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Simulation and Numerical Analysis of Crashworthiness Behaviour of Thin Walled Structure

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ABSTRACT:

The purpose of this work is to find the specific energy absorption (SEA) of a steel tube using ABAQUS/CAE V6.10. Crashworthiness of a structure is characterised by the absorbed more amount of energy while the structure is subjected to an impact. A material structure should safeguard the occupants during an impact. The specimen geometry, processing conditions, and testing speed are the dependent parameters of energy absorption. It aims to determine the generalized mathematical model to evaluate the SEA and also find the parameter that is most likely affects SEA. Simulations are also carried out in ABAQUS/CAE to validate the developed numerical analysis.

KEYWORDS:

Thin walled structures; Crashworthiness; Crushing; ABAQUS/CAE; Specific energy absorption

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1. Introduction

Crashworthiness objective is to safeguard the occupant during an impact load that is acting on its structure by absorbing the impact energy. A great deal of research and development has been carried out in the past decades to design safer automobiles. The factors considered for safety criteria, the crashworthiness has having vital attention due to its numerous functions. The objectives of the crashworthy structures are to (i) absorb the impact energy, (ii) keep the occupant compartments safe and (iii) ensure acceptable deceleration levels for driver and passengers during the crash event. Geometry of material structure is a great influence on specific energy absorption (SEA) while an impact load is acting over its surface. Degree of curvature greatly influences the SEA. SEA of flat segments is slightly lower than the curved specimens. SEA is a structure property that is highly geometry dependent. This analysis was carried out in standard/explicit solver.

Modeling strategies require the control parameters that cannot be measured experimentally; those values are need to be calibrated by trial and error, and may have no physical significance. In this work, Johnson-cook material model was used as failure criterion and the constant values are taken from the previous work [11]. Many researchers have done to improve the energy absorption of these structures due to axial impact. However in the context of a vehicle collision, the vehicles energy absorbers are commonly subjected to both axial and oblique loads. Comparisons with axial loading conditions, relatively few numbers of studies have been carried on the energy absorption of thin walled tubes under off-axis loading conditions. The SEA capability is higher in the axial load when compared to the oblique loading condition, so in this paper a cylindrical thin walled tube was applied by an axial impact load. The SEA was calculated from the load vs. displacement curve which was obtained from the simulation. Impact load is applied on the inertia point on the axis of the cylindrical tube. In this study ABAQUS/explicit is used to solve the crush simulation of thin walled tube. ABAQUS/explicit is having the wide range of material failure models to solve these kinds of high energy impact loads. Nowadays researchers trend to move towards the computer simulation instead of experimental work. Finite element (FE) analysis is mostly used to optimize the work. In this study, FE analysis shows the response of the object when it was subjected to impact and also a generalized equation have been established to find out the SEA of a material structure using design of experiments software. The most influencing parameters were considered to develop the SEA numerical model.

2. FE modeling

ABAQUS/CAE version 6.10 has been chosen for this study. Because ABAQUS/CAE having the ability to solve the transient dynamic of the structures which subjected to impact loading. It is possible to solve complicated, very general, three-dimensional non-linear problems with collapsible bodies in ABAQUS/explicit. Problems involving stress wave propagation can be far more computationally efficient in ABAQUS/explicit. In implicit method the simulations takes several orders of magnitude. An explicit problem requires a small time step increment. ABAQUS/explicit is suitable to model high velocity dynamics. In this study, FE model of Steel tubes was developed using the non-linear FE code ABAQUS-explicit. The entire structure was modelled by using thin shell element which is having 4 nodes and 5 integration points along the axis direction. Johnson-cook material model was selected for steel tube. The both upper and lower plates are made up of rigid planner elements. A deformable 3D extruding surface was selected for making of steel tube specimen.

Both the plates are assembled on the upper and lower ends of the tube. The meshing was carried out with element size of 5mm for tube specimen, 10mm for upper and lower rigid plates. The tie constraint was applied between the bottom surface of the tube and the top surface of the bottom plate. This results in arresting all degrees of freedom at the bottom surface of the steel tube. The upper plate has free to move in axial direction and the remaining degrees of freedom were constrained. Two reference points were defined on the rigid plates. The first one on the bottom plate in which the reaction force was calculated. Another one was defined on the upper plate in which the inertia and velocity were defined. Load is applied at the inertia point and it is located at the center of the top plate. This can equally distribute the impact load to the wall surface of steel tube. The time period for crushing is given by 0.05sec and the velocity is given around 15m/sec. More time periods take more time to solve and it also depends upon the mesh size and number of integration points.

Rigid bodies are efficient analytical means for crushing simulations. The mass for shell element is given by 275 kg. The wide range of elements is available in ABAQUS which is used to create the models. Shell element has been used to model the thin-walled tube. The shell element is less than 1/20 of the height. Fig. 1 shows the FE model of the crushing tube. All the tubes have been modelled in this FE simulation were generated by using the S4R element. This element is a 3D doubly curved four noded shell element. Each node has 3 axes of displacement and 3 rotation degrees of freedom. The crushed model is shown in Fig. 2. While the two rigid plates modelled by using 3D 4 noded rigid element R3D4 type.

Fig. 1: Steel tube geometry and FE mesh of the model

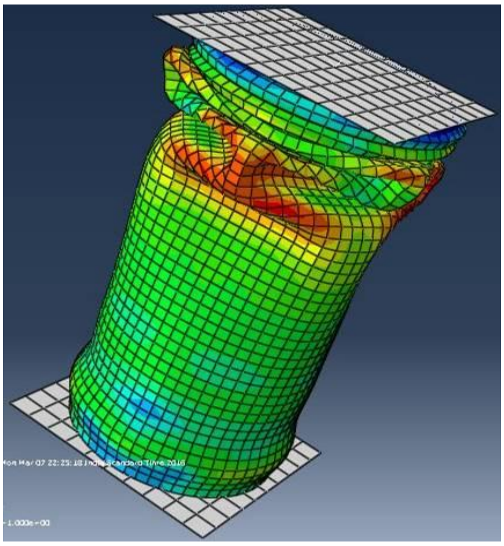

Fig. 2: Crushed FE model

3. Derivation of SEA

The essential feature of a member which subjected to a high velocity impact load is SEA that can be expressed as the Energy Absorbed (EA) per unit mass as,

$$SEA = EA/m \tag{1}$$

Where, m is the original non-deformed mass (before impact). The SEA of a structural component is a measure of energy absorption capability of the particular component. The higher value of SEA is an indicator of the light weight crush members. The SEA calculation done through the displacement vs. load graph which is shown in Fig. 3 based on the simulation results. The crushing speed decreases from the initial impact speed to rest as the specimen absorbs the energy. Area under the load curve is given by,

$$w = \int_0^{S_b} \overline{P} ds = \overline{P}(s_b - s_i) \tag{2}$$

Where w is the total energy absorbed in crushing of the tube specimen, $\overline{P}$ is the average crush load and S_b & S_i are the crush distances (mm). The SEA (E_s) is given by,

$$E_s = \frac{w}{m} = \frac{\overline{P}(s_b - s_i)}{V\rho} = \frac{\overline{P}(s_b - s_i)}{AV\rho} \tag{3}$$

Where A and L are the cross section area and crushed length respectively.

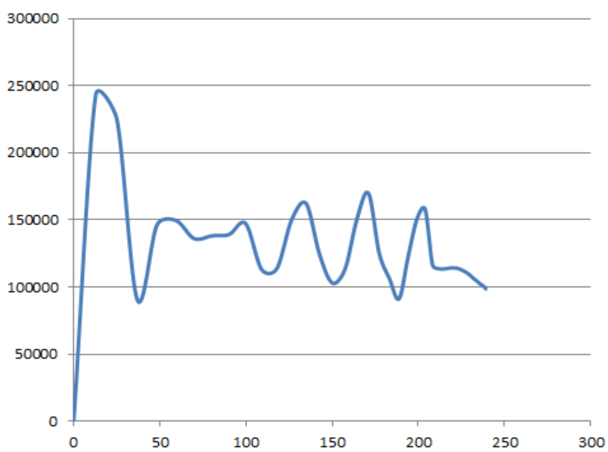

Fig. 3: Displacement (mm) vs. Load (N) curve

When S_i value is much lesser then the S_b, Eqn. (3) is simplified to,

$$E_s = \frac{\bar{P} S_b}{AL\rho} = \frac{\bar{P} k}{A\rho} = \frac{\bar{\sigma} k}{\rho} \tag{4}$$

Where $S_b/L = k$ is a measure of collapsibility of the tube and $\bar{\sigma}$ is the mean crush stress.

4. Optimization

The generalized equation for SEA was formed by using response surface methodology (RSM). Design expert V8.0.7.1 has been used for making the generalized equation. There are 3 stages and three levels of thickness (A), diameter (B) and length (C) were used to model this problem. These are the majorly influencing parameters for the SEA. SEA response from the RSM is given in Table 1. Totally 27 iterations were run to form the final equation. The final equation in terms of coded factors is given by,

$$\begin{aligned} SEA &= 25.31 + 2.13A + 0.53B - 0.09C + 0.50AB \\ &- 0.14AC + 0.070BC - 0.16A^2 + 1.44B^2 - 0.25C^2 \end{aligned} \tag{5}$$

The final equation in terms of actual factors is given by,

$$\begin{aligned} SEA &= 3117.21 - 74.43A - 68.13B + 0.34C \\ &+ 1.25AB - 0.04AC - 4A^2 + 0.36B^2 - 0.00063C^2 \end{aligned} \tag{6}$$

The model F - value of (23.52) implies that the model is significant. There is only a (0.01%) chance that a large "model F-Value" could occur due to noise. In this case A, B, AB, B^2 are the significant model terms. Reduction of model may improve the significance results. The obtained model shows 92.57% of R squared value, which confirms the reliability of the developed model.

Table 1: SEA response

Run	A: Thickness mm	B: Dia mm	C: Length mm	SEA
1	2.2	91	350	27.80
2	1.8	89	330	23.81
3	2	91	350	24.59
4	2	93	350	26.87
5	2.2	93	370	29.27
6	2	89	330	26.90
7	1.8	93	330	23.53
8	2.2	89	370	27.41
9	1.8	89	370	24.86
10	2.2	93	350	29.46
11	2	91	330	25.78
12	2	93	370	27.37
13	2.2	93	330	28.99
14	2.2	89	330	26.99
15	1.8	91	370	22.58
16	2	89	370	24.97
17	1.8	91	350	23.31
18	2.2	89	350	27.67
19	2	93	330	28.18
20	1.8	93	370	24.44
21	1.8	91	330	22.24
22	2	91	370	23.99
23	2.2	91	370	27.47
24	1.8	93	350	24.93
25	2	89	350	26.24
26	1.8	89	350	24.56
27	2.2	91	330	27.55

Table 2 shows the results of Analysis of Variance (ANOVA). The predicted and obtained values, as shown in Fig. 4, are in good agreement, which confirms that the RSM model created will be able to obtain the values at any intervals or range.

Table 2: ANOVA

Analysis of variance table					
Source	Sum of squares	df	Mean square	F-value	p-value Prob > F
Model*	103.19	9	11.46	23.51	< 0.0001
A - Thick	81.74	1	81.74	167.67	< 0.0001
B - Dia	5.13	1	5.13	10.53	0.01
C - Length	0.14	1	0.14	0.30	0.59
AB	2.98	1	2.98	6.11	0.02
AC	0.23	1	0.23	0.48	0.49
BC	0.05	1	0.05	0.12	0.73
A^2	0.15	1	0.15	0.31	0.58
B^2	12.35	1	12.35	25.34	0.01
C^2	0.37	1	0.37	0.77	0.39
Residual	8.28	17	0.48		
Cor Total	111.47	26			

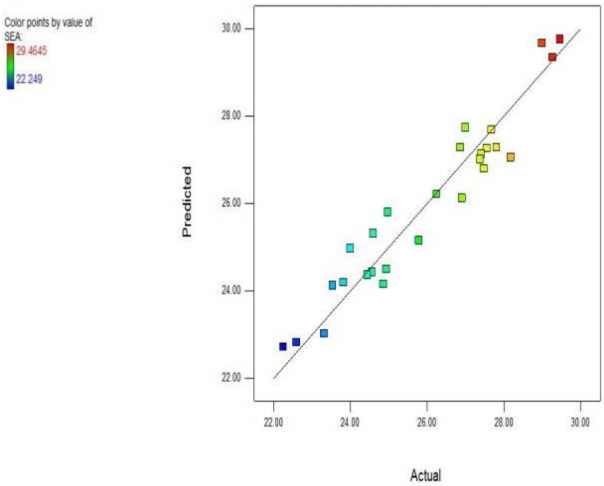

Fig. 4: Predicted vs. Actual values

5. Conclusions

The largest issue involved with the FE modeling of thin walled tube is attributed to the stability. The root cause of the excessive element distortion/rotation errors was never solved. Further research needs to be carried out in eliminating this error if the model is to become stable. Even though physical deformation predicted by the model closely matched, it proved difficult to predict the actual failure modes. SEA lies within an interval in all the simulated cases. The higher thickness, larger diameter and medium length show a good absorption of energy. It is to be noted that for the same thickness and diameter but with higher length there is a decrease in the energy absorption capability.

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