Chosen aspects of the crush energy determination

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Abstract

The paper presents a simulation method of the rigid barrier frontal impact car crash test. LS-DYNA software was applied. The simulations were performed for the Dodge Neon, model 1996. The car motion and the force history during the rigid barrier impact at different velocities were analyzed. Moreover, laboratory results obtained for the fixed barrier frontal impact crash test were examined and compared to the results obtained from a FEM model for a chosen car. The dependence of car stiffness coefficient on car structural parameters was also analyzed. Moreover the crush energy and stiffness coefficient distribution was examined.

Key words:

Passive car safety, car accident reconstruction, crush energy, crash test, stiffness coefficient, finite element method.

1. Introduction

There are two complimentary tools, which might be applied by car researchers nowadays: full-scale crash tests and simulations obtained by means of software and powerful hardware. Most of the simulations rely on Finite Element Method (FEM) models.

The possibility of computational analysis reduces the number of necessary laboratory test and the design phase time, they can lead to better (optimal) results at lower costs. Those advantages enable to improve the quality of products and hence the car safety is finally also improved.

The simulations of the crash test are often used in the automotive industry. Highly optimized software and powerful hardware allow for very effective and accurate crash test analyses. In order to limit the time needed to create car models, FEM commercial codes are being integrated with CAD software and graphical preprocessors, which enable effective visualization and interpretation of the results. Nevertheless, despite all improvements, the model must be always validated with experimental results and all limitations of FEM-based analysis should be considered by the user.

2. FE model of the Dodge Neon

The crash test analysis was performed by means of LS-DYNA software, which is based on the finite element method [1,2]. The FE model of Dodge Neon used in analysis was prepared by National Crash Analysis Center (NCAC) at the George Washington University [3]. It is one of the many models available for free download from the NCAC server [4]. The files can be directly exported to LS-DYNA [2]. Fig. 1 presents the FE model of the Dodge Neon. The density of applied mesh and used parts (marked by means of colors) are shown as well. For each part the material properties and the FE type were defined. The comparison between the technical data of the real car and its model are presented in Table 1. More detailed data (number of nodes, elements and parts etc.) are attached in Table 2.

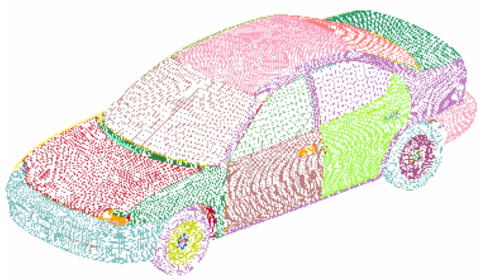

Fig. 1. FE model of Dodge Neon prepared by NCAC [4]

Table 1. Technical data of tested car and its FE model

	FE model [3]	Test vehicle [3]
Vehicle mass [kg]	1333	1354
Wheelbase [mm]	2648	2642
Distance between front wheel and center of gravity [mm]	1044	1022

Table 2. Basic FE model data

Number of nodes	283859
Number of solids	2852
Number of beams	122
Number of shells	267786
Number of elements	270768
Number of parts	336
Number of material model	10

The simulation considered a fixed barrier frontal impact. The ground and the vertical wall were represented by fixed elements. The friction coefficient of 0.9 between the ground and the wheel was modeled.

On the very beginning of the simulation the frontal parts of the car body were located 314 [mm] from the wall. The initial conditions consider linear velocity of the car body. The rotational movement of the wheels was included as well. The total analysis time was approximately 0.2 [s]. The total crush time of the car body was approximately equal to 0,08 [s]. The deformation obtained on the very end of the simulation might be assumed to be the permanent deformation. The integration time step used in the simulation was related to the shortest wave propagation time across the finite element [2], and was equal to 1.0E-06 [s].

3. Chosen simulation results

The simulations were performed for few values of the initial impact velocity [8]. The fixed barrier frontal impact was carried out for 35, 40, 50, 60 and 70 [km/h].

The car body deformation was calculated on the basis of the distance between two nodes. One of them was located close to the center of gravity of car body and the other on the front bumper. The deformations in the time domain are presented in Fig. 2

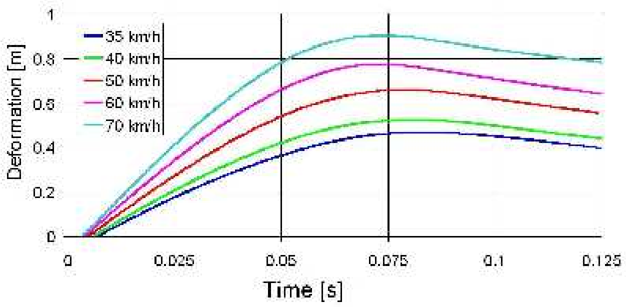

Fig. 2. Deformation process of the Dodge Neon in the time domain obtained at different impact velocities

On the basis of the results presented in Fig. 2 the crush characteristics were obtained. The total wall force versus the deformation for few impact velocities of Dodge Neon are presented in Fig. 3.

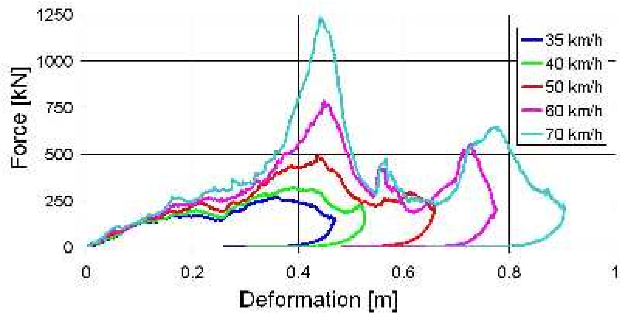

Fig. 3. Deformation process of Dodge Neon

The analysis of the dependence between the impact velocity and maximal deformation was performed on the basis of simulation. The obtained results are presented in Fig. 4.

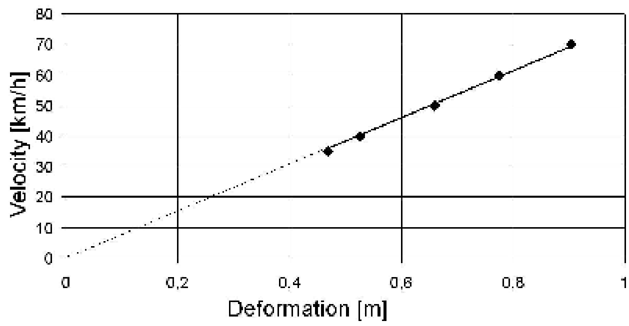

Fig. 4. Impact velocity plotted against the maximal deformation

The obtained results proved the proportional relation between the maximal deformation and the impact velocity. For the velocity range used in the simulation the linear dependence was obtained. On the basis of the results presented above the relationship (1) might be introduced.

$$\Delta x_{\max} = \alpha v \qquad (1)$$

where:

v – impact velocity [m/s],
Δx_{max} – maximal deformation [m],
α – coefficient of dependence between impact velocity and deformation, for the analyzed vehicle approximately equal α = 0,046 [s].

On the basis of the α coefficient the k factor (equivalent car body stiffness coefficient) can be found. Its value is obtained by means of equation (2), which represents the total kinetic energy dissipated in the plastic deformation.

$$\frac{1}{2} m v^2 = \frac{1}{2} k \Delta x^2_{\max} \qquad (2)$$

and

$$k = \frac{m}{\alpha^2} \tag{3}$$

where:
m – car mass.

Similar results like the presented in Fig. 4 were obtained by K.I. Campbell in the early seventies. He has performed the fixed barrier crash test of General Motors vehicles (full sized, mass 2041 [kg]) [5,6]. According to the data [5] it could be noticed that for car used by K.I. Campbell factor α = 0,065 [s].

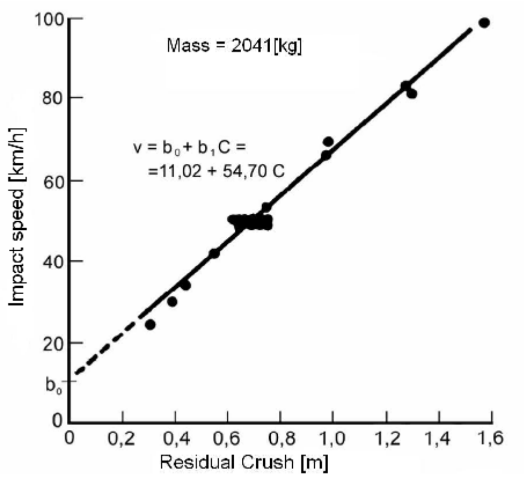

Fig. 5. Impact speed plotted against the measured residual crush for a frontal-fixed barrier impact [5]

Figures 6, 7 and 8 present views of the FE model of the car body. The deformation was obtained for the fixed barrier frontal impact with the initial velocity 50 [km/h].

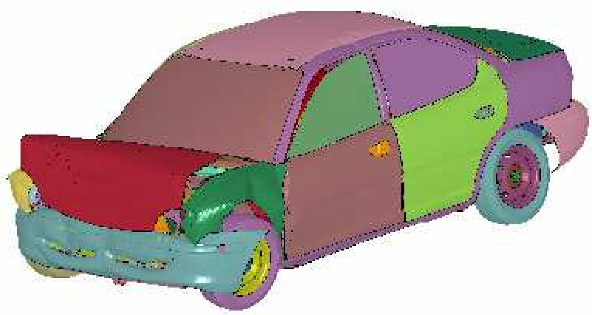

Fig. 6. Car body deformations (impact velocity 50[km/h])

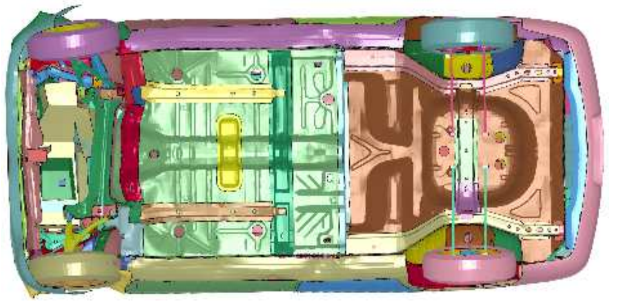

Fig. 7. Car chassis deformation (impact velocity 50[km/h])

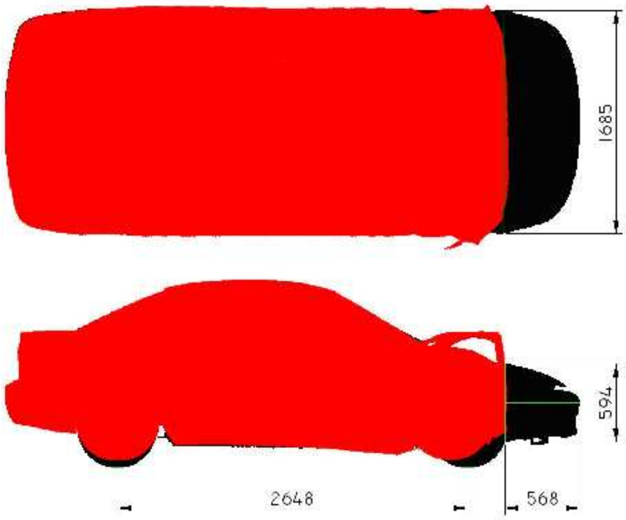

Fig. 8. Deformation area (impact velocity 50[km/h])

Fig. 9 presents the side view of the car body deformation. The results were obtained for the frontal impact velocity 35, 40, 50, 60 and 70 [km/h]. The deformed car bodies overlaid on the not deformed body enable us to estimate the results of the impacts.

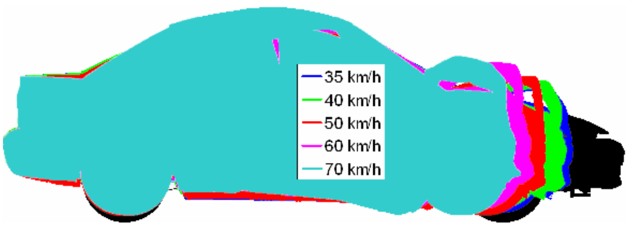

Fig. 9. Car body deformation corresponding to different initial impact velocities

3. Deformation energy and car body stiffness coefficient

The influence of car geometrical parameters on the crush energy and car body strength was determined based on the crash test data. The fixed barrier frontal impact characterized by 56 [km/h] impact velocity were used in the analysis.

Digitalized sensor signals for the corresponding crash test were downloaded from NTHSA database [7]. The signals from the load cell located on the fixed barrier and from the accelerometers mounted to the car body were used for analysis. Fig. 10 presents an example of force signals (in time domain) obtained for the Dodge Neon crush test. The sensors are numbered according to Fig. 15.

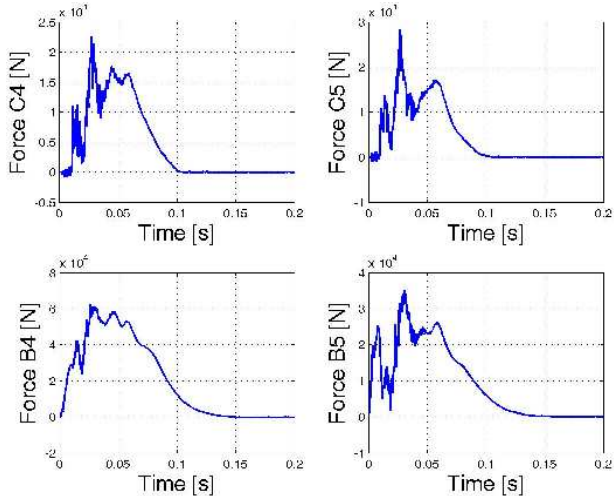

Fig. 10. An example of the force signals history obtained during the Dodge Neon crash test

The deformation process and velocity history of the car during the impact were obtained on the basis of the signal generated by the accelerometer, which was located close to the car body center of gravity. The relationships (4,5) were applied.

$$v(t) = \int_{t_O}^{t} a(t)dt \text{ [m/s]} \tag{4}$$

$$s(t) = \int_{t_O}^{t}\int_{t_O}^{t} a(t)dtdt \text{ [m]} \tag{5}$$

where:

s(t)- car body displacement [m],

v(t)- car body velocity [m/s],

a(t)- car body acceleration [m/s^2].

The displacement, velocity and acceleration of the Dodge Neon center of gravity is presented in Fig. 11.

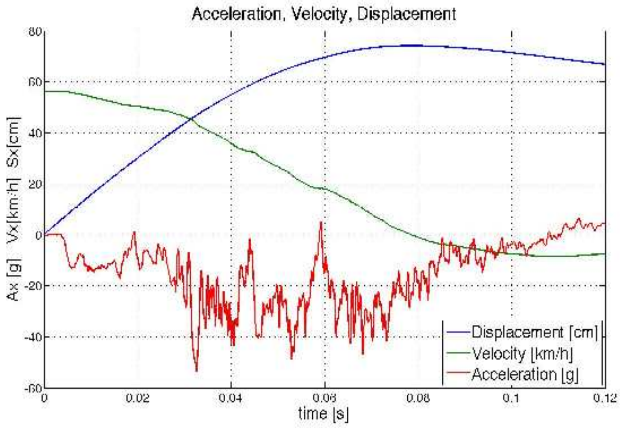

Fig. 11. Displacement, velocity and acceleration of the Dodge Neon center of gravity

The local characteristics of force in the deformation domain were determined on the basis of displacements (Fig. 11) and forces (Fig. 10). Examples of obtained results are presented in Fig. 12.

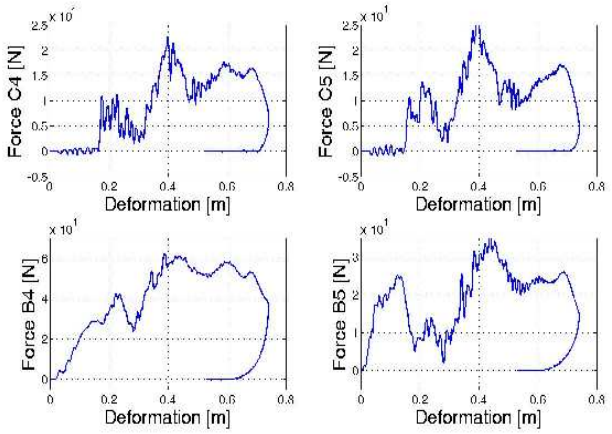

Fig. 12. Force as a function of deformation for Dodge Neon crush process

On the basis of local force characteristics the global force history was obtained. The relationship (6) were applied.

$$P_c(f) = \sum_{i=1}^{n} P_i(f) \tag{6}$$

where:

$P_i(f)$ – force history in function of deformation for the i-th load cell sensor.

The global force characteristics for the front part of the Dodge Neon car body is presented in Fig. 13.

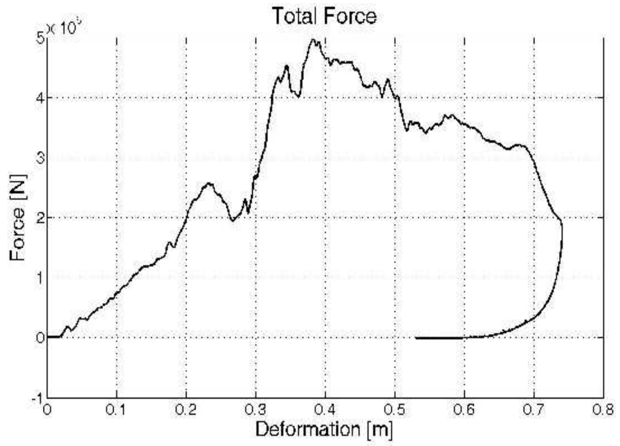

Fig. 13. Deformation characteristic of the Dodge Neon car body

The deformation energy was obtained on the basis of the force history in function of the deformation. The formula (7) was used.

$$Wd_f = \sum_{i=1}^{n} \left[P(f_{i+1}) + P(f_i) \right] \cdot \frac{f_{i+1} - f_i}{2} \text{ [N·m]}. \quad (7)$$

The average car body stiffness coefficient (for maximal deformation) was obtained on the basis of the determined crush energy. The relationships (8,9) were applied.

$$k_{f\max} = \frac{C_{f_{\max}}}{b_{f\max} \cdot h_{f\max}} \text{ [N/m}^3\text{]} \quad (8)$$

where:

$$C_{f\max} = \frac{2 \cdot Wd_{\max}}{f_{\max}^2} \text{ [N/m]} \quad (9)$$

Wd_{fmax} – crush energy for maximal deformation f_{max} [Nm],
f_{max} – maximal deformation ,
b_{fmax} – car body width for maximal deformation f_{max} [m],
h_{fi} – car body height for maximal deformation f_{max} [m].

Fig. 14a and Fig. 14b present the results of the analysis of the determined average car body stiffness parameters. Calculations were performed for approximately 50 different cars. The crash analysis were performed for the front part of the vehicles. The downloaded data correspond to the cars which are well-known in Europe. The results were presented as functions of the wheel base (Fig. 14a) and of the car mass (Fig. 14b).

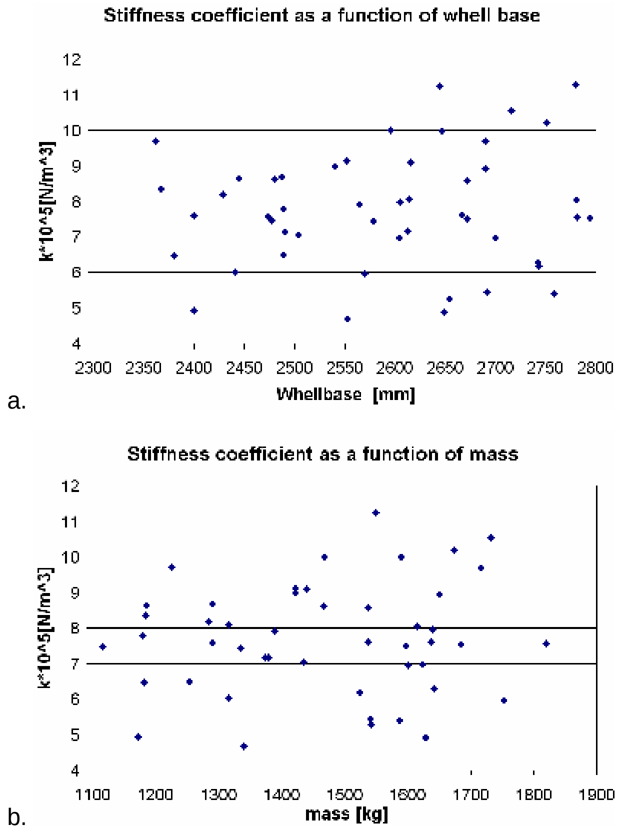

Fig. 14. Stiffness coefficients of car bodies as functions of: a. wheel base of the cars, b. masses of the cars.

The obtained results (Fig. 14a and 14b) prove low sensitivity of the car body stiffness coefficient on the investigated technical parameter for the analyzed group of vehicles.

4. Analysis of the car body stiffness coefficient and crush energy distribution

The car body stiffness coefficient and crush energy distribution were analyzed as well. The crash test data for the Dodge Neon vehicle and the corresponding results obtained in FEM simulation were analyzed. The computational analyses were performed for the 50 and 60 [km/h] impact velocity.

Fig. 15 presents the distribution of the load cells sensors used in the crash test.

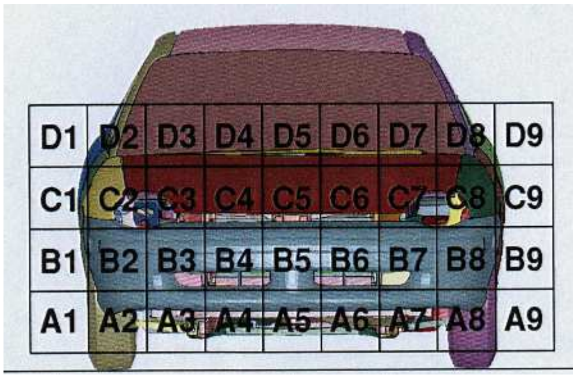

Fig. 15. Load cells localization

On the basis of the Dodge Neon crash test data [7] the distributions of the crush energy and the stiffness coefficient were presented in Figures 16 and 17. The numbering of sensors and the row naming correspond with Fig. 15.

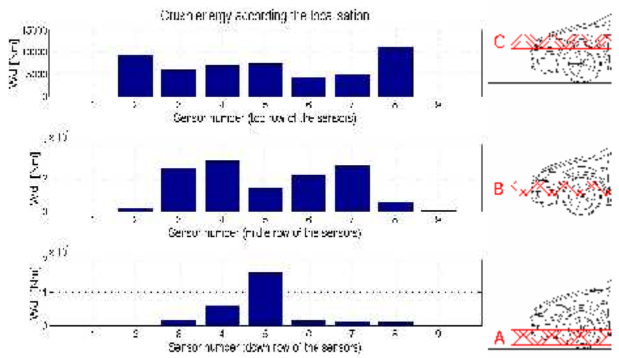

Fig. 16 Distribution of the deformation energy

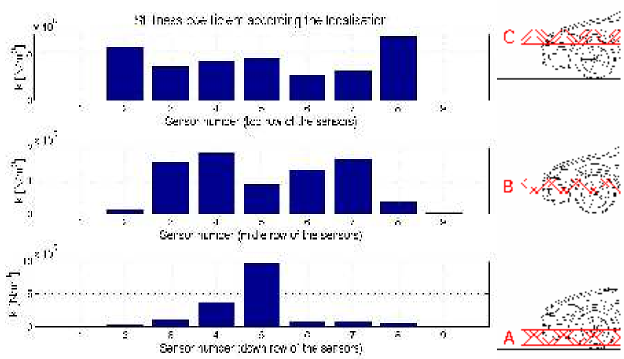

Fig. 17. Distribution of the stiffness coefficient

The distribution (for the whole car height) of the value of the stiffness coefficient is presented in Fig. 18. The results were obtained on the basis of the crash test data.

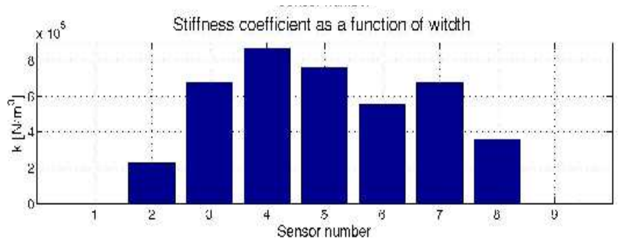

Fig. 18. Distribution (for the whole car height) of stiffness coefficient

The comparison of the results obtained by means of the FE analysis and in the crash test is presented in Fig. 19. The graph presents the distribution of the crush energy.

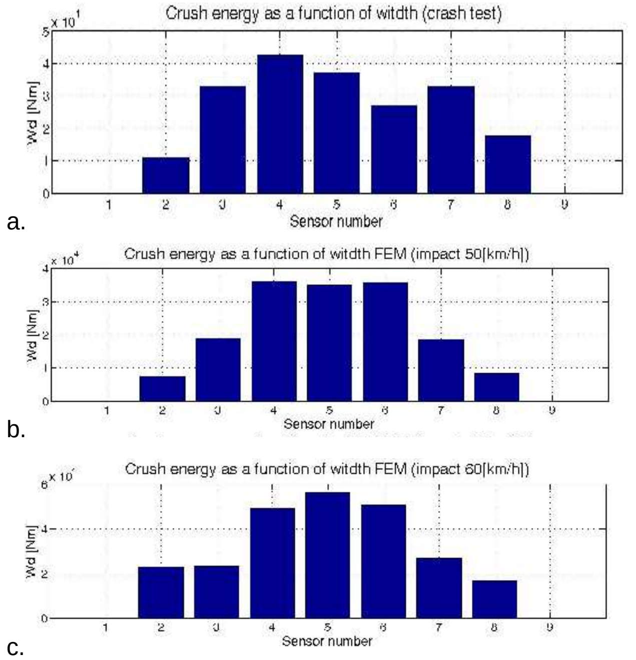

Fig. 19. Crush energy distribution: a. crash test V= 56[km/h], b. FE analysis V=50 [km/h], c. FE analysis V=60 [km/h].

The similarity of the results obtained by means of two mentioned methods is satisfactory.

5. Conclusions

1. The research proved the similarity of the results obtained in the crash tests and by means of the FE analysis. The simulation method of the frontal impact against a rigid wall and chosen results of calculations show that the FE analysis is useful in car research. Force and deformation observed during the impact enable to obtain the stiffness parameters of the car body. These parameters are important for accident reconstruction performed by forensic experts. The stiffness coefficients obtained by means of the analyzed approach can be used for verification of the parameters used in accident reconstruction.
2. For the analyzed car the linear dependence between crush depth and impact velocity was obtained.
3. The global and local stiffness coefficient are characteristic features of car body structure. Relatively large varieties of values of these parameters were observed for the analyzed vehicles.
4. Low sensitivity of the car body stiffness coefficient on the investigated technical parameter for the analyzed group of vehicles was observed.
5. Precision of the determination of the global and local stiffness coefficients influences the precision of estimation of the deformation energy and of the pre- and post-accident parameters of motion.

References

1. LS-DYNA Keyword user's Manual (Nonlinear Analysis of Structures), Livermore Software Technology Corporation, Livermore, California, March, 2006.
2. LS-DYNA Theoretical Manual, Livermore Software Technology Corporation, Livermore, California, May, 2006.
3. FHWA/NHTSA National Crash Analysis Center, "Finite Element Model of Dodge Neon", Model Year 1996, Version 7, http://www.ncac.gwu.edu/.
4. http://www.ncac.gwu.edu/vml/models.html
5. Wojciech Wach "Metody energetyczne w analizie zderzeń pojazdów" Zeszyty Naukowe Politechniki Świętokrzyskiej, Mechanika 2006.
6. Jeremy Daily, Russell Strickland, John Daily, „Crush Analysis with Under-rides and the Coefficient of Restitution", Institute of Police Technology and Management's 2006,
7. http://www-nrd.nhtsa.dot.gov/database/aspx/vehdb /testseries.aspx
8. Wiesław Grzesikiewicz, Lesław Kwaśniewski, Jarosław.Seńko, "Symulacja zderzenia samochodu ze sztywną przeszkodą", Wyd. Autostrady, 7/2007

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Ausgewählte Aspekte der Deformationsenergiebestimmung

Zusammenfassung

Im Beitrag wird eine Simulationsmethode zur Untersuchung von Frontalaufprallen gegen ein starres Hindernis dargestellt. Es wurde die LS-Dyna Software angewendet. Die Simulationen wurden für den Dodge Neon, Baujahr 1996, durchgeführt. Analysiert wurden die Fahrzeugbewegung, der Kraftverlauf während des Aufpralls bei verschiedenen Geschwindigkeiten. Darüber hinaus wurden die Resultate der Laboruntersuchungen des Aufpralls getestet und mit den mithilfe eines FEM Modells gewonnenen Resultaten verglichen. Es wurde auch die Abhängigkeit des Fahrzeugsteifheitskoeffizienten von den Fahrzeugstrukturparametern analysiert sowie die Deformationsenergie und der Steifheitskoeffizient untersucht.