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Failure Analysis of an Energy Absorber Subjected to Pressure Pulse

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In the paper, an influence of the amplitude of pulse loading and the thickness of susceptible parts taking over the pressure pulse on the absorber operation is investigated. The presented absorber consists of six truncated conical shells covered with a thin plate. Elements of the absorber are made of an isotropic material with a bilinear strain–stress curve. A nonlinear contact problem for the triangular loading pulse was solved with the ANSYS code.

Keywords: Absorber, finite element method, conical shell, dynamic response

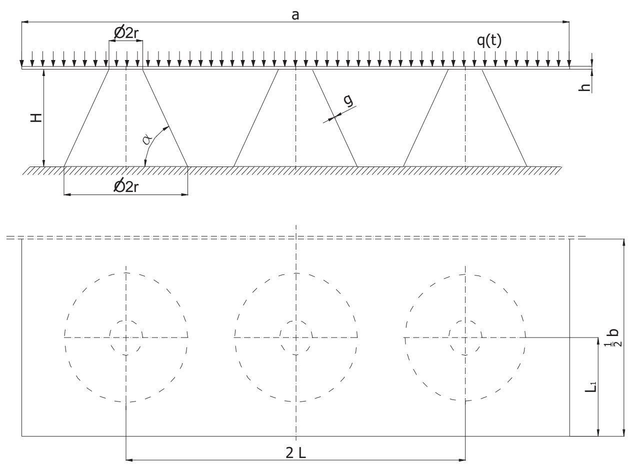
1. Introduction

An absorber as a structural part taking over the energy which comes from external impact loading is an integral element of machines and vehicles in which high safety of operation should be maintained. Its principle of operation is based on converting, partially or completely, the energy of pulse loading into the energy of plastic strain.

In the literature, most of works devoted to investigations of absorbers concerns energy absorbing pipes which have different shapes of cross–sections. The first research involving an analytical description related to pipe absorbers with a circular cross–section was initiated by Runtz and Hodge [3]. In [5], pipe absorbers with an elliptical cross–section were analysed, however, in [6] a pipe with the polyurethane filler was considered. The authors of that paper found a dependence of the force on shortening. In the present paper, numerical calculations are presented for an absorber built of truncated conical shells subjected to the pressure pulse. Pressure was applied directly against the plate which simply lay on upper edges of conical shells (Fig. 1). The pressure transferred to each shell, neglecting the inertia forces of the plate, was determined on the basis of the force equilibrium equation acting in the plate (Fig. 1):

$$q_{shell}(t) = \frac{q(t)ab}{6\pi r^2} \approx 64q(t) \tag{1}$$

The following thickness of the conical shell was considered: 0.25 [mm] or 0.5 [mm], whereas the thickness of the plate was equal to 0.25 [mm], 0.5 [mm] or 1 [mm]. The obtained computational results were compared with the results for a single shell without an external plate which was loaded with pressure on its upper surface.



R [mm]	28
r[mm]	5
α [°]	60
a [mm]	198
b [mm]	152
L [mm]	61
L_1 [mm]	38
H [mm]	40
h [mm]	0.25 0.5 1
g[mm]	0.25 0.5

Figure 1 Structure of the absorber – dimensions and the loading method

1.1. *Boundary and initial conditions*

The truncated conical shells were simply supported on their lower edges. It means that displacements of those edges along all directions were equal to zero:

$$u_x = u_y = u_z = 0 \tag{2}$$

1.2. *Shape of the pressure pulse and maretial propertties*

In the paper, an influence of the pulse shape on the behavior of shells is investigated. A triangular shape of the pulse, which is displayed in Fig. 2, was assumed.

A bilinear strain–stress curve of the ultimate strength is shown in Fig. 3. This material property was assumed for all parts of the absorber.

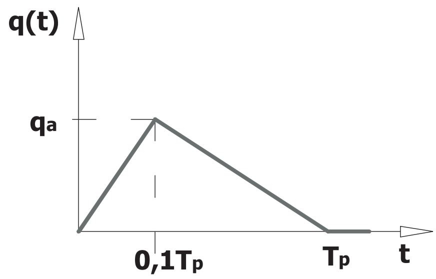


Figure 2 Pressure pulse shape: q(t) – pressure, T_p – duration of the pulse, q_a – amplitude of pressure the pulse

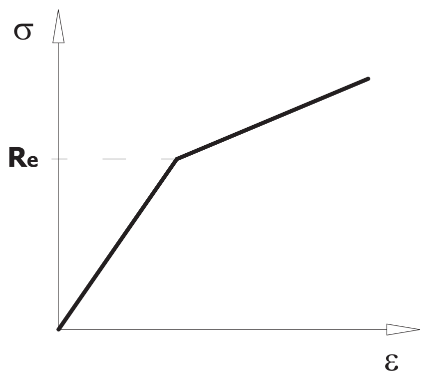


Figure 3 Bilinear strain–stress curve of the ultimate strength: R_e – yield stress

2. Discription of the numerical model and the computation course

The calculations of that problem were performed using the finite element method with the ANSYS 11.0 code. In the numerical computations, the same material for conical shells and the plate was considered. The issue should be treated as the nonlinear one because contact elements: CONTA175 and TARGET170, which were applied between the upper part of shells and the surface of the plate, were taken into account. With regard to a possibility of an interaction of the lateral wall of shells, contact elements were assumed on those surfaces as well. Elements of the SHELL43 type were attributed to conical shells, however, the whole volume of the plate was divided into solid elements of type 92.

In Tab. 1 material constants and physical properties used in the numerical computations are included.

3. Results of the calculations

The results of numerical computations for conical shells are illustrated as a dependence upon time of: the maximal effective stress σ_{eff} (Fig. 5), the maximal

Table 1 Material constants of conical shells and the plate: E –Young modulus; E_t – tangential modulus; R_e – yield stress; μ – friction coefficient

E [MPa]	ν g[t]	E_t [MPa]	R_e [MPa]	μ [-]
200000	0,3	10^3	200	0,1

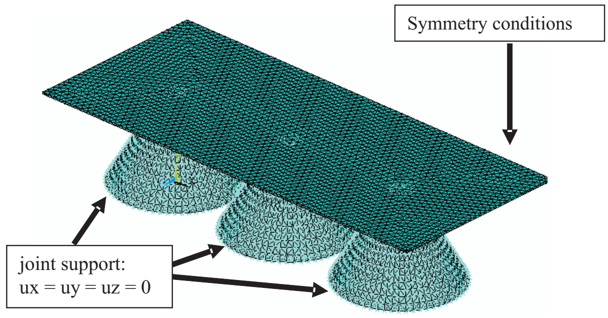


Figure 4 Numerical model of the absorber in the ANSYS code

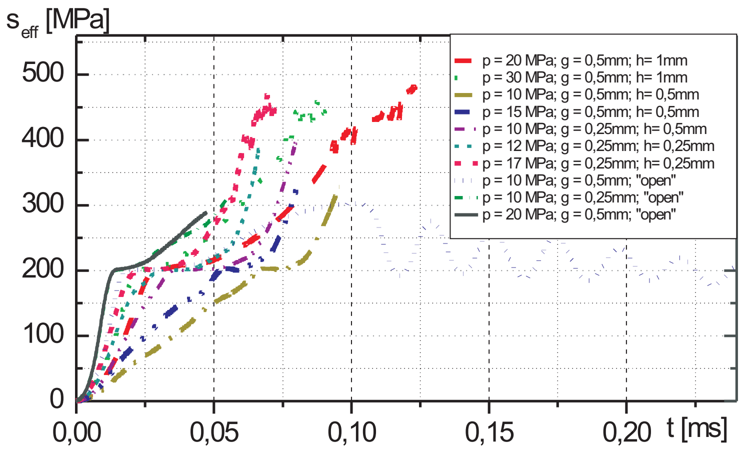


Figure 5 Curves of the maximal effective stress in the whole structure versus time for different pressure amplitudes

summary displacement u_{sum} (Fig. 6), the reaction of ground R_{sum} (Fig. 7), and the strain energy E_{st} (Fig. 8). The same thickness of all shells equal to 0.5 [mm] or 0.25 [mm] was assumed, whereas for the external plate the thickness amounted to 1, 0.5 or 0.25 [mm], respectively. The calculations were conducted also for a single truncated conical shell without an upper plate for the thickness 0.25 and 0.5 [mm] (See Fig. 1), further referred to as an ”open” one. An analysis of significant deformations of the structures was performed for different values of the pulse amplitude. The majority of the range of the obtained curves was not finished within the assumed duration of the pulse because large strains of the investigated structure did not allow for reaching the convergence of the solution. From the curves of the maximal effective stress (Fig. 5) it results that the yield stress is reached faster for the plate–shell structure together with the growth in the loading amplitude. However, one can observe that for shells of the same thickness, the effective stress reaches faster the yield stress for "pen" shells at lower amplitudes of the pressure pulse. These phenomena can result from the fact that "open" shells have lower stiffness than the "covered" ones. It follows that the external plate loaded with pressure acts mainly along the upper edges of the truncated conical shells and does not act directly in its cover.

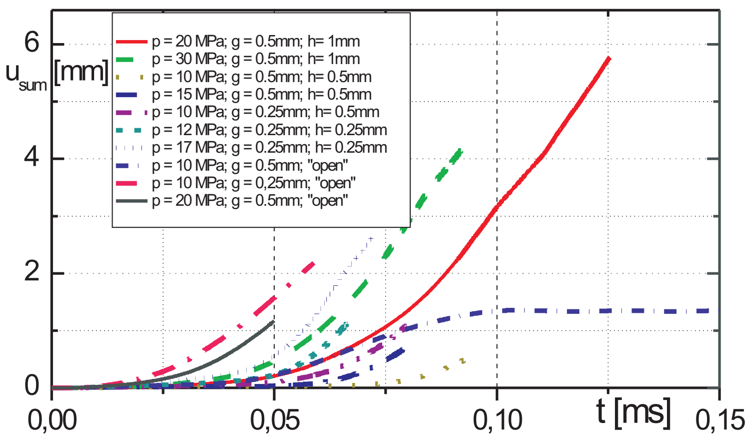


Figure 6 Curves of the maximal summary displacement in the whole structure versus time for different pressure amplitudes

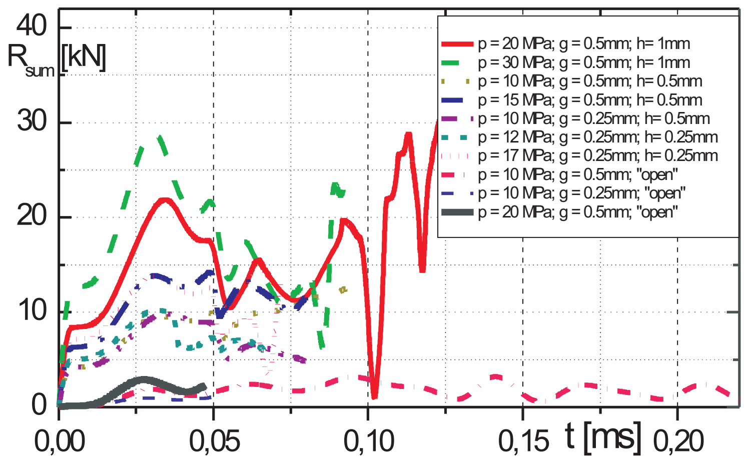


Figure 7 Curves of the ground reaction versus time for different pressure amplitudes

The thickness of the plate (located on the top of conical shells) can be treated as an additional parameter of stiffness of the whole structure. On the basis of the obtained results, it can be stated that the growth in the plate thickness causes the time prolongation of the shell reaction connected with its reaching the yield stress. In case of the summary displacements (Fig. 6), it can be noticed that the growth in the pressure amplitude for the "covered" shells influences an occurrence of large strain. Although "open" shells can carry a considerably lower pressure amplitude than the "covered" ones (about 60 times lower according to equation 1), they show a faster response to the pressure pulse, which can be rather connected with higher stiffness. A plot of the ground reaction (Fig. 7) was drawn for "covered" shells (for three conical shells) and for a single "open" shell. From the obtained graphs, it results that the thickness of the "covered" shells and the thickness of the upper plate does not affect the value of the ground reaction in the initial phase of the absorber destruction (up to 0.05 [s]). On the basis of the shapes of curves, it can been seen that the ground reaction in the first state of loading (before the stress exceeds the yield stress at any point of the structure) is equal to the pressure applied to the plate, though in all cases curves are rather chaotic.

The shapes of curves result from of a dynamic stability loss at high pressure amplitudes where the whole structure vibrates and the intensity of those vibrations is so large that after some time, in spite of a decrease in loading (according to the pulse shape), it is not possible for the structure to achieve the equilibrium state. In Fig. 8, curves of the summary strain energy depending upon time are shown. In this case the curves for appropriate pressures are similar to the curves determined by the summary displacements (see Fig. 6).

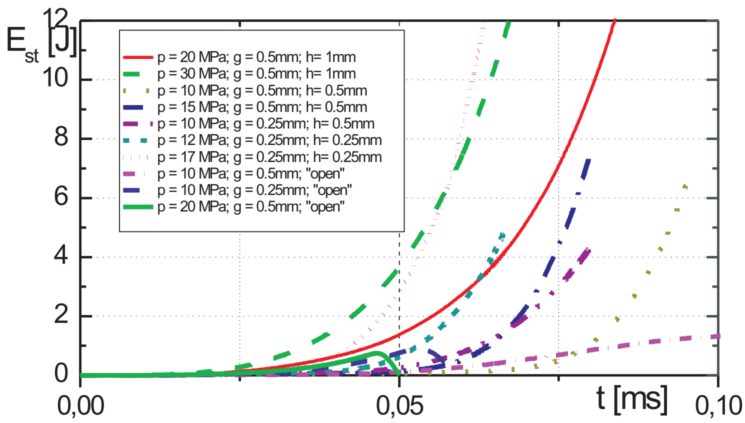


Figure 8 Curves of the summary strain energy versus time for different pressure amplitudes

4. Final conclusions

On basis of the analysis of the structure behavior, one can state that an application of the plate taking over the energy of the pulse increases the stiffness of the whole structure. Additionally, the use of the upper plate causes that the yield stress in the structure occurs later and decreases the velocity of the growth in displacements. It is possible to control such a structure through variations in thickness of the plate and shells. A change in those parameters does not influence the temporary value of the ground reaction in the initial phase of the absorber destruction.

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