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DYNAMIC RESPONSE OF CONICAL AND SPHERICAL SHELL STRUCTURES SUBJECTED TO BLAST PRESSURE

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Abstract. *The thin conical and spherical shell structures subjected to pulse loading were considered. The influence of shell geometrical parameters and pulse loading parameters such as its amplitude, direction and duration on the dynamic response was analyzed. The calculations were conducted for two material models: linearly elastic and elastic plastic. To solve the problem of dynamic response of shell structures the ANSYS software based on finite element method was employed.*

1 INTRODUCTION

The conical, spherical and of other shapes shell structures thanks to their advantages (lightweight, high strength, forming facility, etc) are very often used as a supporting structures in aerospace and automotive industry, shipbuilding and also mechanical and civil engineering. Recently such structures are also used as ballistic protections. This was the cause why the authors paid the attention to these structures and analyzed their behavior under pulse loading (blast pressure). It is well known that the thin-walled or/and thin shell structures are good energy absorbers especially when they loose their stability in elastic range and then plastic deformation appears leading to failure. To understand the behavior of thin shell structures subjected to blast pressure (pulse loading) the static buckling, dynamic buckling and dynamic response has to be investigated.

Since the 60's of previous century the problems of dynamic buckling of thin-walled and shell structures have been widely investigated by many authors. The well known papers dealing with dynamic buckling are written by Volmir [1], [2] Budiansky, Roth and Hutchinson [3], [4], [5] and recently by Simitses at el. [6], [7], [8], [9].

Volmir [2] proposed the criterion which allows to find critical pulse load leading to dynamic buckling. This criterion states that dynamic critical load corresponds to the amplitude of pulse load at which the maximum deflection is equal to some constant value k (usually shell thickness). The other criterion was formulated by Budiansky and Roth [3] and later by Budiansky and Hutchinson [5]. Their dynamic buckling criterion states that dynamic stability loss occurs when the maximum deflection grows rapidly with the small variation of the load amplitude.

Simitses [7] classified the dynamic buckling problem depending on the type of structures (plates, shells) and their static postbuckling behavior. He said that for plate structures which have stable postbuckling path rather dynamic response could be analyzed instead of dynamic buckling – critical condition leading to failure should be determined on the base of special criterion (two of them are mentioned above). He proposed the methodology to determine the critical condition for dynamic buckling of structures with static snap-through buckling. It should be also mentioned that Simitses in his two publication [6] and [9] described dynamic buckling problem for plate and shell structures based on more than 120 positions of literature concerning with this problem.

The dynamic buckling problem was investigated not only for steel but also for composite structures [9], [10], [11], [13]. These works deal with structures used in mechanical or civil engineering as a supporting structures but the structures used as a part of ballistic protections have many times smaller overall dimensions [14]. The dynamic buckling and dynamic response of thin-walled plate and shell structures was also investigated by many others authors [15], [16], [17].

It should be mentioned that the analysis of structures built of thin shells subject to pulse loading still needs a special attention with results interpretation.

The dynamic pulse buckling occurs when the loading process is of intermediate amplitude and the pulse duration is close to the period of fundamental natural flexural vibrations (in range of milliseconds). In such case the effects of dumping are neglected [9].

Similarly as in other papers dealing with dynamic buckling [4], [5], [6], [7], [10] the Dynamic Load Factor (DLF) was introduced. DLF is defined as a ratio of pulse load amplitude to the static upper critical load. The initial imperfection form of shell surface is assumed to be the same as the mode corresponding to the upper buckling load. The amplitude of the initial geometrical imperfection was normalized by thickness of analyzed structure.

2 PROBLEM FORMULATION

The present paper deals with a dynamic response of shell structures in shape of hemisphere and truncated cone closed by spherical cap or by flat circular plate (Fig. 1). The height of analyzed structures is kept constant while the wall thickness and the angle of incline of cone generator may vary.

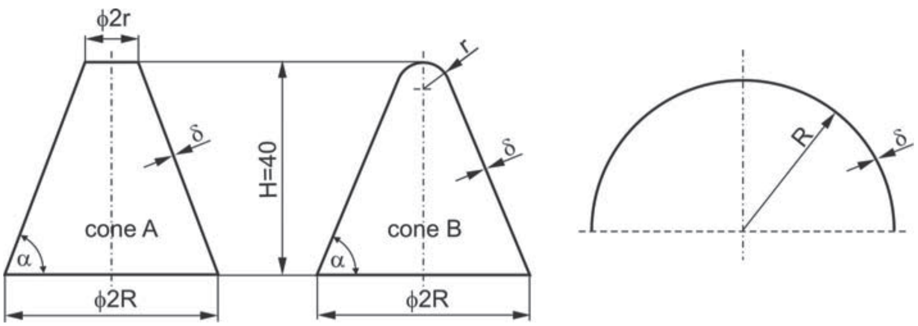

Fig. 1. Analyzed cones and sphere shells

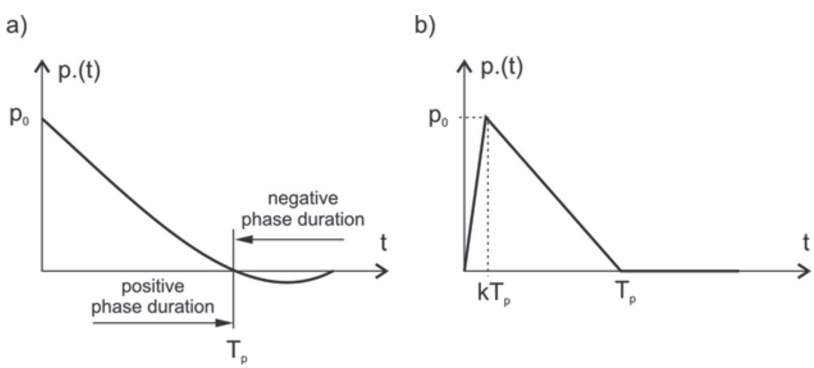

Fig. 2. Blast pressure diagram(a) and its approximation (b)

All taken into consideration structures are loaded by triangular pulse pressure of finite duration (Fig.2b). The shape of the pulse simulates the blast pressure (Fig.2a). The considered structures have such dimensions that their period of fundamental natural vibrations is close to the pulse duration T_p (Fig.2).

The different ways of pressure distribution were analyzed: the constant pressure in a given moment of time; the linear or sinusoidal distribution of pressure in given moment. The variable distribution of pressure value is used for modeling the directional wave of blast pressure (Fig. 3) defined by angle of inclination β.

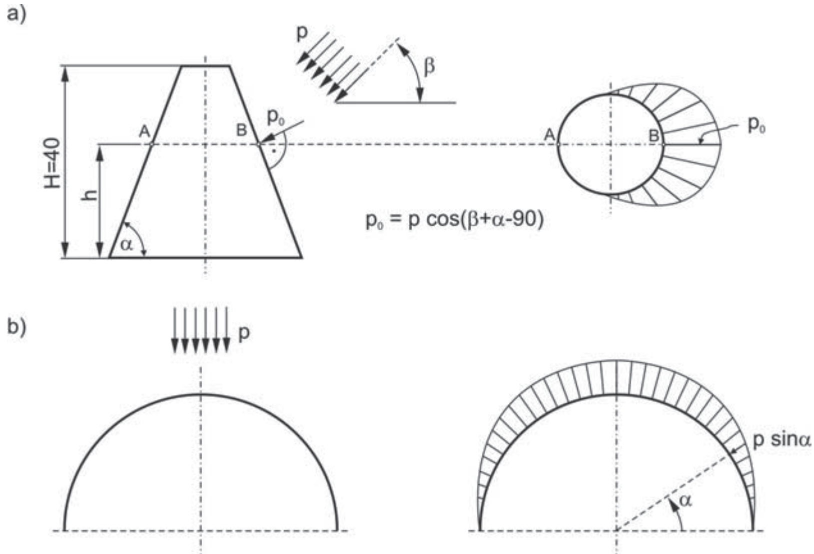

Fig. 3. The pressure value distribution depending on blast pressure direction

For all analyzed structures the material properties was assumed as for steel and they are as follows: Young's modulus $E = 2{\cdot}10^5$ MPa, Poisson's ratio $\nu = 0.3$, yield limit $R_e = 205$ MPa. The calculations are conducted for two material models: linearly elastic and elastic plastic with bilinear or multi-linear characteristic.

The influence of type of shell structures, their geometrical parameters and pulse loading parameters like direction, amplitude and duration on the dynamic response was analyzed.

In the first step of investigation the static buckling and modal analysis was performed. For static buckling problem the critical pressure and buckling mode was found, and from modal analysis the natural frequency was calculated. The upper buckling pressure values are used to define the DLF value applied in dynamic response analysis. Whereas the natural frequency is used to calculate the vibration period which should be close to pulse duration if the dynamic buckling problem is to be considered.

3 FINITE ELEMENT MODEL

To solve the problem of dynamic response of shell structures the ANSYS [18] software based on finite element method was employed. The structures under analysis were treated as simply supported along all bottom edges. For discretization the four-node shell element SHELL 43 [18] of six degrees of freedom was employed. Figs. 4 and 5 present correspondingly the finite element model (way of discretization) for conical and spherical shells. For hemispherical shell the symmetrical and non-symmetrical way of mesh was employed.

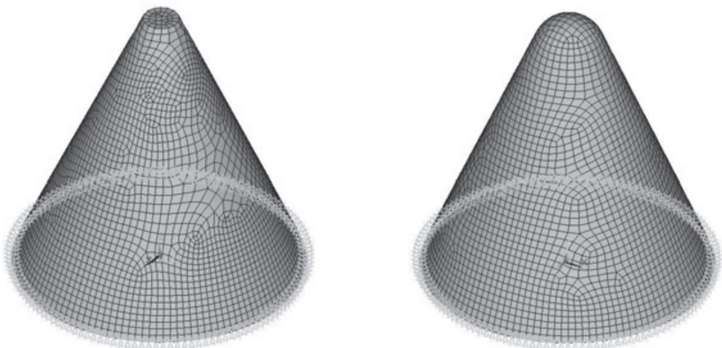

Fig. 4. Finite element models of analyzed cone shells

a) b)

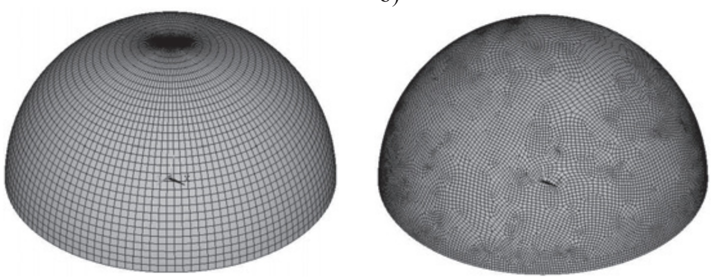

Fig.5. Finite element models of hemispherical shells with symmetric (a) and non-symmetric (b) mesh

a) b)

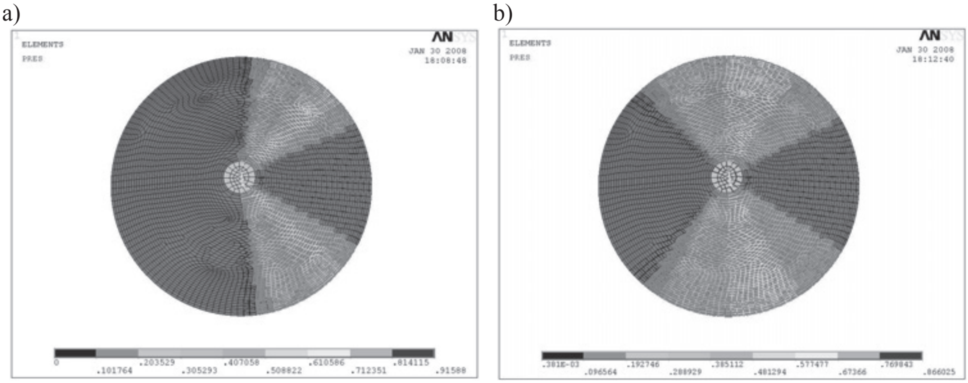

Fig.6. Exemplary pressure distribution for blast pressure inclination β = 0 (a) and β = 60 (b)

For uniform load distribution the load was applied to all elements with the same value at a given moment. Whereas, for non-uniform pressure distribution the load value at a given moment was applied to each element as calculated depending on element position in space and blast pressure direction. To calculate the element pressure value the APDL [18] (Ansys Parametric Design Language) was used. The exemplary pressure distribution for different blast pressure inclination are presented in Fig. 6.

In the numerical analysis the dynamic responses of shell structures loaded by pulse pressure were searched for. At the first stage the modal analysis was performed in aim to determine the period of natural frequency T_{ps}. Next, by liner stability analysis, using eigenvalue method, the critical static load and corresponding buckling mode were determined. The buckling eigen-mode with amplitude w_0 in non-dimensional form w_0/δ was assumed as the initial imperfection shape of shells. The amplitudes of the pulse pressure were applied as multiples of the static critical load. The structural dynamic analysis, which allowed to find the response of a structure for pulse loading, was conducted using the “Full Transient Dynamic Analysis” [18].

In the dynamic analysis the equilibrium equation was complemented by time dependent term and has the following form:

$$\{P\} = [M]\cdot\{\ddot{u}\} + [C]\cdot\{\dot{u}\} + [K]\cdot\{u\} \tag{1}$$

where [M] is a structural mass matrix and [C] is a structural damping matrix. In the dynamic buckling problem damping can be neglected [8] and then Equation (1) can be written as follows:

$$\{P\} = [M]\cdot\{\ddot{u}\} + [K]\cdot\{u\} \tag{2}$$

Substituting time derivative of displacement $\{\ddot{u}\}$ by increment of displacement $\{u\}$ in consecutive discrete instant of time *t* the new equilibrium equations included inertia forces are obtained for each time step. For the equation obtained for each instant of time the solution algorithms used in static analysis can be employed. In ANSYS software the Newmark method is used to integration over the time and for solving equations in consecutive instant of time the Newton-Raphson algorithm is used.

4 EXEMPLARY RESULTS OF CALCULATION

A lot of calculations for hemispherical shells and truncated cone shells closed by spherical cap or by flat circular plate was conducted. Some exemplary results obtained in the elastic range from static buckling, modal and dynamic response analysis are presented below.

4.1 Hemispherical shells

The hemispheres under analysis have radius *R* equal to 40 mm and different shell thickness *δ*. During analysis there was noticed that the buckling mode depends on the way of discretization. Completely different shape of deflection was obtained using symmetrical (Fig. 7a) and non-symmetrical (Fig. 7b) mesh. The axisymmetrical mesh forced the axisymmetrical buckling mode. Also it should be mentioned that consecutive critical pressure are close to each over – first ten critical pressure causing buckling differ less than 3% but all of them have different modes.

a) b)

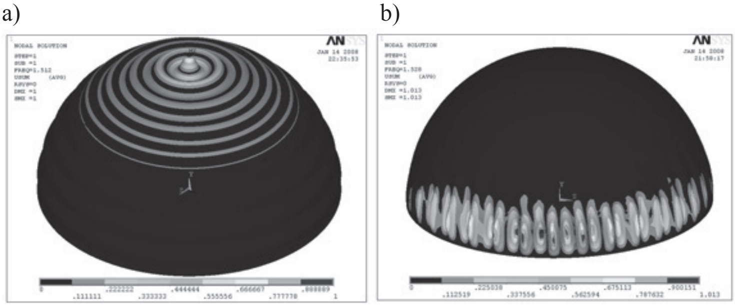

Fig.7. Critical static buckling mode for symmetric (a) and non-symmetric (b) mesh

The relation between static upper buckling pressure and shell thickness for hemisphere with non-symmetrical and symmetrical (Fig. 5) mesh is almost the same. The maximal equivalent (Huber – Misses stress) stresses for hemisphere with shell thickness $\delta \leq 0.2$ mm subjected to critical pressure are greater than 500 MPa. It means that such thin shells are disqualified because then structures under analysis loss their stability in plastic range. Further only hemispherical shells with thickness $\delta \geq 0.2$ mm are considered because such structures loss their stability in the elastic range.

The relation between dimensionless maximal deflection (w/δ) and DLF (Dynamic Load Factor) for hemispherical shell with wall thickness $\delta = 0.3$ mm subjected to uniform pressure pulse with duration equal to the period of natural vibration ($T_p = 0.09$ ms) and parameter describing pulse shape k = 0.5 was calculated and presented in Fig. 8. The shape of initial imperfection is the same as buckling mode and the amplitude of initial imperfection is assumed as 1/100 of hemisphere thickness.

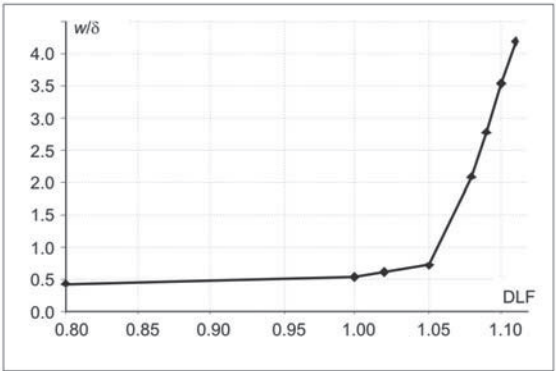

Fig. 8. Dimensionless deflection as a DLF function for hemispherical shell with $\delta = 0.3$ mm

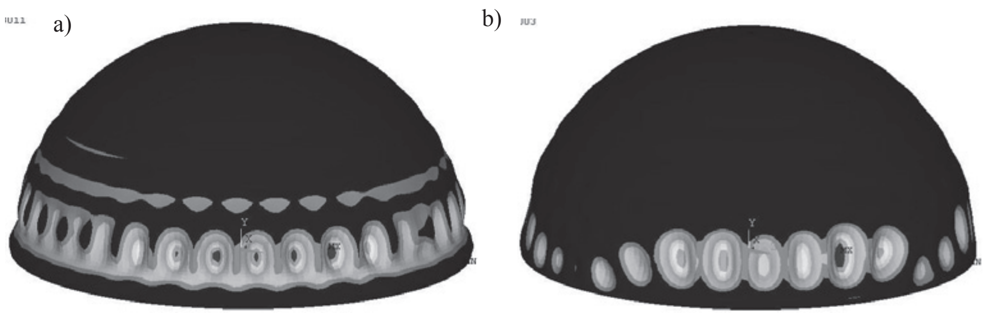

Fig. 9. Deflection in time of hemisphere for DLF = 1.08 a) t = 0,22 ms; b) t = 0.28 ms.

On the basis of the results presented in Fig. 8 the critical value of DLF can be found. According to Volmir criterion the $DLF_{cr} = 1.06$ and according to Budiansky-Hutchinson criterion the critical DLF equals 1.1. It means that amplitude of triangular pressure pulse greater about 10% than static critical pressure leads to dynamic buckling according to Volmir or Budiansky-Hutchinson criterion.

In Fig. 9 the shape of shell deflections for different moment of time are presented.

4.2 Conical shells

The static critical pressure and natural frequency as a function of shell thickness for two different cones (Fig. 1) with r = 5 mm and R = 10 mm or 25 mm are presented in Fig. 10. According to the obtained results it can be said that they do not depend on the shape (spherical cap or flat circular plate) which closes considered truncated cones. The smaller bottom radius R the higher static critical pressure. For both types of analyzed cones with R = 10 mm the natural frequencies do not depend on wall thickness – the cones have global vibration mode. Taking above into account it can be said that the cones with higher bottom radius are better as energy absorbers. The shell thickness should be chosen (the same as for hemispheres) to ensure that dynamic buckling occurs in elastic range.

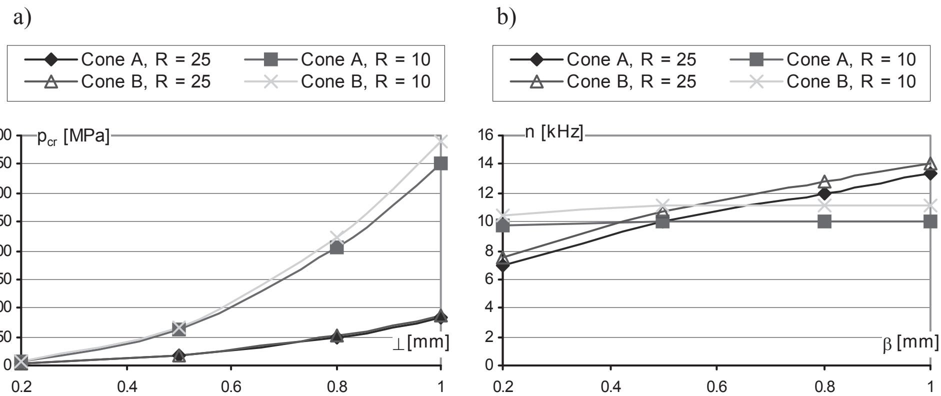

Fig.10. Critical pressure p_{cr} (a) and natural frequency (b) as a function of conical shell thickness δ

The relation between dimensionless displacement (w/δ) and Dynamic Load Factor for both type of cones (A and B) with radiuses R = 20 mm and r = 5 mm, shell thickness δ = 0.2 mm subjected to uniform pressure pulse loading with period of pulse duration T_p equal to 0.3 ms and parameter describing pulse shape (Fig. 2) k = 0.5. The shape of initial imperfection is the same as buckling mode and the amplitude of initial imperfection is assumed as 1/100 of shell thickness.

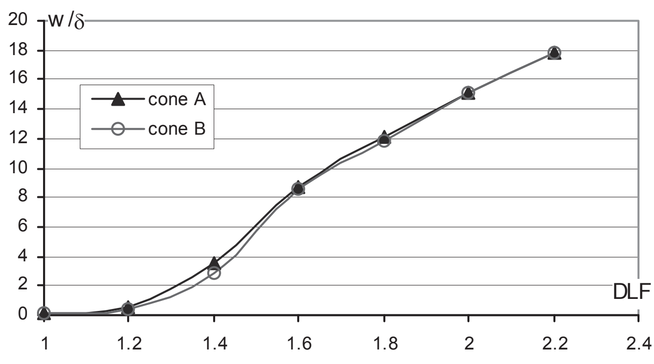

Fig. 11. Dimensionless deflection as a DLF function for conical shell (δ = 0.2 mm)

For both types of cones similar results were obtained. According to Volmir criterion the DLF_{cr} = 1.25 and according to Budiansky-Hutchinson criterion the critical DLF equals 1.5.

5 CONCLUSION

The analysis of the calculation results shows that the dynamic response of conical and spherical structures subjected to pulse pressure of finite duration strongly depends on geometrical parameters. Also material model and the direction of pulse loading have the influence on dynamic response of shell structures – the results of calculations which prove the above sentence will be presented during conference. The proper selection of geometrical parameters for conical or spherical structures (rational design) may reduce the structure sensitivity to pressure direction or/and distribution and allows to control the level of energy absorption.

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