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NUMERICAL SIMULATION OF ENERGY ABSORPTION IN POLYURETHANE FOAMS UNDER IMPACT

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Abstract

Selection of proper mechanical characteristics of the padding material of the head rest, namely polyurethane foam, is particularly important in the design process of the vehicle seat head restraint. In the paper the code of numerical simulation of flexible polyurethane foam absorber response against impact of a rigid body (passenger's head) is presented. The code is based on the lump mass method. Basics of the numerical algorithm based on this method are presented. Material constitutive model of flexible polyurethane foam (FPF) derived on the basis of former experimental tests and used in the algorithm is described. Comparative results of numerical simulations together with results of experiments carried out on drop hammer rig with flat impact or are shown. The results are presented on diagrams of deceleration versus time. Also comparative Finite Element simulations are carried out. The FE simulation was performed using the implicit analysis, taking into account the multi-linear, hyper-elastic material model and large strain (non-linear geometric relations). The influence of damping coefficient on the deceleration characteristics was investigated. Some final conclusions concerning the applicability of the numerical code based on the lump mass method, as well as its advantages in comparison with the FE software are derived.

Keywords: *polyurethane foams, energy absorption, lump mass method, finite element method*

1. Introduction

Flexible polyurethane foams are commonly used materials in automotive applications, especially for internal cockpit parts and seats (Fig. 1). Selection of proper mechanical characteristics of the padding material of the head rest, namely polyurethane foam, is particularly important in the design process of the vehicle seat head restraint. The primary function of the headrest is to support and cushion the head, protecting an occupant from injury in a front and a rear-end collision (Fig. 1).

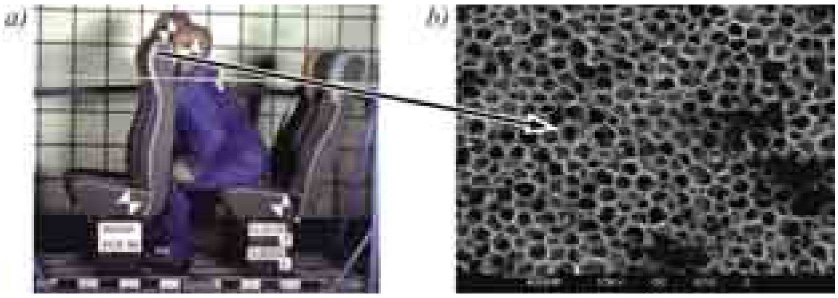

Fig. 1. Impact of a head to a head restraint during bus crash test (a), cellular structure of the polyurethane foam (b)

The head-rest should be able to absorb the kinetic energy of the head, exerting a possibly low force, which would not cause a damage of the brain or skull. Certain biomechanical criteria, including HIC (Head Injury Criterion), as well ECE regulations [1] should be here fulfilled. The whole energy of impact should be absorbed by the padding material, but on the other extreme (because of HIC criterion) a maximum impact load exerted by the head-rest can not exceed a certain limit value. Another biomechanical criterion states that a continuous deceleration of a head over 80g should be no longer than 3 ms (3ms Criterion) [2].

In Fig. 2a WSTC Curie (Wayne State Tolerance Curve) [2] is presented, which indicates a limit between a safe and unsafe area (possibility of a passenger to be injured). The limit diagram shows the deceleration versus the time of deceleration impulse. Selection of a head-rest padding material is of crucial importance as far as the head deceleration in terms of time is concerned (Fig. 2b). In order to fulfil requirements coming from certain biomechanical criteria, a clear understanding of a padding material (polyurethane foam) response to an impact compressive force is necessary. This can be realized on the basis of experimental data. However, experimental tests are very expensive and their results can not be generalized.

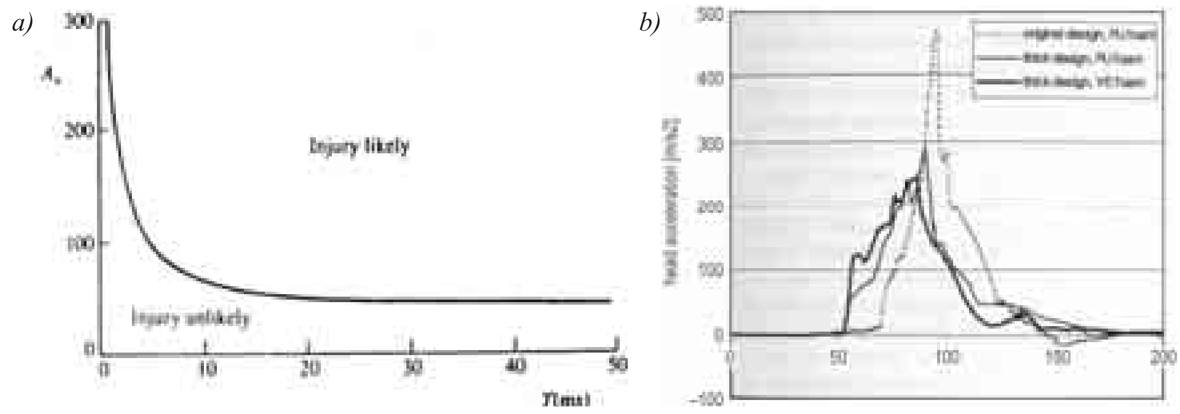

Fig. 2. WSTA curve (a), head deceleration for three different padding materials for impact velocity 30 km/h (b)

Thus, it is desirable to elaborate a computational tool (numerical code), which enables to simulate an impact of a polyurethane foam absorber by a rigid body. This code might be applied in design process of the head-rest (selection of a proper padding material) and would allow one to eliminate or limit expensive experimental tests.

2. Constitutive material model

On the basis of experimental investigations into FPF response to dynamic (impact) compressive force, performed by the authors [6-8], an analytical constitutive model was derived. It assumes the density components, substrates proportion and strain rate to be separate functions. The following interpolation function was derived:

$$\sigma = f_0(\varepsilon)\cdot G(\rho)\cdot H(i)\cdot M(\varepsilon,\dot{\varepsilon}), \tag{1}$$

where:

$f_0(\varepsilon)$ – a “shape” function, represented by the 4th order polynom:

$$f(\varepsilon_1) = \sum_{n=1}^{4} A_n\, \varepsilon_0^n , \tag{1a}$$

$H(i)$ – substrate proportion function,

$G(\rho)$ – density function,

$M(\varepsilon,\dot{\varepsilon})$ – strain rate function.

Coefficients A_n were determined on the basis of compression static tests. The shape function (1a) was assumed to be a "base function" and the specimen, representing this approximation – the "base specimen" [9]. In the same way functions $H(i)$ and $G(\rho)$ were determined.

On the basis of relations proposed by other authors [4] the strain rate function $M(\varepsilon,\dot{\varepsilon})$ was assumed in the following exponential form:

$$M(\varepsilon,\dot{\varepsilon}) = \left[\frac{\dot{\varepsilon}}{\dot{\varepsilon}_0}\right]^{n(\varepsilon)}. \tag{1b}$$

In order to determine coefficient n, dynamic compression impact tests of FPF specimens were carried out on the special experimental stand, namely drop hammer rig [8, 9].

For all tested specimens of different ρ and i, the stress-strain relation in tension was linear. Thus, on the basis of static tensile tests the following constitutive relation for tension was derived:

$$\sigma_1 = E_R \cdot \varepsilon_1. \tag{2}$$

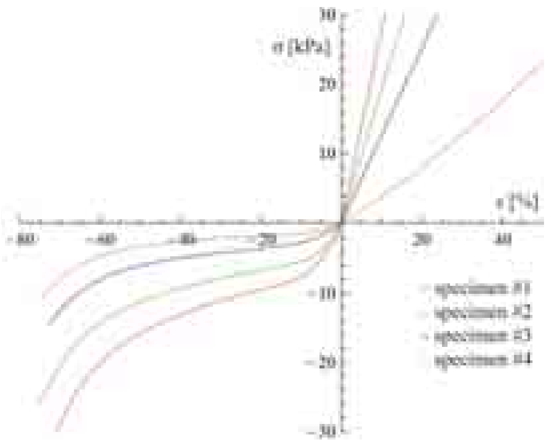

Fig. 3. Static tensile/compressive test diagrams [3, 4]

In order to derive the strain-rate function $M(\varepsilon,\dot{\varepsilon})$, the time-deceleration diagrams, obtained from the impact tests were used. The impact tests were performed on different specimens, of different substrates proportion, and density, as well as for each type of the specimen – for different velocities. Impact tests will be described in paragraph 5. Experimental time-deceleration diagrams were converted into σ–ε diagrams and compared with static compression tests diagrams. The exemplary comparative diagrams, representative for all tested specimens, are shown in Fig. 7.

The range of velocities, which could be realized on the drop-hammer rig, did not allow one to determine definitely the exponent n in (9). In the range of applied velocities the strain-rate function was of constant value, namely $M(\varepsilon,\dot{\varepsilon}) = 2$.

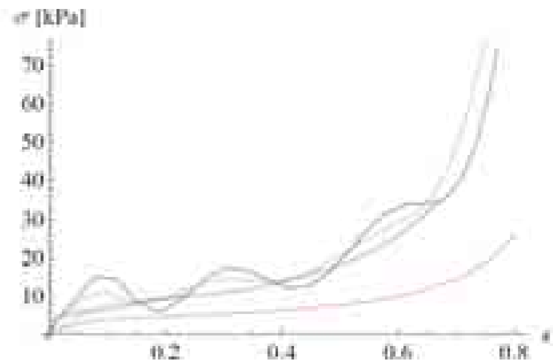

Fig. 4. Comparison of static and impact tests results for the same specimen. Red line – static test, other lines – dynamic tests

3. Algorithm based on the Lumped Mass Method (LMM)

Numerical code, which allows one to simulate an response of the foam energy absorber to the impact of a rigid body (passenger's head) has been based on the Lumped Mass Method [2, 3]. It consists in division of 2D material continuum [3, 4] into small elements, each containing a mass point – "lumped mass" (Fig. 5a) of certain mass (i, j). The mass points act on each other by means of forces, coming from the deformation, which is determined by the constitutive material model.

The model of the absorber (Fig. 5) has been divided into a finite number of mass points, which results in the generation of LMM mesh (Fig. 6). It was assumed in the analysis, that the impactor is a perfectly-rigid body and the contact surface of the impactor and the absorber is flat (Fig. 6). For each mass point (i, j) the following equations of motion were formulated:

$$m\,\ddot{x} = \sum_{k=1}^{4} n^{(k)}\,\sigma^{(k)}\,S^{(k)} - C\,\dot{x}\,, \tag{3}$$

where:

m – is the lumped mass,
$\mathbf{x}$ (x, y) – position vector,
$\mathbf{n}$ (n_x, n_y) – a unit normal vector of the element face,
$\sigma^{(k)}$ – Cauchy stress tensor,
$S^{(k)}$ – k-th face area after deformation,
C – damping coefficient.

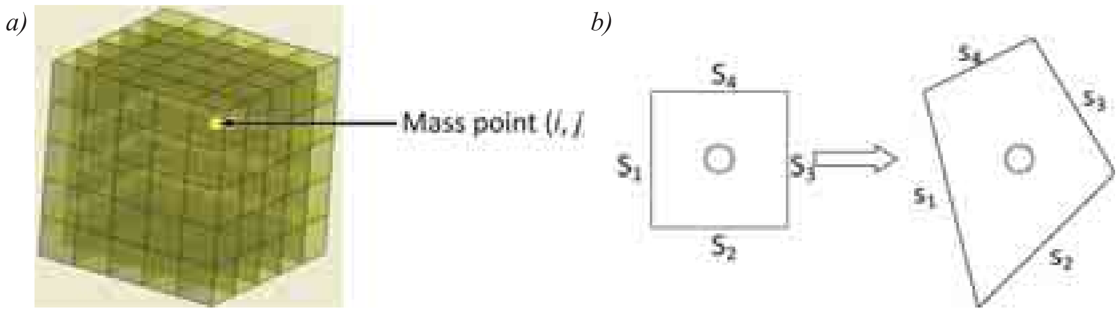

Fig. 5. LMM discrete model: model 3D (a), model of deformation (b)

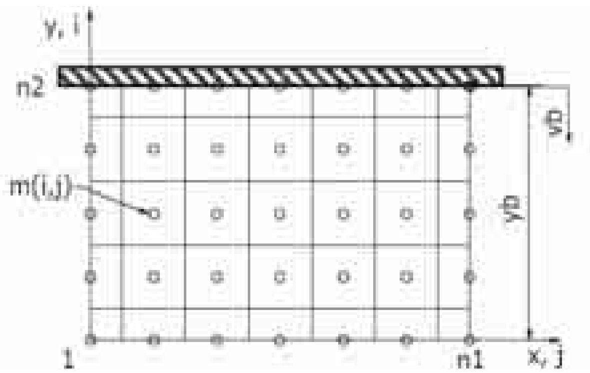

Fig. 6. LMM mesh – model 2D

After resolving the components of the Cauchy stress tensor, equations (3) take form:

$$\begin{aligned} m(i,j)\,\ddot{x}(i,j) &= T_x(i,j) - C\,\dot{x}(i,j)\,, \\ m(i,j)\,\ddot{y}(i,j) &= T_y(i,j) - C\,\dot{y}(i,j)\,, \end{aligned} \tag{4}$$

where surface forces acting on the mass point (i,j) (Fig. 5, 6) are given by the following relations:

$$T_x(i,j)=\sum_{k=1}^{4}[\sigma_{11}(i,j,k)\,n_x(i,j,k)+\sigma_{21}(i,j,k)\,n_y(i,j,k)]\;S(i,j,k)\,,$$
$$T_y(i,j)=\sum_{k=1}^{4}[\sigma_{12}(i,j,k)\,n_x(i,j,k)+\sigma_{22}(i,j,k)\,n_y(i,j,k)]\;S(i,j,k)\,. \tag{4a}$$

Using the theory of Finite Deformation in order to define the components of the Piola-Kirchhoff tensor, and subsequently – the assumed material constitutive relation (1) – in order to define the current components of the Cauchy tensor, the simultaneous equation of motion (4) with the surface forces given by (4a) was solved in the iterative procedure, using the Runge-Kutta numerical integration procedure of 4th order.

The detailed algorithm of the numerical procedure is shown in the block diagram in Fig. 7. The input parameters of the numerical simulation are: material properties (stress-strain relation), shape and dimensions of the foam specimen (FPF), impact velocity, shape and dimensions of the hammer (impactor). The output quantities from the numerical simulation are: acceleration, velocity and position of each discrete mass point of the numerical model, in terms of time.

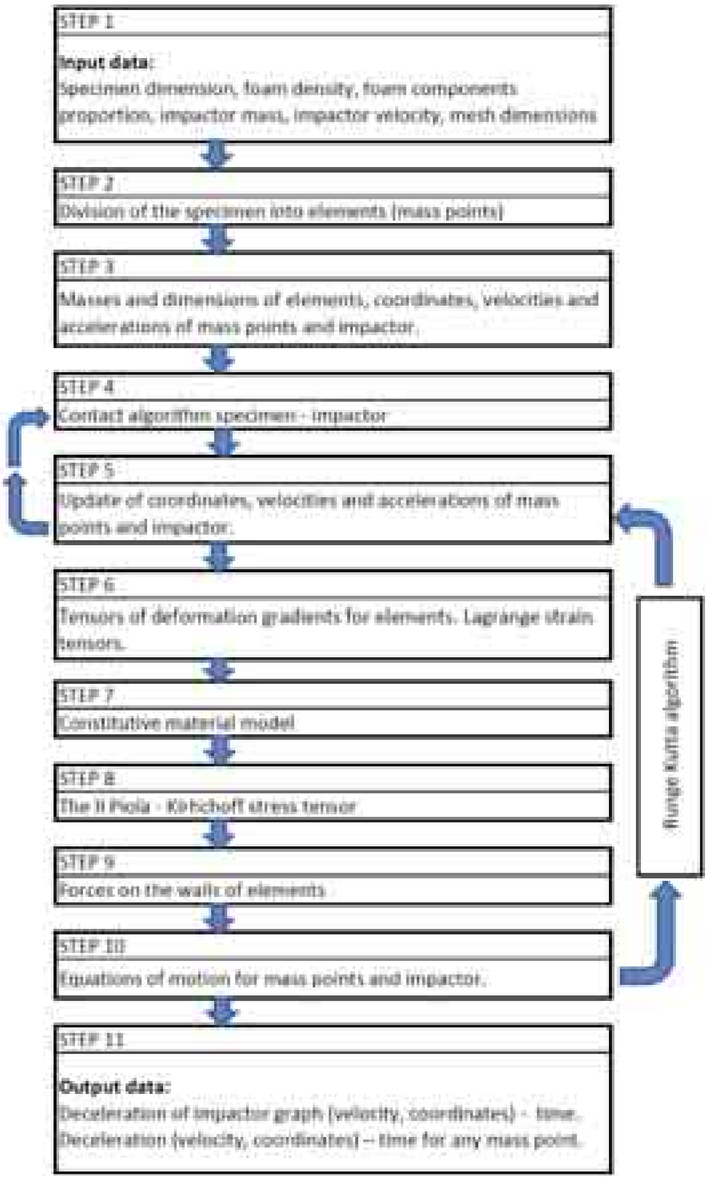

Fig. 7. Algorithm of the numerical code LMM

4. FE model

Finite Element analysis was performed using the commercial package ANSYS 11.0. The implicit analysis was applied for the integration of equations of motion. In FEM model the element type Solid187 with six degrees of freedom in each node (three displacement parallel to local coordinate axes). The element formulation is based on the logarithmic strain and true stress measures and it is well suited for nonlinear large strain applications. The element allows applying different material characteristics, taking into account nonlinear (in-elastic) effects.

The multi-linear hyper-elastic material model of the foam has been used in the FE calculations. The actual, compressive stress-strain diagram was approximated with four linear segments (Fig. 8), which was in good agreement with the experimental stress-strain diagram (Fig. 3). The material of the impactor was assumed to be a linear, elastic one (steel).

The full transient analysis procedure was applied to register the foam response to the impact load depending on time. The time of impact was defined as about 12 ms with the initial velocity 4.96 m/s. In the numerical simulation two planes of symmetry were used, so that the quarter of the model was taken into account (Fig. 9a).

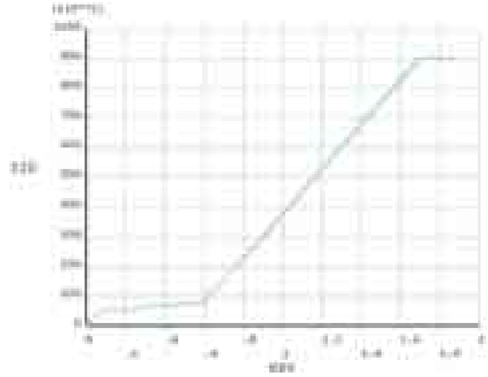

Fig. 8. Material model of FpF

a) b)

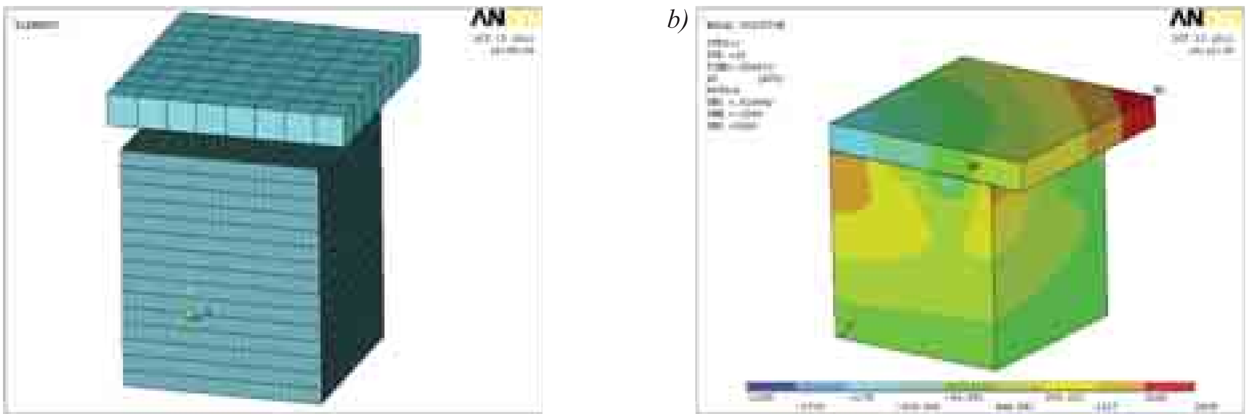

Fig. 9. FE theoretical model: FE mesh (a), deformation patterns (b)

5. Numerical results – comparative analysis

In Fig. 10 comparative diagrams of numerical simulations results obtained using the LMM code and experimental impact tests results for three different foam specimens and three different impact velocities are shown.

Experimental impact tests were carried out on the drop hammer rig [8, 9]. Diagrams present the impactor acceleration in terms of time. Numerical calculations were carried out for the different damping coefficient for foam and ram.

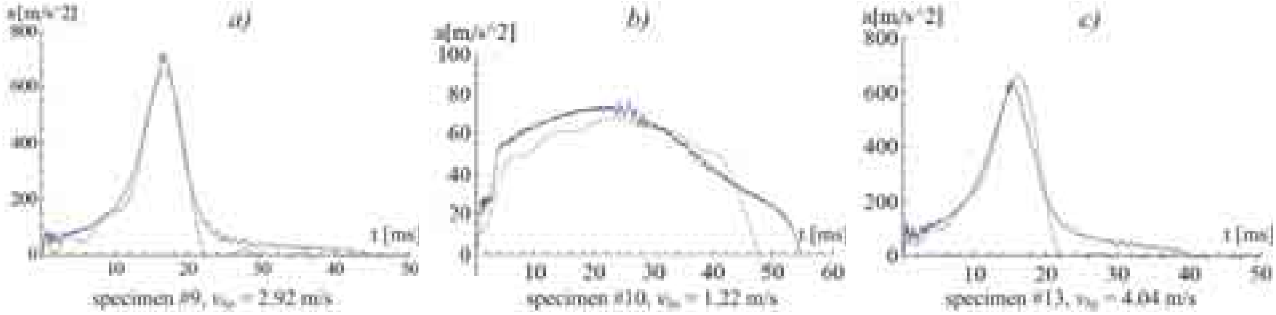

Fig. 10. Results of numerical LMM simulations (green curves) and experimental results (blue curves). Specimens: a) No. 9 – substrate proportion 1/2, ρ = 72 g/dm^3, b) No. 10 – substrate proportion 4.5/10, ρ = 72 g/dm^3, c) No. 13 – substrate proportion 1/2, ρ = 65 g/dm^3

Analogous comparative diagrams for the specimen No. 9 (substrate proportion 1/2, ρ = 72 g/dm^3, impact velocity v_0 = 4.96 m/s) obtained using the LMM numerical code, FE simulation and experimental results are shown in Fig. 11.

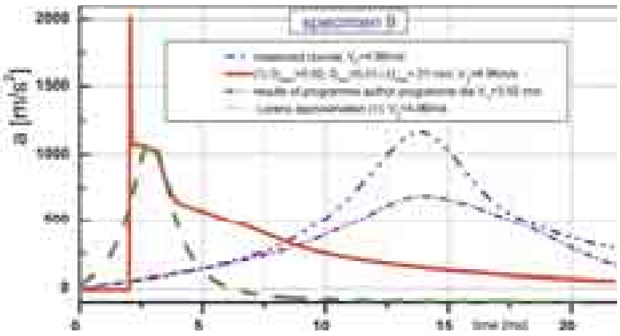

Fig. 11. Results of LMM numerical simulation (authors' code) and FEM (ANSYS)

The discrepancies among maximum decelerations depending on the assumed damping coefficient amount about 15 % (in the case from Fig. 11 – 2%). However, we should underline, that the time interval of braking assumed in FE simulation as well as maximum deformation of the foam specimen were much shorter than those taken into account in LMM simulation and obtained in the experiment (several times). It is illustrated by the "shift" of the FE curve with respect to the LMM and experimental curves (Fig. 11). This discrepancy is probably caused by the model of the finite element (typical solid model, which does not take into account the compressibility of the foam and the approximated material model).

In foam cells actually are filled with air, which induces very large final deformation (about 90% of the initial dimension). In the future FE model a new finite element and a more exact material model should be introduced.

Figure 12 presents the influence of damping effect on the results of FE simulations. The lower is the damping coefficient, the larger is the rate of change of deceleration during impact.

6. Final conclusions

The agreement between theoretical LMM simulation and experimental results is very good. Thus, the proposed, relatively simple lump mass model aided by the experimentally determined material characteristics and the original numerical code based on this model can be used to simulate the head to seat headrest impact and to design a headrest as an energy absorber. One of the main advantages of the elaborated numerical code is a short time of calculation, very much competitive with commercial FE codes and, on the other hand, allowing one to perform several "numerical experiments" for different FPF parameters and different impact parameters. These "numerical

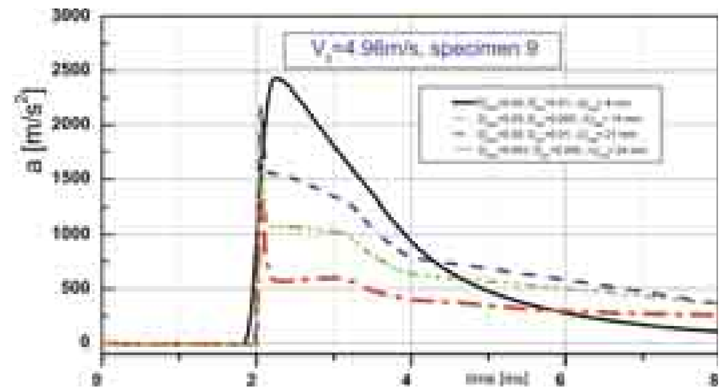

Fig. 12. Influence of damping coefficient on the results of FE simulations

can replace very expensive "real" experimental tests, particularly in the initial phase of head rest design process, when a designer has to select a most proper material of the headrest, according to the normative regulations.

The structure of numerical simulation LMM code is such one, that it enables an ease extension and addition of a block, in which a selection of a proper material of the headrest from a given data base, according to certain head injury criterion (e.g. HIC or 3ms) is carried out. The numerical code can be also completed with module, in which different shapes of impactors (different contact algorithms) are introduced.

Obtained FE results are not in a satisfactory agreement with the experimental and LMM results, except the maximum deceleration and the general character of the deceleration – versus time – curves (Fig. 11). Thus, in further research it is necessary to apply a more advanced FE model (e.g. based on the Ogden material model) or to use a numerical code, which allows one to perform the explicit analysis (e.g. LS-DYNA).

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