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COMPARATIVE STUDY OF THE DEFORMATION STATE FOR THE OPTIMIZED DESIGN SPHERE-CYLINDER SHELL GRP UNDER INTERNAL PRESSURE

Dora FLOREA

STUDIU COMPARATIV AL STĂRII DE DEFORMAȚII PENTRU DESIGNUL ÎNVELITORII SFERĂ-CILINDRU GRP SUB PRESIUNE INTERNĂ

Acest articol prezintă un studiu relativ la deplasarea totală a unei învelitori cilindru-sferă GRP de grosime constantă supusă unei presiuni interne, comparativ cu deplasarea totală pentru opt profile optimizate ale aceluiași tip de învelitoare supusă la aceeași presiune internă. Concluzia este că deplasarea totală este mică pentru învelitoarea optimizată prin metoda propusă de autoare [4], pentru valoare minimă a tensiunii echivalente von Mises semnalată la analiza stării de tensiuni, deplasarea totală $\delta^*_{max\ cylinder}$ = 0,673 mm reprezentând aproximativ jumătate din valoarea deplasării totale pentru învelitoarea neoptimizată $\delta^{i}_{maxcylinder}$ = 1,244 mm.

This article presents a study relative to the total displacement of a cylinder-sphere covers GRP constant thickness subjected to internal pressure, as compared to the total displacement profiles optimized for eight of the same type of parcel subject to the same internal pressure. The conclusion is that the total displacement is small for optimized by method of covering proposed by author [4], for the minimum value of the voltage equivalent to von Mises condition analysis of reported tensions, the total displacement $\delta^*_{max\ cylinder}$ = 0.673 mm representing about half of the total displacement no optimization $\delta_{imaxcylinder}$ = covering 1.244 mm.

Cuvinte cheie: deplasare totală, învelitoare cilindru-sferă, GRP, profil optimizat

Keywords: total displacement, covers the cylinder, the sphere, GRP, optimized profile

1. Introduction

At the designing of the shell under pressure, the principal condition is that the shell not change the elements size subject the different loads, and for the high temperature the principal condition is the plastic deformation to be under dangerous limit [7]. So it is very important to know the displacement of the shell under the load.

In this paper is an analysis the displacement for the number of seven optimized designs of the sphere-cylinder shell GRP under internal pressure comparative with the same shell but the constant thickness. For the study used the graphics of displacement belongs the nodes of the finite elements used the FEA analysis with programmer NASTRAN V4.0. For the stress analysis used ax symmetrical linear elements CTRIA3 and quadratic elements CTRIA6, with 6 respective 12 degrees of freedom, corresponding the displacement's u_i, v_i in each node [1].

2. Theoretical consideration

2.1. The characteristics of the finite elements used to FEA analysis

The problem of the stress distribution in the body of the axisymmetric revolution under axisymmetric load is proving to practical interest considerable. In the same situation are the displacement components in a section through the symmetrical axis strain from structure. Because of the symmetry, two displacement components u, v in any section of the body plan, belong the symmetry axis, defined complete the specific deformation and the stress state [6].

In figure 1 is presents a revolution body with the symmetrical axial load.

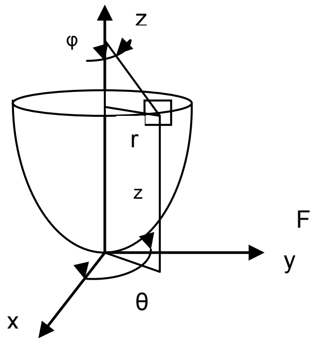

Fig.1 Element of an axisymmetric shell

Analysis made with the finite elements of special type-axisymmetric elements of the ring formed with constant transversal section. For this element, the nodes are nodal circle with the center on the symmetry axis z. The transversal section of the ring element is defined in a transversal section which has the symmetrical axis. Because of the symmetry not exist the displacement on the circumferential direction. The displacement vector has the components u and v on the directions r and z.

2.2. Displacement function

If used a triangular element with nodes i, j, m in the inverse hour hand, the nodal displacement it defined with two components:

$$a_i = \begin{vmatrix} u_i \\ v_i \end{vmatrix} \tag{1}$$

$$a^e = \begin{vmatrix} a_i \\ a_j \\ a_m \end{vmatrix} \tag{2}$$

and the displacement of triangular element through the vector:
The field of the displacement [6] in the interior of element is defined with linear polinom, and so the field of displacement is:

$$U = \begin{vmatrix} u \\ v \end{vmatrix} = \left[IN_i, IN_j, IN_m\right]a^e \tag{3}$$

$$N_i = (a_i + b_i r + c_i z)/2\Delta \qquad N_j = (a_j + b_j r + c_j z)/2\Delta \tag{4}$$

$$N_m = (a_m + b_m r + c_m z)/2\Delta$$

where I is the identity matrix and 2Δ is the area of the triangle ijm.

2.3. Strain

It consider three components of the strain, although exist all the components of the strain in axisymmetric dispacement. Figure 2 show and define this strain and the associate stresses [6]. The strain vector is defined through the list of the strain components involveed and defines them in terms of the displacement of a point.

(5)

$$\varepsilon = \begin{vmatrix} \varepsilon_z \\ \varepsilon_r \\ \varepsilon_\theta \\ \gamma_{rz} \end{vmatrix} = \begin{vmatrix} \frac{\vartheta v}{\vartheta z} \\ \frac{\vartheta u}{\vartheta r} \\ \frac{u}{r} \\ \frac{\vartheta u}{\vartheta z} + \frac{\vartheta v}{\vartheta r} \end{vmatrix} = Lu$$

Used the displacement functions defined with (3) results:
In the relation (6) B_i is presented in (7).

$$\varepsilon = B a^e = [B_i, B_j, B_m] a^e \qquad (6)$$

2.4. Stress

The relation between the stress and strain in the case of isotrope matherial [6] is presented with the relation (8). In the FEM analysis it determines first the nodal displacement and the field displacement in the internal element, then the stress σ and strain ε**.** In the figure 2 is an illustration the strains associate stresses in the analysis of axisymmetric shell.

3. Theoretical results with programme NASTRAN V4.0

The structure sphere-cylinder shell with the initial profile studed is show in figure 3 and the profile optimized through the method J. Leach is show in the paper [4] and the profiles optimized through the method Dora A.T. Florea studied in this paper was show in the papers [2], [3].

$$B_i = \begin{vmatrix} O & \frac{\vartheta N_i}{\vartheta z} \\ \frac{\vartheta N_i}{\vartheta r} & O \\ \frac{1}{r} N_i & O \\ \frac{\vartheta N_i}{\vartheta z} & \frac{\vartheta N_i}{\vartheta r} \end{vmatrix} \tag{7}$$

$$\sigma = \begin{vmatrix} \sigma_r \\ \sigma_z \\ \sigma_\theta \\ \sigma_{rz} \end{vmatrix} = \frac{E}{(1+\mu)(1-2\mu)} \begin{vmatrix} (1-\mu) & \mu & \mu & 0 \\ \mu & (1-\mu) & \mu & 0 \\ \mu & \mu(1-\mu) & 0 & 0 \\ 0 & 0 & 0 & \frac{1-2\mu}{2} \end{vmatrix} \begin{vmatrix} \varepsilon_r \\ \varepsilon_z \\ \varepsilon_\theta \\ \gamma_{rz} \end{vmatrix} \tag{8}$$

The stress and strain state it achieved with the programme NASTRAN V4.0 for the shell with the matherial characteristic E=7E+3 (module Young of elasticity), μ = 0.34 (Poisson = s coefficient) and the function parameters p = 0.4 MPa (pressure). The programmer NASTRAN V4.0 materializes the displacement diagram for the nodes of finite elements and these used to study the displacements in the profiles. The results for the total displacement (total translation) for the eight models are presented in the Table 1.

From the Table 1 it show that the maximum value of the displacement is 1.365 mm and this is in the model 7 optimized with the first theory of resistance, following the model 2 optimized with the four theory of resistance, following the model 2 optimized with the four theory of resistance in which the maximum value of the total displacement is 1.249 mm, than initial model 1 which has a maximum displacement of 1.244 mm. All these displacements with maximum values are on the interior contour in the cylinder. The maximum value of the variation band of the total transition is in the initial profile (model 1) and is 0.695 mm.

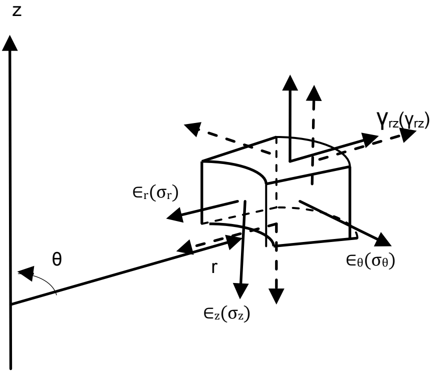

Fig. 2

Strains and stresses in the element of axisymmetric shell

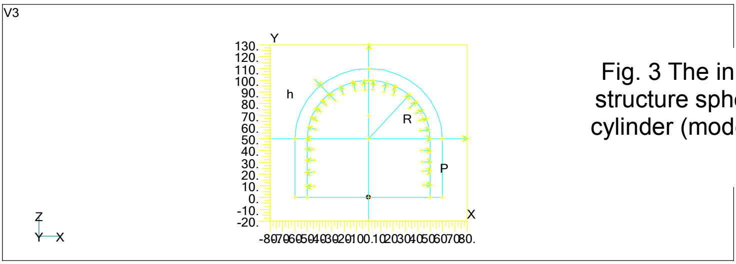

Fig. 3 The initial structure sphere-cylinder (model 1)

The minimum values for the maximum total displacement are in the profile - model 3 optimized with the four theory of resistance for a small value of a equivalente von Mises stress through the method Dora A.T. Florea [4] and is 0.673 mm on the interior contour.

The small values for the total displacement are and the models 5 and 6 (optimized with the five theory of resistance, variants 2 and 3) in which the maximum total translation is 0.674 mm.

Relative to the profile J. Leach (model 8) is remark that the maximum displacement is 1.246 mm and with a band of the displacement of 0.291 mm. The displacements in the sphere are more small than from cylinder and in the joint are sudden displacement with diminished on the cylinder. The difference the maximum displacements between cylinder 1.246 and sphere 1.075 on the interior cylinder is 0.171 mm so the profile J. Leach occupy the two place.

The minimum value of the difference displacement between cylinder-sphere on the interior contour is 0.127 mm which is achieved of the optimized profile (model 3), the same value is for the two variants studded (model 5, 6). So the model 3, 5, 6 are the first place, because this minimum difference of the displacement mark a uniform displacement for the models.

It remark that the model 2 (Table 1) after the optimized the initial profile (model 1) presents a variation band of the total displacement decreased at half for the interior and exterior contour, the maximum value of the total displacement is 1.249 mm beeing the results of the imposed condition certain to realize a equivalents von Mises stress equal with the maximum value belong initial model, so the total displacement is 1.249 mm keep neighbouring the total displacement 1.244 mm of the initial model 1 (Table 1).

The conclusion is that model 3 (Table 1) presents more small value of the maximum total displacement 0.673 mm and so the variants 2 and 3 (model 5 and 6) with 0.674 mm displacement, the difference of the displacement between cylinder and sphere beeing 0.127 mm.

Table 1

No.	Model	Total displacement		Band of the variation of displa-cement col.(3-4) cont.int./ cont.ext.	Maximum value of the displa-cement (interior contour of cylinder)	Maxi-mum value of the displa-cement (Interior contour of sphere)
		interior contour maxi-mum/ minimum	exterior contour maxi-mum/ mini-mum			
0	1	2	3	4	5	6
1	initial [2]	1.244 0.549	1.229 0.541	0.695 0.648	1.244	0.667
2	optimized I A [3]	1.249 0.903	1.235 0.895	0.346 0.340	1.249	1.028
3	optimized II A [3]	0.673 0.496	0.658 0.486	0.177 0.172	0.673	0.546
4	optimized II A (1) [3]	0.871 0.438	0.853 0.441	0.433 0.412	0.871	0.55
5	optimized II A(2) [3]	0.674 0.492	0.659 0.482	0.182 0.177	0.674	0.547
6	optimized II A(3) [3]	0.674 0.492	0.659 0.482	0.182 0.177	0.674	0.547
7	optimized B [3]	1.365 1.027	1.35 1.017	0.338 0.333	1.365	1.17
8	J. Leach [5]	1.246 0.955	1.235 0.946	0.291 0.289	1.246	1.075

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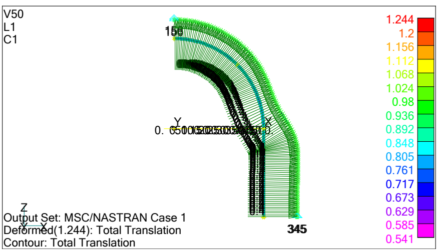

Fig. 4 Total displacement for the model 1

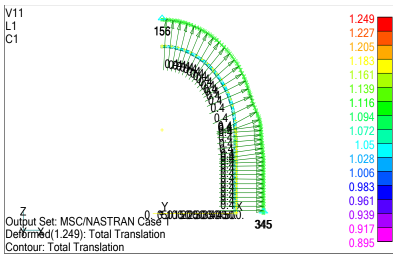

Fig. 5 Total displacement for the model 2

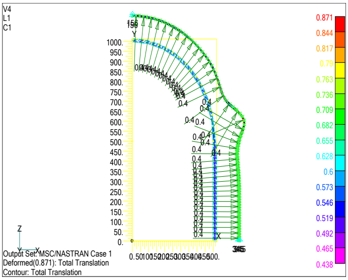

Fig. 6 Total displacement for the model 3

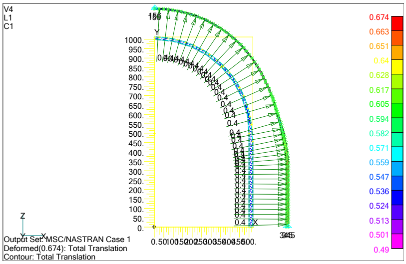

Fig. 7 Total displacement for the model 4

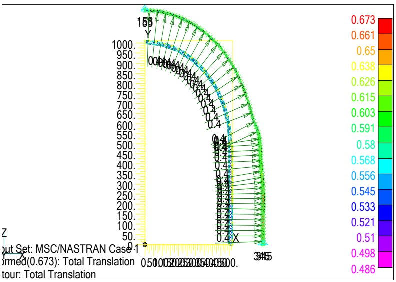

Fig. 8 Total displacement for the model 5

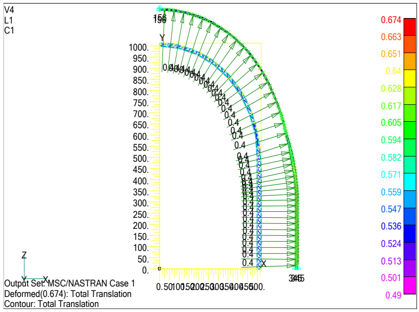

Fig. 9 Total displacement for the model 6

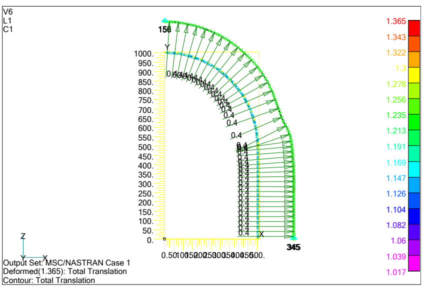

Fig.10 Total displacement for the model 7

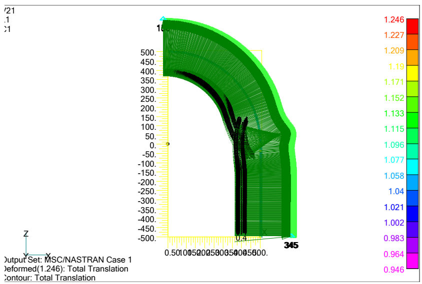

Fig.11 Total displacement for the model 8

Fig.12 Graph of the total displacement for the model 3

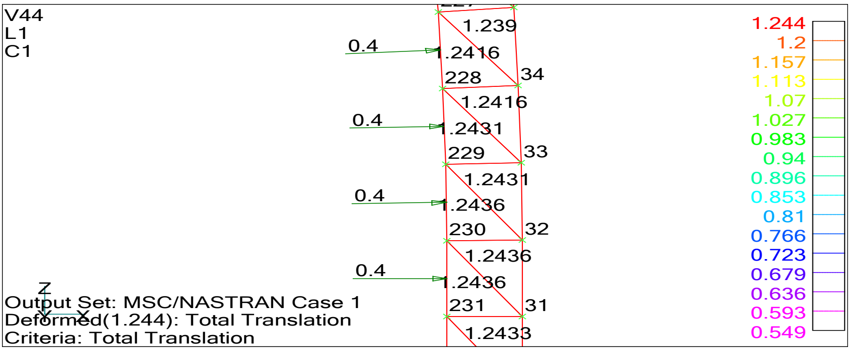

Fig.13 Element 197 with nod 230 with the displacement 1.244 for the model 1

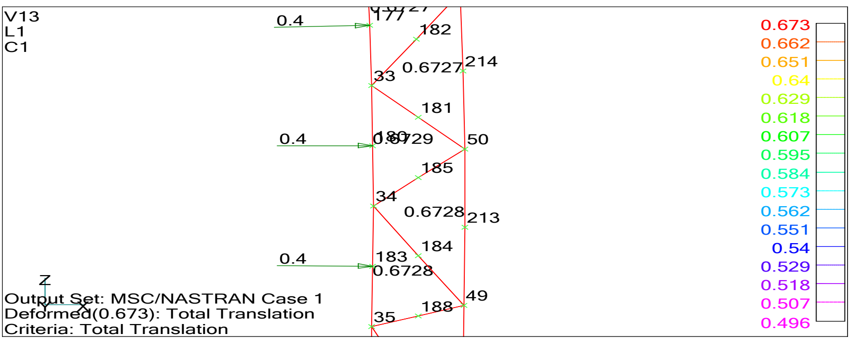

Fig.14 Element 16 with node 34 for the model 3 (Table 1) with the displacement 0

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OPTIMIZATION IN THE ELASTIC LINEAR FIELD OF THIN SHELL CYLINDER-SPHERE GRP UNDER SMALL INTERNAL CONSTANT AND UNIFORM PRESSURE

Dora FLOREA

OPTIMIZAREA ÎN CÂMP ELASTIC LINIAR A UNEI ÎNVELITORI CILINDRU-SFERĂ GRP SUPUSĂ UNEI PRESIUNI INTERNE CONSTANTE ŞI UNIFORME

Articolul prezintă un studiu cu privire la optimizarea unui profil de învelitoare cilindru-sferă GRP (Glass Reinforsed Plastic), calculată în domeniul elastic linear pentru o valoare mică de presiune constantă și uniformă. Se prezintă rezultatele cercetărilor făcute de J.Leach [4] pentru câteva profile de învelitoare, relativ la starea de tensiuni și deformaţii utilizând programul BOSOR [4] bazat pe metoda diferenţelor finite. În continuare se prezintă rezultatele cercetărilor făcute de autoare relativ la trei învelitori cilindru-sferă, optimizate prin impunerea a trei tensiuni echivalente von Mises constante pe întreaga învelitoare, utilizând metoda propusă de autoare [2] care are la bază ecuaţiile de echilibru ale teoriei clasice în domeniul elastic linear. Analiza stării de tensiuni și deformatii este realizată cu ajutorul analizei FEM utilizând programul NASTRAN V4.0 [5], pentru cele trei situaţii de optimizare ale învelitorii cilindru-sferă prin programul OPTIMISES [3] rezultate prin impunerea a trei tensiuni echivalente von Mises σ_{Vme} = 5 MPa, σ_{Vme} = 9,007 MPa, respectiv σ_{VMe} = 17,96 MPa.

La sfârșit se face o analiză cuprinzând rezultatele opţinute relativ la tensiunile circumferenţiale, echivalente von Mises, și relativ la deplasarea totală, pentru profilele optimizate, evidenţiind rezultatele obţinute prin metoda propusă de Dora Florea [3].

The article presents a study about optimizing a parcel profile cylinder-sphere GRP (Glass Reinforsed Plastic), calculated in linear elastic for a small value constant and uniform pressure.