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Construction Methods and Comparative Evaluation of Metal Deployable Load-carrying Shell Structures

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ABSTRACT: This study describes the main regularities for the construction of deployable metal shells of a cylindrical type. The comparative analysis of the efficiency of compact folding methods of thin metal shells of cylindrical and conical types, capable of maintaining the initial load-carrying capacity after unfolding without applying the additional strengthening methods, is presented. The range of possible parameters of long-length structures, constructed on the basis of rigid deployable pressurized shells, is characterized. The paper gives the qualitative and quantitative evaluation of basic functional characteristics of folding inflatable metal structures of cylindrical and conical types under the action of a complex of characteristic external loads, close to the maximum allowable ones. The result of experimental operability confirmation of the proposed structures' configurations is presented.

KEYWORDS: Transformable-volume structures, Load-carrying shells, Foldable shells, Deployable structures, Inflatable structures, Surfaces of zero total curvature.

INTRODUCTION

The increasing distribution of deployable inflatable structures in the field of space activities is explained, first of all, by their high mass-size characteristics and the ability to vary geometric dimensions within wide limits without sacrificing functional characteristics. The field of application of such structures includes the problems of taking off a load from the spacecraft surfaces (inflatable booms and trusses), the maintenance of a spatial shape of large-sized objects with a low rigidity (for example, folding antenna reflectors), the formation of tanks for storage of different substances, etc. However, the structures based on inflatable booms, being not subjected to the action of earth gravity, may experience under the conditions of space environment the inertial loads of the similar intensity and the cyclic temperature effects, which cause a complex stress-strain state of the load-carrying shell.

Thus, an undoubted advantage of deployable inflatable booms is their ability to withstand a load at the sufficient rigidity. This property contradicts both the requirements to a high compactness and also the simplicity of the technology of a compact folding of a long-length shell, and is solved in the known inflatable booms made of polymer materials due to different special

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methods of imparting the rigidity to a soft shell (Schenk *et al.* 2014). On the other hand, the known deployable structures (for example, telescopic masts) with a high load-carrying capacity based on a sectional metal shell (Rohweller 2002) are far superior to the mentioned inflatable booms by the specific weight and possess a much lower compactness in the folded state. A possible compromise can be found in pressurized long-length booms, combining the general principles of folding the soft inflatables with a rigidity of a metal shell and called transformable-volume structures (TVS) (Paton *et al.* 2015). Methods for improving the functional characteristics of such structures using the example of a multi-section shell of conical type are considered, in particular in (Lobanov *et al.* 2016). Nevertheless, it is of interest to compare the effectiveness of conical-type metal TVS and TVS with a cylindrical shell, which is the basis for most known inflatable booms made of polymer materials. In the present work, the authors try to answer the question whether the higher rigidity of cylindrical shell is a sufficient argument to the benefit of its selection for construction of metal inflatable booms. For this purpose, the main principles of construction of metal multi-sectional TVS of two above-mentioned types are considered.

THEORETICAL BASIS

The compact folding of the cylindrical shell, possessing the highest resistance to the action of non-axisymmetric loads and being the basis of the majority of the known inflatable booms, can be represented by replacement of its surface with some approximated surface formed by a plurality of simple plane figures conjugated on the adjacent sides. In case of a thin metal shell with a constant thickness at $\delta/R \leq 0.001$, where δ is the wall thickness, R is the radius of median surface of the cylindrical shell, the proximity of their total area to the area of the initial cylinder is the criterion of approximation to isometric transformation, which occurs without tension or compression of its material. The fulfillment of this condition is, finally, expressed in reducing the probability of a local buckling of the load-carrying shell under the service conditions.

Figure 1 shows two cases of the approximation (triangulation) of the cylindrical thin metal shell surface with height H and ratio $H/R = 2$ to the cylindrical polyhedron surface, which represents a variant of the known "origami folding". The variant of a triangulated cylinder with the ratio $\delta/R = 0.001$ and the ratio of the total area of triangles to the area of median surface of the equivalent cylinder å(g,n) / $S_{CYL} > 3.0$ (Fig. 1a, the stainless steel shell with $\delta = 0.15$ mm) may represent only a hinged system with rigid triangular plates joined between each other using flexible inserts by analogy with the known models (Guest and Pellegrino

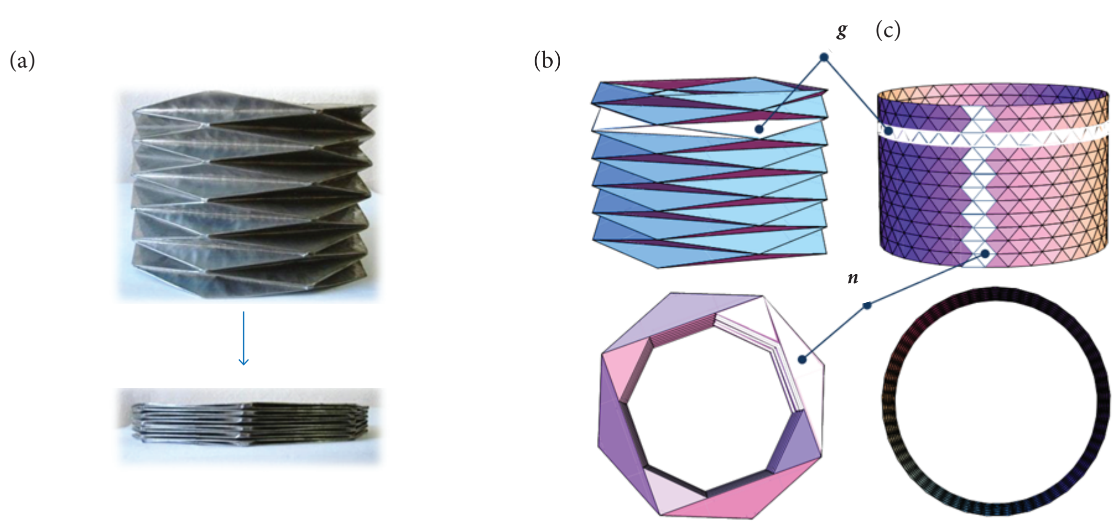

Figure 1. Approximation of cylinder surface by a cylindrical polyhedron with the equal number of girths g = 12 and sectors (a, b) n = 4 and (c) n = 33.

1994; Zawidzki 2016). On the other hand, the methods based on the assumption of "rigid folding" (Saito *et al.* 2016) cause doubts about the possibility of creating a pressurized metal shell required for inflatable structures. The folding of the cylindrical shell of revolution, made of a metallic sheet, requires increasing the number of substituting triangles for approximation of areas å (*g*, *n*) and S_{CYL} (Fig. 1c). At the same time the number of folds and junctions of their intersection, i.e., stress concentrators, increases, where the probability of violation of the shell mechanical integrity is high. In this case, a contradiction arises both as to a high compactness of folding, caused by the values of minimum admissible radii of sheet metals bending, and also as to its technological feasibility.

Maximum approximation of a triangulable surface to the initial conical surface at the simultaneous minimizing the number of intersection points of folds can be carried out using the property of the ruled surfaces of zero Gaussian curvature to be isometrically "unfoldable to plane" without tensions and compressions. At the same time, the meridians and parallels of surfaces may correspond to bending lines and remain unchanged in the process of movement.

A cylindrical surface can be moved to the plane by bending the surface fragments along the lengths of straight lines; in a cone, it can be also made along the lines of closed circumferences at a mirror reflection. For the median surface of a cylinder, this movement may be represented by transformation through a double ruled surface of the negative Gaussian curvature, i.e. hyperboloid of one sheet (Fig. 2a), given in the parametric form as (Eq. 1):

$$\begin{aligned} x(u,v) &= cos[v]\,cosh[u] \\ y(u,v) &= sin[v]\,cosh[u] \\ z(u,v) &= sinh[u] \end{aligned} \tag{1}$$

where: $x(u,v)$, $y(u,v)$, $z(u,v)$ are the component functions of parameters u, v, determining, respectively, the shape of meridian and the rotation angle of its plane relative to the axis of a surface symmetry, $0 < v < 2\pi$. In this case, on the hyperboloid surface the arbitrary rectilinear generatrices G can be preset (Fig. 2a), the length of which will remain unchanged at the change of parameter u (Eq. 1) under the condition G < D, where D is the diameter of hyperboloid bases. In the similar metal shell, the rectilinear generatrices G may correspond to rectilinear generating lines of folds, which are formed at the relative rotation and converging the bases C of the initial cylinder (Fig. 2a). The formation of folds will be considered below by the example of a numerical model of the cylindrical shell folding process.

In case of a cone, the median surface can be transformed relative to the parallel planes $\gamma_1, \gamma_2, \ldots \gamma_n$ which are perpendicular to the axis of rotation z by a successive mirror reflection. The formation of the median surface fold bent with non-zero radii (Fig. 2b) is described in (Lobanov *et al.* 2016). The successive formation of concentric folds allows transforming the whole initial conical surface into a flat corrugated disc.

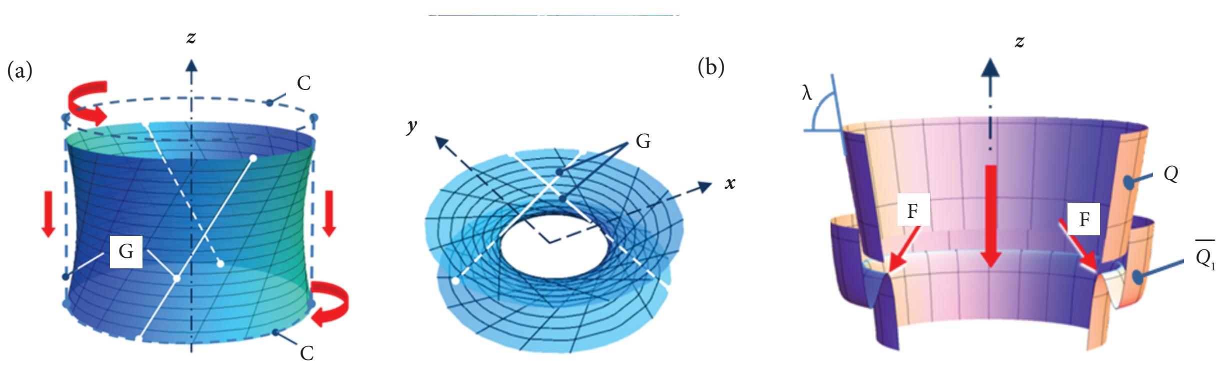

Figure 2. The scheme of the transformation of a shell median surface: (a) cylindrical shell and (b) conical shell. F = fold of the median surface for the model of bending using thin-sheet metal.

For the structure of sheet metal, the selection of the number of generatrices G and, correspondingly, the folds of a cylindrical shell are determined by the ratio of geometric sizes (H, R, δ) and also by elastic-plastic properties of the structural material. The same parameters also determine the selection of distance between the secant planes γ_1, γ_2 and, accordingly, the number of concentric folds of the conical shell.

METHODOLOGY

The required approximation to isometric transformation was practically achieved using a sufficiently broad range of austenitic steels, aluminum and titanium alloys with the ratio of yield strength $R_{P0,2}$ to the tensile strength σ_U in the range of $R_{P0,2}/\sigma_U = 0.3…0.8$ (Paton *et al.* 2015). The stress-strain curve of the material should not possess a clearly expressed yield plateau, determining the increase in the number of Lüders lines at material loading and, as a consequence, the regions with geometric imperfections in the zones of action of tensile and compressive forces. As the shells material, the stainless steel AISI 321 ($Rp_{0.2} = 205$ MPa, $\sigma_u = 548$ MPa, $R_{P0,2}/\sigma_U \approx 0.37$) was selected, providing a lower deformability of the structure due to the value of elasticity modulus, being 1.5 to 2 times higher as compared with that of titanium and aluminum. The choice of material of the examined structures also included a number of requirements, defined by their service conditions. The authors investigated shells with a thickness of $\delta = 0.1$ to 0.2 mm. The tendency to maximum reduction in structure weight in the specified range does not contradict the possibility of making the vacuum-tight welded joints, subjected to subsequent technological deformations and loads during service under extreme conditions.

Thus, the problem of optimal type determination of a load-carrying deployable metal shell structure is reduced to searching for the opportunity to make the transformation, close to isometric one, and the search for the most favorable combination of its strength and stability to the action of typical loads, as well as weight and compactness in the folded state.

CYLINDRICAL SHELL

The application of thin-sheet metals as a material for shells and the use of the considered transformation methods impose the limitations on the ratio of their height and the diameter of their bases. It ultimately requires the sectioning of a long-length shell structure. The sectioning of a lengthy shell is a positive factor because circumferential joints of the adjacent sections of the structure contribute to the increase in its spatial rigidity without the decrease in its compactness. The transformation of the cylindrical shell of thin-sheet metal similar to Fig. 2a leads to the formation of the so-called hyperboloid folds on its lateral surface. The bases profiles of a shell are not the plane curves at any moment of deformation, i.e. the adjacent sections cannot be joined between each other and require the rigid fixing on a plane circular contour. In this case, for the complete transformation of thin steel shell folds, the value H/R corresponds approximately to the empirical ratio (Eq. 2):

$$0.6 < \frac{H}{R} < 1.2 \tag{2}$$

Beyond the specified range, the influence of edge effect near the boundary of a rigid shell contour leads to characteristic distortion of the straight-linear folds of the shell and makes the folding process unpredictable. It is obvious that the upper boundary of the range (Eq. 2), corresponding to the highest compactness of the shell in the folded state, is preferable.

The geometrical parameters of hyperboloid folds of the sheet metal cylindrical shell with rigid contours of bases can be determined due to the feature, arising in it during torsion (for example, Teng and Rotter 2004). Figure 3 presents the fragments of the finite-element model (FEM) of the transformation of a stainless steel shell with rigid circular contours of bases with ratio $H/R = 1.1$ and thickness of material $\delta = 0.15$ mm at $\delta/R < 0.001$. FEM of the structures considered in this paper were executed in a three-dimensional statement using ANSYS software. Four-node shell finite elements of SHELL181 type, having three linear and three angular degrees of freedom were used. It is accepted that the geometrical arrangement of FEM nodes coincides with the medium surface of simulated shells. Taking account of the existing non-linear dependence between the stresses and deformations, the elastic-plastic model of material was taken. The model is based on Prandtl-Reis law of flow with the yield criterion of von Mises and multi-linear kinematic hardening rule. Geometric non-linearity of structure behavior is considered by introducing the quadratic terms into expressions for deformations. Computation of numerical models was made by a multi-step iteration method of Newton-Raphson with a correction of the solution at each step. As the main convergence criterion at each step of problem-solving the control of the balance of forces in system nodes was taken.

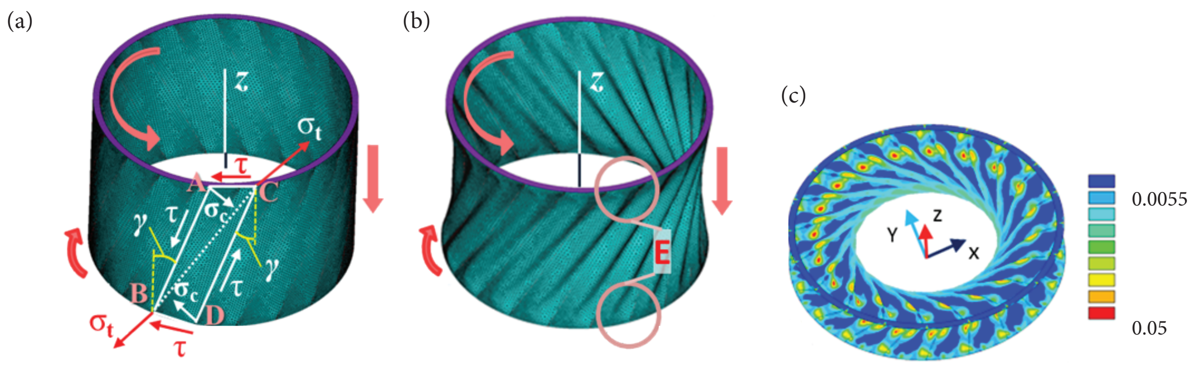

Figure 3. The transformation of a cylindrical shell into hyperboloid folds:
(a) internal forces in the initial shell; (b) the initial stage of shell transformation; and (c) distribution of equivalent strains σ_e (Von Mises) in the median surface of the shell at the final stage of transformation.

The fragments E (Fig. 3b) illustrate the formation of an action zone of edge effect in the metal shell at the rigid fixing along the base contour. As a result of the linear movement of bases along the axis z and their simultaneous multi-directional rotation around the same axis, the internal forces – tangential stresses, σ_t and σ_c – normal stresses arise in the shell wall. The visual representation of their orientation is shown in Fig. 3a.

During buckling the shell acquires a new state of equilibrium, the qualitative view is shown in Fig. 3b. The endpoints of the adjacent forming folds (B, C) move along the arcs of bases in the opposite directions, the conditional line BC approaches the shell rotation axis, and the acting loads cause the formation of tensile stresses σ_t towards the conditional line BC. The distance between the points A and D decreases, and the acting loads are causing the formation of the compressive stresses σ_c normal to BC. As a result, the surfaces ABC and CBD are formed, conjugated with the bending line BC relatively to which their mutual rotation with the further formation of a hyperboloid fold occurs.

The buckling mode of the shell wall is characterized by the formation of a plurality of regular folds on its surface, the rectilinear generatrices of which AB, CD, etc. are inclined relative to the axis of rotation z at the angle γ (Fig. 3a). At the initial stage of the process of transformation of a metal shell the geometrical imperfections may be preset on its surface, thus initiating the position and direction of rectilinear generatrices BC. It is obvious that the direction of mutual rotation of bases contours of a double ruled surface determines also the direction (left or right) of folds inclination. The number of folds in the circumferential direction n_f and the value of inclination angle of their generating lines γ to the meridian of a steel shell can be determined in accordance with Eqs. 3 and 4 (Volmir 1967):

$$n_f = 4.2\sqrt[8]{1-\mu^2}\sqrt{\frac{R}{H}}\sqrt[4]{\frac{R}{\delta}} \tag{3}$$

where $\mu = 0.3$ is the Poisson's ratio;

$$\gamma = arctg(1.73\sqrt[4]{\frac{R\times\delta}{H^2}}) \tag{4}$$

It should be noted that the theoretical value of the angle γ corresponds to the initial form of shell buckling and, as will be shown further, can be forcibly changed to increase its compactness in a full-scale experiment.

The numerical simulation of geometric transformation of the considered cylindrical shell (see Fig. 3) demonstrates the compliance of the folds number $n_f = 20$ with Eq. 3. The values of equivalent strains (von Mises) do not exceed $e_e = 0.017$. Nevertheless, at the local areas located in the direct vicinity of folds apexes and cylinder bases, their values reach $e_e = 0.046$ (Fig. 3c). In the process of folding the stress-strain state of the shell is characterized as uniform, the values of equivalent stresses σ_e (von Mises) on the median surface are changed in the range from $\sigma_e = 0.0$ MPa to $\sigma_e = 78.0$ MPa. The exceptions are local zones located along the lines of folds formation and concentrators near their apexes, where stresses reach $\sigma_e = 159$ MPa and $\sigma_e = 280$ MPa, respectively.

It can be seen that multiple impositions of folds with a thickness, determined by minimum bending radii, and the existence of the edge effect zone are the main factors reducing compactness of the cylindrical shell in the folded state. The considered further conical shell is deprived of the mentioned disadvantages and may be an object of comparative analysis of the main functional characteristics of the two considered types of metal deployable inflatable structures.

CONICAL SHELL

The variations of geometric parameters of a truncated conical shell have more degrees of freedom as compared to the cylindrical shell considered above. However, its relative height is also limited which, in the case of a lengthy structure, requires joining the separate sections along the contours of larger and smaller bases. The height HC of a truncated conical section is determined by the inclination angle λ (see Fig. 2) and the ratio of its greater and lesser bases radii RC/rC. The condition λ → max increases the structure stability to non-axisymmetric loads, but it considerably complicates technical feasibility of the metal shell folds forming process. The actual maximum value λ is limited, in particular, by the values of relative circumferential deformations in the shell during its folding, which at λ ≤ 65° to 67° are incompatible with the required approximation to the isometric transformation. Similarly to Fig. 2b, the conical shell of thin-sheet metal can be transformed by a successive formation of circular folds at a certain ratio of their width and material thickness.

The width, or pitch distance of corrugation, b, for the flattened shell, should be between 26 δ and 29 δ for thin shells made of AISI 321 steel (Paton et al. 2015). The precise choice of the parameter b value in the specified range, the same as the value of the parameter H/R (see Eq. 2) is performed on the basis of numerical analysis of the shell stress-strain state in the process of folding and it is confirmed experimentally. Figure 4 illustrates the result of numerical modeling of geometric transformation to the compact form of the conical shell rigidly fixed along the contour of a small base with the thickness δ = 0.15 mm and preliminary selected relations (rC + RC) / 2 = R, b = 28 × δ, where RC, rC are truncated conical section greater and lesser bases radii, R is the radius of cylindrical shell. The values of equivalent strains (von Mises) in the formed folds do not exceed ee = 0.011, and in the process of folding, they do not exceed the values ee = 0.023 in the apexes of the formed folds. The maximum values of equivalent stresses (von Mises) on the median surface of the shell (σe = 141 MPa) arise within the limits of the forming folds.

Further, the determination of the optimal ratio R_C/r_C for the conical section of a multi-sectional shell is the result of optimization, realized using a dynamic finite-element modeling, and which is briefly summarized in (Paton *et al.* 2015). In different fields of application, implying the effect of axial loads on a multi-sectional shell or the use of single sections, the high transformation coefficients K_T, i.e., ratio of the length in compact and transformed state, can be achieved (up to $K_T = 100$ and higher) in conical deployable shells. Nevertheless, in the considered case of a long-length structure, at most variants of combinations of the parameters R_C, r_C and δ, the ratio of bases should not exceed the values $R_C/r_C = 1.6$. At its higher values in the zones adjacent to the small bases of conical sections with the radius r_C, the clearly expressed stress concentrators are observed under the action of non-axisymmetric loads on a long-length multi-sectional structure. The reduction of their values requires the selection of the greater thickness of a structural material, which leads to an undesirable increase in weight and reduction in compactness of the

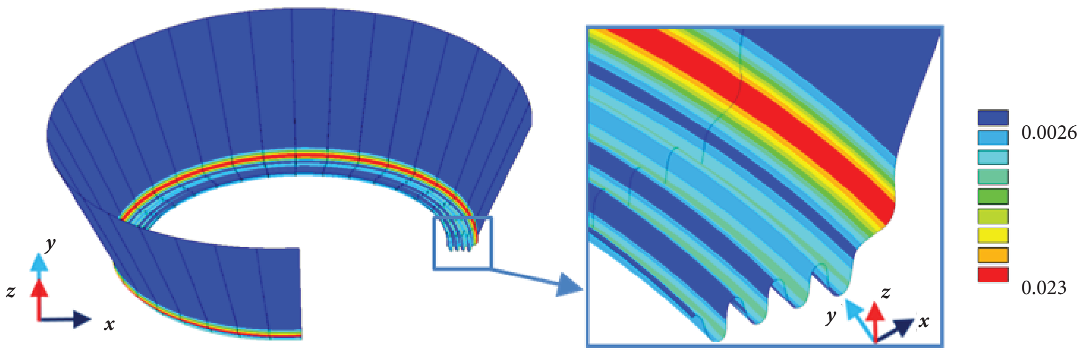

Figure 4. Distribution of equivalent strains (von Mises) **σe** on the median surface of the conical shell fragment transformed to the compact form.

structure. The comparison of these parameters, as well as functional characteristics related to them on the example of equivalent configurations of TVS of conical and cylindrical types are considered by the authors below.

During the experimental investigation of the kinetics of the process of deploying of the multi-sectional TVS of two types the absence of clearly expressed stress concentrators in shell folds is confirmed by the uniformity of their successive unfolding. This parameter can be controlled by a steepness of curve of pressure growth in internal volume of structure shells. Moreover, the correct selection criterion of geometric parameters of TVS sections is the perpendicularity of plane of circumferential connecting elements to a longitudinal axis *z* at all the stages of deployment. Carried out in the final part of the research, the experimental testing of the deployment process of two types TVS was performed under conditions of the Earth gravitation without unweightlessness of structures. The similar conditions allowed better evaluation of the TVS' vertical stability. The next full-scale experiments on deployment should be performed in zero gravity conditions at the absence of temperature gradients, for example, at the shade side of the Earth orbit and at the absence of dynamic conditions of the base station operation.

RESULTS AND DISCUSSION

COMPARATIVE ENGINEERING EVALUATION OF LONG-LENGTH METAL TVS OF CYLINDRICAL AND CONICAL TYPES

For a comparative analysis of structures of two types an example of the most unfavorable combination of loads combined with the space environment factors (SEF), acting on TVS at their rigid cantilevered fixing along the C contour (Fig. 5) on the outer surface of the base spacecraft (for example, International Space Station – ISS), was considered. The values and directions of effects linear (a_X = +12 m/s^2, a_Y = +12 m/s^2, a_Z = +9 m/s^2) and angular (e_X = +1,4 rad/s^2, e_Y = +1,4 rad/s^2, e_Z = +0,4 rad/s^2) accelerations of mass center of the structure, in combination with the calculated values of temperature effects –43 to +63 ° C (Lobanov *et al.* 2016) meet the requirements to designing and service of the equipment located on the outer surface of ISS. In the calculation results presented below, the Z axis corresponds to the axial direction of the structures, and the X and Y axes correspond to the orthogonal radial directions.

At the mentioned effects two comparison multi-sectional structures with the shells of a minimum permissible thickness δ = 0.1 mm have the best combination of weight and size characteristics and compactness (see Eqs. 2, 3, 4) at the minimum possible equal length and weight. At the same time, the shells of TVS meet the accepted criterion of strength, i.e., the condition $Rp_{0.2} > f \times \sigma_e$, where $Rp_{0.2}$ is the conventional yield strength of the material obtained experimentally ($Rp_{0.2}$ = 205 MPa), taking account of the safety factor f = 3.0. It is assumed that while changing the length of structures, at the other equal external factors, the qualitative character of relations of their design characteristics remains unchanged. The values of these characteristics, corresponding to the above-mentioned equations, are given in Table 1.

Table 1. Design characteristics of cylindrical and conical TVS.

Parameters	Cylindrical TVS	Conical TVS
Length	1500 mm	1500 mm
Number of section	8	9
Max/min diameter	340 mm/340 mm	400 mm/250 mm
Shell thickness	0.1 mm	0.1 mm
Weight	1.5 kg	1.5 kg
Transformation coefficient, KT	8	50

Figure 5 shows the values of equivalent stresses (von Mises) σ_e at the median surface of shells of both types under the influence of the mentioned above combination of inertia and thermal loads at the rigid fixing along the contour C under the conditions of inner and outer vacuum. The values of equivalent stresses σ_e at the area of the rigid fixing of the conical shell (σ_e = 34 MPa) is lower than those of the cylindrical one (σ_e = 41 MPa). The maximum values of equivalent stresses (σ_e = 60 MPa) are concentrated in the zone of smaller bases contours conjugation of the first two conical sections on the rigid fixing side. In the cylindrical shell, the maximum values of equivalent stresses do not exceed σ_e = 41 MPa and are observed in the contour of the rigid attachment.

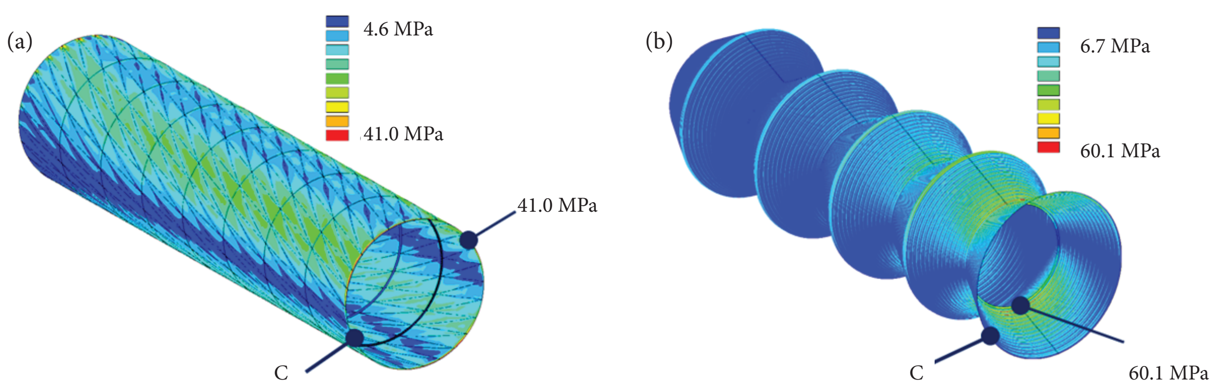

Figure 5. Distribution of equivalent stresses σ_e (von Mises). (a) cylindrical shell; (b) conical shell.

The problem of the presence of stress concentrators in the contours of TVS conjugation is not decisive and can be solved in a designing way, in particular, by changing the cross-section of circular joining elements. The calculated stress-strain state parameters of structures and their functional characteristics are shown in Table 2.

Table 2. Stress-strain state parameters and functional characteristics of TVS of equivalent configuration.

Parameters	Cylindrical TVS	Conical TVS
Deviation from the symmetry axis at the action of load	4.7 mm	14.9 mm
Equivalent stresses (von Mises) at the zone of rigid attachment	41.0 MPa	34.0 MPa
Natural frequency	21.06 Hz	14.91 Hz
Buckling load factor (BLF)	1.2	2.7

It can be seen that the structure of a conical type has a higher deformability; however, it is capable of compensating the higher load values without buckling. At equal values of external effects the conical TVS has the less (by 20%) rigidity, but essentially more than six times larger compactness. The results of the calculations show also an increase in the spatial rigidity of a conical TVS after unfolding at the formation of residual plastic deformations on its surface in the apexes of folds, performing the function of circular stiffeners (bulkheads). In contrast, in a completely unfolded cylindrical TVS, the presence of residual deformations in the apexes of straight folds leads to a slight and irrelevant increase (by 3% to 5%) in the maximum values σ_e under the action of the considered combination of loads. Taking into account this fact, the further experimental investigation of the kinetics of the unfolding process of two structures, the orientation of folds of a cylindrical TVS was subjected to change. It was also assumed that the increase in the inclination angle γ (see Fig. 3c) will allow reducing the negative influence of the edge effect zone on the compactness of the structure without changing the algorithm of its folding. Thus, the increase in γ results in the reduction of the fold deformations in the edge effect zone (see Fig. 3b), which facilitates the increase in compactness of a shell in the folded state. On the other hand, the increase in γ and the deviation from the theoretical initial buckling mode of the shell leads to radial bending and, finally, to fracture of folds AB, BC, and CD (Fig. 3a). The values being experimentally confirmed for the investigated shells allow the folds at a complete unfolding to undergo longitudinal bending without buckling. At folding of a flat BC section into a straight fold the wavelike deformations, perpendicular to it, are observed, which has no negative effect neither on the process of folding, nor on the rigidity of a cylindrical TVS in the unfolded state (Eq. 5).

$$\gamma_E \le arctg\frac{0.7R}{H} \tag{5}$$

EXAMPLE OF THE EXPERIMENTAL RESULTS

At the compact folding of initial smooth shells, manufactured with thin-sheet metal (stainless steel, δ = 0.1 mm) by means of microplasma welding of vacuum-tight butt joints, the algorithm of volume deformation was realized according to the corresponding schemes of Figs. 3 and 4. Its principle is the same for both types of shells and consists in providing the uniform pressure on the outer surface over the whole length of straight and circular generating lines, along which the bending is performed. The difference of folding processes in both types of shells consists in the different complexity of technological realization of the mentioned algorithm. In the conical shell, a uniform pressure is applied alternately at each next forming fold with a larger radius. In this case, the fixation of the only smaller base (r_C) is sufficient at a rigid contour. In the cylindrical shell the formation of all the folds occurs simultaneously. Accordingly, the rigid guides (punches), creating a forming force, should move along the complicated trajectories during the entire folding process, because the shell bases fixed on the rigid contours approach each other, performing a different-directional rotation. The used algorithms of shell folding represent the variants of the technology of thin metal shells cold volumetric deforming. However, the principle difference is the scheme of applying the forming force, at which the folds are formed smoothly and successively at almost absence of tension and compression of the shell material.

The cylindrical and conical sections are connected by the contours of the corresponding bases in the vacuum-tight structure (Figs. 6a-i 6b-i) by means of microplasma welding. Flanging is formed around each of the sections. It represents a miniature flange, the plane of each is normal to the longitudinal axis z. At the width of about $50 \times \delta$ all the adjacent flanging of TVS sections can be joined simultaneously by two welded joints, using the slot and edge weld, made by the "flange welding". As a result, the produced single element of rigidity ("circumferential bulkhead") allows distributing the load between the welds in the zones of stress concentrators of the TVS circumferential joints.

It should be noted that at the even number of a cylindrical TVS sections the contours of its outer bases at unfolding do not perform rotation relatively to the longitudinal axis z on the condition of alternation of left and right directions of folds (Fig. 6a). Thus, the authors consider the variants of inflatable shells capable of axially symmetric unfolding. The given property allows also joining and simultaneously unfolding the several long-length shells forming a spatial load-carrying structure.

Figures 6a and 6b illustrate the stages of the unfolding of vacuum-tight multi-sectional TVS under internal pressure. The given curves of pressure growth characterize the unfolding process kinetics. The presented structures correspond to the numerical models, given in item 4.1 at the value γ_E = 32° (see Eq. 5), which allowed increasing the compactness of a cylindrical TVS to the value K_T = 11. After depressurization in the shell inner volume (extreme phases of unfolding in Fig. 6) the axial compression ΔL_{CON} = 20 mm was noted in the conical TVS, while, in the cylindrical TVS it is ΔL_{CYL} = 2 mm, which characterizes its significantly lower damping capacity.

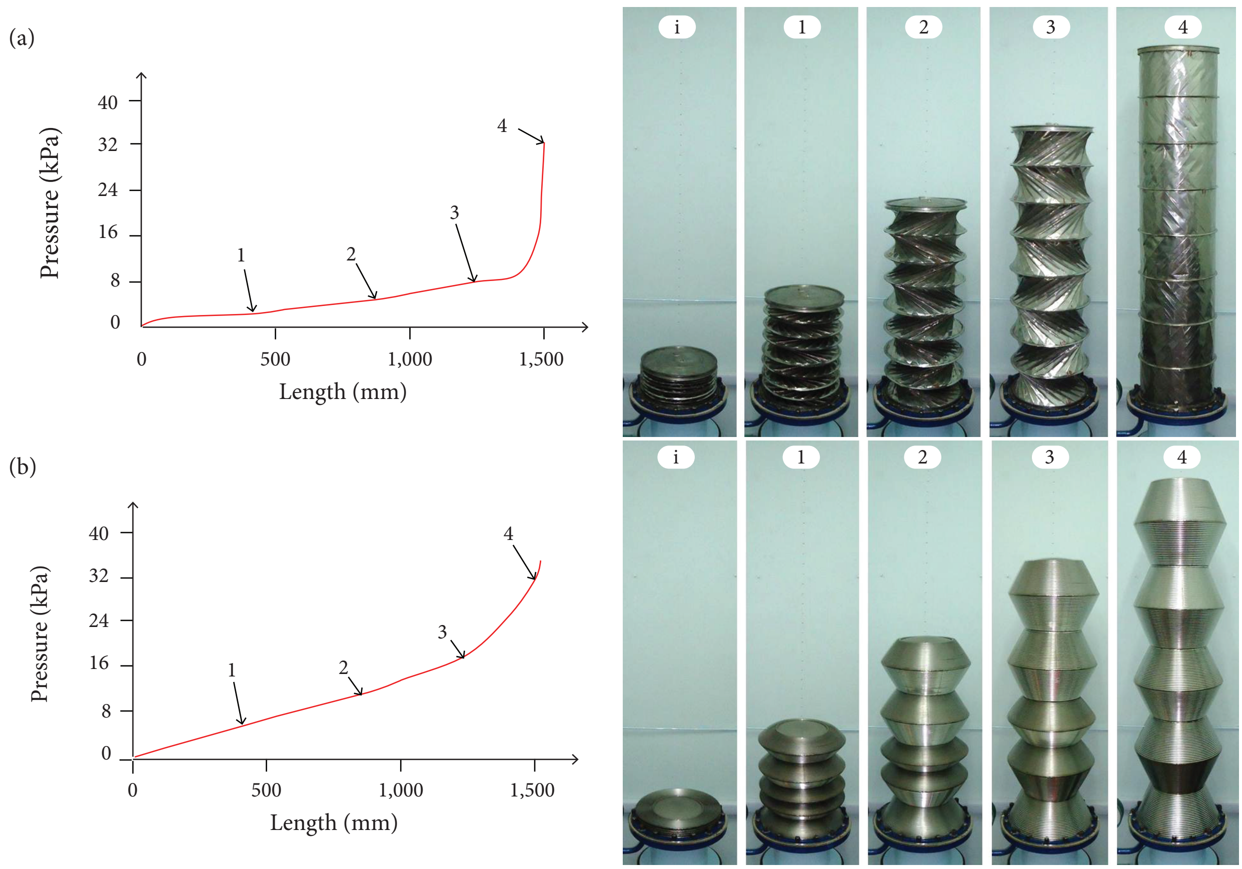

Figure 6. Results of the experiment on unfolding (a) cylindrical and (b) conical TVS.

At the inner pressure $P \geq 24$ kPa for the cylindrical TVS (Fig. 6a) and $P \geq 32$ kPa for the conical TVS (Fig. 6b) the relative speed of their linear unfolding decreases to 0.5 mm/Pa. Moreover, the further unfolding is realized due to elastic deformations in the apexes of residual folds and is not appropriate. The deformations of folds of the conical TVS symmetrical in the circumferential direction can eliminate the problem of a non-uniform unfolding, noted, in particular, in Senda *et al.* 2006.

Thus, TVS does not require additional strengthening. Moreover, in case of TVS of a conical type, the spatial rigidity of the shell after unfolding is increased due to the formation of circular stiffeners in the apexes of circular folds. It is evident that application of TVS as the load-carrying trusses or long-length elements of complex spatial structures does not limit the range of their application variants at space activities. For example, while creating the structures with other geometric proportions and character of acting loads, for example, TVS of a tank type, on-planet habitats, etc., the above-mentioned problems of shell stability are secondary. The presented result is not an illustration of the obvious advantages of TVS of cylindrical or conical types. However, it allows simplifying the choice of one of them with regard to specific service conditions. The obtained calculation and experimental data make it possible to conclude that the high spatial rigidity of cylindrical TVS is not evidently the decisive advantage even in case of their application as load-carrying elements of space structures. The decisive advantage to the benefit of cylindrical TVS can be only definite conditions of their launching to the orbit and applications. For example, at other equal conditions, the cylindrical TVS have a smaller maximum diameter and large inner volume.

CONCLUSIONS

The study presents the algorithm of creation and gives the comparative characteristic of metal load-carrying deployable structures, based on shells of zero Gaussian curvature. The comparative evaluation of methods of their compact folding is given providing the reversibility of folding process without the loss of spatial rigidity of metal shells after their complete deploying. Also, the result of experimental approbation of their operability is shown. The evaluation of the influence of technological deformation process of shells of both types at folding on their rigid and strength characteristics in the unfolded state was carried out. The comparative calculated evaluation of the main functional properties of TVS of both types having equivalent basic weight and size characteristics was made at other equal values of external effects. It was shown that the approaching of multi-sectional TVS to the cylindrical shape, which provides a high rigidity to it, at other equal conditions (mass, maximum length) leads to significant decrease in its compactness and stability. Thus, the higher rigidity of cylindrical shells can be a key parameter only when they are chosen for the construction of metal inflatable booms, having increased deformability requirements. For similar cases, the increase in compactness of the cylindrical TVS at the change of the theoretical value of the inclination angle of straight folds of its surface was confirmed experimentally in the presented work.

The presented results illustrate the possibility of creating the long-length load-carrying inflatable shells with a significant spatial rigidity, not requiring strengthening after unfolding and possessing at the same time the weight and size characteristics which are at the level of the known analogs of soft materials.

AUTHOR'S CONTRIBUTION

Conceptualization, Lobanov L, Volkov V and Yakimkin A; Methodology, Volkov V and Yakimkin A; Investigation, Volkov V and Yakimkin A; Writing, Volkov V; Funding Acquisition, Lobanov L; Resources, Volkov V and Yakimkin A; Supervision, Lobanov L.

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