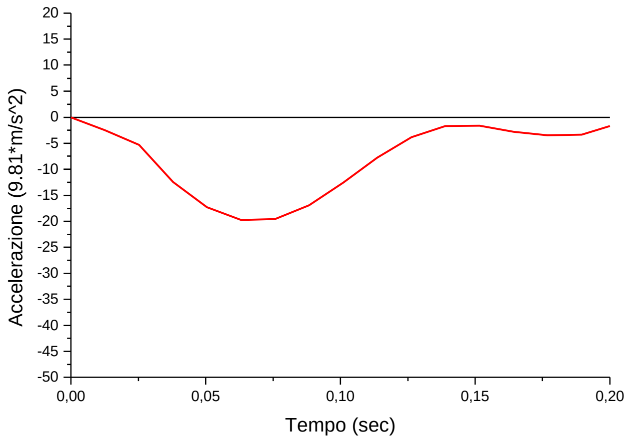

Figure 17. Acceleration in Overlap 50% offset collision (simulated).

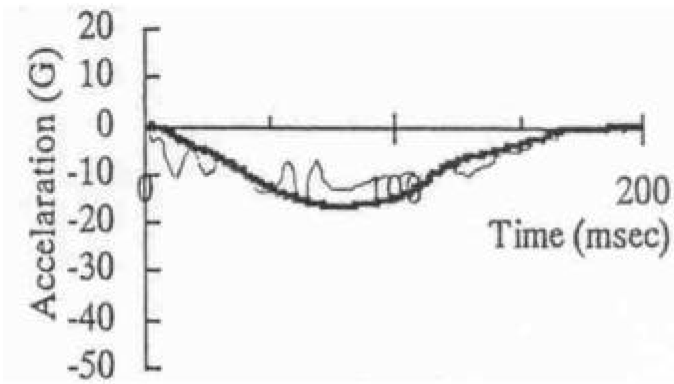

Figure 18. Acceleration in Overlap 50% offset collision (experimental and calculated in [2]).

The agreement between the values of the maximum permanent strain is also verified (experimental = 0.26m, simulated 0.27m).

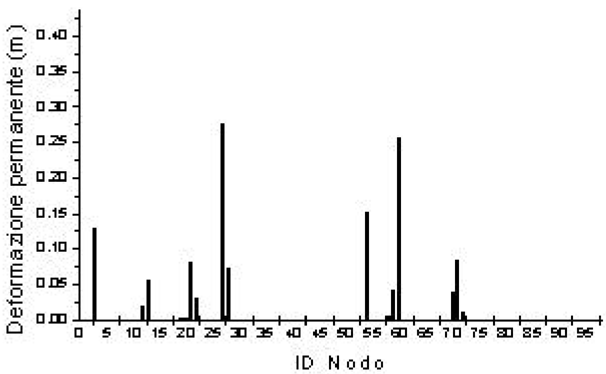

Figure 19. Permanent strain versus identifying node number.

The figures 20, 21 show the deformed configuration of vehicles A and B. Figure 22 shows a frame of the simulation during the evolution of collision. Note the sign made by the slip of the tire.

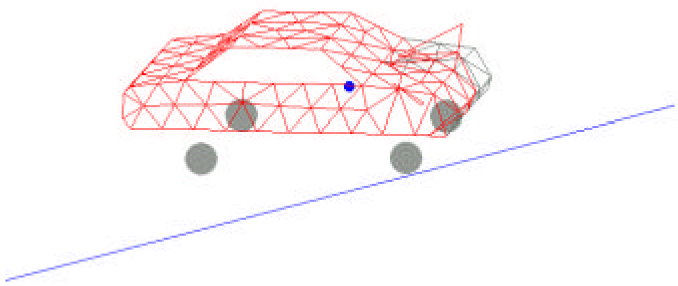

Figure 20. Deformed configuration of vehicle A.

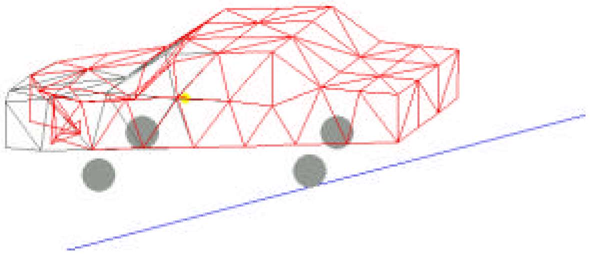

Figure 21. Deformed configuration of vehicle B.

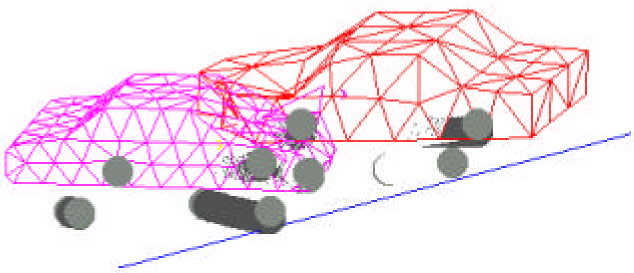

Figure 22. Overlap 50% offset collision simulation.

CONCLUSION

In this work a simulation tool was proposed for numerical simulation of complex interactions and dynamics of vehicles involved in an accident. To achieve this aim many solution modules were developed. The motion equations solver permits to study the dynamic of a generic rigid body. The explicit integration procedure has been used to successfully solve two rigid motion problems: a spinning top and a vehicle. The handling model realized is able to reproduce, with satisfactory result, the three dimensional dynamic of a four independent wheels vehicle in a change line test. Furthermore the elastic-plastic model permit to solve, with a good accuracy, a great variety of body interaction problems. All the tools, used simultaneously, permit to simulate the behavior of road accident. As far as the handling model is concerned, a generic input trajectory can be imposed to the vehicle setting steering angles, throttle and brake actions, obviously the results are affected by vehicle and tires parameters (inertial values, geometry, etc.). The latter can easily identified by means of road test measures. Furthermore the availability of real crash test data permits to determine the mesh node density, stiffness and damping constants to use in a collision point model. Proposed model, while based on a drastic simplification of the dynamic problem, after a proper parameters setting, is able to reproduce accurately the dynamic of an accident with little computation efforts. For this reason the numeric model is suitable to reconstruct accident scene with a trial and error method that requires a lot of simulations to reach convergence.

ACKNOWLEDGMENTS

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REFERENCES

[1] “Meccanica dell’autoveicolo” G.Genta, Ed. Levrotto e Bella, Torino.
[2] “Three dimensional behavior simulation of vehicle to vehicle collision by dynamic model”, M.Abe, M.Morisawa, T.B.Sato, K.Kobayashi, JSME Review 20 (1999)
[3] International Standard ISO 7401 “Road vehicles – Lateral transient response test methods”, 1995
[4] Dario Del Pin “Sviluppo di un metodo di calcolo 3d multi body elastoviscoplastico per la simulazione del moto e dell'urto di veicoli” – Thesis in Mechanical Engineering, Tor Vergata University
[5] Paolo Sassaroli “Simulazione dell’urto di veicoli in condizioni piane” Thesis in Mechanical Engineering, Tor Vergata University
[6] Working Model 3D 4.0 - User Manual