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Finite element analysis of EPS foams used in bicycle helmets

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EXTENDED ABSTRACT

Introduction

Task of increasing protective properties of bicycle helmets is in the main scope of not only manufacturers of protective equipment, but also engineers and scientist all over the world. Recently, expanded polystyrene (EPS) foams became widely used as energy absorption material during impact, not only in bicycle helmets, but also in other protective equipment [1]. It is due to the fact of their unique material properties and low cost. In this paper, the parametrical study on EPS foams of different densities with and without polycarbonate (PC) outer shell is performed. Numerical simulations were performed using Abaqus software, where we conducted tests at different velocities and different impact angles. Output variables were translational velocities and accelerations and angular accelerations at the centre of gravity of head model, because these values are used in head injury criterions used for evaluation of possible head injuries that can occur as a result of such an impact. Comparison of peak values of accelerations, or velocities and duration of linear and angular accelerations at the centre of gravity of the head model was carried out. Based on the results some conclusions, concerning influence of density, presence of outer shell, impact angle, etc. were drown.

Finite element model and simulation parameters

For all the numerical analysis, ABAQUS was used as a solver and also as modeller because of the simple geometry of all instances. Numerical tests were done to assess suitability of material model with comparison to real data obtained from experimental compression test, which was carried out according to standards. Abaqus/Implicit was used for compression test simulation, where we modelled block of EPS with dimensions 50x50x23 mm and densities of 40, 80 and 120 kg/m^3 using CRUSHABLE FOAM material model for isotropic material with volumetric hardening [2, 3]. This kind of material model must be used with linear elastic material model, where we have to define Young's modulus and Poisson's ratio. Poisson's ratio is for crushable foams considered to be equal to zero [4]. Young's modulus values we obtained from experimental data and they can be found in Table 1.

Table 1 Young's modulus values for EPS of different densities

EPS density [kgm^{-3}]	40	80	120
Young's modulus	11	44	48

Further input data for this model were compression yield stress ratio k and hydrostatic yield stress ratio k_t, as we did not tested EPS in hydrostatic tension we used input data adopted by Mills **Chyba! Nenašiel sa žiaden zdroj odkazov.** , k=1.933 and k_t=1, which showed as reliable also for our material models. We also had to provide input for the hardening law by specifying value of yield stress in uniaxial compression as a function of the absolute value of the axial plastic strain. It was inserted in tabular way and those data were obtained from compression tests. For the dynamic analysis we had to take into consideration, that as a strain rate increase, EPS show increase in the yield stress. This increase is significant also in our case, so we defined strain-rate dependent material behaviour in Abaqus and we used tabular input of yield ratio, where rate-dependent behaviour is specified by giving a table of the ratio $r = \frac{\bar{\sigma}_c}{\sigma_c}$ as a function of the absolute value of the axial plastic strain, where r is the uniaxial compression yield stress ratio, σ_c is uniaxial compression yield stress at a given value of ε^{pl}_{axial} for the experiment with the lowest strain rate [3]. Input data for strain-rate dependency where obtained from experimental results.

Geometry of headform, anvil and EPS foam. The aim was to set all the parameters for this numerical study in such a manner that it will be efficient for computation as we had to run numerous set of analysis, our FEA assembly was based on experimental assembly, which consist of three main parts: headform, EPS foam sample and anvil.

Headform: As a Hybrid III 50th Male Dummy head (used in experiments) is much stiffer than EPS foam we considered it as a rigid body, and in our case we have decided to model a simplified spherical model, using analytical rigid part, which is computationally more efficient than using discrete rigid part, but at cost of being not able to fully copy headform geometry of Hybrid II 50th Male Dummy head. We decided to use this simplification as it was parametrical study, where we were focused on influence of changing parameters of EPS foam on linear and angular acceleration. Radius of spherical model was 85.7 mm and weight was 4.5 kg. As a reference point for this rigid body we set the centre of mass.

Anvil: Rigid steel anvil was also modelled as analytical rigid part as its stiffness is also much higher than stiffness of EPS. On anvil we put a reference point where we restrained all degrees of freedom with implied boundary condition.

EPS foam sample: EPS was modelled directly in Abaqus. It was modelled using deformable solid part, where we used material model for crushable foams with aforementioned properties.

Polycarbonate (PC) shell: The outer shell is modelled as linear elastic material and is perfectly bonded to outer surface of EPS foam. Its thickness is 0.5 mm. Material properties of PC were based on the data found in literature so Young's modulus was 7250 MPa, Poisson's ratio was 0.3 and density was 1200 kg/m^3 [7].

Meshing. As analytical parts does not need to be meshed, only EPS foam and PC shell was meshed. For EPS foam linear C3D8R elements were used, with distortion control on (we used default value of 0.1), which does not allow elements to invert during large deformations. Element size was set to 4 mm, with 11 elements throughout the thickness of EPS foam. These values were set according to mesh sensitivity analysis. Shell was modelled with four node doubly curved thin shell, reduced integration, hourglass control, finite membrane strain model elements S4R. Element size was set to 3 mm. On the Fig. 1 are shown meshed EPS foam and meshed PC shell.

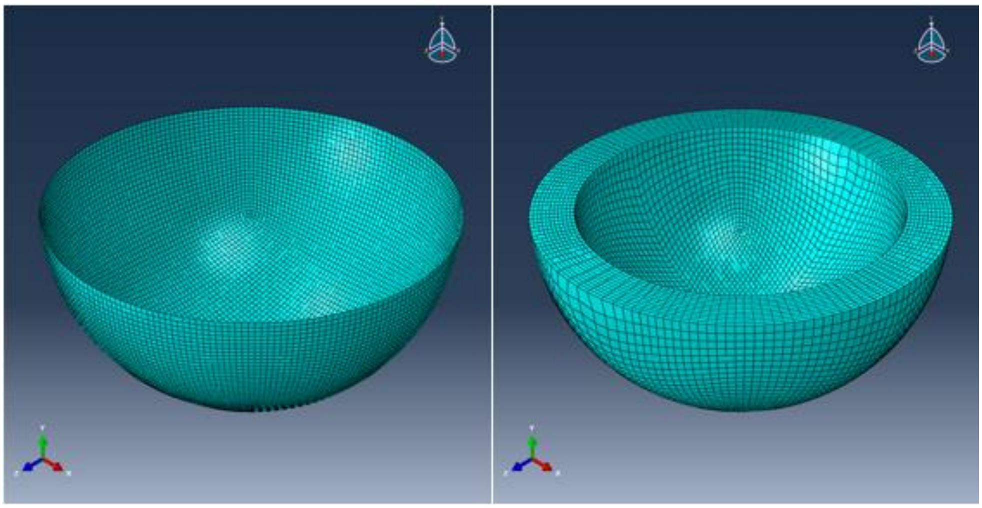

Fig. 1 Meshed PC shell (on the left) and meshed EPS foam (on the right)

Contact conditions. We used general contact which was satisfactory for our analysis. Friction in tangential direction was set to *f=0.2* for PC to anvil contact and *f=0.37* for EPS to anvil contact. In normal direction we used hard-contact option. EPS foam was connected to headform using coupling, where we restrained all the degrees of freedom (except rotations) for the surface which was in contact with headform.

Simulation parameters. To evaluate the energy absorption performance of the EPS foam with different densities of 40, 80 and 120 kg/m^3, impact simulations were performed with three impact positions: high energy (referred to 6500 mm/s), medium energy (referred to 5425 mm/s) and low energy (referred to 3962 mm/s). These impact positions were implied at the model in form of predefined field, that was defined on the whole model except the anvil and should simulate the state right before the impact. Whole time of simulation was 12 ms. For complete parametrical study we measured rotational and linear accelerations at the centre of mass of headform model as an output variables. In the Table 2 you can see all the parameters that were combined in this study [4, 5, 6].

Table 2 Numerical simulation parameters, that were involved in the study

<table>
<tr><th>Parameter</th><th colspan="6">Modification of parameter</th></tr>
<tr><td>EPS density [kgm^{-3}]</td><td colspan="2">40</td><td colspan="2">80</td><td colspan="2">120</td></tr>
<tr><td>Outer PC shell</td><td colspan="3">yes</td><td colspan="3">no</td></tr>
<tr><td>Impact velocity [ms^{-1}]</td><td colspan="2">3.96</td><td colspan="2">5.42</td><td colspan="2">6.5</td></tr>
<tr><td>Impact angle</td><td>90</td><td>75</td><td colspan="2">60</td><td>45</td><td>30</td></tr>
</table>

On the Fig. 2, it is shown basic setup for impact simulations, where all parameters are in bold and the output variables are in italic. Anvil and simplified head model were created as rigid bodies as their stiffness is much higher than stiffness of EPS.

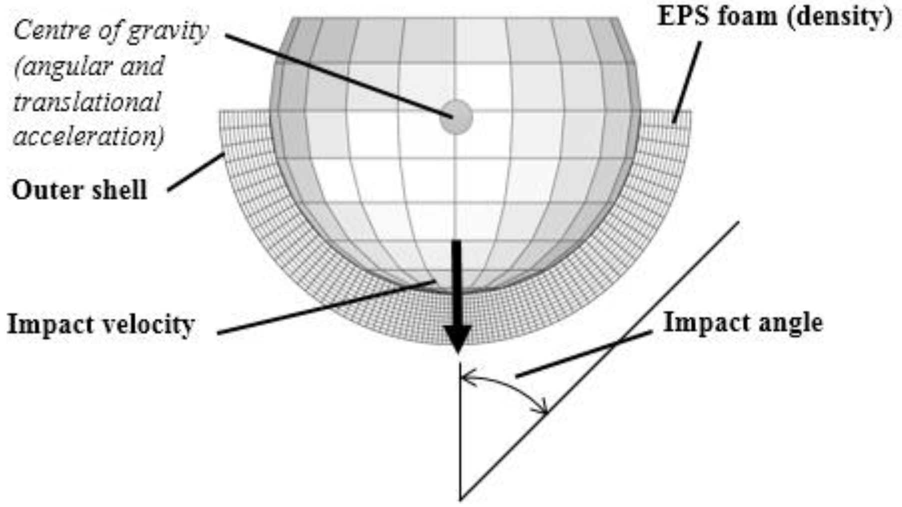

Fig. 2 Example of numerical model assembly with highlighted parameters of interest. Side view, EPS foam is meshed and in side section view.

On the Fig. 3 is shown assembly in Abaqus environment with predefined field of initial velocity of whole model and with boundary conditions implied on anvil. The anvil was rotated to match all the angles of impact that were of our particular interest.

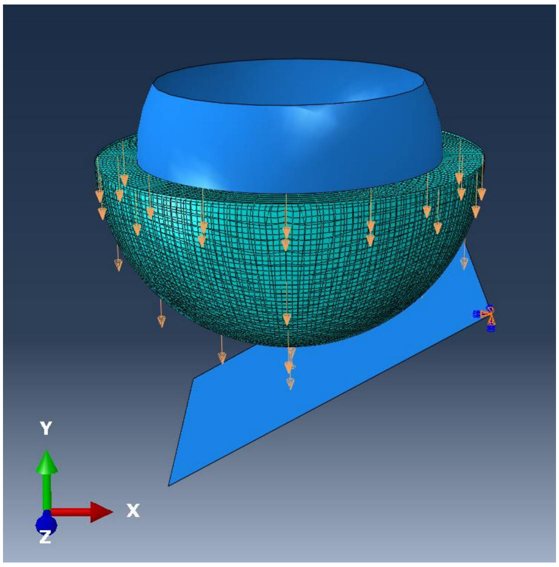

Fig. 3 Model in Abaqus environment with implied boundary conditions

Results

For complete evaluation of helmet quality is necessary to use one of the head injury criterions, which relate linear or rotational accelerations, or linear and rotational velocities, or other quantities like von Mises stress, principal strains or strain rate (depending on used criterion) with possibility of severe head injuries like skull fracture, focal injuries or diffuse axonal injury. Nowadays most of those criterions use either linear or rotational accelerations for their input (HIC, GAMBIT, HIP or some other), so for our parametrical study with simplified model we decided that we will focus only on linear accelerations in X and Y direction and angular acceleration around Z axis as all the other accelerations for our study are negligible. More information about head injury criterion can be found in literature [8]. There are two main factors which are crucial for head injury criterions. It is peak value of acceleration and its duration. In Table 3, 4 and 5 are shown all the peak values of accelerations.

Table 3 Resluts of peak angular accleration around z axis in rads^{-2}

	Shell	yes								
	EPS density	40			80			120		
	Impact velocity	L	M	H	L	M	H	L	M	H
Impact angle	90	41.2	206.8	36.4	7.4	13.4	20.0	47.5	30.0	18.0
	75	4691.1	6748.9	8157.3	7167.2	9976.4	11765.2	9176.2	12309.3	14756.6
	60	6801.9	9126.3	10950.0	10432.1	14435.2	17925.9	13006.2	18291.3	22147.0
	45	5769.0	7869.0	8949.0	8988.0	12344.1	15210.7	11575.1	16207.0	19108.5
	30	3811.8	4932.6	5676.6	6371.5	8720.7	10157.6	8225.0	11294.9	13515.1
	Shell	no								
Impact angle	90	27.9	38.2	89.3	53.0	39.9	54.1	86.0	107.5	92.8
	75	2361.5	3295.0	4256.5	4209.4	5612.5	6702.0	5922.6	7854.7	9255.2
	60	4546.9	6289.5	7878.6	7933.0	10697.3	12742.3	11032.2	14715.4	17537.0
	45	6182.0	8555.6	10448.6	10738.8	14468.3	17336.3	14538.6	19521.7	23273.2
	30	5425.37	7148.37	8293.41	9868.10	13120.6	15573.1	12915.1	17579.8	20727.1

Table 4 Resluts of peak linear accleration in x direction in ms^{-2}

	Shell	yes								
	EPS density	40			80			120		
	Impact velocity	L	M	H	L	M	H	L	M	H
Impact angle	90	1.3	3.3	0.9	0.2	0.4	0.7	0.9	0.7	0.8
	75	396.6	556.6	713.5	590.5	834.3	1014.5	768.9	1066.3	1291.0
	60	564.1	738.1	879.1	862.9	1176.4	1420.1	1056.8	1447.6	1742.1
	45	479.7	685.5	835.4	729.4	1021.3	1247.6	932.6	1313.3	1580.2
	30	424.7	582.8	715.5	676.2	933.5	1118.1	865.2	1202.4	1436.4
	Shell	no								
Impact angle	90	0.9	1.3	2.3	1.4	1.9	2.3	3.2	2.5	3.1
	75	250.7	385.3	717.4	429.4	586.8	708.3	569.8	775.0	932.9
	60	411.5	592.8	829.6	723.0	981.3	1176.5	963.4	1300.5	1559.0
	45	417.2	566.7	703.0	755.6	1006.4	1181.3	1004.8	1349.1	1591.3
	30	307.8	425.7	513.3	540.9	739.1	889.3	711.9	982.4	1177.7

Table 5 Results of peak linear acceleration in y direction in ms^{-2}

	Shell	yes								
	EPS density	40			80			120		
	Impact velocity	L	M	H	L	M	H	L	M	H
Impact angle	90	1328.8	1988.3	2574.4	1930.9	2829.8	3476.9	2382.6	3388.3	4172.6
	75	1210.5	1791.6	2354.0	1750.6	2564.8	3147.7	2164.9	3080.4	3820.5
	60	1008.7	1459.2	1851.3	1483.3	2117.6	2617.3	1865.2	2632.7	3219.3
	45	706.0	1010.0	1215.5	1061.7	1493.9	1825.6	1350.8	1894.0	2294.3
	30	363.3	494.4	611.3	574.8	795.2	957.0	733.0	1024.6	1228.5
	Shell	no								
Impact angle	90	1056.5	1722.3	3891.5	1705.3	2416.0	2972.2	2209.9	3116.7	3800.6
	75	989.1	1570.0	2757.5	1598.7	2269.9	2785.0	2074.1	2913.7	3568.1
	60	806.3	1219.7	1722.2	1324.4	1860.6	2295.5	1723.4	2402.7	2934.9
	45	594.0	856.7	1089.5	995.3	1391.7	1688.6	1301.1	1821.1	2187.8
	30	356.3	496.7	601.2	625.8	859.0	1034.9	818.9	1137.2	1361.8

After running complete set of sets few conclusions can be made based on these results. As we expected with increasing impact velocity and with increasing density of EPS foam both translational and linear accelerations are increasing. On the other hand peak duration with changing velocity is not varying too much, but with increasing density the peak duration is decreasing. This decrease is significant from EPS40 to EPS80, where peak duration is approximately 35 % shorter, whereas from EPS80 to EPS120 it is only around 12 %. There are few other interesting trends that are similar for EPS of all the densities and in this paper they are shown on the example of EPS80 on the following Fig.4. Based on this graphs we can see that for EPS without shell the most critical angle of impact is 45° on the other hand for EPS with shell critical angle of impact is 60° when combination of angular and linear acceleration has the highest values. Another important thing is that for impact angles 30° and

45°, EPS with shell has a lower values of peak accelerations that is because of lower friction coefficient between PC shell and steel anvil, then between EPS and anvil. Peak duration is affected by presence of shell in positive way (it means that peak duration is shorter) and with decreasing impact angle it decreases peak duration remarkably as it is shown on Fig. 5, where you have comparison of angular acceleration curves for EPS 80 at high impact velocity with and without shell.

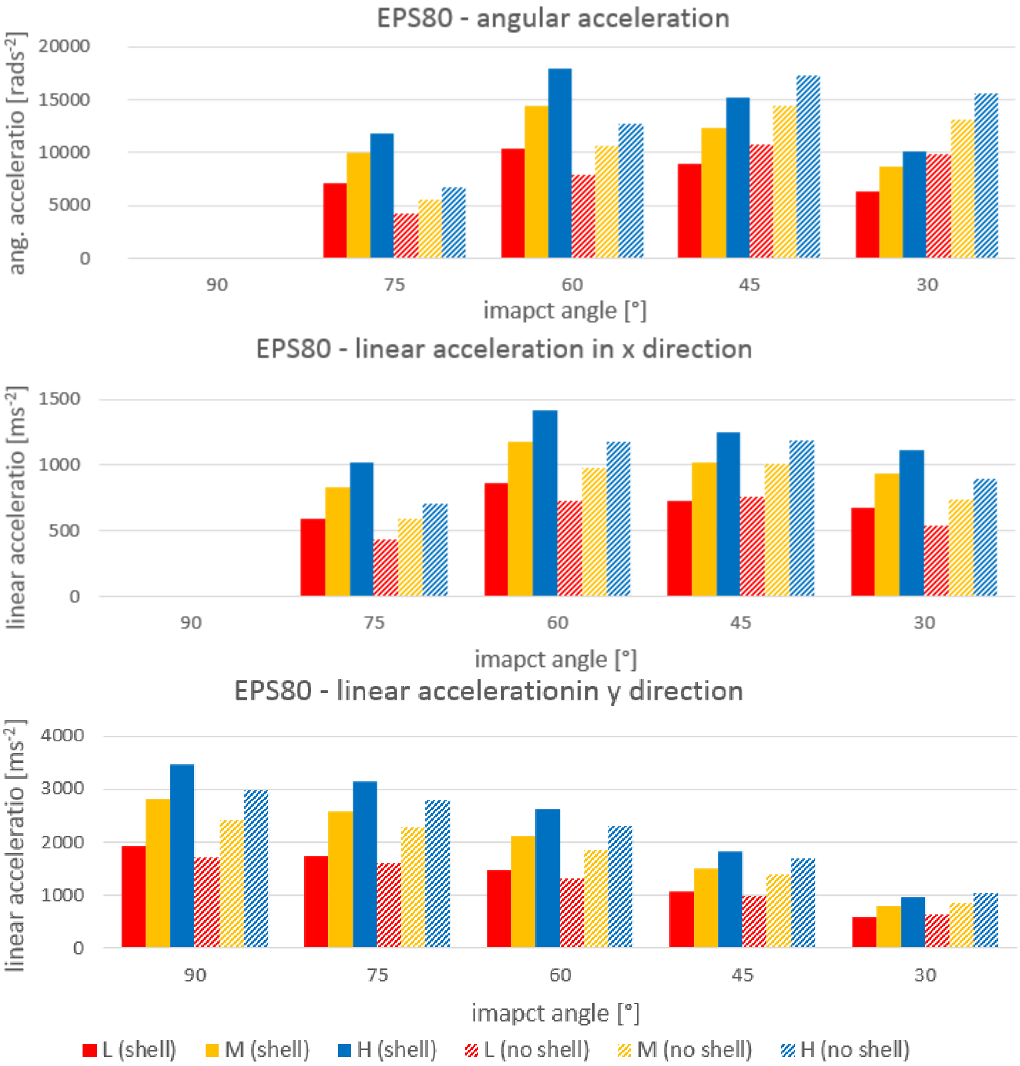

Fig. 4 Comparison of peak accelerations for EPS80

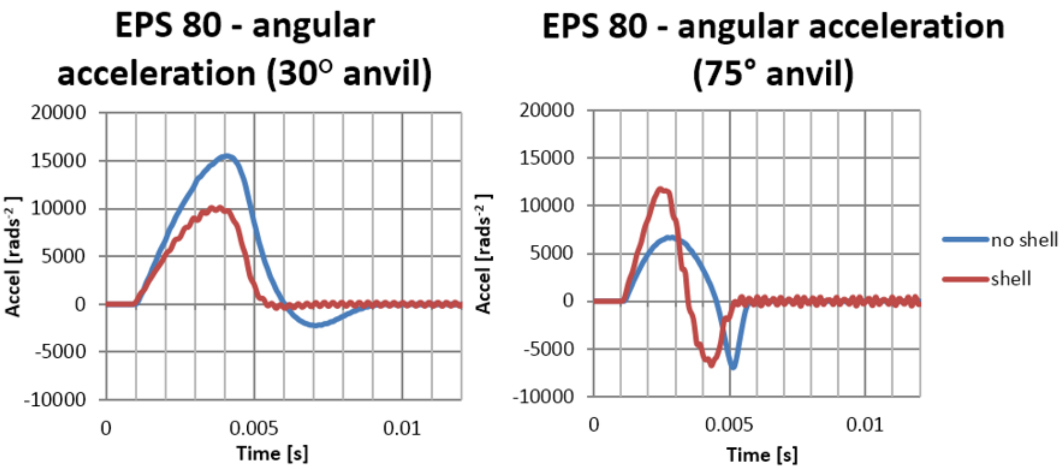

Fig. 5 Comparison of peak duration for EPS80 at high impact velocity

On the Fig. 6 and Fig. 7 is comparison of stress distribution at the cross-section of EPS80 and on the bottom part of EPS80, respectively shell in case with shell on EPS (Fig. 6) in the moment of peak value of accelerations.

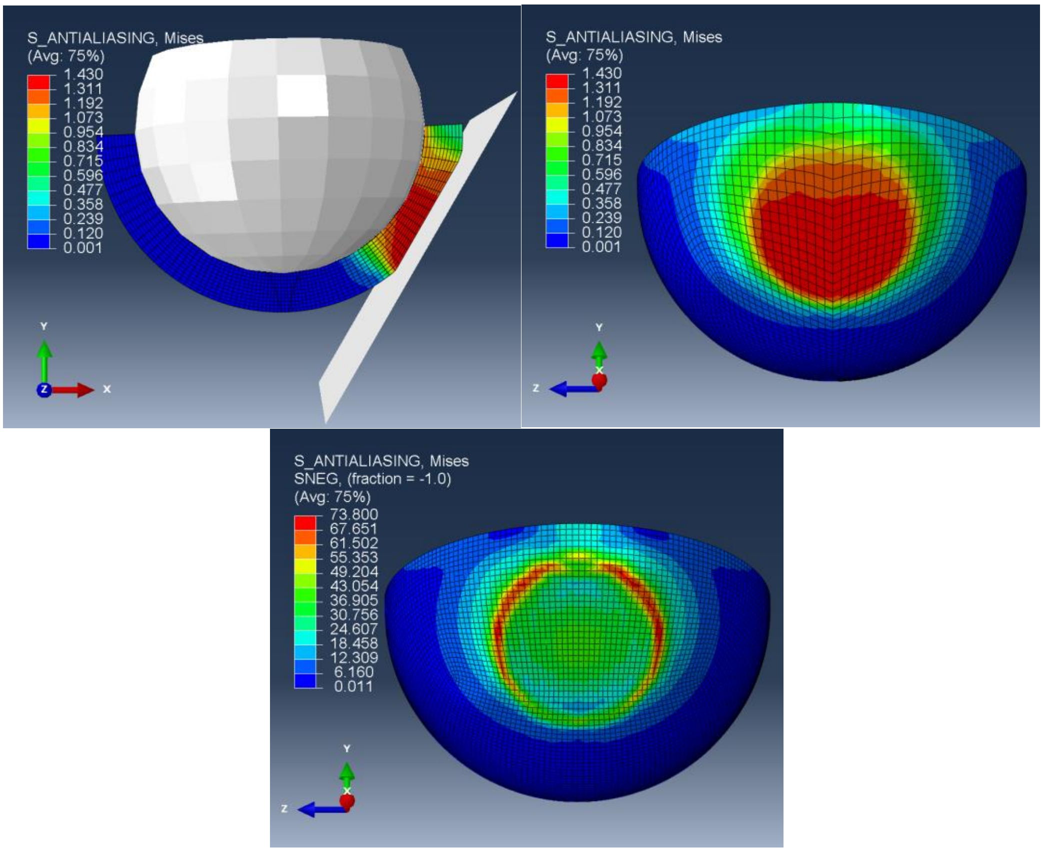

Fig. 6 Stress distribution for EPS 80 with shell. Cross-section (upper left picture), bottom view (upper right picture, bottom view of shell (lower picture)

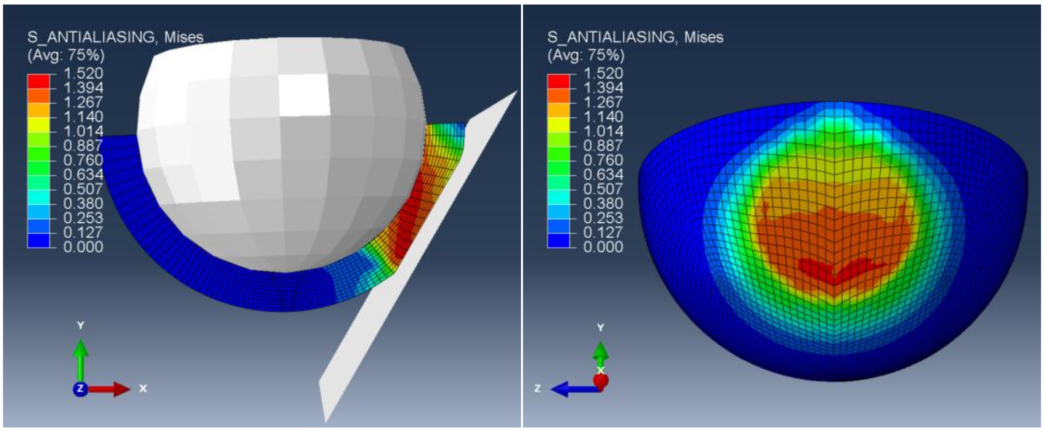

Fig. 7 Stress distribution for EPS 80 withou shell. Cross-section (left picture), bottom view (right picture)

Conclusions

Based on this simulations we got global insight on how different conditions affect duration and peak value of angular and translational acceleration at centre of gravity of simplified head model. Based on this observations we can differ more significant from less significant parameters. This work should lead to deeper understanding of the relations among parameters and can lay foundation for further research on other parameters as friction between head and

EPS foam, influence of retention straps, different materials from which outer shell is made of, etc.

Acknowledgement

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References

[1] B. Depreitere et al., Bicycle-related head injury: a study of 86 cases. In: Accident Analysis and Prevention, Vol. 36, No. 4 (2004), p. 561–567.

[2] V.S. Deshpande, N.A. Fleck, Multi-axial yield behaviour of polymer foams. In: Acta Materialia, Vol. 49 (2001), p. 1859-1866.

[3] ABAQUS. Abaqus analysis user‘s guide 6.13. [online]. Vélizy-Villacoublay: Dassault Systemes SIMULIA Ltd, 2013. Information on:
< http://129.97.46.200:2080/v6.13/books/usb/default.htm>

[4] N.J. Mills, A. Gilchrist, Finite-element analysis of bicycle helmet oblique impacts. In: Journal of Impact Engineering. Vol. 35, No. 9 (2008), p. 1087-1101.

[5] Q.H. Shah, A. Topa, Modelling large deformation and failure of expanded polystyrene crushable foam using LS-DYNA. In: Modelling and Simulation in Engineering. (2014), p. 1-7.

[6] K. Vanden Bosche, et al, Combination shear-compression testing of foam materials for their application in bicycle helmets or other complexly loaded structures. In: 18th International Conference on Composite Materials. Jeju, (2011).

[7] M. A. Forero Rueda, L. Cui, M.D. Gilchrist, Optimization of energy absorbing liner for equestrian helmets. Part I: Layered foam liner. In: Materials and Design. Vol. 30, (2009). p. 3405-3413

[8] W. Goldsmith, The state of head injury biomechanics: past, present and future (part 1). In: Critical reviews in Biomedical Engineering, Vol. 29, No. 5 (2001), p. 441–600