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DEVELOPMENT OF DIMENSIONALLY STABLE STRUCTURES OF MULTILAYER PIPELINES AND CYLINDRICAL PRESSURE VESSELS FROM CARBON FIBER REINFORCED PLASTIC

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В рамках безмоментної теорії циліндричних тонких оболонок досліджено пружне деформування багатошарових труб і посудин тиску. Передбачається, що труби і посудини тиску виконані перехресної спіральної намотуванням армованої стрічки з вуглепластика на металеву оправлення.

Виконано аналіз залежностей пружних деформацій від кутів армування. Отримано співвідношення для осьових і окружних деформацій стінки в залежності від структури пакета шарів, кутів армування при статичному навантаженні. Розглянуто відокремлена і комбінована дія внутрішнього тиску і температури. Для відокремленої дії навантажень побудовані графіки залежностей деформацій від кута намотування.

Досліджено композитні труби, виготовлені з вуглепластика КМУ-4Л, а також складові метало-композитні труби. Результати, отримані для теплових навантажень, добре узгоджуються з даними відомого експерименту і рішення. Залежно від параметрів навантажень визначені композитні та метало-композитні структури з розміростабільними властивостями.

Показано, що розміростабільні структури можуть бути використані для вирішення проблеми компенсації пружних деформацій трубопроводів. З цією метою за допомогою програмного комплексу ASCP виконаний варіантний аналіз модельної конструкції. Шляхом порівняльного аналізу трьох варіантів конструкції отримані структури пакетів шарів і схеми армування, щоб забезпечити значне зниження навантажень на опорні елементи. На прикладі трубопроводу з протікаючою рідиною показано, що застосування розміростабільних багатошарових труб дозволяє виключити деформації вигину і помітно знизити рівень робочих зусиль і напружень.

Розміростабільні багатошарові труби з композитів відкривають нові підходи до проектування трубопроводів і посудин під тиском. З'являються можливості створення конструкцій з наперед заданими (не обов'язково нульовими) полями переміщень, узгодженими з полями початкових технологічних переміщень, а також з переміщеннями сполучених пружних елементів і устаткування при зміні режиму роботи. Область застосування подібних конструкцій не обмежується «гарячими» трубами. Отримані результати можуть знайти застосування в кріогенній техніці

Ключові слова: композити, вуглепластики, вуглецеві волокна, схеми армування, конструкції трубопроводів, розміростабільність, пружні деформації

1. Introduction

The main loads acting on pipelines and pressure vessels are temperature effects, associated with changes in temperature conditions, and internal pressure of a working medium [1]. Temperature effects and internal pressure cause deformations of a shell not only in the axial direction. The cylindrical shell is also deformed in the circumferential direction

[1]. In this case, points of the middle surface receive radial displacements. In places of inhomogeneities: zones of interface of shells of different thickness; zones adjacent to rigid flanges, frames, etc., the development of radial displacements is constrained. As a result, wall bending occurs and significant local bending stresses arise. In certain conditions, local forces and stresses lead to wall rupture and detachment of reinforcement elements.

Therefore, when designing pipelines and pressure vessels, the problem of compensation of elastic deformations takes an important place [1, 2]. For deformation compensation, special devices are traditionally used: floating or sliding supports, flexible inserts in the form of bellows, lens or packed expansion joints. With the same purpose, curvilinear sections in the form of L-, V- or U-shaped bends are "inserted" into the line. In this case, the greater the flexibility of the bends, the higher the ability of the structure to compensate elastic deformations, the lower the level of working forces and stresses.

The introduction of additional elements and units in the line increases dimensions and weight, the cost of the structure as a whole. Fundamentally new approaches to the design of pipelines and pressure vessels open up with the use of modern structural materials [3, 4]. The use of multilayer structures with dimensionally stable properties becomes an effective means of "compensation" of elastic deformations [5, 6]. Selection of materials [7], development of layer package structures and methods for designing such structures with weight and dimension limitations, without reducing the product strength, seems to be a rather urgent problem.

2. Literature review and problem statement

The most promising materials for dimensionally stable structures are fiber-reinforced composites (FRC) [8]. Moreover, most of such structures are used in the rocket and space industry [9].

In [10], rod systems from FRC are considered. In [11], methods of optimization and design of dimensionally stable structures based on composite plates are presented. Matrix and binder selection questions are discussed in [6, 10] (carbon fiber reinforced plastic), [12] (metal composite) and [13] (combined metal and FRC). In [14], the technique for measuring the deformation of composite specimens for designing dimensionally stable structures at low temperatures is proposed.

Systematization of the results of these studies [5] suggests that existing approaches to solving the problem of compensation of deformations of composite structures rely on uniaxial self-compensation. Package structures and reinforcement schemes that ensure dimensional stability have been developed only under temperature loads. There are practically no studies on the creation of composite pipelines and pressure vessels with dimensional stability under the combined effect of temperature and internal pressure [8].

All this suggests that it is expedient to conduct a study on the development of new layer package structures and design techniques of dimensionally stable structures from FRC under the combined effect of temperature and internal pressure.

3. The aim and objectives of the study

The aim of the study is to develop structures of multilayer cylindrical shells with axial dimensional stability.

To achieve this aim, it is necessary to accomplish the following objectives:

– to choose package structures, reinforcement schemes and determine package parameters that provide geometric stability under two-component static loading;

– to investigate two-way reinforced and combined multilayer pipes, made by winding of reinforced tape on a thin-walled metal mandrel;

– to perform a design justification of dimensionally stable structures.

4. Derivation of relations for elastic deformations of the wall and analysis of the effect of reinforcement parameters

We consider the deformation of thin-walled pipelines and cylindrical pressure vessels under the action of temperature and internal pressure.

Fig. 1 shows the pipe section as a thin-walled cylindrical shell. The wall of the shell is formed by two-way spiral winding of two symmetric fiber (filaments or tapes) systems on the mandrel. Fibers make up angles $\pm\theta$ with the shell generator.

The number of the unidirectional layers is $2k+1$. The inner layer (technological mandrel) is considered homogeneous and isotropic, the unidirectional layers from FRC (monolayers) – orthotropic and linearly elastic. The bonds of fibers and binder, as well as individual layers with each other, are assumed to be ideal.

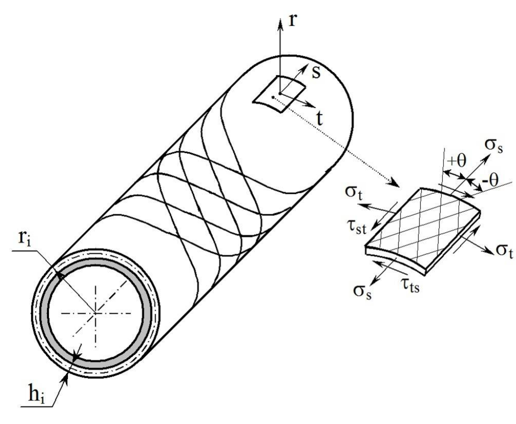


Fig. 1. Thin-walled multilayer shell element

We use two right orthogonal systems of coordinate axes. Coordinates 1, 2, 3 (natural coordinates) are associated with the axes of the elastic symmetry of the monolayer: axis 1 is directed along the fibers, axis 2 – perpendicular to the fibers in the monolayer plane. Coordinates *s*, *r*, *t* are combined with the middle surface of the shell (Fig. 1).

We assume that the monolayer "works" under conditions of plane stress state (radial stresses are neglected). In this case, the relation between deformations and stresses has the following form [15]:

$$\begin{Bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{Bmatrix} = \begin{bmatrix} \frac{1}{E_1} & -\frac{\nu_{21}}{E_2} & 0 \\ -\frac{\nu_{12}}{E_1}T & \frac{1}{E_2} & 0 \\ 0 & 0 & \frac{1}{G_{12}} \end{bmatrix} \begin{Bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{Bmatrix} + \begin{Bmatrix} \alpha_1 \\ \alpha_2 \\ 0 \end{Bmatrix} \Delta \qquad (1)$$

or in the matrix form:

$$\{\varepsilon_{12}\}=\left[S^0\right]\{\sigma_{12}\}+\{\alpha_{12}\}\Delta T.$$

Here $\{\varepsilon_{12}\}$, $\{\sigma_{12}\}$, $\{\alpha_{12}\}$ are the vectors of deformations, stresses and coefficients of thermal expansion, respectively; $[S^0]$ is the monolayer elastic compliance matrix; ΔT is the temperature variation (measured from the initial level, corresponding to the stress-free state); E_1, E_2, G_{12}, ν_{12}, ν_{21}, α_1, α_2 are the technical thermoelasticity constants (effective thermoelastic constants: elastic moduli, shear modulus, Poisson's ratios and thermal expansion coefficients). In this case, $\nu_{12}E_2=\nu_{21}E_1$. A homogeneous field is assumed. The relations inverse to (1) are:

$$\begin{Bmatrix}\sigma_1\\ \sigma_2\\ \tau_{12}\end{Bmatrix}=\begin{bmatrix}g_{11}^0 & g_{12}^0 & 0\\ g_{12}^0 T & g_{22}^0 & 0\\ 0 & 0 & g_{66}^0\end{bmatrix}\begin{Bmatrix}\varepsilon_1\\ \varepsilon_2\\ \gamma_{12}\end{Bmatrix}-\begin{Bmatrix}\beta_1\\ \beta_2\\ 0\end{Bmatrix}\Delta \quad (2)$$

or

$$\{\sigma_{12}\}=\left[G^0\right]\{\varepsilon_{12}\}-\{\beta_{12}\}\Delta T,$$

where $[G^0]$ is the monolayer stiffness matrix, $\{\beta_{12}\}$ is the temperature stress vector.

The coefficients of the matrix $[G^0]$ and the vector $\{\beta_{12}\}$ are related to the technical thermoelasticity constants by the following relations:

$$g_{11}{}^0=E_1/(1-\nu_{12}\nu_{21}),\quad g_{12}{}^0=\nu_{21}E_1/(1-\nu_{12}\nu_{21}),$$

$$g_{22}{}^0=E_2/(1-\nu_{12}\nu_{21}),\quad g_{66}{}^0=G_{12},\quad g_{21}{}^0=g_{12}{}^0,$$

$$\beta_1=(\alpha_1+\nu_{21}\alpha_2)\,E_1/1-\nu_{12}\nu_{21}),$$

$$\beta_2=(\alpha_2+\nu_{12}\alpha_1)\,E_2/(1-\nu_{12}\nu_{21}).$$

Let us rewrite the dependences (1) and (2) from the natural coordinate system 1, 2, 3 to coordinates s, r, t. As a result, we get

$$\{\varepsilon_{st}\}=[S]\{\sigma_{st}\}+\{\alpha_{st}\}\Delta T, \quad (3)$$

$$\{\sigma_{st}\}=[G]\{\varepsilon_{st}\}-\{\beta_{st}\}\Delta T. \quad (4)$$

Transformations of matrices when the axes of coordinates s and t are rotated relative to the normal to the surface r have the known form [15]:

$$[S]=[L_1]\left[S^0\right][L_1]^{\mathrm{T}},\quad [G]=[L_2]\left[G^0\right][L_2]^{\mathrm{T}},$$

$$\{\alpha_{st}\}=[L_1]\{\alpha_{12}\},\quad \{\beta_{st}\}=[L_2]\{\beta_{12}\}. \quad (5)$$

Here $[L_1]$ and $[L_2]$ are rotation transformation matrices:

$$[L_1]=\begin{bmatrix}m^2 & n^2 & -mn\\ n^2 & m^2 & mn\\ 2mn & -2mn & m^2-n^2\end{bmatrix},$$

$$[L_2]=\begin{bmatrix}m^2 & n^2 & -2mn\\ n^2 & m^2 & 2mn\\ mn & -mn & m^2-n^2\end{bmatrix}, \quad (6)$$

In this case, $m=\cos\theta$, $n=\sin\theta$. The index "T" denotes the matrix transpose operation.

Note that, according to (5), with arbitrary reinforcement, the angular deformations γ_{st} of the unidirectional layer are related to normal stresses σ_s and σ_t, and the linear deformations ε_s and ε_t – to shear stresses τ_{st}. In the particular case of symmetric winding for the system composed of two monolayers with angles $\pm\theta$, the coefficients

$$s_{16}=s_{26}=0,\ g_{16}=g_{26}=0,\ \alpha_{st}=0,\ \beta_{st}=0.$$

As a result, these relations disappear.

We define the linear deformations ε_s and ε_t of the section of the multilayer pipe reinforced with the symmetric filament system. To do this, using the method of sections, we write the following equilibrium equation [4]:

$$\sum_i \sigma_{s_i} h_i r_i=\frac{1}{2}pR^2,$$

$$\sum_i \sigma_{t_i} h_i=pR. \quad (7)$$

Here $i=1, 2,\ldots, k+1$ is the sequence number of the layer (composed of two monolayers), h_i and r_i are the thickness and radius of the middle surface of the layer, R is the pipe hole radius, $p=p_m+\rho_f\times v_m^2$ is the intensity of internal pressure. In this case, p_m is the static component, $\rho_f\times v_m^2$ is the dynamic component of pressure, equal to twice the velocity head (ρ_f and v_m are the density and velocity of the fluid flow). It is believed that the pipe end sections are closed with plugs.

Substituting the dependences (4) into the equation (7), we obtain

$$\varepsilon_s\sum_i g_{11_i}h_i r_i+\varepsilon_t\sum_i g_{12_i}h_i r_i-\Delta T\sum_i \beta_{s_i}h_i r_i=\frac{1}{2}pR^2,$$

$$\varepsilon_s\sum_i g_{21_i}h_i+\varepsilon_t\sum_i g_{22_i}h_i-\Delta T\sum_i \beta_{t_i}h_i=pR.$$

Where

$$\varepsilon_s^p=\frac{pR}{2}\left[\frac{R\sum_{i=1}^{k+1}(g_{22_i}h_i)-2\sum_{i=1}^{k+1}(g_{12_i}h_i r_i)}{\sum_{i=1}^{k+1}(g_{11i}h_i r_i)\sum_{i=1}^{k+1}(g_{22i}h_i)-\sum_{i=1}^{k+1}(g_{21i}h_i)\sum_{i=1}^{k+1}(g_{12i}h_i r_i)}\right]\approx$$

$$\approx\frac{pR}{2}\left[\frac{\sum_{i=1}^{k+1}(g_{22_i}h_i)-2\sum_{i=1}^{k+1}(g_{12_i}h_i)}{\sum_{i=1}^{k+1}(g_{11i}h_i)\sum_{i=1}^{k+1}(g_{22i}h_i)-\left(\sum_{i=1}^{k+1}(g_{12i}h_i)\right)^2}\right],$$

$$\varepsilon_s^{\Delta T}=\Delta T\left[\frac{\sum_{i=1}^{k+1}(\beta_{S_i}h_i r_i)\sum_{i=1}^{k+1}(g_{22_i}h_i)-\sum_{i=1}^{k+1}(\beta_{t_i}h_i)\sum_{i=1}^{k+1}(g_{12_i}h_i r_i)}{\sum_{i=1}^{k+1}(g_{11i}h_i r_i)\sum_{i=1}^{k+1}(g_{22i}h_i)-\sum_{i=1}^{k+1}(g_{21i}h_i)\sum_{i=1}^{k+1}(g_{12i}h_i r_i)}\right]\approx$$

$$\approx\Delta T\left[\frac{\sum_{i=1}^{k+1}(\beta_{S_i}h_i)\sum_{i=1}^{k+1}(g_{22i}h_i)-\sum_{i=1}^{k+1}(\beta_{t_i}h_i)\sum_{i=1}^{k+1}(g_{12_i}h_i)}{\sum_{i=1}^{k+1}(g_{11i}h_i)\sum_{i=1}^{k+1}(g_{22i}h_i)-\left(\sum_{i=1}^{k+1}(g_{12i}h_i)\right)^2}\right],$$

$$\varepsilon_t^p=\frac{pR}{2}\left[\frac{2\sum_{i=1}^{k+1}\left(g_{11i}h_ir_i\right)-R\sum_{i=1}^{k+1}\left(g_{21i}h_i\right)}{\sum_{i=1}^{k+1}\left(g_{11i}h_ir_i\right)\sum_{i=1}^{k+1}\left(g_{22i}h_i\right)-\sum_{i=1}^{k+1}\left(g_{21i}h_i\right)\sum_{i=1}^{k+1}\left(g_{12i}h_ir_i\right)}\right]\approx$$

$$\approx\frac{pR}{2}\left[\frac{2\sum_{i=1}^{k+1}\left(g_{11i}h_i\right)-\sum_{i=1}^{k+1}\left(g_{12i}h_i\right)}{\sum_{i=1}^{k+1}\left(g_{11i}h_i\right)\sum_{i=1}^{k+1}\left(g_{22i}h_i\right)-\left(\sum_{i=1}^{k+1}\left(g_{12i}h_i\right)\right)^2}\right],\qquad(8)$$

$$\varepsilon_t^{\Delta T}=\Delta T\left[\frac{\sum_{i=1}^{k+1}\left(\beta_{t_i}h_i\right)\sum_{i=1}^{k+1}\left(g_{11i}h_ir_i\right)-\sum_{i=1}^{k+1}\left(\beta_{S_i}h_ir_i\right)\sum_{i=1}^{k+1}\left(g_{21i}h_i\right)}{\sum_{i=1}^{k+1}\left(g_{11i}h_ir_i\right)\sum_{i=1}^{k+1}\left(g_{22i}h_i\right)-\sum_{i=1}^{k+1}\left(g_{21i}h_i\right)\sum_{i=1}^{k+1}\left(g_{12i}h_ir_i\right)}\right]\approx$$

$$\approx\Delta T\left[\frac{\sum_{i=1}^{k+1}\left(\beta_{t_i}h_i\right)\sum_{i=1}^{k+1}\left(g_{11i}h_i\right)-\sum_{i=1}^{k+1}\left(\beta_{S_i}h_i\right)\sum_{i=1}^{k+1}\left(g_{12i}h_i\right)}{\sum_{i=1}^{k+1}\left(g_{11i}h_i\right)\sum_{i=1}^{k+1}\left(g_{22i}h_i\right)-\left(\sum_{i=1}^{k+1}\left(g_{12i}h_i\right)\right)^2}\right].$$

In the particular case of an isotropic body with k=1, the formulas for linear deformations (8) acquire the known form [1]:

$$\varepsilon_s^p=p\frac{1-2\nu}{2E}\frac{R}{h},$$

$$\varepsilon_t^p=p\frac{2-\nu}{2E}\frac{R}{h},$$

$$\varepsilon_s^{\Delta T}=\varepsilon_t^{\Delta T}=\alpha\ \Delta T,\qquad(9)$$

where E, ν, α are the elastic modulus, Poisson's ratio and thermal expansion coefficient, respectively.

In the case of an isotropic body, the equality $\varepsilon_s=0$ holds: under the action of internal pressure, if ν=0.5; under the action of temperature, if $\alpha=0$.

For the pipe made of two-way reinforced material containing two monolayers, we define the angles $\pm\theta^*$, ensuring ε_s=0. Consider internal pressure loading (temperature load is considered in [15]) on the pipe. In this case, based on (8), from the condition ε_s^p=0 for k=1, it follows that:

$$g_{22}-g_{12}\left(2+\frac{h}{R}\right)=0.\qquad(10)$$

The coefficients g_{12} and g_{22} are determined by the formulas (5):

$$g_{22}=g_{11}^0n^4+2\left(g_{12}^0+2g_{66}^0\right)m^2n^2+g_{22}^0m^4,$$

$$g_{12}=g_{12}^0\left(m^4+n^4\right)+\left(g_{11}^0+g_{22}^0-4g_{66}^0\right)m^2n^2.\qquad(11)$$

Substituting relations (11) into equality (10) and introducing a new variable $\lambda=n^2$, we obtain the algebraic equation of the following form

$$a\lambda^2+b\lambda+c=0,\qquad(12)$$

where

$$a=3\left(1+\frac{h}{3R}\right)\left(g_{11}^0+g_{22}^0-2g_{12}^0-4g_{66}^0\right),$$

$$c=g_{22}^0-2g_{12}^0\left(1+\frac{h}{2R}\right),$$

$$b=2\left[\begin{array}{l}3g_{12}^0\left(1+\frac{h}{3R}\right)+6g_{66}^0\left(1+\frac{h}{3R}\right)-\\-g_{11}^0\left(1+\frac{h}{2R}\right)-2g_{22}^0\left(1+\frac{h}{4R}\right)\end{array}\right].$$

The solution of the quadratic equation (12) determines the angles $\pm\theta^*$, providing dimensional stability in the s direction.

5. Results of numerical experiment on thermal power loading of versions of multilayer pipe reinforcement schemes

For the sample made of two-way reinforced material based on unidirectional *KMU-4l* carbon fiber reinforced plastic (Russian Standard GOST 28006-88) with the characteristics: E_1=140.0 GPa, E_2=9.8 GPa, G_{12}=6.1 GPa, α_1= =-0.24×10^{-6} K^{-1}, α_2=37.0×10^{-6} K^{-1}, ν_{12}=0.26, inner diameter d=96 mm, wall thickness H=2.4 mm, we obtain λ_1=0.6255827 and λ_2=0.0199187. Where $\theta_1^*=\pm52^o16'$ and $\theta_2^*=\pm8^o07'$.

Thus, under internal pressure loading on the sample, two different winding angles θ^*, providing geometrical stability are obtained. Moreover, the angle θ_1^* is quite close to the angle $\theta_0=\pm54^o44'$, corresponding to the equilibrium reinforcement scheme [4].

Fig. 2 shows the graphs of the linear deformations ε_s and ε_t of the sample against the winding angle, constructed by the formulas (8). Graphs in Fig. 2, *a* correspond to the effect of temperature ΔT=1 K. The points on the graph are the experimental results [5, 15], averaged in the temperature range of 25÷150 °C. Graphs in Fig. 2, *b* correspond to the action of internal pressure p=1 MPa.

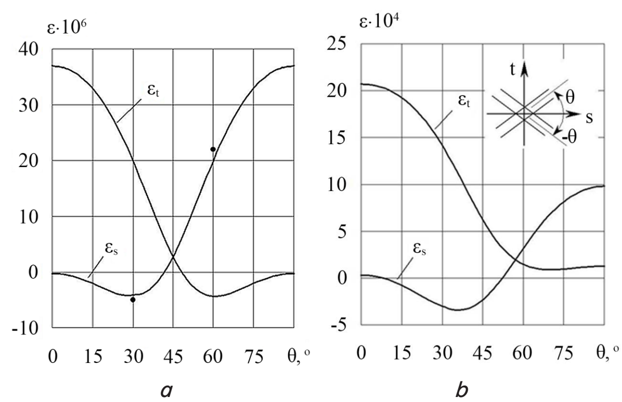


Fig. 2. Change of linear deformations from the winding angle for the carbon fiber reinforced plastic sample: *a* — temperature loading; *b* — pressure loading

Table 1 shows the calculated values of the angles θ^*, providing the dimensional stability of the sample in the axial direction, under two-component static loading. The combined effect of temperature and internal pressure is taken into account. We state that the calculated angles for various combinations of loads are located in the interval of ±41°47' and ±52°16'. The boundaries of the interval are limited by the angles, calculated under the partial action of loads.

Table 1

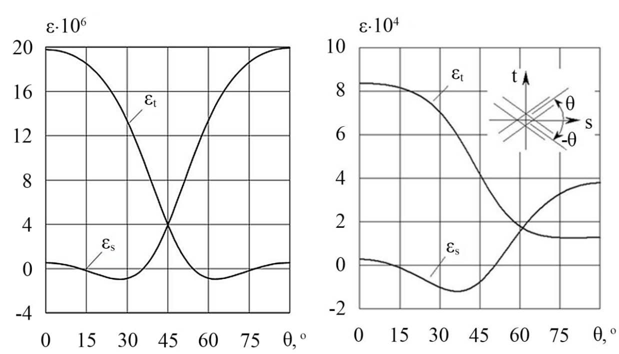
Angles θ^* ensuring axial dimensional stability

Temperature, K	Pressure, MPa					
	0	1.0	2.0	3.0	4.0	5.0
0	–	52.28	52.28	52.28	52.28	52.28
20	41.78	48.65	50.08	50.70	51.05	51.27
40	41.78	46.89	48.65	49.54	50.08	50.44
60	41.78	45.85	47.64	48.65	49.30	49.76
80	41.78	45.16	46.89	47.94	48.65	49.16
100	41.78	44.67	46.31	47.37	48.11	48.65

We investigate the deformation of the combined multilayer pipe, made by two-way spiral winding of reinforced tape on the mandrel. The mandrel is made of technical titanium VT1-0 with the parameters [16]: E=1.121×10^5 MPa, α=8.0×10^{-6} K^{-1}, ν=0.32. The tape – from KMU-4l carbon fiber reinforced plastic. The inner diameter of the pipe is d=96 mm, mandrel thickness is h_1=0.3 mm, and wall thickness is H=2.4 mm.

Fig. 3 shows the graphs of the linear deformations ε_s and ε_t against the winding angle. Fig. 3, *a* corresponds to the effect of temperature ΔT= 1 K, Fig. 3, *b* – to the action of internal pressure p=1 MPa.

Using the ASCP software package [1] and the finite element [17], we analyze the stress-strain parameters of the pipeline with the flowing fluid. The design scheme of the pipeline is shown in Fig. 4 (dimensions in mm). We consider the mode of undisturbed internal flow. Fluid parameters: ρ_f=1.02 g/cm^3, v_m=10 m/s, p_m=0.898 MPa, ΔT=100 K. Forces of friction of the liquid against the pipe walls are neglected. Let us compare three versions of structures differing in pipe design. Version I is a single-layer structure made of VT1-0 titanium (inner diameter d=96 mm, wall thickness H=2.4 mm). Version II is a multilayer combined structure made by two-way spiral winding of KMU-4l carbon fiber reinforced plastic on the VT1-0 titanium mandrel (d=96 mm, winding angles θ=±60°, mandrel thickness h_1=0.3 mm, wall thickness H=2 4 mm). Version III is a multilayer combined dimensionally stable structure. It differs from version II in winding angles – $\theta=\theta^*$=±12°48'.



a *b*

Fig. 3. Change of linear deformations from the winding angle for the combined sample: *a* — temperature loading; *b* — pressure loading

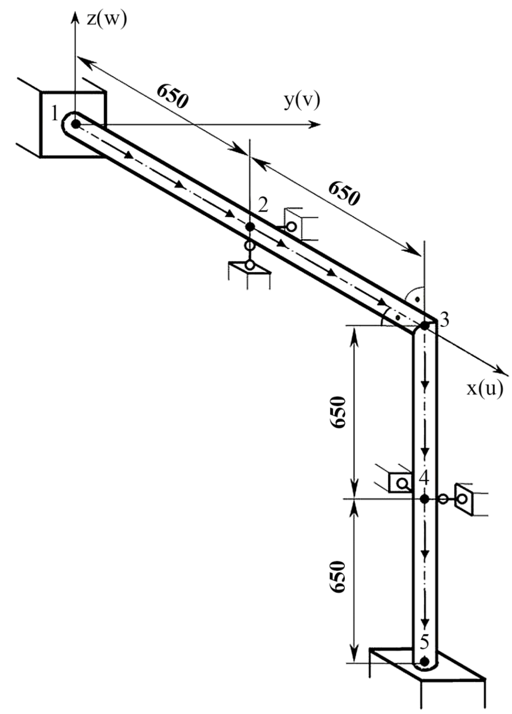


Fig. 4. Pipeline design scheme

Table 2 gives estimated displacements, longitudinal forces, maximum values of the bending moments and loads on supports for each version of the structure.

Table 2

Pipeline stress-strain parameters

Version of the structure	Displacement p. 3, mm		N, kN	M_y^{max}, kNm	Load on supports 2 and 4	
	u	w			$R_y^{(2)}$, kN	$R_x^{(4)}$, kN
I	1.036	1.036	4.388	–1.112	4.560	4.560
II	1.870	1.870	6.226	–0.3987	1.594	1.594
III	1×10^{-4}	1×10^{-4}	7.238	–7.102×10^{-5}	2.552×10^{-4}	2.552×10^{-4}

Each of the design versions of the pipeline gives noticeably different values of displacement and support reactions.

6. Discussion of the results of design justification of dimensionally stable structures

The results of numerical simulation (Fig. 2, 3, Tables 1, 2) indicate that the proposed design technique for multilayer composite pipelines allows creating dimensionally stable structures with two-component loading.

Indeed, from the analysis of the graphs in Fig. 2, it follows that:

– At temperature load, dimensional stability of the sample (thermal stability) is ensured in both axial and circumferential directions ($\varepsilon_s^{\Delta T}$=0 for θ^*=±41°47', $\varepsilon_t^{\Delta T}$=0 for θ^*=±48°13'). In this case, in the region of angles θ<41°47' with increasing temperature, the sample length decreases, and in the region θ>48°13' its diameter decreases.

– Under the internal pressure, dimensional stability of the sample is ensured only in the axial direction. At the same time, in the region of angles θ≥±60°, the circumferential deformations are rather small.

From the graphs in Fig. 3 it can be seen that the winding angles corresponding to ε_s=0 are: under the action of temperature $\theta^*_1=\pm12^o46'$ and $\theta^*_2=\pm35^o58'$, under the action of internal pressure – $\theta^*_1=\pm12^o53'$ and $\theta^*_2=\pm50^o34'$. Note, in the region of $\theta^*_1<\theta<\theta^*_2$, the internal pressure causes a decrease in the pipe length.

Analysis of the results of the computational experiment on modeling the stress-strain state of the pipeline with the flowing fluid showed the following. For version III of the pipeline, the estimated displacements, bending moments and loads on the supports are practically zero. Obviously, the use of dimensionally stable multilayer pipes eliminates axial deformations. As a result, the forces and stresses in the pipes associated with the bending deformations disappear, the supporting structure is unloaded.

Single-layer metal and combined pipes from carbon fiber reinforced plastic are in tension. Moreover, the tensile force in the pipe with a dimensionally stable structure (version III) is equal to the compressive force in the channel, i. e., $N\approx P$, where $P=pA_o$=7.238 kN. Here A_o is the area of the pipe opening. In this case, the expansion loads from the pipe are not transferred to the fittings.

Of course, this method of compensation of deformations has some limitations:

– According to the used hypotheses of the theory of thin shells, the normal stresses in the direction of the normal are considered to be zero. Only interlayer shear stresses are taken into account. In this, connections between the individual layers, as well as the connections between the fibers and the binder, are assumed to be ideal. In a real structure, this circumstance may require additional technological or design solutions to ensure joint deformation of the multilayer wall and exclude delamination.

– The problem considers the pipeline as an engineering object with low thermal conductivity. A homogeneous temperature field is assumed, that is, the same temperature is obtained in all layers. In real structures, when filling the pipeline, a temperature gradient and thermal shocks are obtained. But due to the small wall thickness of the pipeline and slow filling (quasi-static process), we can assume that the temperature field is aligned in thickness, and in the first approximation it can be considered as homogeneous.

– Internal pressure is considered as an action of a stationary internal fluid flow. In actual pipeline designs, hydraulic shocks and vibrations may occur during operation. Therefore, it is impossible to refuse all compensators, but their number can be reduced.

– The physicomechanical properties of materials vary with temperature. Therefore, the proposed pipeline designs will have dimensionally stable properties within the preservation of the properties of materials. For other temperature ranges, there will be other layer packages and reinforcement schemes.

Note, dimensionally stable multilayer pipes from FRC open up new approaches to the design of pipelines and pressure vessels. There are opportunities to create structures with preassigned (not necessarily zero) displacement fields [18]. The coordination of these fields with the fields of the initial technological displacements, as well as with the displacements of the coupled elastic elements and equipment when the operation mode changes. The scope of such structures is not limited to "hot" pipes. The results can be used in cryogenic engineering. Thus, such structures can be used in products of aviation and rocket-space technology, oil and gas industry, that is, in structures where the operation of pipelines is characterized by an increased level of tension with stringent requirements to overall and mass characteristics.

7. Conclusions

1. The obtained calculated relations for linear deformations of multilayer cylindrical shells under separate and combined temperature and internal pressure loading allow estimating the influence of winding angles and package parameters on the values of elastic deformations. Analysis of dependencies and strain values revealed the package structures, reinforcement schemes and package parameters that ensure the geometric stability of the pipeline in the axial direction under two-component static loading.

The presented structures were obtained on the basis of technical VT1-0 titanium and KMU-4l carbon fiber reinforced plastic. However, this is not the only combination of materials. Structures composed of stainless steels, niobium, organoplastic, and other materials have similar properties (dimensional stability).

2. The study of two-way reinforced and combined multilayer pipes, made by winding of reinforced tape on a thin-walled metal mandrel, allowed obtaining the dependences of the elastic strains on the reinforcement parameters. The possibility of self-compensation of axial deformations both under separate action of temperature and internal pressure, as well as with their combined action is demonstrated.

3. The calculated substantiation of the obtained dimensionally stable structures of multilayer cylindrical shells was performed. Using the example of the pipeline with the flowing fluid, the advantages of dimensionally stable structures are demonstrated. Thermal power loading of several versions of reinforcement schemes showed: the use of dimensionally stable multilayer pipes eliminates axial deformations. As a result, the forces and stresses in the pipes associated with the bending deformations disappear, the supporting structure is unloaded.

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