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CHAPTER 1

ANALYSIS OF THE TEMPERATURE EFFECT ON THE STRESS-STRAIN STATE OF THE TANK CAR BOILER DURING STEAMING

ABSTRACT

Uninterrupted transportation of bulk fuel and lubricant cargoes is one of the main tasks facing the railway transport of Ukraine. At the same time, the nomenclature (light and dark petroleum products, fuel oils) of bulk cargoes and the limited number (due to the obsolescence of the existing fleet of tank cars and the inability to renew them in military conditions) of the corresponding rolling stock require constant and high-quality cleaning of tank car boilers. This is due to the need for prompt use of tank cars for the transportation of various types of cargo. Modern boiler cleaning technologies are associated with the application of temperature loads to their boilers, namely with washing and steaming operations. This directly affects the stress-deformed state of tank car boilers. Which, in turn, determines the need for scientific research to determine the temperature effect on the stressed-deformed state of the tank car boiler during washing and steaming operations. It was the solution of such a scientific and applied task that became the goal of research, the results of which are presented in this section of the monograph.

To achieve the set goal, the following scientific and applied tasks were defined and solved. The existing information sources on the relevant topic were analyzed. Next, the technical description and requirements for modern designs of tank cars and their boilers are presented. Then, the used and adapted moment theory of shells is presented. On the basis of this mathematical theory, the load values at the control points of the tank car boiler are determined. Based on these results, the calculation model built in a modern computer computing and software complex was adjusted. Which was calculated using the finite element method and brought to an adequate level. The developed adequate finite element model includes the optimal number of finite elements and nodes: 10 182 768 elements and 18 655 084 nodes. Tetrahedrons and triangles are used as finite elements. With the use of the developed adequate finite-element model, temperature load calculations and simulation of the application of hot steam were carried out. At the same time, the temperature of hot steam is 160 °C, and the calculation period is 20 minutes. As a result of the calculations, it was found that the maximum heating temperature of the boiler is 71.3 °C when the temperature is loaded with steam. On the basis of the obtained results of temperature simulations, the input data for determining the stress-strain state of the tank car boiler were formed. The results of the calculations made it possible to establish that the maximum stress values are 173 MPa and do not exceed the permissible values. That is, when washing and steaming operations are carried out, the strength condition is fulfilled.

The obtained results of simulations of temperature loads and the stress-strain states corresponding to them will be useful in conducting further research and development works on the selected topic. In addition, the obtained results and achievements can be used in educational activities in the preparation of students of various levels of education.

KEYWORDS

Transport mechanics, railway transport, temperature influence, stress-strain state, cars, tank cars.

Today, one of the priority areas of railway transport is the need for stable transportation of bulk fuel and lubricant cargoes. At the same time, the nomenclature of bulk cargoes is wide and the demand for transportation of specific cargo is constantly changing. That is, there is a constantly changing need to transport: light and dark petroleum products, lubricants, fuel oils, and others. The above determines the need for particularly careful technical maintenance of the fleet of tank cars. Including washing and steaming operations in accordance with the technical regulations.

In general, the temperature of the steam used for washing and steaming works is 160 °C. A thorough scientific research analysis of the stress-deformed state of the tank car boiler during washing and steaming operations was not carried out. And this is exactly the goal set in this study.

The purpose of the manuscript is to highlight the results obtained from the study and analysis of the temperature effect on the stressed-deformed state of the tank car boiler during washing and steaming operations. At the same time, the operating temperature of the hot steam was considered to be 160 °C, which corresponds to the existing operating value.

To achieve the set goal, a number of the following scientific and technical problems were defined and solved:

1) analysis of information sources in which issues of temperature influence on machine-building structures, features of relevant mathematical and computer modeling, simulations are covered;

2) technical description and requirements for modern designs of domestic tank cars and their boilers. Such descriptions are determined in order to take into account the relevant features when creating a calculation model and conducting computational computer simulations;

3) adaptation and application of moment theory of shells for mathematical determination of load values at control points of tank car boiler construction;

4) creation of a spatial calculation computer model of a tank car boiler in a modern software complex. Adjusting the adequacy of the developed finite-element calculation model;

5) determination of features and implementation of computer simulations. Namely, the determination of the features of applying temperature loads, fixing the calculation model, setting the external

and initial temperatures, etc. At the same time, temperature loads arising from the action of hot steam of 160 °C are considered. Obtaining and analyzing the obtained results of temperature loads;

6) calculation of the stress-strain state of the tank car boiler according to the calculated case of temperature loads. Analysis of the obtained pictures of stress-strain states and comparison of calculated results with permissible values, which are taken in accordance with the characteristics of the material;

7) analysis of the obtained results and formulation of general conclusions.

The search for information sources was aimed at finding works that are devoted to highlighting the achievements and results of research on: determination of temperature effects on mechanical engineering structures, mathematical and computer modeling of processes in mechanical systems, analysis of stress-strain states in transport structures and, in particular, in freight cars.

In work [1], it was found that a specific factor in the production of cellular aggregate for transport structures made of polymer materials is multi-stage impregnation with the composition of the dressing and then the binder in the final operations, followed by drying and heat treatment of the cellular blocks. Which leads to uneven heat transfer of the binder from the central part of the panel to its peripheral end zones. The regularities of this non-uniform heat migration of the binder along the length of the cellular channel have been created and analyzed. It was found that these phenomena are caused by the hydrodynamic movement of the binder, which is mainly caused by temperature gradients. Taking this into account, a method of determining the thickness of the binder layer along the channels of the honeycombs was created with the given input data of the change in density and surface tension along the length of the honeycomb aggregate cell.

In the scientific work [2], the approach to mass optimization of shell transport structures was further developed. Which includes significantly improved components of fragments of known analogues, as well as new fragments that were not taken into account before. This approach made it possible to solve the complex multi-parameter problem of optimal design of the considered class of shell transport structures with almost no loss of accuracy. For this, the optimization process was divided into several stages according to the justified significance levels of the parameters included in the objective function - minimum mass. The analysis of the effectiveness of the reinforcement structure of the load-bearing skins and the preliminary optimization of the properties of the cellular aggregate were carried out, which significantly simplified the choice of their optimal parameters. It is shown that with a minimal gain in mass due to the optimal reinforcement scheme, which is approximately 5 % compared to a quasi-homogeneous shell, there is a real risk of doubling the mass of the shell when choosing a significantly suboptimal shell structure.

Article [3] is devoted to highlighting the results of the analysis and substantiating the fields of tolerances for the relevant types of technological defects in transport engineering structures. The tolerances for the deviation of the thickness of the formed product from the design value were determined. It was found that the input control determines the deviation of the thickness from the nominal value for a single-layer semi-finished product realized in the prepreg. Deviation in thickness from the nominal value includes components that arise during its formation. These components are associated with integral deviations of the technological mode of formation (including temperature) from that regulated by the relevant documentation. Analytical dependences on the reasonable assignment of tolerance fields for the physical and mechanical characteristics of the material with deviations in thickness, in the presence of local integrity violations in the form of voids, were obtained. In contrast to the existing models, the obtained dependencies made it possible to assess the quality of technological processes of forming semi-finished products and products from polymer composite materials according to the level of defects of the considered class.

The authors of the study [4] proposed a new approach to diagnosing the technical condition of structures that transmit mechanical energy. At the same time, the impact on their stress-strain state was considered. And dependencies were developed accordingly, the publication is devoted to highlighting the features of which.

The work [5] presents the design calculation of the automatic electro-hydraulic drive for the rotary movement of technological equipment. At the same time, special attention was paid to the influence of temperatures on the corresponding properties. However, such studies did not consider the possibility of using the developed method for non-load-bearing transport metal structures.

Research [6] is devoted to the modeling of the dynamics of a node of an element of a general design by the method of spectral analysis. At the same time, the influence of various factors on the stress state of the general structure was investigated. However, special attention was not paid to the issue of temperature influence, which is initially explained by the lack of such a task.

The authors of the article [7] studied, among other things, the temperature effect on the driving components of the power plants of transport. For this purpose, appropriate mathematical and computer models were developed and researched. However, their decisive component is compliance with the tasks of electrical equipment research.

In the scientific and applied work [8], the main attention is also paid to the tasks of researching the electrical equipment of vehicles. At the same time, the results of the conducted simulations, on the diagnosis of thermal effects, will be useful in solving similar problems for other structural components.

The article [9] presents the results of an analysis of the influence of the existing stress-deformed state of freight cars and the possibility of extending their service life. However, the possibility of their accident-free operational temperature loads was not considered.

The scientific study [10] is devoted to the analysis of the influence of the bending deformations of the car body on the indicators of interaction with the railway track. However, not enough attention has been paid to temperature deformations.

Article [11] highlights the results of research into the possibility of extending the service life of the bodies of universal semi-cars that have exhausted their regulatory resource. However, it is not specified how their stressed-strained state will be formed from possible temperature loads from the corresponding load.

The authors of the scientific publication [12] presented the results of computational modeling of the dynamic load of containers placed on a platform car during railway ferry transportation. Which corresponds to the second calculation mode. However, another case of specific perception of loads, namely from temperature influence, was not considered.

Modeling and simulation of the decision support system structure of an intelligent locomotive and evaluation of the quality of operation are given in [14]. It presents complex mathematical models and a proposed computer method for solving them. However, the issue of the possibility of applying appropriate approaches to calculated mathematical strength cases is not given.

Article [15] is devoted to the issues of theoretical and practical determination of the parameters of the on-board capacitive energy storage of metro rolling stock. However, the direction of their influence on the carrying capacity of the car body, or their temperature factors, were not investigated.

In works [17–19], the issue of safety of sea cargo transportation is comprehensively investigated. At the same time, the article [17] proposed an integrated approach to assessing the vulnerability of critical ship equipment and systems. And the work [18] gives the results of the study of the environmental efficiency of the operation of ships from the point of view of ensuring the efficiency of cargo transportation. And finally, in article [19], the issue of environmental efficiency of ship operation from the point of view of ensuring the efficiency of cargo transportation is considered. However, the considered models of stress-deformed vessels did not include the modes of temperature loads on their cases.

Summarizing the above, it can be noted that in all the above-mentioned scientific works, the peculiarities of the temperature effect on the stress-strain state of load-bearing systems of vehicles, and in particular the tank car boiler during steaming, were not investigated.

1.1 REQUIREMENTS FOR MODERN DESIGNS OF DOMESTIC TANK CARS AND THEIR BOILERS

One of the common models of tank cars was chosen for the study. It has the characteristics listed in **Table 1.1**.

Table 1.1 Characteristics of a typical tank car

Parameter name	Designation	Value
Inner radius of the boiler, m	R	1.60
Length of the cylindrical shell, m	L	9.775
Elliptical bottoms: – internal height of the oval part, m – height of the cylindrical part, m	 H_{ov} H_c	 0.64 0.06
Length of the car along the coupling axes of auto couplings, m	$2L_c$	12.02
Slope depth, m	H_{sl}	0.03
Slope volume, m^3	V_{sl}	0.02
Number of axles in the cart	n	2
Carrying capacity, t	Q_{rp}	70.5
Container mass and container tolerance, t	$T \pm \Delta T$	23.3
Acceleration of free fall, m/s^2	g	9.81

The image of the tank car boiler under study is shown in **Fig. 1.1**.

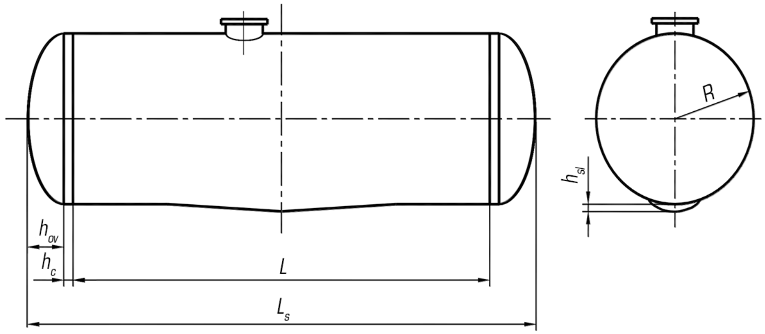

Fig. 1.1 Tank car boiler with elliptical bottoms

The tank car is intended for the transportation of petroleum products. Tank cars should be used as intended in accordance with the rules of technical operation of railways and operational documentation. The use of tanks for the transportation of other dangerous goods must be agreed with the manufacturing plant, the main organization of the product being transported, and the bodies that control traffic safety and operation at the discharge-filling points. Operational loads should not exceed those established by current regulatory documents. It is not allowed to replace elements of the tank car with others, which differ in design or materials from those provided in the manufacturer's drawings, during operation, without its consent.

Sampling from tanks containing a dangerous petroleum product must be carried out without releasing the product into the atmosphere, unless it is stipulated by the standard for the transported product.

Tank cars containing dangerous goods must have markings that characterize the transport danger of the goods. The marking should contain: danger sign, UN serial number, emergency card number.

The manufacturer guarantees compliance of tank cars with the requirements of the standard, subject to compliance with the rules of operation, maintenance and repair. The warranty period of operation of tank cars is established in the technical conditions for specific models of tank cars, but not less than 18 months from the day of commissioning. The warranty period for replaceable components, parts and accessories for tank cars is established by relevant regulatory documents for specific products.

Tests of tank cars should be performed under the values of climatic factors of the external performance environment "U" in accordance with the current regulatory documents.

Components of tank cars and tank cars as a whole that are manufactured must be checked for compliance with the requirements of regulatory and technical documentation during technical control.

During the control of tank cars, measuring devices should be used in accordance with the requirements of the technical documentation.

Fitting into the overall dimensions is checked by passing the tank cars through the overall frame.

Assembly and installation of the boiler, frame, carts, ladders, platforms, pouring and pouring fittings, safety devices, performance of the self-clutch mechanism, braking and other equipment of tank cars, tightness of the boiler with pouring and pouring fittings and safety devices, marking, color and quality of painting are controlled visually and by measurements during tests of tank cars in accordance with the design documentation and technical conditions.

The quality control of the welded joints of the boiler is carried out in accordance with the current regulatory documents. Mechanical tests of welded joints of boilers of tank cars are performed in accordance with current regulatory documents.

The quality of welded joints of boilers of tank cars operating under excessive pressure over 0.07 MPa should be tested in accordance with current requirements.

Reliability requirements are monitored based on the results of reliability tests according to a specially developed and agreed program and methodology.

The capacity of the safety valves must be selected according to the calculation in accordance with the current regulatory documents.

Incoming control of materials and component products, which are received for the manufacture of tank cars, must be carried out in the order established by the manufacturer in accordance with the current requirements.

Production, testing and acceptance of the boiler of tank cars must comply with the requirements of current regulatory documents.

The boiler should consist of a shell, two bottoms, a hatch, a hatch for installing a safety inlet valve, and a drain device. The material used for the production of the main elements of the boiler is in accordance with the current regulatory documents.

The boiler must be equipped with a fence, external stairs with a platform that allow for maintenance, repair and control of tank car draining and filling operations, as well as internal stairs fixed to the manhole shell for inspection and repair inside the boiler.

The deviation of the internal diameter of the shell of the tank car boiler from the nominal should be within the tolerance of ±6 mm.

Bumps, scratches and other defects (pits, sinks, etc.) are not allowed on the surface of the tank car boiler, if their depth exceeds the minus limit deviations provided by the relevant standards or technical conditions for the supply of metal.

The deviation from the straightness of the generator tank car boiler should not exceed 2 mm per 1 m of length, but not more than 20 mm over the entire length of the shell, without taking into account the local deviation from straightness in the welds.

Local dents and convexities with a height of no more than 5 mm per 1 m of length are allowed on the body of the tank car boiler, but no more than 2 units on each side of the shell.

The displacement of the edges of the sheets in the butt joints, the accuracy of the docking of the ends of the shells with the bottoms must be ensured within the tolerances for the displacement and removal of the edges provided by the current regulatory documents.

Welding of the tank car boiler and its elements, as well as welding of parts to the boiler, must be carried out by welders certified in accordance with current regulatory documents.

In the butt welds of the tank car boiler, a joint removal of the edges (inside and outside) is allowed within 10 % of the sheet thickness plus 3 mm, but not more than 5 mm, while the removal of the edges in the longitudinal seams is determined by the template, the length of which (by chord) is equal to 1/3 of the shell radius. The clumsiness of ring seams is determined by a ruler 300 mm long.

The tank car boiler must be symmetrically located on the pivot beams. Permissible displacement of the longitudinal axis of the boiler relative to the longitudinal axes of the backbone beams in the horizontal plane is no more than 5 mm.

The tank car boiler must be equipped with a drain-filling and safety valve.

Draining and filling fittings must include: a draining device, which must ensure the draining and filling of the product at the draining-filling points, the preservation of the product during the transportation of the tank car and include three sequentially installed locking devices:

– internal (main) locking device to ensure complete merging of the cargo and sealing of the tank car boiler during movement;

– additional locking device – disk valve for sealing the tank car boiler in the event of a malfunction of the main locking device;

– external shut-off device for sealing the tank car boiler in the event of a malfunction of the internal (main) and additional shut-off devices.

The distance from the surface of the drain device to the level of the rail head must comply with the current regulatory documents.

The body of the draining device before welding to the tank car boiler shell must be inserted inside flush with the internal surface of the boiler. Then it should be ensured that the joint is fully welded over the entire thickness of the tank car boiler shell.

Draining the product from the tank car boiler must be provided with a slope to the drain device with a height difference of 25...30 mm.

The cover of the hatch of the tank car boiler must ensure the density of the boiler (taking into account water hammer and steam pressure) at a pressure of at least 0.25 MPa (2.5 kgf/cm^2). A sealing ring must be installed on the cover.

There should be a device for installing a locking and sealing device on the cover of the tank car's boiler hatch.

The hatch cover of the tank car must be equipped with a device for detonating the cover in the event of its freezing or the presence of a vacuum in the boiler, as well as to ensure safe opening of the cover in the presence of excess pressure in the boiler.

Safety fittings must include a safety inlet valve.

The safety-inlet valve must be installed in the upper part of the boiler and serves to exclude a possible increase in pressure inside the boiler beyond the permissible level, as well as to prevent the formation of a vacuum. The safety-inlet valve cannot act to eliminate the vacuum after steaming operations.

The design of the safety inlet valve and its placement on the boiler should exclude the possibility of accidental release or theft of the product from the tank. The safety valve must be sealed after adjustment.

1.2 ADOPTION AND USE OF MOMENT THEORY OF SHELLS FOR MATHEMATICAL CALCULATION

When calculating for all types of loads, except for the calculation of bottoms from internal pressure, the tank car boiler is considered as a closed cylindrical shell, freely supported at the ends on diaphragms that are absolutely rigid in their plane and absolutely flexible perpendicular to this plane. Such diaphragms are the bottoms of the tank car boiler. To calculate the bottom and the adjacent zone of the cylindrical part of the boiler, the actual shape and elasticity of the tank car may be taken into account.

Let's highlight an infinitesimally small elementx of the middle surface of the shell with two longitudinal and two transverse sections *ABCD* (**Fig. 1.2, *a***). This element has the dimensions $dx = R_1 d\alpha$ and $ds = R_1 d\beta$, where α and β are the dimensionless coordinates of the cylindrical system. The coordinate $\alpha = x/R_1$ coincides with the generator *x*, and $\beta = s/R$ with the arc *s* of the cross section of the cylindrical part of the boiler.

Instead of the removed parts of the shell, it is possible to apply force factors to the longitudinal and transverse sections of the element *ABCD* (**Fig. 1.2, *b***): normal $T_1(\alpha,\beta)$ and $T_2(\alpha,\beta)$, transverse $Q_1(\alpha,\beta)$ and $Q_2(\alpha,\beta)$, shear $S_1(\alpha,\beta)$ and $S_2(\alpha,\beta)$ forces, bending $M_1(\alpha,\beta)$ and $M_2(\alpha,\beta)$ and torsional $M_{T1}(\alpha,\beta)$ and $M_{T2}(\alpha,\beta)$ moments.

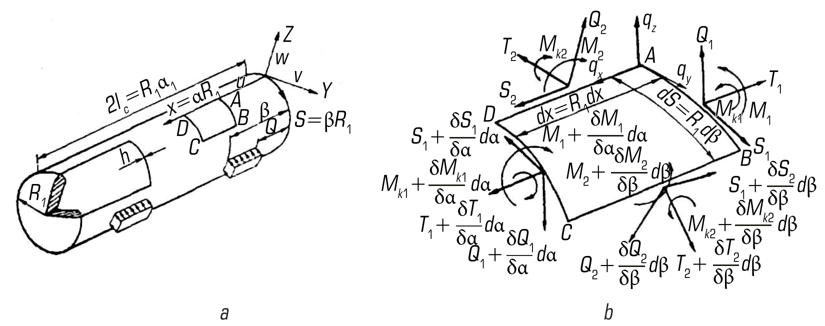

Fig. 1.2 Calculation scheme of the shell and its element

All these force factors, which determine the stress state of the shell, are related to the length unit of the corresponding side of the analyzed element:

$$S_1(\alpha,\beta) = S_2(\alpha,\beta) = S(\alpha,\beta), \tag{1.1}$$

$$M_{T1}(\alpha,\beta) = M_{T2}(\alpha,\beta) = M_T(\alpha,\beta). \tag{1.2}$$

Let's take the positive reference direction of the coordinates as shown in **Fig. 1.2, *a***. Along these three mutually perpendicular directions, let's decompose the complete displacement of the point, marking its components: longitudinal (along the generator) through $u(\alpha,\beta)$, tangential (along the tangent to the arc of the circle) – $v(\alpha,\beta)$ and radial (along the internal normal) – $w(\alpha,\beta)$.

These movements of the middle surface determine the deformed state of the shell. In addition, the angles of rotation of the longitudinal $\theta_2(\alpha,\beta)$ and $\theta_1(\alpha,\beta)$ transverse planes of sections are considered.

The external load can also be distributed along these three directions. The components of surface loading of the boiler shell have the following designations: longitudinal through $q_x(\alpha,\beta)$, tangential $q_y(\alpha,\beta)$ and radial $q_z(\alpha,\beta)$. The parentheses (α,β) indicate that the considered parameters are, in the general case, functions of two variables α and β, and are omitted in the future to shorten the record. Equilibrium equations are formulated to determine the unknown force factors acting on an infinitesimal element of the shell. Since these equations are not sufficient to solve the statically indeterminate problem under consideration, the conditions of non-discontinuity of deformations are used, i.e. connections between components of movements and deformations (geometric relations) and connections of deformations with force factors (physical relations).

The complete system of equations, consisting of equilibrium equations and strain continuity equations, is equivalent to the following equation in partial derivatives of the eighth order:

$$\nabla^2\nabla^2\nabla^2\nabla^2\Phi + 4k^4\frac{\partial^4\Phi}{\partial\alpha^4} = 0, \tag{1.3}$$

where $\nabla^2 = \partial^2/\partial\alpha^2 + \partial^2/\partial\beta^2$ – Laplace differential operator; $\Phi(\alpha,\beta)$ – enabling function.

The part of the expression of equation (1.3) containing $4k^4$ is calculated by the formula:

$$4k^4 = \frac{12(1-\mu^2)R_1^2}{h_1^2}, \tag{1.4}$$

where μ – Poisson's ratio; h_1 – thickness of the wall of the cylindrical part of the boiler.

Through the enabling function Φ, all power factors and displacements of the boiler shell can be determined. However, the calculation of the boiler based on the given equations of the moment theory of shells for all types of loading, especially taking into account the different thicknesses of the sheets that make up the boiler, is a rather difficult and time-consuming task. Therefore, it is advisable to use electronic computing machines to solve it. The machine calculation algorithm in matrix form can be constructed as follows.

Equation (1.3) by substitution is reduced to the form:

$$\Phi = \sum_{m=1}^{\infty} f_m(\beta)\sin\lambda\alpha, \text{ then}$$

$$\sum_{m=1}^{\infty}\left[\frac{d^8 f_m(\beta)}{d\beta^8} - 4\lambda_m^2\frac{d^6 f_m(\beta)}{d\beta^6} + 6\lambda_m^4\frac{d^4 f_m(\beta)}{d\beta^4} + \lambda_m^4\left(\lambda_m^4 + 4k^4\right)f_m(\beta)\right]\sin\lambda_m\alpha = 0, \qquad (1.5)$$

where m – row member number; $2l_c$ – length of the cylindrical part of the boiler; $f_m(\beta)$ – coefficient of the number of the series of the function, which depends on the coordinate β.

Condition (1.5) is fulfilled for any values α only if for each number the expression in square brackets is equal to zero.

Using this condition, it is possible to obtain a system of independent differential equations of the eighth order with respect to the coefficients $f_m(\beta)$ of the function series Φ. In the future, the index m is lowered, but it should be borne in mind that all calculated dependencies refer to the same number m of the member of the series.

The given expression of the function Φ has an important property: the forces and movements calculated based on it satisfy the boundary conditions adopted above, i.e. the resting of the ends of the boiler on the indicated diaphragms. These conditions mean that when $\alpha = 0$ and $\alpha = \alpha_1 = 2l_c/R_1$, $M_1 = T_1 = w = v = 0$.

To solve ordinary differential equations (1.5), let's formulate a characteristic equation of the form:

$$\left(\gamma^2 + \lambda^2\right)^2 + 4k^4\lambda^4 = 0. \qquad (1.6)$$

The roots of equation (1.6) are of the complex form:

$$\gamma_j = \pm c_l \pm i d_l, \qquad (1.7)$$

where $j = 1, 2, 3, \ldots, 8$; $l = 1, 2$.

Real parts c_l and d_l complex roots are calculated according to the formulas obtained from expression (1.6) after transferring its second term to the right part, extracting from the right and left parts the roots of the fourth and then the second degree, taking into account the rules used when performing these operations on complex numbers:

$$c_l^2 = \frac{\lambda}{2}\left\{(\lambda \pm k) + \left[(\lambda \pm k)^2 + k^2\right]^{\frac{1}{2}}\right\}, \qquad (1.8)$$

$$d_l = \frac{\lambda k}{2c_l}. \qquad (1.9)$$

In formula (1.8), the plus sign corresponds to the index $l=1$, and the minus sign corresponds to $l=2$.

The general integral of each of the homogeneous equations (1.5) has the form:

$$f(\beta)=C_1\varphi_1(\beta)+C_2\varphi_2(\beta)+C_3\varphi_3(\beta)+C_4\varphi_4(\beta)+C_5\varphi_5(\beta)+$$
$$+C_6\varphi_6(\beta)+C_7\varphi_7(\beta)+C_8\varphi_8(\beta), \tag{1.10}$$

where $\varphi_j(\beta)$ – partial integrals of the homogeneous equation.

It is advisable to present them in the form of:

$$\left.\begin{aligned}
\varphi_{1,5}(\beta)&=e^{c_{1,2}(\beta-\beta_0)}\cos d_{1,2}(\beta-\beta_0);\\
\varphi_{2,6}(\beta)&=e^{c_{1,2}(\beta-\beta_0)}\sin d_{1,2}(\beta-\beta_0);\\
\varphi_{3,7}(\beta)&=e^{-c_{1,2}(\beta+\beta_0)}\cos d_{1,2}(\beta+\beta_0);\\
\varphi_{4,8}(\beta)&=e^{-c_{1,2}(\beta+\beta_0)}\sin d_{1,2}(\beta+\beta_0).
\end{aligned}\right\} \tag{1.11}$$

In formula (1.11), β_0 means half of the central angle corresponding to the arc of the panel from which the cylindrical part of the boiler can be assembled.

Representing force factors M_1, M_2, T_1, T_2, Q_2 and displacements v, w, θ_2 in series by $\sin\lambda\alpha$, and S, M_k, Q_1, u and θ_1 series by $\cos\lambda\alpha$ using the formulas of V. Vlasov [20], through the known coefficients $f(\beta)$, it is possible to obtain the coefficients of the series of the listed force factors and displacements.

It is advisable to present the coefficient formulas in matrix form. Then, if to denote the vector of coefficients of force factors on the longitudinal edges $-\beta_0$ and β_0 the panel through the expression:

$$\vec{S}=\left\|S(-\beta_0)T_2(-\beta_0)Q_2(-\beta_0)M_2(-\beta_0)S(\beta_0)T_2(\beta_0)Q_2(\beta_0)M_2(\beta_0)\right\|, \tag{1.12}$$

and the vector of arbitrary integrations by $\vec{C}_j$, then it is possible to write:

$$\vec{S}=\|A\|\vec{C}_j. \tag{1.13}$$

Similarly, if the vector of displacement coefficients of the longitudinal edges:

$$\vec{u}=\left\|u(-\beta_0)v(-\beta_0)w(-\beta_0)\theta_2(-\beta_0)u(\beta_0)v(\beta_0)w(\beta_0)\theta_2(\beta_0)\right\|, \tag{1.14}$$

then it can be represented as:

$$\vec{u}=\|B\|\vec{C}_j. \tag{1.15}$$

In formulas (1.13) and (1.15) $\|A\|$ and $\|B\|$ are matrices of transformation of the vector of arbitrary integrations into vectors of forces and displacements. At the same time:

$$\left.\begin{aligned}\|A\| &= \|A_d\|\|A_{sq}\|;\\ \|B\| &= \|B_d\|\|B_{sq}\|,\end{aligned}\right\} \tag{1.16}$$

where $\|A_d\|$ and $\|B_d\|$ – diagonal matrices of the eighth order; $\|A_{sq}\|$ and $\|B_{sq}\|$ – square matrices of the eighth order.

In diagonal matrices, the non-zero elements are, respectively:

$$\left.\begin{aligned}&\frac{Eh_1}{R_1}\lambda^3,\ \frac{Eh_1}{R_1}\lambda^4,\ -2\frac{D}{R_1^3}\lambda^2k^2,\ 2\frac{D}{R_1^2}\lambda^2k^2,\\ &\frac{Eh_1}{R_1}\lambda^3,\ \frac{Eh_1}{R_1}\lambda^4,\ -2\frac{D}{R_1^3}\lambda^2k^2,\ 2\frac{D}{R_1^2}\lambda^2k^2,\\ &\lambda,\ 1,\ 2\lambda^2k^2,\ 2\frac{\lambda^2k^2}{R_1},\ 1,\ 2\lambda^2k^2,\ 2\frac{\lambda^2k^2}{R_1},\end{aligned}\right\} \tag{1.17}$$

where $D=\dfrac{Eh_1^3}{12\left(1-\mu^2\right)}$ – cylindrical stiffness of the shell; E – modulus of elasticity of the boiler material; μ, h_1 and R_1 have the previous values.

Denote by $\beta=-\beta_0$ the function $\varphi_j=p_j$, and by $\beta=\beta_0$ the function $\varphi_j=\delta_j$. Taking this into account, square matrices can be represented in the form:

$$\|A_{sq}\| = \begin{Vmatrix}(\alpha_s p_1-\beta_s p_2)(\alpha_s p_2+\beta_s p_1)-(\alpha_s p_3+\beta_s p_4)-(\alpha_s p_4-\beta_s p_3) & \dots\\ \vdots \quad \vdots \quad \vdots \quad \vdots & \dots\\ (\alpha_s \delta_1-\beta_s \delta_2)(\alpha_s \delta_2+\beta_s \delta_1)-(\alpha_s \delta_3+\beta_s \delta_4)-(\alpha_s \delta_4-\beta_s \delta_3) & \dots\\ \vdots \quad \vdots \quad \vdots \quad \vdots & \dots\end{Vmatrix}, \tag{1.18}$$

$$\|B_{sq}\| = \begin{Vmatrix}(\alpha_u p_1-\beta_u p_2)(\alpha_u p_2+\beta_u p_1)-(\alpha_u p_3+\beta_u p_4)-(\alpha_u p_4-\beta_u p_3) & \dots\\ \vdots \quad \vdots \quad \vdots \quad \vdots & \dots\\ (\alpha_u \delta_1-\beta_u \delta_2)(\alpha_u \delta_2+\beta_u \delta_1)-(\alpha_u \delta_3+\beta_u \delta_4)-(\alpha_u \delta_4-\beta_u \delta_3) & \dots\\ \vdots \quad \vdots \quad \vdots \quad \vdots & \dots\end{Vmatrix}. \tag{1.19}$$

In matrices (1.18) and (1.19) to shorten the record, only the first halves of the two terms corresponding to the effort S and displacement u at the edges of the panel $-\beta_0$ and β_0. The next four elements are the same as the first, but in them the coefficients α_s and β_s must be replaced by α_{1s} and, α_{1s} and the coefficients α_u and β_u by α_{1u} and β_{1u}. In addition, the functions $p_{1,\,2,\,3,\,4}$ and $\delta_{1,\,2,\,3,\,4}$ are replaced by $p_{5,\,6,\,7,\,8}$ and $\delta_{5,\,6,\,7,\,8}$, respectively. The other six terms, marked in the matrices with dots, correspond to the written terms if the coefficients and with indices are replaced by coefficients α and β with indices S, on the coefficients with indices T, Q and M, and coefficients with indices u are replaced by coefficients with indices v, w and θ.

The values of the coefficients are given in **Table 1.2**. In addition, in the matrices $\|A_{sq}\|$ and $\|B_{sq}\|$, the third, fourth, seventh and eighth elements of the terms with the coefficients having indices T, M, v and θ, must have signs reversed to the signs of these elements in the terms with indices S and u.

Table 1.2 The values of the coefficients

Index	Coefficients			
	α	β	α_1	β_1
S	c_1	d_1	c_2	d_2
T	1	0	1	0
Q	$\left(d_1^3-3c_1^2d_1\right)+$ $+(2-\mu)\lambda^2 d_1$	$\left(c_1^3-3c_1d_1^2\right)-$ $-(2-\mu)\lambda^2 c_1$	$-\left(d_2^3-3c_2d_2^2\right)-$ $-(2-\mu)\lambda^2 d_2$	$-\left(c_2^3-3c_2d_2^2\right)+$ $+(2-\mu)\lambda^2 c_2$
M	$-2c_1d_1$	$\left(c_1^2-d_1^2\right)-\mu\lambda^2$	$2c_2d_2$	$-\left(c_2^2-d_2^2\right)+\mu\lambda^2$
u	$\left(c_1^2-d_1^2\right)+\mu\lambda^2$	$2c_1d_1$	$\left(c_2^2-d_2^2\right)+\mu\lambda^2$	$2c_2d_2$
v	$-\left(c_1^3-3c_1d_1^2\right)+$ $+(2+\mu)\lambda^2 c_1$	$\left(d_1^3-3d_1c_1^2\right)+$ $+(2+\mu)\lambda^2 d_1$	$-\left(c_2^3-3c_2d_2^2\right)+$ $+(2+\mu)\lambda^2 c_2$	$\left(d_2^3-3d_2c_2^2\right)+$ $+(2+\mu)\lambda^2 d_2$
w	0	1	0	1
θ	$-d_1$	c_1	d_2	$-c_2$

Let's use the displacement method to calculate the tank car boiler. Let's present the cylindrical part of the tank car boiler divided into separate panels, which are the main elements of the calculation scheme. As the boundaries of these elements, it is expedient to take the lines of transition from one thickness of the sheet to another, intersections of sudden changes in load, generating lines along which concentrated forces are applied. For the hydrostatic pressure of the liquid, which is variable along the arc of the section of the cylindrical part of the tank car boiler, it is advisable to choose the width of the panels in such a way that it is possible to accept the load on it as constant.

Let's apply bonds to the longitudinal edges $-\beta_0$ and β_0 of selected panels, and then one by one, let's assign to these bonds forced displacements u, v, w and θ_2, which change along the generating line according to the law $\cos\lambda\alpha$ or $\sin\lambda\alpha$ with a unit amplitude. These movements will be single.

For each main element from equation (1.15) let's find the vectors:

$$\vec{C}_j=\|B\|^{-1}\vec{u}, \tag{1.20}$$

where $\|B\|^{-1}$ – inverse matrix $\|B\|$.

In a vector $\vec{u}$, one of the components in sequence is equal to one, and all others are equal to zero.

The vector of reactions of the longitudinal edges at all considered single displacements according to equations (1.13) and (1.20) is:

$$\vec{S} = \|A\|\|B\|^{-1}\vec{u}. \tag{1.21}$$

So, the reaction matrix of the main element can be presented in the form:

$$\|r\| = \|A\|\|B\|^{-1}. \tag{1.22}$$

Formulas (1.13) and (1.15) are valid for homogeneous equations. In the case of an inhomogeneous equation, when the right-hand side of equation (1.3) is not equal to zero, these formulas will have the form:

$$\left.\begin{aligned} \vec{S} &= \|A\|\vec{C}_j + \vec{S}_{part}; \\ \vec{u} &= \|A\|\vec{C}_j + \vec{u}_{part}, \end{aligned}\right\} \tag{1.23}$$

where $\vec{S}_{part}$ and $\vec{u}_{part}$ are, respectively, the vectors of partial values of force factors and displacements determined by the right-hand side of the inhomogeneous equation, which depends on the nature of the surface external load.

So, if the power element of the boiler shell is loaded with surface radial forces, then the vectors $\vec{S}_{part}$ and $\vec{u}_{part}$ also have non-zero components:

$$\left.\begin{aligned} T_{2part} &= \frac{Eh_1}{\Delta R_1}Z; \\ M_{2part} &= -\frac{D\mu\lambda^2}{\Delta R_1}Z; \\ u_{part} &= \frac{\mu}{\lambda\Delta}Z; \\ w_{part} &= \frac{1}{\Delta}Z, \end{aligned}\right\} \tag{1.24}$$

where Z – coefficient of expansion in a series along $\sin\lambda\alpha$ the radial load, which is transmitted to the main element of the shell of the tank car boiler:

$$\Delta = \frac{Eh_1\left[h_1^2\lambda^4 + 12R_1^2\left(1-\mu^2\right)\right]}{12R_1^4\left(1-\mu^2\right)}. \tag{1.25}$$

If the main element is loaded with surface tangential loads, then the non-zero components will be:

$$S_{part} = -\left[1 + \frac{h_1^2}{12\left(1-\mu^2\right)R_1^2}\lambda^4\right]\frac{Eh_1}{\Delta R_1 \lambda}Y, \tag{1.26}$$

$$v_{part} = 2\left[\left(1+\mu\right) + \frac{h_1^2}{12\left(1-\mu^2\right)R_1^2}\lambda^4\right]\frac{1}{\Delta\lambda^2}Y, \tag{1.27}$$

where Y – coefficient of expansion into a series of tangential load transmitted to the main element of the shell of the tank car boiler.

Components of vectors of private solutions for the main element loaded with surface longitudinal forces:

$$\left.\begin{aligned} N_{2\,part} &= -\frac{Eh_1^3\mu\lambda^3}{12R_1^3\left(1-\mu^2\right)\Delta}X; \\ M_{2\,part} &= -\frac{Eh_1^3\mu^2\lambda}{12R_1^2\left(1-\mu^2\right)\Delta}X; \\ u_{part} &= \frac{1+\dfrac{h_1^2}{12R_1^2}\lambda^4}{\Delta\lambda^2}X; \\ w_{part} &= \frac{\mu}{\lambda\Delta}X, \end{aligned}\right\} \tag{1.28}$$

where X – coefficient of expansion in a series by $\cos\lambda\alpha$.

With the known law of load distribution along the coordinate lines α, the calculation of expansion coefficients Z, Y and X is carried out according to the Euler-Fourier formulas [21, 22].

The tank boiler is a symmetrical structure with respect to the average section, symmetrically loaded by external forces – support, hydrostatic and internal pressures, and for frameless tank cars – by longitudinal tensile and compressive forces. Therefore, members of the series with odd numbers are kept in the series, i.e. $m=1, 3, 5$.

For bearing radial and tangential loads, uniformly distributed over the sections of the boiler supports, the series coefficients are determined by the formulas:

$$\left.\begin{aligned} Z &= \frac{4q_{z1}}{m\pi}; \\ Y &= \frac{4q_{y1}}{m\pi}, \end{aligned}\right\} \tag{1.29}$$

where q_{z1} and q_{y1} – the intensities of radial and tangential loads uniformly distributed over the entire length of the main element, respectively.

For an inhomogeneous equation, the vector of arbitrary integrations is found from formula (1.23) taking into account the obtained values $\vec{u}_{part}$, that is, from the expression:

$$\vec{C}_j = \|B\|^{-1}\vec{u} - \|B\|^{-1}\vec{u}_{part}. \quad (1.30)$$

Therefore, the vector of reactions on the longitudinal edges of the main element subjected to external forces, according to the first formula (1.23) and equation (1.22), will be equal to:

$$\vec{S} = \|A\|\|B\|^{-1}\vec{u} - \|A\|\|B\|^{-1}\vec{u}_{part} + \vec{S}_{part} = \|r\|\vec{u} - \|r\|\vec{u}_{part} + \vec{S}_{part}. \quad (1.31)$$

The second and third components of formula (31) at a given load represent an eighth-order vector:

$$\vec{S}_{\Sigma part} = -\|r\|\vec{u}_{part} + \vec{S}_{part}. \quad (1.32)$$

Formula (1.31) is convenient for machine calculations of boiler shells made of panels of the same length, but of different thicknesses.

A boiler composed of panels with different radii of curvature and made of different materials can be calculated in a similar way.

Consider the calculation sequence of a boiler with a symmetrical design and loading, the cylindrical part of which consists of six panels (**Fig. 1.3**).

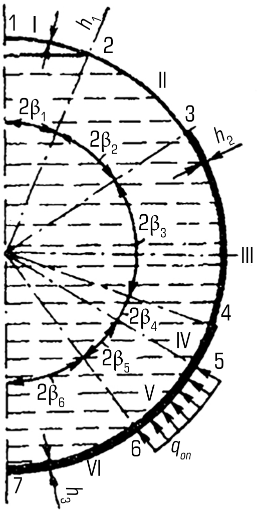

Fig. 1.3 Cross section of the boiler shell

In **Fig. 1.3**, panel numbers are indicated by Roman numerals. Nodal intersections are indicated by Arabic numerals. The length of the arc of the *i*-th panel is equal to $2R\beta_i$. Having constructed reaction matrices $\|r\|$ and vectors $\vec{S}_{\Sigma part}$ for each of the main elements, let's fulfill the condition of conjugation of the panels that make up the shell. These conditions are expressed in the equality of forces along the panel joining lines. For this, it is necessary to equate the last four components of the vector $\vec{S}$, i.e. $S(\beta_0)$, $T_2(\beta_0)$, $Q_2(\beta_0)$ and $M_2(\beta_0)$ respectively to the first four components of the vector of the next panel, i.e. $S(-\beta_0)$, $T_2(-\beta_0)$, $Q_2(-\beta_0)$ and $M_2(-\beta_0)$. As a result of such superimposition of vectors, let's get a canonical system of equations of the displacement method.

Such a canonical system must satisfy the boundary conditions at the initial edge of the first and the final edge of the last panel.

According to the conditions of symmetry on the generators of the boiler, which coincide with the upper and lower ends of the vertical diameter $v=\theta_2=Q_2=S=0$. Therefore, in the canonical order system *n*, let's cross out the second, fourth, $n-2$, and *n*-th columns, as well as the second, fourth, $n-2$, and *n*-th row. After solving such a system that has an order $n-4$, let's determine the vectors $\vec{v}$ of each panel of the calculation scheme, and based on them and formula (1.31) – the vectors of power factors $\vec{S}$. Let's multiply the obtained vectors of force factors and displacements by the diagonal matrix $\|\alpha_i\|$ of the eighth order, the elements of which are $\cos\lambda\alpha$, $\sin\lambda\alpha$, $\sin\lambda\alpha$, $\sin\lambda\alpha$, $\cos\lambda\alpha$, $\sin\lambda\alpha$, $\sin\lambda\alpha$, $\sin\lambda\alpha$.

At certain values of the argument α_i, after summing over the number of retained members of the series *m*, let's determine the force factors and displacements in the cross-sections of the shell of the tank car boiler:

$$\left.\begin{aligned}\vec{S}(\alpha_i) &= \sum_{m=1}^{m}\|\alpha_i\|_m \vec{S}_m;\\ \vec{u}(\alpha_i) &= \sum_{m=1}^{m}\|\alpha_i\|_m \vec{u}_m.\end{aligned}\right\} \qquad (1.33)$$

In formula (1.33) $\vec{S}(\alpha_i)$ and $\vec{u}(\alpha_i)$ are, respectively, the vectors of force factors and displacements in the intersections α_i, and the indices *m* emphasize that the matrices and vectors in the right-hand side of the equalities depend on the number of the member of the series.

In addition to the forces that are components of the vector $\vec{S}$, normal forces $T_1(\alpha_i)$ and bending moments $M_1(\alpha_i)$ on the cross-sections are important for assessing the stress state of the boiler. The coefficients of the series of these power factors are determined from the elasticity ratios using the formulas:

$$\left.\begin{aligned}T_1 &= \mu T_2 - \frac{Eh_1}{R_1}\lambda u;\\ M_1 &= \mu M_2 + \lambda^2 \frac{D}{R_1^2}\left(1-\mu^2\right)w.\end{aligned}\right\} \qquad (1.34)$$

So:

$$\left.\begin{aligned} T_1(\alpha_i) &= \sum_{m=1}^{m} T_{1m} \sin\lambda\alpha_i; \\ M_1(\alpha_i) &= \sum_{m=1}^{m} M_{1m} \sin\lambda\alpha_i. \end{aligned}\right\} \tag{1.35}$$

The normal stresses on the transverse σ_1 and longitudinal σ_2 surfaces of the sections of the cylindrical part of the boiler are determined by the formulas:

$$\left.\begin{aligned} \sigma_1 &= \frac{T_1}{h_1} \pm \frac{6M_1}{h_1^2}; \\ \sigma_2 &= \frac{T_2}{h_1} \pm \frac{6M_2}{h_1^2}. \end{aligned}\right\} \tag{1.36}$$

The specified mathematical approach to determining the stresses in the control locations of the tank car boiler was implemented with the help of the modern Mathcad computer software complex. Solving the relevant equations made it possible to obtain stress values, which became the basis for adjusting the adequacy of the developed calculated finite-element model.

1.3 ANALYSIS OF THE TEMPERATURE EFFECT ON THE STRESS-STRAIN STATE OF THE TANK CAR BOILER DURING STEAMING

Modeling/simulation of the temperature effect of hot steam at a temperature of 160 °C on a tank car boiler. Such a temperature effect is standard during a typical washing-steaming process of tank car boilers. At the same time, the time of the washing-steaming technological operation is from 15 minutes to 1 hour, depending on the external temperature and the initial temperature of the tank car boiler with cargo residues.

In the case under investigation, the following input data are taken into account: steaming time is 20 minutes; the temperature of the supplied steam is 160 °C; the external temperature is equal to 20 °C; the temperature of the tank car boiler with cargo residues is 20 °C.

At the first stage, a calculation model of a tank car boiler was created in a modern software computing environment, in natural size (**Fig. 1.4**).

At the same time, the above technical characteristics and requirements for its design are taken into account. Steel 09G2D was chosen as the material. Taking into account the peculiarities of the given task, the loading hatch was not taken into account in the model of the tank car boiler. Fixation of the calculation model is carried out by zonal abutments on the stud nodes, which are given below.

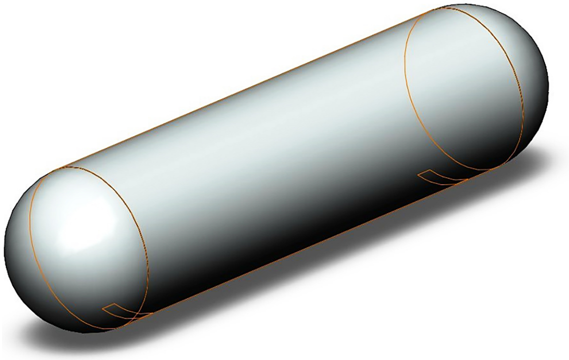

Fig. 1.4 Calculated computer model of a tank car boiler

At the second stage, the adequacy of the developed calculation model was adjusted by comparing theoretically (mathematically, based on the method of moment theory of shells) obtained stress values at the control points of the tank car boiler with those determined by computer modeling. Namely, the values were taken in the control locations of the bottoms and the shells. On the basis of the performed tuning, it was found that an adequate calculation model will have the optimal parameters of the finite element mesh presented below. The number of mesh elements was 10 182 768, nodes – 18 655 084. The maximum size of the mesh element is 11 mm, the minimum is 0.55 mm, the maximum aspect ratio of the elements is 41.031, the percentage of elements with an aspect ratio less than three is 99.3, more than ten is 0.0215.

The finite element model of the tank car boiler is shown in **Fig. 1.5**.

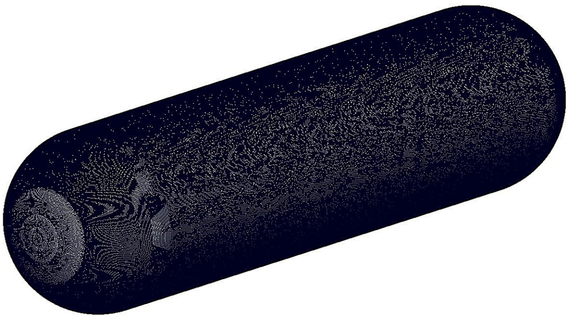

Fig. 1.5 Computational finite-element model of a tank car boiler

At the next (third) stage of research, temperature analysis was carried out. Temperature loads were applied to all internal faces of the tank car boiler. At the same time, the temperature effect was equal to 160 °C, and the convection coefficient was selected from **Table 1.3**.

Table 1.3 The values of the convection coefficient

No.	Average value	Convection coefficient (W/m²·K)
1	Air (natural convection)	5–25
2	Air/superheated steam (forced convection)	20–300
3	Steam (condensing)	4 000–20 000

The calculated model with applied temperature loads is shown in **Fig. 1.6**.

At the fourth stage, the application of loads from the external temperature, which is equal to 20 °C, to the external faces of the calculation model was simulated. At the same time, the convection coefficient was also chosen from **Table 1.3**. The model with applied external temperature loads is shown in **Fig. 1.7**.

Later, the characteristics of the transient process of temperature analysis were specified. Namely, the process time is 20 minutes, and the process step is 1 minute.

At the fifth stage of the research, the initial temperature of the tank car boiler with cargo remains was set at 20 °C.

The obtained temperature load modeling data made it possible to determine temperature analysis patterns at all 20 calculation steps. At the same time, it was found out that the construction undergoes the greatest temperature effect in the fifth step (**Fig. 1.8**) and is 71.3 °C.

At the sixth stage of research, in order to find out the stress-deformed state of the tank car boiler due to temperature effects, a new study was performed. This is how the calculation model was first attached (**Fig. 1.9**).

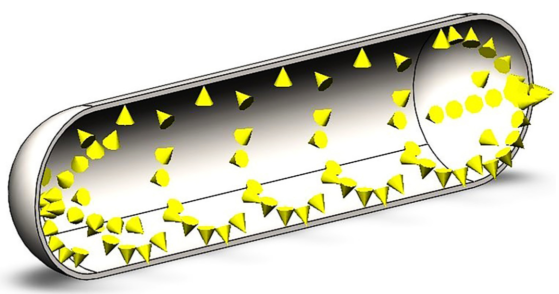

Fig. 1.6 Section of the calculation model of the tank car boiler with applied loads from the action of hot steam

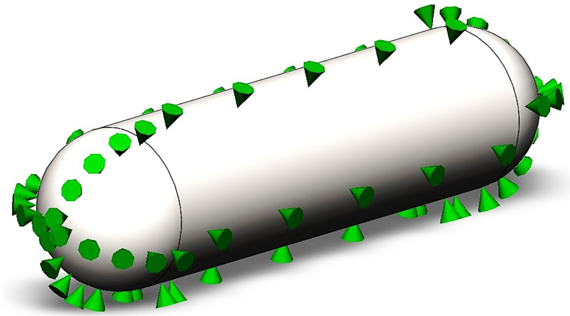

Fig. 1.7 Calculation model of a tank car boiler with applied loads from external temperature

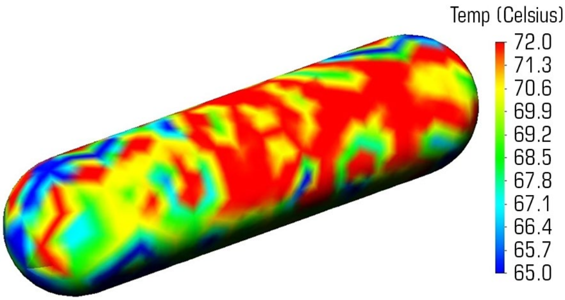

Fig. 1.8 Chart of temperature loads of a tank car boiler

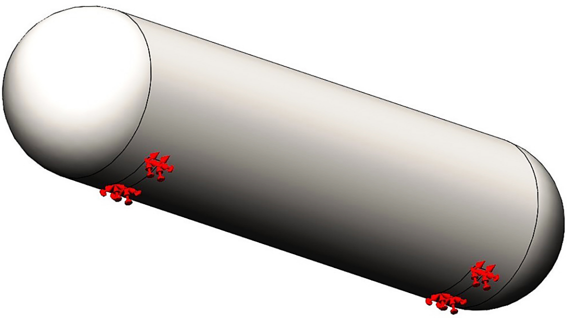

Fig. 1.9 Schematic of fixing the tank car boiler

After that, the previously obtained temperature loads were transferred to be taken into account in static studies. Temperature loads corresponding to the fifth calculation step were chosen as the calculation case. As a result of the calculations, the picture of the stress state due to the action of temperature loads is presented in **Fig. 1.10**.

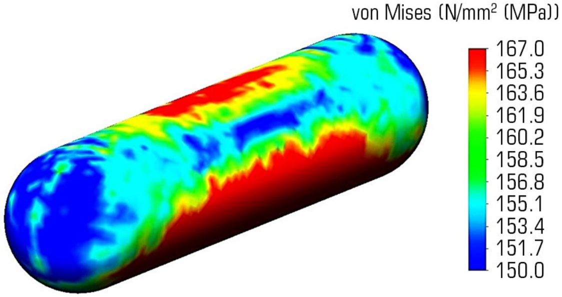

Fig. 1.10 Picture of the stress state of the calculation model of the tank car boiler under the influence of temperature loads

It was found that the maximum stress value is 173 MPa and does not exceed the permissible values.

CONCLUSIONS

The article presents the results of a scientific and applied research on determining the temperature effect on the stressed-deformed state of the tank car boiler during washing and steaming operations.

During the creation of the estimated finite-element model of the tank car boiler, the following basic requirements for its design were taken into account. The quality of welded joints of boilers of tank cars operating under excessive pressure over 0.07 MPa should be tested in accordance with current requirements. The boiler should consist of a shell, two bottoms, a hatch, a hatch for installing a safety inlet valve, and a drain device. The material used for the production of the main elements of the boiler is in accordance with the current regulatory documents. The deviation of the internal diameter of the bottom of the tank car boiler from the nominal should be within the tolerance of ±6 mm. Bumps, scratches and other defects (pits, sinks, etc.) are not allowed on the surface of the tank car boiler, if their depth exceeds the minus limit deviations provided by the relevant standards or technical conditions for the supply of metal. The deviation from the straightness of the generator tank car boiler should not exceed 2 mm per 1 m of length, but not more than 20 mm over the entire length of the shell, without taking into account the local deviation from straightness in the welds. Local dents and convexities with a height of no more than 5 mm per 1 m of length are allowed on the body of the tank car boiler, but no more than 2 units on each side of the shell. In the butt welds of the tank car boiler, a joint removal of the edges (inside and outside) is allowed within 10 % of the sheet thickness plus 3 mm, but not more than 5 mm, while the removal of the edges in the longitudinal seams is determined by the template, the length of which (by chord) is equal to 1/3 of the shell radius. The clumsiness of ring seams is determined by a ruler 300 mm long. Longitudinal welds of adjacent shells and bottom welds must be offset relative to each other by at least 100 mm. The main dimensions of tank car boiler bottoms must comply with current regulatory documents, while the reduction in sheet thickness in the bottoms after stamping should not exceed 15 % of the nominal sheet thickness.

The moment theory of shells used and adapted to the given tasks is presented. On the basis of this mathematical theory, the load values at the control points of the tank car boiler are determined. Based on these results, the calculation model built in a modern computer computing and software complex was adjusted. It was calculated using the finite element method and brought to an adequate level.

The developed finite element model includes the optimal number of finite elements and nodes. Tetrahedrons and triangles are used as finite elements. The optimal number of finite elements includes 10 182 768 elements and 18 655 084 nodes.

As a result of the calculations, it was found that with a steam temperature load of 160 °C, the highest stress temperatures in the boiler are 71.3 °C.

The results of the calculations made it possible to establish that the maximum stress values are 173 MPa and do not exceed the permissible values. That is, when washing and steaming operations are carried out, the strength condition is fulfilled.

The obtained results of simulations of temperature loads and the stress-strain states corresponding to them will be useful in conducting further research and development works on the selected topic. In addition, the obtained results and achievements can be used in educational activities in the preparation of students of various levels of education.

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