CONTRIBUTION OF A SEAT SUSPENSION TO REDUCE CRASH INJURIES WITH A MULTIBODY APPROACH

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ABSTRACT. The purpose of this study is to investigate the contribution of a seat suspension in absorbing a part of the impact energy. Standard test conditions were considered for frontal direction of crash impacts.

A first research project has lead to the development of an active seat suspension according to ride comfort motivations, i.e. to isolate the occupant from vertical vibrations. This study has been performed to demonstrate the capability of an active isolation device to optimize seat car vibratory isolation (Périsse, 1997). In this paper, a passive damper seat suspension is studied.

A multibody approach was chosen to model the seat and the occupant because of its reliability to resolve crash computational problems with hightly nonlinear mechanical elements. A particular attention was paid to the influence of seat belt positions, the stiffness and damping characteristics of the suspension.

The simulation results demonstrate the seat suspension capability to answer to crash injuries criteria and, in some cases, its usefullness to reduce the head acceleration levels. Finally, it was shown that a seat suspension can be considered as a crash absorbing device.

Introduction

Many studies have shown the interest of energy absorbing structures in seats in reducing the load on the occupant. Winter (1984) showed that the design of the seat influences significantly several injury criteria. He also pointed out that a proper design of energy absorbing structures in the frontal part of the seat improves the protective capacity of the seat. Most of these studies have dealt with specific problems such as submarining or head restraint capability (Drazetic, 1996). However, few studies tackle the case of seat suspension, firstly designed for ride comfort, to improve seat crashworthiness.

The purpose of this work is to investigate the biomechanical motion of the occupant, sitting in a car seat with passive suspension, in a standard crash test, in order to suggest reasonable solutions to crash injuries. The seat suspension was originally designed to perform an active isolation from vertical ground vibrations (Périsse, 1997). The active control was replaced by a passive damper but the kinematics of the system was unchanged. The goal was to study numerically the capability of this system to absorb a part of the impact energy during a crash.

The study was conducted using the VeDyAC multibody program (Giavotto, 1983). The model uses a 50th percentile HIII dummy. The system is submitted to a 20 g pulse deceleration of mid severity corresponding to a frontal impact of a car at 50 km/h.

Different cases were examined: seat without suspension, seat suspension with different damping factor and stiffness values, different foam seat characteristics, belt fixed on the seat, belt fixed

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on the vehicle. In this paper, the results on belt position influence and seat suspension capacity are presented.

Multiboby modelisation with VeDyAC

For vehicle crash analysis, multibody programs show great advantages over finite element methods. Some features of multibody programs are :

- theory that treats large elastic-plastic deformation ;
- nonlinear conditions required to simulate internal and external contact/rebound between structural parts ;
- low number of parameters and elements ;
- structural optimization ;
- capability to model a large variety of structural types ;
- accurate and efficient numerical methods to solve nonlinear equations of motion (low CPU time).

VeDyAC program was used to analyze the response of a car seat suspension to general automotive crash test conditions. VeDyAC (*Vehicle Dynamics And Crash*) is a numerical program developed in the 80's by the Department of Aerospace Engineering of the *Politecnico di Milano* and the *SWOV* in Netherlands [3]. It is basically oriented for general purpose road accident dynamic simulation. It was aimed to produce a software sufficiently modular and flexible to be employed with models of different kind and complexity. It is based on Lagrangian formulation of the dynamical motion of a rigid body.

The main components of the VeDyAC multibody program are : *rigid bodies*, *nodes*, *deformable elements* and *contact surfaces*. The motion of each body present in the system is computed by resolving the dynamical equation of motion in time domain. Each body has 6 degrees of freedom (*dof*) and is described by its mass, its inertia and its nodes (center of mass and connection point coordinates to other bodies). Each deformable element is massless and is composed of two nodes representing the connecting set to the bodies. The force-displacement laws are elastic and elastoplastic for each *dof*. Deformable elements are chosen according to the mechanical characteristic of each link between the bodies and the mode of deformation (torsion, bending, yielding,...).

The available outputs are produced by a post-processor program. Such outputs basically contain:

- displacements, velocities and accelerations bodies in the system ;
- loads in each deformable elements ;
- contact loads between subsystems ;
- head injury criteria (HIC) acceleration severity indices.

Motion equations are numerically time integrated, taking into account non-linear elements in the model, with a predictor-corrector algorithm (Belytschko, 1992).

Model of the car seat suspension

The seat suspension model

The system studied here is based on an active seat suspension schematized in Figure 1. This prototype has been patented with Bertrand Faure company (Périsse, 1997). The seat suspension prototype includes a kinematic guide $\boxed{3}$ and a spring stiffness system $\boxed{5}$. The kinematic guide offers a pure vertical movement based on the Watt six bars mechanism principle. The stiffness device is composed of another bars mechanism which gives the suspension vertical rigidity by using two horizontal traction springs $\boxed{6}$. A servo-motor can adjust the springs initial tension to maintain the same vertical static position whatever the load on the seat (not represented on the scheme). The electromechanical actuator $\boxed{8}$ is replaced by a viscous damper.

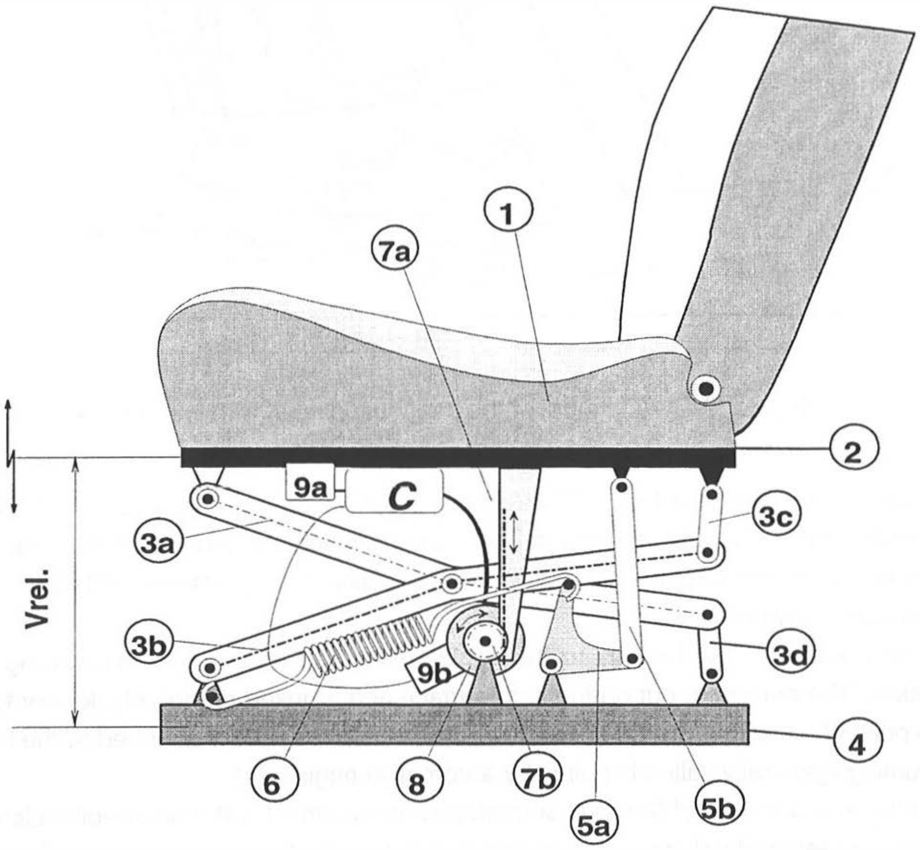

Fig. 1: prototype scheme of the active seat suspension.

The seat suspension model can be divided in four subsystems:

1. the vehicle floor or the test sled ;
2. the suspension ;
3. the seat (denoted $\boxed{1}$ in Figure 1) ;
4. the occupant represenrted by a dummy.

The whole multibody model of the seat suspension is presented Figure 2. It consists in 32 rigid bodies divided in 4 substructures, 11 of wich belong to the dummy, 4 to the seat and 11 to the suspension. Rigid bodies are defined with 355 nodes. Their spatial positions are represented in a cartesian coordinate system.

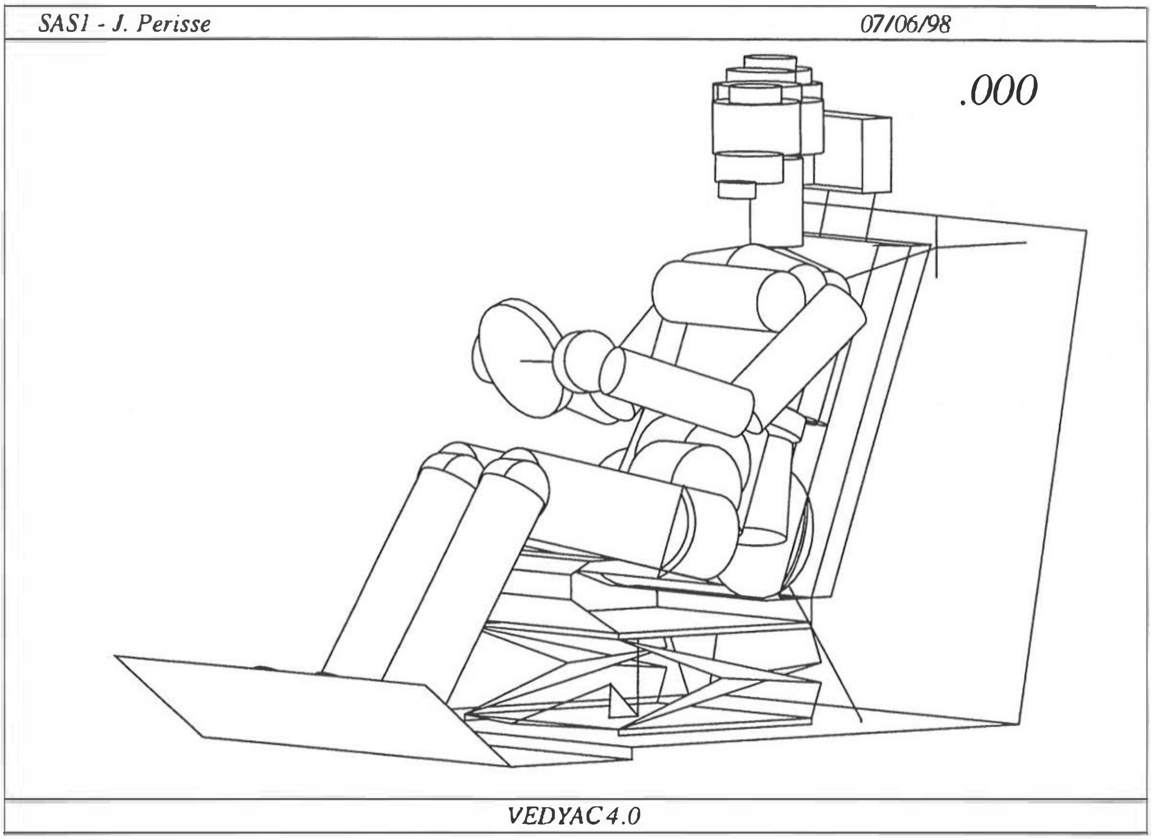

Fig. 2: VeDyAC model of the seat suspension with HIII dummy.

Bodies are interconnected with 67 deformable elements which represent mobilities between each substructure. Dummy-seat interactions are represented with 25 contact elements. Deformable elements are characterized by elastic or elastoplatic joints connected connected with two nodes belonging to the bodies.

The suspension is fixed rigidly to the vehicle floor and the seat to the moving part of the suspension. The dummy is put on the seat surface and is linked to the vehicle – or to the seat – with a 3 points belt: one attached to the left shoulder and the others attached to the hauch points of the dummy (generally called H points by automotive engineers).

Stiffness and damping of the seat suspension are modeled with one specific element, represented by a spring and a damper connected in parallel. The headrest of the seat is modeled with a contact element representing properly the crushing properties.

Dynamic test requirements for car seats

Numerical tests use a 50th percentile anthropometric dummy which is required in automotive crash tests. Postural angles of the dummy are presented Figure 3. They correspond to standard postural angles. The seat base, i.e. the vehicle sled, is submitted to a deceleration frontal pulse with a peak value of 20 g occuring at 20 ms after the impact (see Figure 4). The potential injury due to head impact is classically determined by *Head Injury Criteria* – HIC – which has an acceptable limit of 1000. Other specific injury related limits can be studied : pelvic compressive load and upper torso restraint by the admissible load in a single shoulder belt.

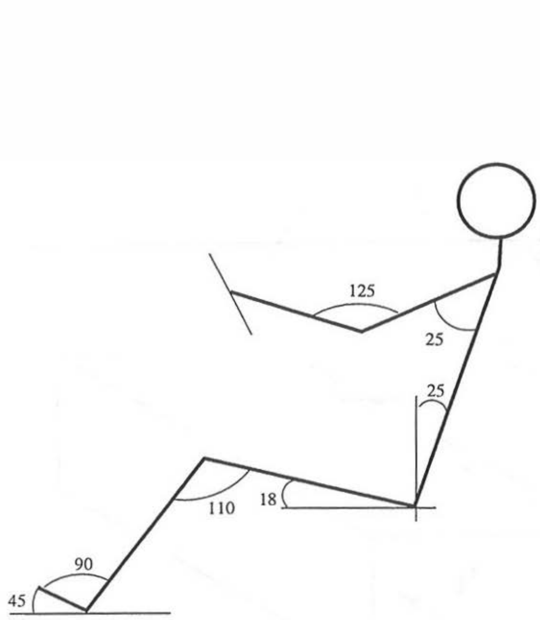

Fig. 3: postural angles of the dummy (in degrees).

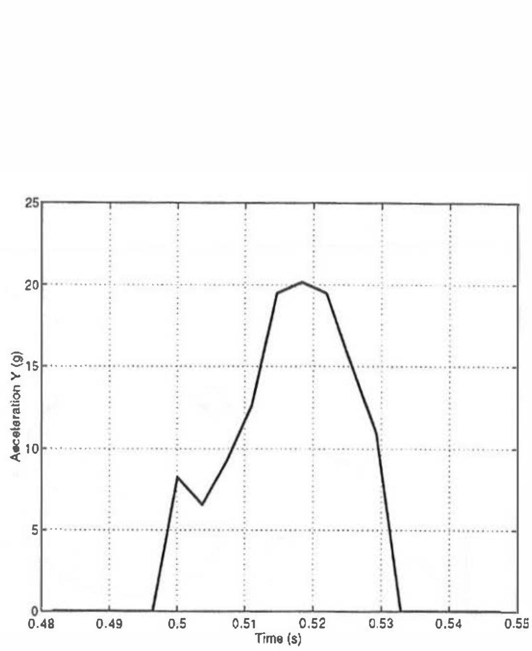

Fig. 4: deceleration curve of the car (in g).

Foam cushion contact model with dummy

In VeDyAC, the modeling of contacts uses specific elements. Contact forces are elasto-viscoplastic and depend on the penetration and the contact area. VeDyAC is able to calculate the contact in four cases: cylinder/plane, cylinder/sphere, sphere/plane and sphere/polyedron. In the seat model, only the cylinder/plane is considered because the HIII dummy is only characterized with rigid cylinders and the seat model with plane surface of contact. In fact, we must be able to represent the contact forces between the dummy and the foam in the back-rest and cushion of the seat. For this cases, the normal force at the interface between two bodies is evaluated with a polytropic law as

$$F_n = P_o.A.\left(\frac{V_o}{V_o - \Delta V}\right)^{\gamma}\left(1 + \tanh\frac{V_n}{V_{no}}\right) \tag{1}$$

with P_o the initial pression of the body, V_o the initial volume of the body, A the area of contact, V_n the normal velocity of penetration, ΔV the variation of volume due to the penetration and $\gamma = 1.4$ the adiabatic constant of the air. Hyperbolic term in the equation takes into account the viscosity effect in the contact with an hysteresis. The tangential force is evaluated with the normal force as

$$F_t = -F_n.\mu_1\mu_2.\tanh\left(\frac{V_t}{V_{to}}\right) \tag{2}$$

with μ_1, μ_2 the friction coefficients of the bodies in contact. V_t is the tangential velocity in the contact used to exhibit a possible hysteresis.

The objective is to evaluate the parameters of the polytropic model in order to correctly represent the forces in the dummy/seat contacts. To validate this approach, the uniaxial compression test of foam sample by a cylinder was studied (see Figure 5).

The vertical displacement of the cylinder U_o is imposed at the center O of the rigid cylinder. The total strength resulting of the contact forces of the foam, in the vertical direction, is F_r. The displacement velocity of the cylinder is small and the viscosity effects are neglected. The radius of the cylinder is equal to 130 mm. The dimensions of the foam sample are $500 \times 500 \times 130$ mm.

The compression of the foam U_z is expressed as

$$U_z = U_o - R + (R^2 - x^2)^{1/2} \tag{3}$$

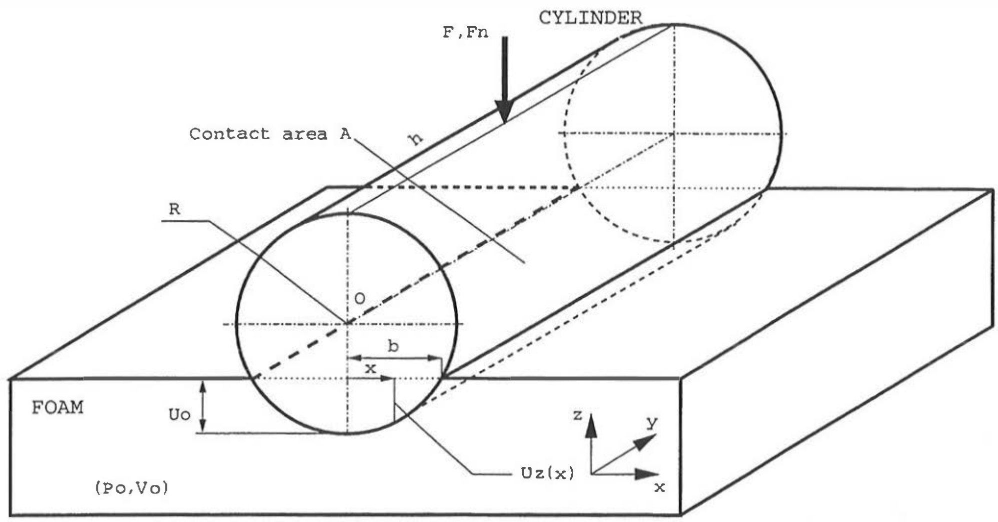

Fig. 5: foam compression by a cylinder.

with the cartesian x the abscissa of a contact point on the cylinder in the half length of contact. The half-distance of contact is b ans is expressed as

$$b = (2.R.U_o - U_o^2)^{1/2} \tag{4}$$

For hypercompressible foam material and cylinder impactor, the normal contact force at a distance x can be represented accordingly the Ogden law [5] by

$$F_r(x) = U_z.(75 - 16.U_z + 1,8.U_z^2) \tag{5}$$

The coefficients in the equation (5) were identified with a compression test on a foam sample. The total vertical foam strength at 0 is calculated by integrating Fr over the horizontal distance of contact as

$$F = 2\int_0^b F_r.dx \tag{6}$$

The unknown parameters of the equation (1) are ΔV and A. These two parameters can be expressed in function of the depth of the contact h, the radius of the cylinder R and the vertical displacement imposed U_o. By assuming that U_o is small comparatively to R, the contact area can be expressed as

$$A = 2.h.\left(2.R.U_o - U_o^2\right)^{1/2} \tag{7}$$

The expression of the volume variation is

$$\Delta_V = h.\left[R^2.\arccos(1 - \frac{U_o}{R}) - (R - U_o).(2.R.U_o - U_o^2)^{1/2}\right] \tag{8}$$

The total normal force on the cylinder F_n is a function of (P_o, V_o, U_o). It leads to minimize the error function $e_{(U_o)} = F_n - F_r$ with the two parameters P_o, V_o. The Figure 6 represents the total vertical strength on the cylinder obtained with the polytropic model compared with the measurement, i.e. the Ogden law. The difference between the two curves is small. By the minimisation computation, we found $V_o = 0.016\, m^3$ and $P_o = 2.58\, 10^3\ Pa$. The vertical force versus the horizontal distance x from the center is shown Figure 7 from $U_O = 0$ to $U_0 = R$. This graph shows the evolution of the load profil when the imposed vertical deformation increases showing the good capability of the polytropic model to represent the foam behaviour with a rigid cylinder. Finally, we have demonstrated that the polytropic model is adapted to the modeling of the contact forces between the dummy and the seat foam.

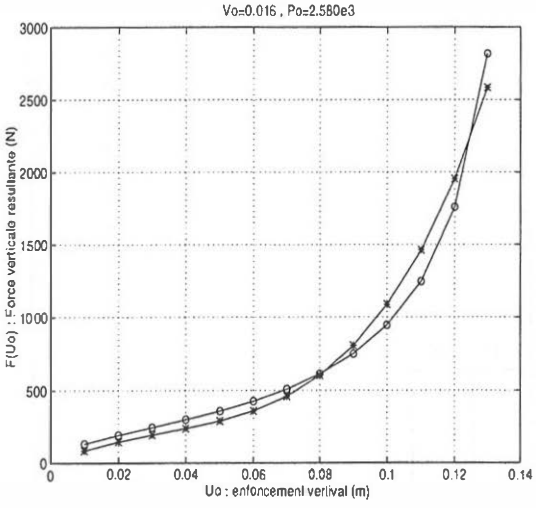

Fig. 6: total vertical load F versus imposed penetration U_o; $*$ measurement, o polytropic model.

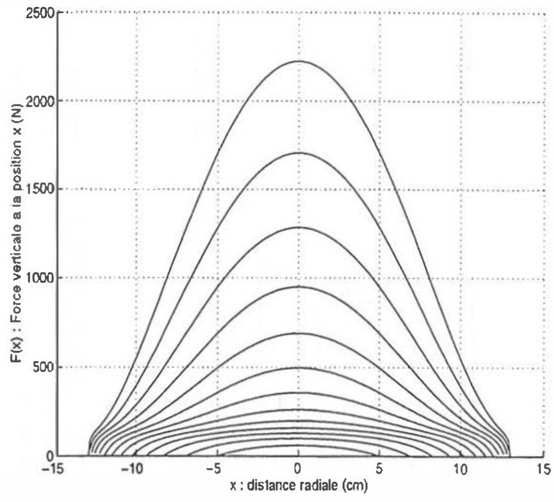

Fig. 7: vertical load versus x distance from the center for different values of U_o (polytropic model).

Numerical results

The numerical computation of the model require the choice of a time step. The integration time step must be optimized to achieve good accuracy and low computation time. A frequency analysis of the model is realized with a specific program which give the number of natutal frequencies by range of frequency (see Figure 8). A time step of Δt=0.0001 s has been chosen. The simulation time is equal to 1.5 s and leads to a CPU time of 10 minutes on a IBM RISC 6000/397 computer.

Before the pulse impact, the vertical static position of the system is reached giving the dummy equilibrium in the seat. The pulse deceleration is then applied on the sled like in experimental test conditions.

Displacement time simulations during the impact are presented Figures 9 and 10. The time step between two drawings is equal to 40 msec. These simulations show the influence of the belt fastener positions on the motion of the occupant. Low transversal motions of the dummy are observed with the three point belt on the seat. Headrest efficiency to decrease head rotation amplitudes is clearly demonstrated. Greater displacements of the suspension with belt fastened on the vehicle can be observed, particularly in the transversal direction.

Figures 11 to 14 compare numerical simulations between the belt fixed on the seat (a) and on the vehicle (b). In the first case, there is a point fixed in the backrest and the others fixed on the seat base conformable to car standards.

Acceleration criteria at the center of gravity (*cog*) of the head is respected; the HIC values are acceptable and are roughly equal in both (a) and (b) cases. However, a lower acceleration peak amplitude is noticed when the belt is fastened on the seat (see Figure 11). Figure 12 shows the rotation of the head *cog* according the yaw axis, i.e. the axis perpendicular to the transversal

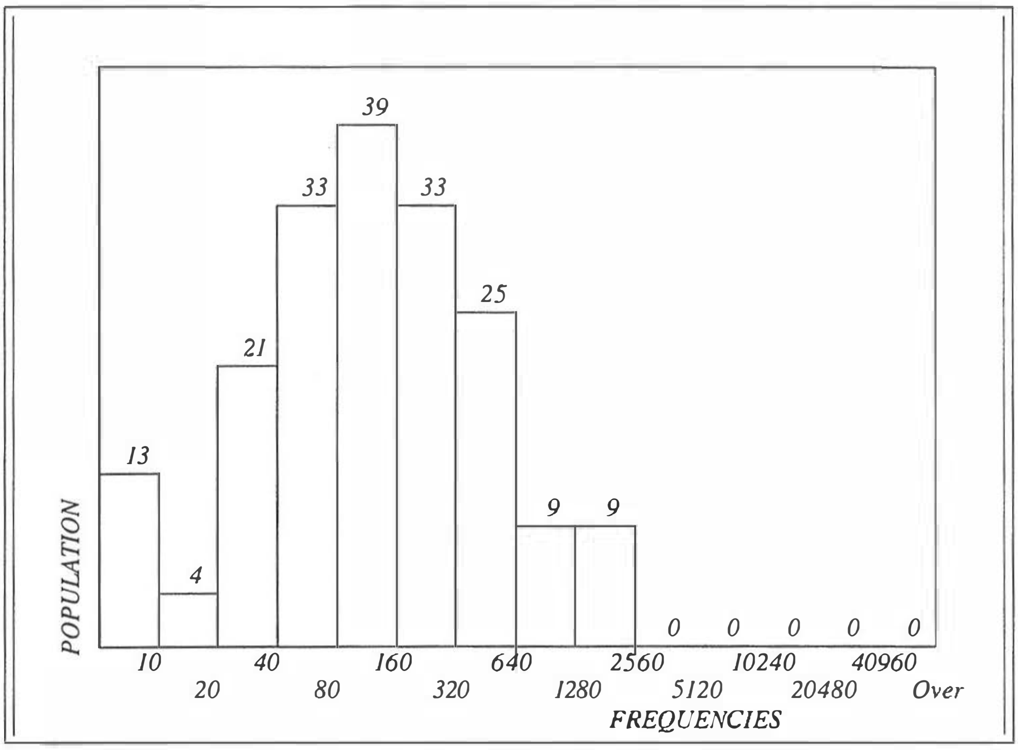

Fig. 8: natural frequencies range in the model.

plane respectively to the impact plane. Low rotation is achieved for the (a) configuration of the belt.

The contact force atthe seat cushion/dummy interface is plotted Figure 13. Maximum amplitude of the vertical load is obtained when the belt is fixed on the vehicle; this is due to a rebound effect of the seat suspension amplified by the belts. At 0.6 s, the force is cancelled and there is no more contact of the occupant with the cushion. The peak value of the contact force is reduced to 250 N when the belt is attached on the seat (for 450 N with the belt attached on the vehicle).

However, greater load is achieved in the connections of the suspension kinematic. The peak force in a joint of the suspension mechanism calculated in (b) is 50% greater than the one calculated in (a). As a part of the energy is absorbed by the belt if they are linked to the vehicle frame, because in (a) the belt is only fixed to the seat, dynamical forces in the suspension increase during the impact.

It can also be noticed on Figures 9 and 10 that belt positions influence the submarining behaviour of the occupant. In (a), the dummy largely penetrates in the seat cushion. However, the seat belt model carried out in this study is not sufficiently accurate to have the true absolute motion of the dummy but only relative difference between the two seat belt configurations.

Figures 15 to 16 show two graphs corresponding to the simulations performed in two different cases: seat with suspension and belt fastened on the seat (c) and seat without suspension (d). Low improvements resulting of the seat suspension are noticed. However, acceleration peak value calculated at the head *cog* is reduced about 10%. Vertical load contact is quite similar between (c) and (d).

Conclusions

In this study, the potential injury-reducing benefits in frontal impacts of well designed seat have been evaluated through mathematical simulations. In these simulations, various seat properties and configurations have been studied and only few of them have been presented.

The results show that low improved protection can be achieved with seat suspension designed for ride comfort purposes. In any cases, occupant security is not deteriorated by the seat suspension to a frontal impact. Particular attention must be paid to the design of the seat belt. Numerical results show that the positions of belt anchorage points should be carefully studied to obtain low loads in the seat suspension and at dummy-seat interfaces.

Basically, the risk of submarining and the risk of head striking in the vehicle interior are influenced by variations in seat design. Head impacts produce acceptable HIC values. Head and torso rotations are reduced with belts fixed on the seat. Seat-dummy interaction loads between back-rest and foam cushion are reduced using belts fixed on the seat.

Crash enhanced performances of seat suspensions could be obtained for vertical impact situations, such as in aircraft crash-landing for example.

The multibody approach, using VeDyAC code, to study seat suspension behaviour to car frontal impact has shown its good flexibility and its computational performance. It allows good reliability and versatility in the model characteristics for parametric analysis. Parameters of each element are very close to design parameters for the seat designer. It provides a very suitable method to study seat crash behaviour and a powerful tool to design new absorbing seat components. Experimental tests should be performed to validate numerical results.

Acknowledgements

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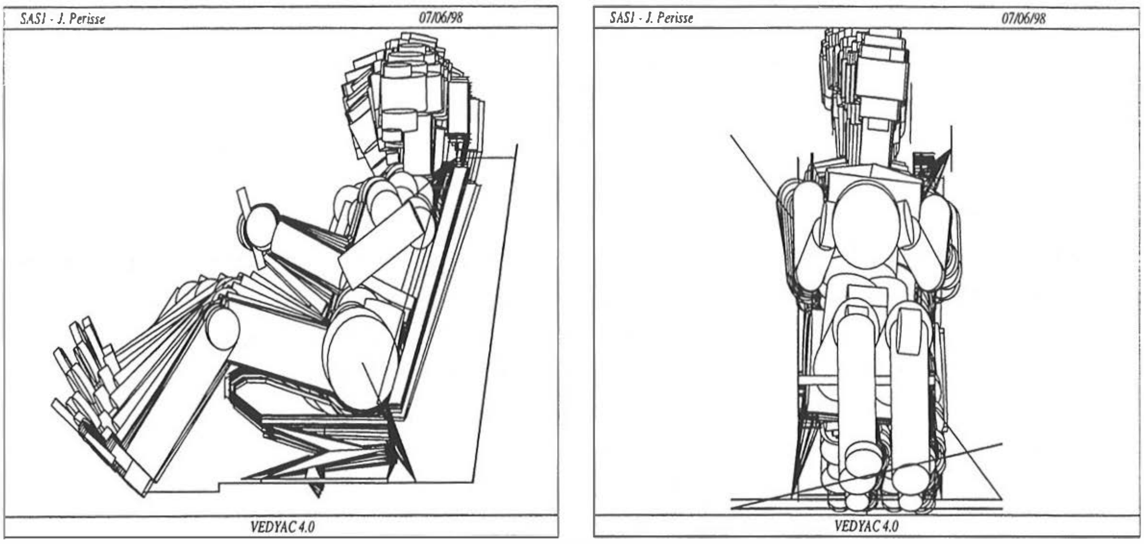

Fig. 9: displacements during the impact (belt on the car).

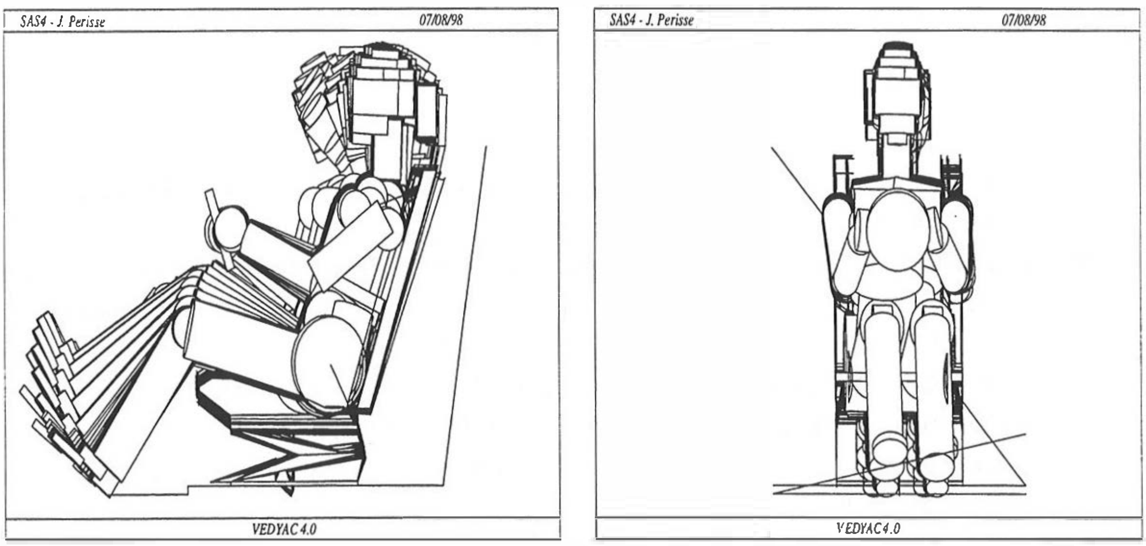

Fig. 10: displacements during the impact (belt on the seat).

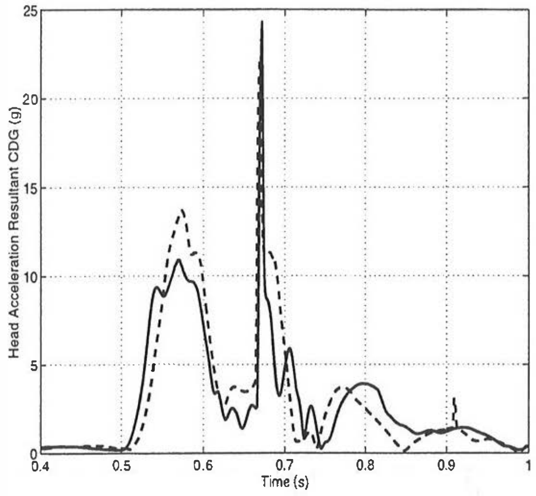

Fig. 11: *cog* acceleration of the head (— belt on the car , — — belt on the seat).

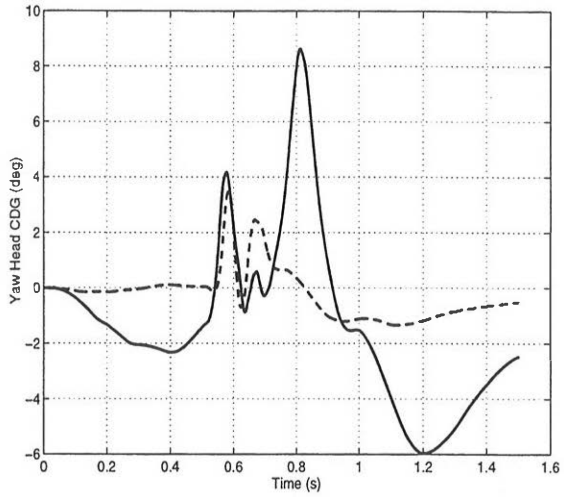

Fig. 12: *cog* rotation of the head (— belt on the car , — — belt on the seat).

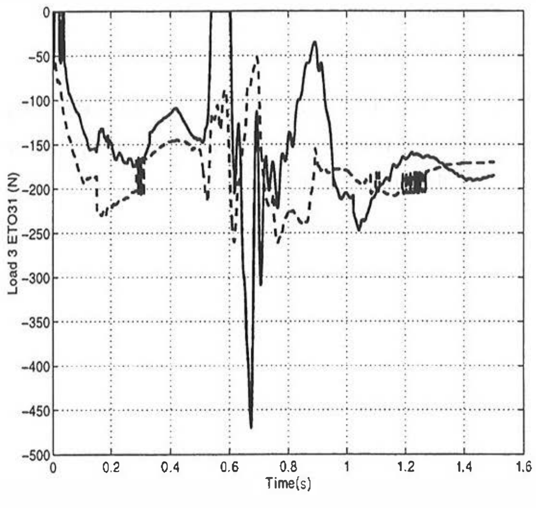

Fig. 13: vertical load at the buttocks dummy/foam cushion interface (— belt on the car, — — belt on the seat).

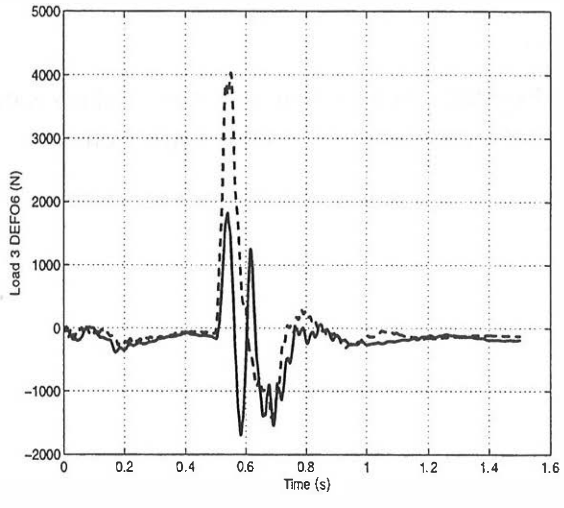

Fig. 14: load in a articulation joint of the suspension mecanism (— belt on the car , — — belt on the seat).

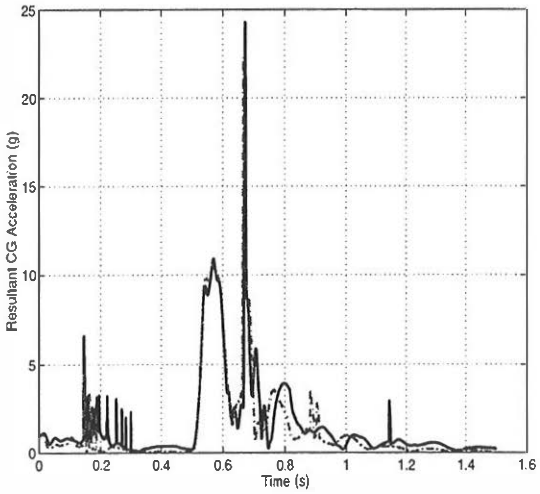

Fig. 15: *cog* acceleration of the head (— without suspension, —.— with suspension).

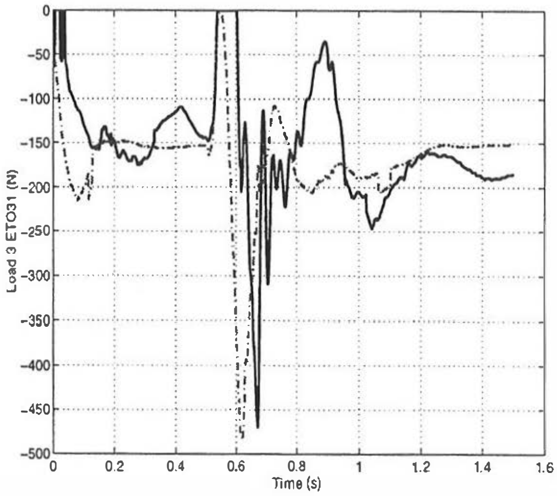

Fig. 16: vertical load at the buttocks dummy / foam cushion interface (— without suspension, —.— with suspension).