BRAZE-WELDED TUBULAR BILLETS FOR PIPELINES AND HIGH-PRESSURE VESSELS*

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Application of two-layer shells for formation of common wall is supposed to be prospective for reduction of weight of pipeline structures as well as pressure vessels. Calculation of three single-type pressure vessel structures, having common wall, which is made as two-layer shell, was carried out. At that, analysis was given for applied materials and their combination used for manufacture of internal and external shells of common two-layer wall of three types of pressure vessels considering their strength indices. It is determined that application of the same steel grade in two-layer structure does not result in efficient loading of common wall. Application of material with higher strength characteristics for manufacture of external shell results in efficient loading of their common wall, reduction of its thickness and growth of its internal stresses, i.e. reasonable loading. Considered are the issues of application of mesh-like material for manufacture of external shell as well as usage of high-frequency pressure braze-welding in production of perspective tubular billets, designed for pipelines and high-pressure vessels. Ref. 12, Tables 7, Figures 8.

Keywords: *pressure braze-welding, tubular billet, thin-wall shell, stresses, simulation model*

Pressure vessel bodies have, as a rule, cylindrical form. Pipes of corresponding dimension-types are used in their manufacture or the bodies are manufactured from tubular billets, including thin-wall ones, designed for production of longitudinal or spiral pipes with continuous weld. The spiral pipes have series of advantages. Since a weld is produced at angle to cylinder axis, its partial unloading from radial stresses, appearing in a wall of pressure vessel body, is provided [1].

Relevant operation of pressure vessel can be achieved in providing of uniform distribution of stresses in its wall. At that, stresses formed in cylindrical wall of the body should not exceed allowable values for material, used in manufacture of the vessel. Obviously, that application of materials with increased strength properties results in reduction of thickness of cylindrical wall of the pressure vessel as well as weight of such commercial products.

Relevant application of material and reduction of thickness of wall of the vessel cylindrical bodies is also possible, if they are produced compound, for example, the body wall is made from two-shells or more. It is a well-known fact that two co-axial shells, fit on with tension, provide more relevant distribution of stresses formed in their common wall [2].

Stresses, appearing as a result of effect of internal pressure in cylindrical wall of vessel body and acting in radial and axial direction, are not uniform. It was determined [2, 3] that the stresses in radial direction 2 times exceed the stresses acting in axial direction. These peculiarities should be taken in account in selection in structural material with corresponding strength characteristics and wall thickness.

There is an experience of manufacture [4, 5] of metal-composite vessels for compressed gases, structure of body of which is in fact two-shell. At that, external load-bearing shell is produced from composite material, namely synthetic hardrail or roving, coiled over internal shell under tension.

Backgrounds of application of high-frequency braze-welding technology in manufacture of proposed prospective, from our point of view, structure of pressure vessels, are considered in present paper. At that, it is assumed to ensure relevant distribution of stresses in the vessel walls, improvement of its weight and strength indices as well as technological advantages in manufacture. Calculation-experimental model of the vessel was used. Such a model structure can be balloon model, cylindrical part of which is produced in form of two coaxial shells, fit on with tension. At that, internal cylindrical shell can have a sealing function, provide resistance in contact with operating medium as well as have good weldabil-

*The work was carried out under the leadership of Profs A.S. Pismenny and O.V. Makhnenko.

ity with bottom parts of the balloon for obtaining of sound, seal and corrosion-resistant welds.

External shell of cylindrical part of the body can be manufactured from material with strength indices higher in comparison with that of internal shell.

Such two-shell cylindrical part of the body should have a series of advantages, namely provide more relevant distribution of stresses forming in their common wall, as well as reduce total weight of the whole balloon due to reduction of thickness of wall of the external shell in the case of its manufacture from higher-strength material.

The work considers three single-type structures of samples of the pressure vessels (balloons), wall of cylindrical body of which represents itself two-layer shell. The internal thin-walled (relation of diameter to wall thickness is 10 and more) shell of cylindrical body has similar geometry in all model structures. The external shell in calculation models is fitted on the internal shell with minimum tension, close to zero, at that its internal diameter matches with external diameter of the internal shell, and its external diameter is calculated for each applied material.

Determined are the stresses forming in external and internal shells as well as total stresses in common wall of the two-shell balloon. The following conditions were maintained at that:

- calculations were carried out taking into account the same internal operation pressure for all three types of balloons ($p_{\text{op. work}}$ = 20 MPa) and the same geometry of wall of the internal shell ($d_{\text{in }1}$ = 147.5 mm; $d_{\text{ex }1}$ = 152 mm), wall thickness of the internal shell is constant (S_1 = = 2.25 mm) for all three types of balloons;
- internal diameter of the external shell corresponds to external diameter of wall of the internal shell and is constant ($d_{\text{ex }1} = d_{\text{in }2}$ = = 152 mm) for all three types of balloons;
- dimensions of wall of the external shell ($d_{\text{ex }2}$, S_2) were calculated based on stresses forming in the external shell and strength indices of used steel grade;
- the minimum value from calculation thicknesses of common balloon wall S_{bal} was used in the balloon structure;
- stresses in the separate shells as well as total stresses in the common vessel wall were determined without tension consideration;
- the calculation was carried out in accordance with acting reference documents and provisions of work [6], at that operating working pressure of balloon model was calculated based on coefficient of safety n = 2.7, on maximum allowable values of stresses forming in metal $[\sigma]_{\text{work}}$ = = $\sigma 0/n$, where σ_0 is the critical dangerous strength. Depending on steel grade $\sigma_0 = \sigma_y$ for soft materials and $\sigma_0 = \sigma_t$ for brittle materials.

Combination of different steel grades for internal and external shells was used for optimizing of the model structure. Strength indices of steel grades used for each shell were respectively laid in the calculation models (three single-type structures of two-shell balloons).

Strength of the bottom parts was not calculated, however, thickness of bottom wall for given structure according to metal technical properties can not be lower than the thickness of body wall of the whole balloon [6].

Weld strength in the calculation models was taken equal the base metal strength. The weld strength for given mathematical calculation model, considering application of different material grades, is provided by means of rise of total thickness of wall of the cylindrical body by value of wall thickness increase, which in all cases should be more than 0.5 mm [6].

Technological peculiarities of production of longitudinal, spiral and circumferential welds were not considered in the calculation models in present work. Effect of stresses formed in internal and external shells as well as technological factors (different grades of materials and other factors) was taken into account based on the results of carried calculations in designing of real models of tubular billet structures and samples of pressure vessels.

Figure 1 shows a sketch of structure of two-layer balloon, consisting of internal S_1 and external S_2 shells, and Table 1 gives the combinations of materials and relation of thicknesses of shells wall for the calculation models.

Table 2 shows the results of calculation of single-type structures of balloon samples 1–3.

Working stresses, forming in metal of internal $\sigma_{\text{work }1}$, external $\sigma_{\text{work }2}$ and common wall of the balloon $\sigma_{\text{work bal}}$ at operating working pressure applied to the balloon $p_{\text{op. work}}$ = 20 MPa, were determined for each sample.

Table 3 gives calculation values of forming working stresses σ_{work}, stresses in internal and external shells, and common wall of the balloon for each sample, from operating working pressure $p_{\text{op. work}}$ applied to the balloon.

Values of maximum allowable working pressures forming in internal $p_{\text{allow }1}$ and external $p_{\text{allow }2}$ shells, as well as $p_{\text{allow bal}}$ in common wall of balloon were calculated based on allowable values of stresses forming in metal (taking into account steel grade), determined as $[\sigma]_{\text{allow}}$ =

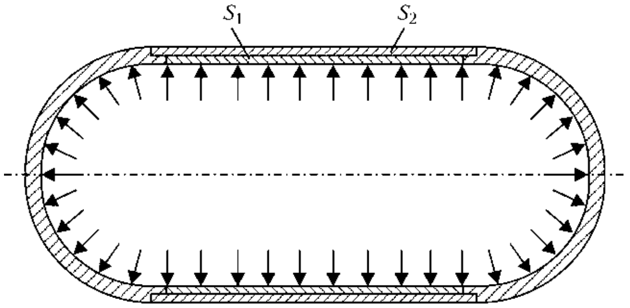


Figure 1. Sketch of two-layer balloon (for designations see the text)

$= 0 / n$, and their percent relationship (Tables 4 and 5) was determined.

Thus, approximately similar pressures are to be applied to the balloon of each sample in order to achieve allowable stresses in metal of the wall of each shell. Their values multiply exceed a nominal internal operating pressure.

The results of calculation show that allowable pressure in the balloon of each sample equals the sum of allowable pressures of internal and external shells. At that, the sum of allowable pressures

Table 1. Combination of materials and relation of thicknesses of shell walls in three models

Number of sample (model)	$S_1 + S_2$ combination of steels	S_1/S_2, mm
1	St.08.kp + high-strength steel	2.25/4.11
2	20 + 20	2.25/13.29
3	St.08kp + St.08.kp	2.25/20.54

Table 2. Calculation values of thickness of external shell wall and common balloon wall

Parameter	Sample 1	Sample 2	Sample 3
Allowable stresses in wall metal, MPa:			
internal shell $[\sigma_1]$	74.0	114.3	74.0
external shell $[\sigma_2]$	370.4	114.3	74.0
Thickness of wall of internal shell S_1, mm	2.25	2.25	2.25
Diameter, mm:			
internal shell $d_{\text{in }1}$	147.5	147.5	147.5
average $d_{\text{av }1}$	149.75	149.75	149.75
external $d_{\text{ex }1}$	152	152	152
Thickness of external shell wall S_2, mm	4.11	13.29	20.54
Diameter, mm:			
internal shell $d_{\text{in }1}$	152	152	152
average $d_{\text{av }1}$	156.11	165.29	172.54
external $d_{\text{ex }1}$	160.22	178.58	193.08
Thickness of balloon common wall $S_{\text{bal}} = S_1 + S_2$, mm	6.36	15.54	22.79
Balloon diameter, mm:			
internal $d_{\text{in }1}$	147.5	147.5	147.5
average $d_{\text{av }1}$	153.86	163.04	170.29
external $d_{\text{ex }1}$	160.22	178.58	193.08
$d_{\text{ex bal}}/S_{\text{bal}}$	160.22/6.36 = 25.19	178.58/15.54 = 11.49	193.08/22.79 = 8.472
S_2/S_1	1.83	5.91	9.1

Table 3. Comparison of calculation stresses formed in metal of internal $\sigma_{\text{work }1}$, external $\sigma_{\text{work }2}$ shell walls and common balloon wall $\sigma_{\text{work bal}}$ with allowable stresses $[\sigma]$ from applied to the balloon working pressure $p_{\text{op. work}} = 20$ MPa

Parameter	Sample 1	Sample 2	Sample 3
Allowable stresses $[\sigma_1]$, $[\sigma_2]$, MPa	$[\sigma_1] = 74$; $[\sigma_2] = 370.4$	$[\sigma_1] = 114.3$; $[\sigma_2] = 114.3$	$[\sigma_1] = 74$; $[\sigma_2] = 74$
Calculation stresses, MPa:			
internal shell $\sigma_{\text{work }1}$	6.7	6.7	6.7
external shell $\sigma_{\text{work }2}$	3.8	1.2	0.84
in common balloon wall, MPa: $\sigma_{\text{work bal}} < \frac{\sigma_{\text{work }1} + \sigma_{\text{work }2}}{2}$	2.4 < 5.2	1.049 < 3.95	0.747 < 3.77

Table 4. Values of allowable stresses in metal of external $[\sigma_1]$ and internal $[\sigma_2]$ shells and common balloon wall $[\sigma_{allow\ bal}]$, MPa

Parameter	Sample 1	Sample 2	Sample 3
Allowable stresses, MPa: wall of internal shell $[\sigma_1]$	74	114.3	74
wall of external wall $[\sigma_2]$	370.4	114.3	74
Comparison of forming calculation stresses $[\sigma_{allow\ bal}]$ in balloon wall, MPa	$[\sigma_{allow\ bal}] > \frac{[\sigma_1]+[\sigma_2]}{2}$ 262.8 > 222.2	$[\sigma_{allow\ bal}] = \frac{[\sigma_1]+[\sigma_2]}{2}$ 114.3 = 114.3	$[\sigma_{allow\ bal}] = \frac{[\sigma_1]+[\sigma_2]}{2}$ 74 = 74

Table 5. Calculation values of forming allowable pressure p_{allow} inside shells and balloon, their comparison with operating working pressure $p_{op.\ work}$ = 20 MPa after achievement of allowable stresses $[\sigma]$ by metal

Parameter	Sample 1	Sample 2	Sample 3
Allowable pressures, MPa: internal pressure $p_{allow\ 1}$	222.37 (10.23 %)	343.56 (15.76 %)	222.37 (11.23 %)
external shell $p_{allow\ 2}$	1950.4 (89.77 %)	1838.52 (84.24 %)	1761.9 (88.94 %)
common balloon wall $p_{allow\ bal}$	2172.7 (100 %)	2179.45 (100 %)	1980.7 (100 %)
$p_{allow\ 1}/p_{op.\ work}$	11.12	17.178	11.12
$p_{allow\ 2}/p_{op.\ work}$	97.52	91.92	88.1
$p_{allow\ bal}/p_{op.\ work}$	108.63	108.97	99.035

in samples 1 and 2 is virtually the same, and reduction of total allowable pressure for sample 3 is caused by the fact that it is referred to the category of thick-wall balloons (relation of external diameter to wall thickness in sample 3 is determined as $d_{ex\ bal}/S_{bal}$ = 193.08/22.79 = = 8.472 < 10 (see Table 2)), that requires additional correction of carried calculations. Figure 2 is made based on the results of Table 5.

Values of limiting pressures forming in internal $p_{lim\ 1}$ and external $p_{lim\ 2}$ shells, respectively, and in common wall of the balloon $p_{lim\ bal}$ were calculated, considering material grade and stresses forming in metal σ_0, determined as limiting values, and their percent relation was found (Tables 6 and 7).

At that, metal of internal, external shells and common wall of the balloon achieves the values of (dangerous) stresses σ_0. Figure 3 is made based on the results of Table 7.

Application of the shell, which represent itself sheet material with mesh-like structure of specific thickness, in nods of which three or more rods intersect, is supposed to be perspective for external shell in the two-layer balloon structure. At that, the stresses are distributed in a formed welded joint in area between internal and external shells as well as in rods of the external shell.

There are nonsolid sheet materials — drawn-punched mesh of different types (TU 14-4-1789–96) as well as mesh hose (TU 26-02-354–85). Production of two-layer structural sheet material

Table 6. Values of dangerous stresses in metal of internal σ_{01}, external σ_{02} shells and common balloon wall $\sigma_{0\ bal}$, MPa

Parameter	Sample 1	Sample 2	Sample 3
Internal shell σ_{01}	200	308.7	200
External shell σ_{02}	1000	308.7	200
Comparison of calculation forming limiting stresses in common balloon wall $\sigma_{0\ bal}$, MPa	$\sigma_{0\ bal} > \frac{\sigma_{01}+\sigma_{02}}{2}$ 709.6 > 600	$\sigma_{0\ bal} = \frac{\sigma_{01}+\sigma_{02}}{2}$ 308.6 = 308.7	$\sigma_{0\ bal} = \frac{\sigma_{01}+\sigma_{02}}{2}$ 199.99 = 200

Table 7. Calculation values of forming limiting pressures p_{lim} inside the balloon, MPa

Parameter	Sample 1	Sample 2	Sample 3
Internal shell $p_{lim\ 1}$	601 (10.24 %)	927.6 (15.763 %)	601 (11.23 %)
External shell $p_{lim\ 2}$	5265.5 (89.75 %)	4964 (84.378 %)	4761.4 (88.94 %)
Total balloon wall $p_{lim\ bal}$	5866.5 (100 %)	5884.68 (100 %)	5353.2 (100 %)

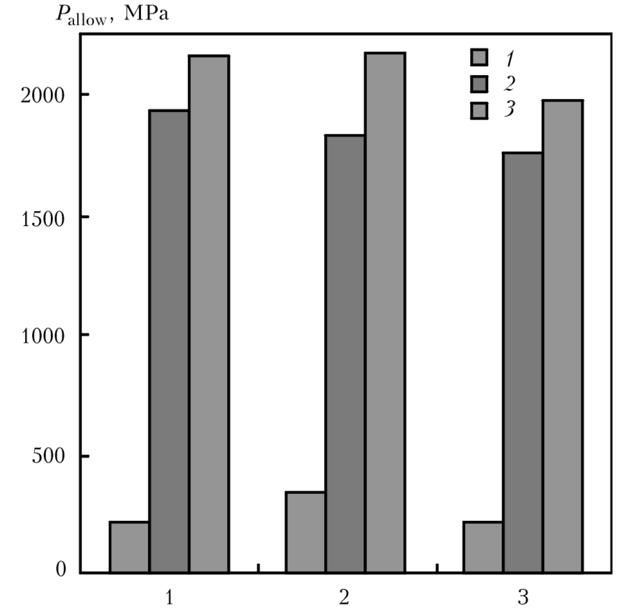


Figure 2. Values of forming allowable pressures in internal (*1*), external (*2*) and common (*3*) wall of the balloon in samples 1–3

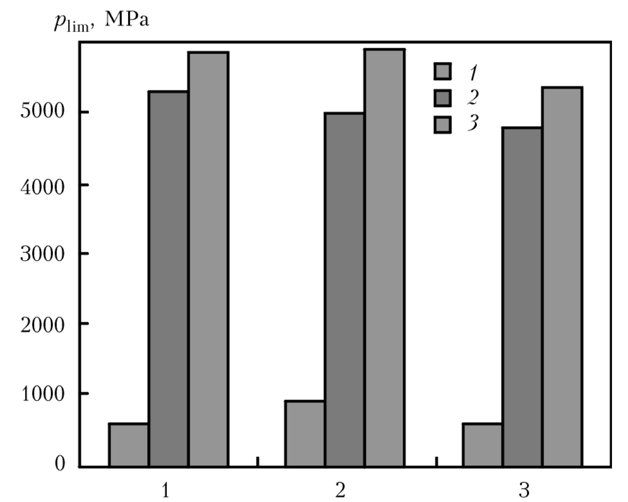
can be perspective in development of tubular billets and balloon bodies (Figure 4). The material represents itself one layer of solid sheet material, joined with another layer (nonsolid sheet material) by welding over the whole adjacent area.

Production of such external shell from steel grades having increased strength allows:

• uniform distribution of loading in «rods» of the shell and partial reduction of its thickness;

• providing of reduction of shell weight by value of volume weight of cell metal of mesh-like sheet material.

It is a well-known fact that the radial (circumferential) stresses, appearing in the cylindrical shell under the effect of internal pressure, is 2 times larger than the axial stresses (stresses along the cylinder generatrix) [1–3]. Positioning



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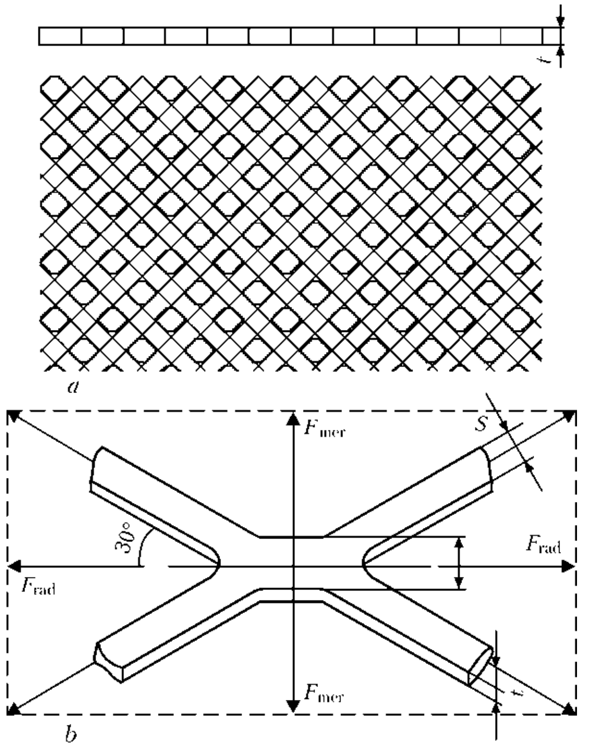
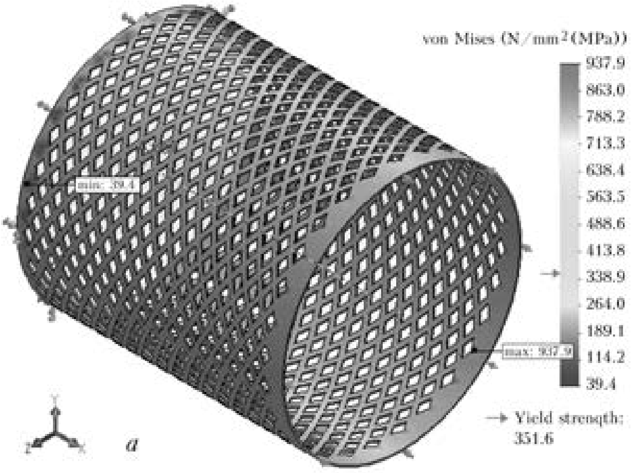


Figure 4. Sheet material with mesh-like structure (*a*) and nod, in which four rods intersect (*b*)

of intersecting rods at 60° angle in its specific putting on cylindrical surface (internal shell) allows developing uniform distribution in radial as well as axial directions in the rods of such shell from effect of internal pressure. At that, the density of distribution along the surface in radial direction will be 2 times higher the density in axial direction (Figure 5).

Computer simulation of internal pressure loading of the model of two-layer balloon, the external shell of which is produced from sheet material with mesh-like structure of specific thickness, confirms the relevance of application of given material (Figure 6). Formation of centers of stress concentrators in the nods of mesh-like material can be eliminated by approximation of rhomb profile of mesh cell to elongated ellipse profile (Figure 7).

Important factor, effecting the application of that or another mesh-like external shell, is determination of sizes of the mesh cell. In the external shell they depend and are determined based on thickness of material, used for manufacture of internal shell and its strength characteristics. Thus, the calculation models of three single type structures of two-layer balloon use external shell cell of 10 mm maximum gap in manufacture of internal shell from steel St.05kp, and that makes 20–15 mm in manufacture of internal shell from steel 20. At that, thickness of external shell *t* and width of rod *s* have the similar values. However, final determination of size of



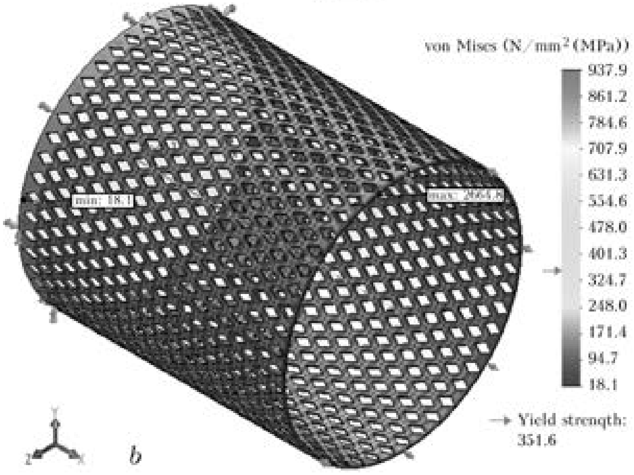


Figure 5. Level and distribution of stresses forming as a result of effect of internal pressure (internal cylindrical shell for the purpose of obviousness is not shown): *a* — external shell, manufactured from sheet material with mesh-like structure of specific thickness, in the nods of which four rods intersect; *b* — external shell, 90° turn

the external shell cell requires testing of samples of two-layer wall and full-scale tests of two-layer balloon.

Development of such multi-layer sheet structures is perspective from point of view of material saving, reduction of specific weight and preservation of strength properties of such welded

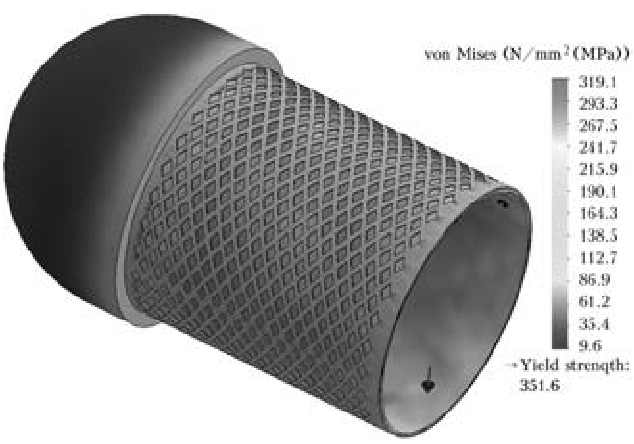


Figure 6. Simulation of loading of model of two-layer balloon by internal pressure

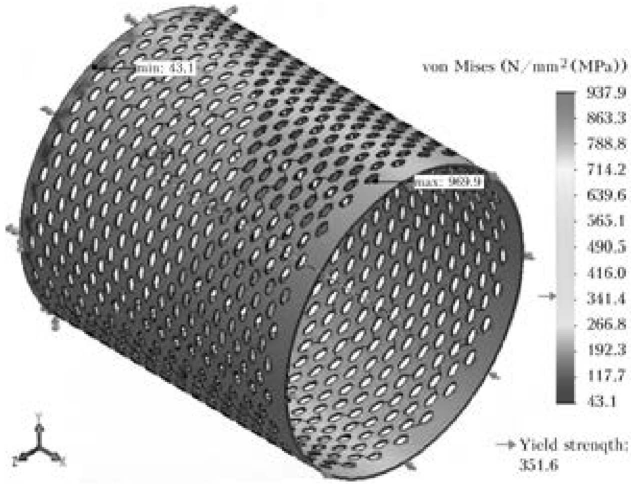


Figure 7. External shell, profile of mesh cell of which approaches to elongated ellipse profile

structure in determined direction. However, it creates the task of development and production of welded joint of the whole area. Formation of a joint between the layers (one sheet to another one over the whole area) can be produced by different pressure welding methods (for example explosion). Press induction braze-welding method can be used for joining of mesh-like material to solid sheet over the whole area. Process of press induction braze-welding is described in [7–10]. Study of the joints, made with soft interlayer in the weld, and brazed joints, produced with upsetting and plastic strain of weld zone [11, 12], proceeded its appearance.

A joint in this work was produced by press induction braze-welding, using activating substances previously applied over the surface of solid sheet and further HFC heating. Weld formation takes place in area of specific length with thermomechanical effect in form of elasto-plastic strain with further pressure solidification, which

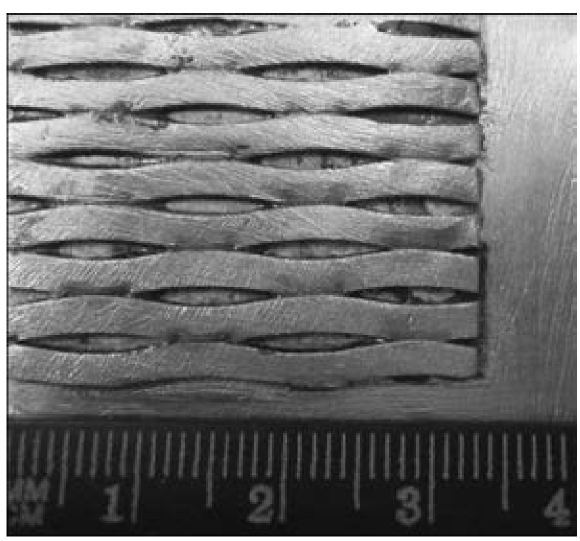


Figure 8. Fragment of whole area joint of mesh-like material to solid sheet material, produced by press braze-welding

approaches to uniform volume distribution. It allows receiving uniform weld metal composition with increase of its strength characteristics. Use of press braze-welding provides for development of multi-layer structures using the metals, welding of which by traditional methods is difficult (Figure 8).

Application of HFC heating in manufacture of two-layer shell of the vessel-balloon bodies allows performing thermal-mechanical shrinkage of the external shell relatively to internal one, i.e. technology of fitting on with tension, that significantly increases structural strength of compound wall of the whole balloon.

It is reasonable when the balloon wall is made compound, i.e. two-layer, in order to receive more uniform distribution of stresses along the thickness of balloon wall.

Manufacture of the internal shell from solid sheet material is caused by general requirement to the balloon, namely sealing and capability to withstand set calculation pressure. The welds should also be seal, dense and maintain set pressure. Designation of the external shell is to provide more uniform distribution of stresses along the total thickness of wall and unload the internal shell.

Conclusions

1. Stresses in the internal shell during working pressure loading of the balloon is higher than that in the external shell of balloon body, at that the balloon common wall is underloaded.

2. Using of the same steel grade in two-layer balloon structure does not result in efficient loading of the balloon common wall.

3. Application of the material with higher strength properties in manufacture of the external shell results in efficient loading of the balloon common wall, that in turn provides for the possibility of reduction of thickness of its wall, growth of internal stresses in it, i.e. relevant balloon loading.

4. Press fit on of the external shell of balloon body on the internal shell allows optimizing loading of the balloon common wall from effect of internal pressure.

1. Pismenny, A.S., Prokofiev, A.S., Gubatyuk, R.S. et al. (2012) Increase of strength characteristics of spirally-welded pipes of structural designation. *The Paton Welding J.*, **3**, 30–34.
2. Pisarenko, G.S., Yakovlev, A.P., Matveev, V.V. (1988) *Handbook on strength of materials*. Ed. by G.S. Pisarenko. 2nd ed. Kiev: Naukova Dumka.
3. Majzel, V.S., Navrotsky, D.I. (1973) *Welded structures*. Leningrad: Mashinostroenie.
4. Forum, A.M. (2009) Our balloons will withstand any storm! *AGZK+AT*, **2**, 31–33.
5. Sakhatov, R.M. (2009) Nonshatterable metal-composite cylinders BMK-300V and others. *Ibid.*, **4**, 51–55.
6. Chernega, V.I. (1976) *Safety servicing of steam boilers, vessels and pipelines* (Transact. of official documents). 2nd ed. Kiev: Tekhnika.
7. Tabelev, V.D. (1991) Peculiarities of joint formation in brazing with plastic deformation of base metal. *Avtomatich. Svarka*, **7**, 5–9.
8. Tabelev, V.D., Kareta, N.L., Panasenko, A.I. et al. (1985) Structure and phase composition of welds made by capillary and pressure brazing. *Ibid.*, **11**, 26–29.
9. Lebedev, V.K., Tabelev, V.D., Pismenny, A.S. (1983) Butt pressure brazing. *Ibid.*, **9**, 25–27.
10. *DSTU 3761.2–98*: Welding and related processes.
11. Bakshi, O.I., Shrou, R.Z. (1962) Strength under static tension of welded joints with soft interlayer. *Svarochn. Proizvodstvo*, **5**, 6–10.
12. Lebedev, V.K., Pismenny, A.S., Kasatkin, O.G. et al. (1990) Physical modeling of upsetting in butt welding and braze-welding of pipes. *Avtomatich. Svarka*, **8**, 17–20.

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