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Load Calculation of the Load-carrying Structure of a Tank Car in Operating Modes

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The article determines dynamic load and strength of the load-carrying structure of a tank car with elastic-friction connections in the boiler bearings and between boiler bearings and the boiler bearer. It was established that the use of elastic-friction connections allows to reduce the dynamic load on the tank car by almost 36% in comparison with the prototype. The results of stress calculation showed that the maximum equivalent stresses in the load-carrying structure of a tank car occur in the interaction area of the center sill with the draw-bar and do not exceed the allowable values. The conducted research will allow to increase the operational efficiency of tank cars by reduction of operating costs, and will also promote the creation of their innovative designs.

Keywords: transport mechanics, tank car, load-carrying structure, dynamic load, strength, fatigue strength coefficient.

Визначення навантаженості несучої конструкції вагона-цистерни при експлуатаційних режимах

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Підвищення ефективності експлуатації залізничної галузі вимагає впровадження інноваційних конструкцій рухомого складу, зокрема вагонів. Найбільш розповсюдженим типом вагону для перевезення наливних вантажів є вагони-цистерни. Несуча конструкція вагонів-цистерн випробовує дії значних циклічних навантажень в експлуатації. Це викликає їх пошкодження. В матеріалах статті наведені результати щодо удосконалення несучої конструкції вагона-цистерни шляхом зменшення динамічної навантаженості посередництвом впровадження пружно-фрикційних зв'язків між котлом та його опорами, а також між опорами та рамою. Для обґрунтування запропонованого рішення використано класичні методи теорії коливань та динаміки вагонів, методи розв'язання диференціальних рівнянь руху, зокрема, метод Рунге-Кутта, реалізований в програмному комплексі MathCad, а також метод скінчених елементів, здійснений в SolidWorks Simulation. Проведено визначення динамічної навантаженості та міцності несучої конструкції вагона-цистерни з пружно-фрикційними зв'язками в опорах котла та між опорами та рамою. Встановлено, що використання пружно-фрикційних зв'язків дозволяє зменшити динамічну навантаженість вагона-цистерни у порівнянні з прототипом майже на 36%. Результати розрахунку на міцність показали, що максимальні еквівалентні напруження в несучій конструкції вагона-цистерни виникають в зоні взаємодії хребтової балки зі шворневою та не перевищують допустимих значень. При цьому проектний строк служби несучої конструкції більше ніж на 20% вищий за строк служби вагона-прототипу. Коефіцієнт опору втомі з урахуванням запропонованих конструкційних рішень склав 4,2, що вдвічі перевищує допустимий. Проведені дослідження дозволять підвищити ефективність експлуатації вагонів-цистерн шляхом зменшення витрат на утримання, а також сприятимуть створенню їх інноваційних конструкцій.

Ключові слова: транспортна механіка, вагон-цистерна, несуча конструкція, динамічна навантаженість, міцність, коефіцієнт опору втомі

Introduction

Ensuring the efficient operation of railroads' rolling stock as a leading branch of the transport network requires putting modern car designs in operation. At the same time, promotion of competitiveness of the railway industry leads to increased requirements not only for technical-and-economic indicators of railroads rolling stock, but also for the possibility of adapting structures to the appropriate operating conditions.

One of the busiest load-carrying types of cars in operation are tank cars, due to the mobility of the goods transported in them. Most of them are tank liquid cargos that have their own degree of freeness due to the tank ullage.

Review of the research sources and publications

Areas for improvement of the design of railway tank cars are discussed in [1]. The paper presents an improved design of a tank car, the outstanding feature of which is that the boiler has conical open-end inserts and a gapless coupler drawbar hook.

Paper [2] considers the options for design and technical solutions which may improve the efficiency of liquid cargo transportation in tank cars. The authors of the article have chosen the most rational schematic construction, which will help increase the productivity of tank cars usage.

It is important to say that the mentioned paper does not propose measures to reduce the dynamic load on the load-carrying structures of tank cars in operation.

The results of a computer-based simulation of the hydrodynamic load of the manhole area of the tank car are covered in the article [3]. The paper presents the study of the load on the joint when changing the height of the part of the manhole shell, located inside the boiler. However, no design solutions have been proposed to reduce the load on the tank car boiler.

Theoretical aspects of determining the residual life of a tank car for dangerous cargo are considered in [4]. The paper presents the structural scrutiny of load-carrying structures of the tank cars, which have depleted their guideline lives. The peculiarities of tank cars' testing technique are highlighted in this research. However, the paper does not specify any solutions for the possibility of reducing the load on the load-carrying structures of tank cars to extend their service life.

Paper [5] considers the load on the load-carrying structure of a tank car under operation conditions, taking into account the shunting collision of the tank car. The authors of this scientific work have determined the influence of liquid cargo on the dynamic load of the boiler.

Also, the study of the impact of liquid cargo on the load of the tank car boiler is carried out in [6]. It was established in this scientific work that the liquid cargo has an impact on the distribution of loads between the front and rear wheel groups of the car. At the same time, the authors of the above-mentioned works did not offer any measures to reduce the load on the tank car in operation.

Papers [7, 8] offer the substantiation of measures on reduction of dynamic loading on load-carrying structures of cars at operational modes. The studies, conducted regarding open railroad freight cars and platform cars, confirmed the feasibility of the proposed solutions. However, no measures to reduce the dynamic load on the load-carrying structures of tank cars were discussed in these papers.

Definition of unsolved aspects of the problem

The load-carrying structure of tank cars is subject to the loads that occur during operating conditions. The most common of these are vertical loads, caused by railway line unevennesses. Due to the cyclical nature of such loads, load-carrying structures of tank cars may suffer damages, and hence it will lead to the necessity of appropriation of additional costs for their maintenance. This necessitates the development and implementation of measures to improve load-carrying structures of tank cars to reduce their dynamic load in operation. In order to reduce the dynamic load on the load-carrying structures of tank cars, the authors propose the use of elastic-friction connections between the boiler and boiler bearings and between boiler bearings and the boiler bearer. However, the implementation of such innovations at the first level requires scientific justification and comprehensive calculations.

Problem statement

The aim of the article is to determine the dynamic load on the load-carrying structure of a tank car with elastic-friction connections in boiler bearings and between boiler bearings and the boiler bearer. To achieve this goal, the following tasks should be completed:

– to determine the dynamic load on the load-carrying structure of a tank car with elastic-friction connections in boiler bearings and between boiler bearings and the boiler bearer;

– to carry out stress calculation of the load-carrying structure of a tank car;

– to calculate the design lifetime, as well as the fatigue strength coefficient of the load-carrying structure of a tank car.

Basic material and results

In order to reduce the dynamic load on the load-carrying structure of a tank car, it is proposed to use elastic-friction connections between the boiler and boiler bearings, as well as between boiler bearings and the boiler bearer (Fig. 1).

To determine vertical accelerations acting on the tank car's boiler, mathematical modelling was performed. The design model of the tank car is shown in Fig. 2.

When composing the differential equations of motion of the tank car, it is stated that it moves empty, as under such conditions the greatest load on the load-carrying structure is observed. It is taken into account that the railway line has viscoelastic characteristics, and the reactions of the railway line are proportional to both its deformation and the speed of this deformation [9].

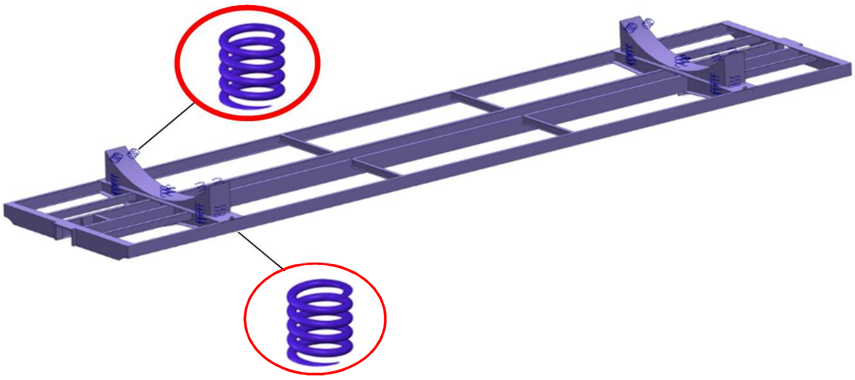

Figure 1 – Arrangement of elastic-friction elements on the boiler bearings and between boiler bearings and the boiler bearer

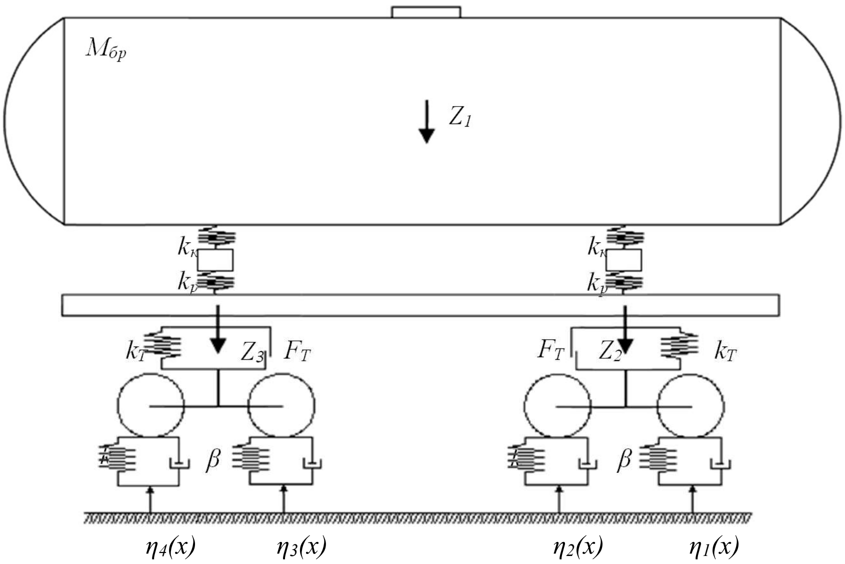

Figure 2 – Calculation scheme of a tank car

Differential equations of motion of the tank car have the following form:

$$M_1 \cdot \ddot{q}_1 + C_{1,1} \cdot q_1 + C_{1,2} \cdot q_2 + C_{1,3} \cdot q_3 = = -F_{TP} \cdot \left(sign\left(\frac{d}{dt}\delta_1\right) + sign\left(\frac{d}{dt}\delta_2\right)\right) \quad (1)$$

$$M_2 \cdot \ddot{q}_2 + C_{2,1} \cdot q_1 + C_{2,2} \cdot q_2 + B_{2,2} \cdot \dot{q}_2 = = F_{TP} \cdot sign\left(\frac{d}{dt}\delta_1\right) + k(\eta_1 + \eta_2) + + \beta\left(\frac{d}{dt}\eta_1 + \frac{d}{dt}\eta_2\right) \quad (2)$$

$$M_3 \cdot \ddot{q}_3 = -C(y_1 - y_2) - M_3 \cdot g \quad (3)$$

where M_i – inertial coefficients of the oscillating system elements (the load-carrying structure of a tank car, two bogies and the boiler);
C_{ij} – elasticity characteristics of the oscillating system elements, which are determined by the values of the stiffness coefficient of the springs;
B_{ij} – scattering function;
q_i – generalised coordinates corresponding to the translational displacement relative to the vertical axis, the car's boiler, the first and second set of wheels;
k – railway line stiffness;
β is the damping coefficient;
F_F – frictional force in the spring grouping;
δi – deformations of springing elements of the spring hanger; $\eta(t)$ is the unevenness of the railway line.

In the equations of motion (1) – (3) we accept:

$Z_1 \sim q_1$ – the coordinate that characterises the translational movements of the load-carrying structure of a tank car relative to the vertical axis;

$Z_2 \sim q_2$ – the coordinate that characterises the translational movements of the first car facing the engine and relative to the vertical axis;

$Z_3 \sim q_3$ – the coordinate that characterises the translational movements of the second car facing the engine and relative to the vertical axis.

The connection of the boiler with boiler bearings, of boiler bearings with the boiler bearer and of the boiler bearer with the travel carriage was described as a serial elastic coupling (Fig. 3):

$$C = \frac{2k_k \cdot 2k_p \cdot 2k_T}{2k_k + 2k_p + 2k_T}\ , \qquad (4)$$

where k_k is the stiffness of the springs between the boiler and boiler bearings;
k_p – stiffness of springs between boiler bearings and the boiler bearer;
k_T – stiffness of the springs of the spring hanger.

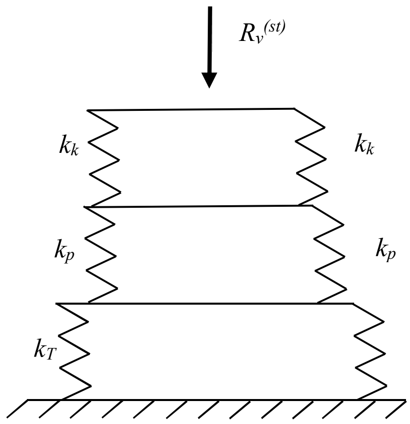

Figure 3 – The scheme of load transfer from the tank car boiler to the bogies ($R_v^{(st)}$ - vertical static load)

It is taken into account that the rigidity of the connection between the boiler and its bearings, boiler bearings and the boiler bearer, between the boiler bearer and the travel carriage is described as parallel coupling. That is, the total stiffness between the boiler and its bearings is $2k_k$, between boiler bearings and boiler bearer – $2k_r$, between the boiler bearer with the travel carriage – $2k_T$.

Railway line unevenness was described by a periodic function [9]

$$\eta(t) = \frac{d}{2}(1 - \cos\omega t)\ , \qquad (5)$$

where d is the unevenness depth, which is given;
ω is the oscillation frequency.

The input parameters to the mathematical model are given in table 1. The calculations were carried out based on the parameters of the car model 18-100.

Table 1 – Input parameters to the mathematical model

Parameter name	Value
LOAD CARRYING STRUCTURE mass, t	 14,9
CARS mass, t spring hanger stiffness, kN/m relative friction coefficient	 4,3 8000 0,1
RAILWAY LINE damping factor, kN s/m stiffness, kN/m unevenness amplitude, m unevenness length, m	 200 100000 0,01 25

The calculation was made for a tank car model 15-1443-06.

The solution of the differential equations of motion (1) – (3) was carried out in the MathCad software package [10 – 12].

The initial motions and velocities are set to zero.

The solution of the model in the MathCad software package was complied in the following form:

$$F(t,y) = \begin{bmatrix} y_2 \\ y_4 \\ y_6 \\ \dfrac{-F_{TP}\cdot\left(sign\left(\dfrac{d}{dt}\delta_1\right) + sign\left(\dfrac{d}{dt}\delta_2\right)\right) - C_{1,1}\cdot y_1 - C_{1,2}\cdot y_3 - C_{1,3}\cdot y_5}{M_1} \\ \dfrac{F_{TP}\cdot sign\left(\dfrac{d}{dt}\delta_1\right) + k\left(\eta_1 + \eta_2\right) + \beta\left(\dfrac{d}{dt}\eta_1 + \dfrac{d}{dt}\eta_2\right) - C_{2,1}\cdot y_1 - C_{2,2}\cdot y_2 - B_{2,2}\cdot y_4}{M_2} \\ \dfrac{F_{TP}\cdot sign\left(\dfrac{d}{dt}\delta_2\right) + k\left(\eta_3 + \eta_4\right) + \beta\left(\dfrac{d}{dt}\eta_3 + \dfrac{d}{dt}\eta_4\right) - C_{3,1}\cdot y_1 - C_{3,3}\cdot y_3 - B_{3,3}\cdot y_6}{M_3} \end{bmatrix}, \qquad (6)$$

$$Z = rkfixed\ (Y_0,\ t_n,\ t_k,\ n,\ F).$$

With $y_1 = q_1$, $y_3 = q_3$, $y_5 = q_5$, $y_2 = \dot{y}_1$, $y_4 = \dot{y}_3$, $y_6 = \dot{y}_5$,

The results of the calculation are shown in Fig. 4, 5.

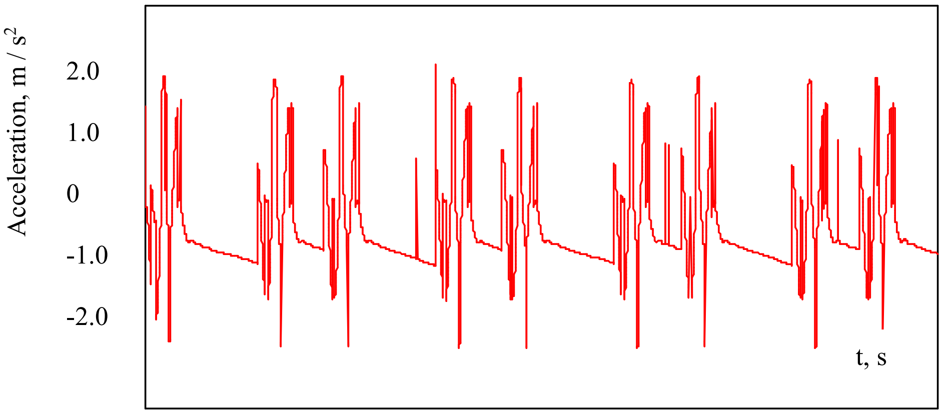

Figure 4 – Acceleration of the load-carrying structure in the centre of inertia

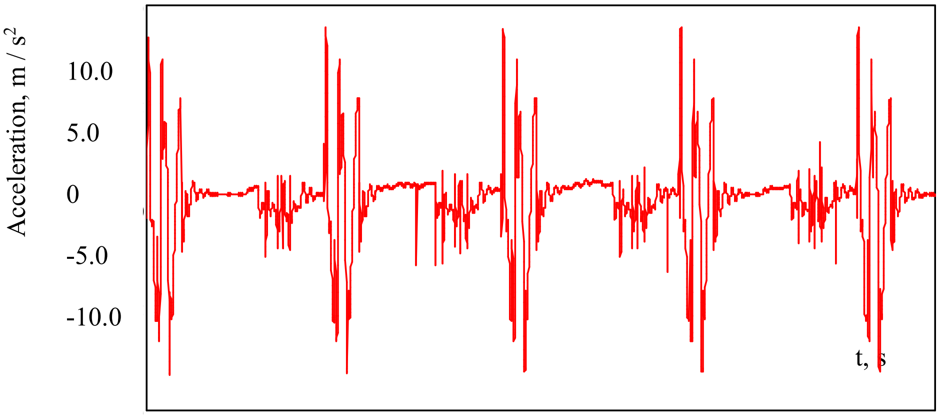

Figure 5 – Acceleration of carts

The maximum vertical acceleration of the load-carrying structure of the empty tank car is 1.35 m/s^2 (0.14 g), and the carts – 8.91 m/s^2 (0.91 g). Based on the calculations, it can be concluded that the run of the car is assessed as "excellent" [13, 14]. At the same time, the use of elastic-friction connections allows to reduce the dynamic load of the tank car by almost 36% in comparison with the prototype. The total stiffness of the elastic connection between the boiler with boiler bearings, of boiler bearings with the boiler bearer and of the boiler bearer with the travel carriage must not exceed 4360 kN/m.

At the next stage of the study, the strength of the load-carrying structure of a tank car was calculated. Graphic work was carried out in the software package "SolidWorks".

The calculation was performed by the finite element method in the "SolidWorks Simulation" software package.

The finite element model of the load-carrying structure of a tank car is shown in Fig. 6. The optimal number of grid elements was determined by the grapho-analytical method [15, 16]. Spatial isoparametric tetrahedra were used as finite elements [17-19]. The number of grid elements was 721195, the number of joints – 232420. The maximum size of the grid element is 40 mm, the minimum – 8 mm, the maximum ratio of the sides of the elements – 93.724, the percentage of elements with the ratio of the sides less than three – 21.8, more than ten – 0.414.

The design model of the load-carrying structure of a tank car is shown in Fig. 7. Determination of the strength of the load-carrying structure of a tank car was carried out for the first design mode – "jerk – tension". It is taken into account that the load-bearing structure is subjected to a longitudinal load P_L, which is applied to the front draft lugs of automatic couplers and is equal to 2.5 MN, liquid cargo pressure is P_P, and vertical load is P_V, which takes into account dynamic load, determined by mathematical modelling. The rate of acceleration, acting on the load-carrying structure of a tank car when moving in the loaded state was about 3.0 m / s^2 (0.3 g).

The fastening of the model was carried out in the areas of resting on the carts. The construction material is grade steel 09G2S. The results of the stress calculation are shown in Fig. 8 and 9.

The maximum equivalent stresses occur in the area of interaction of the center sill with the draw-bar and are about 250 MPa and do not exceed the allowable equivalent stresses [13, 14]. The maximum displacements were 8.3 mm and were concentrated in the manhole area.

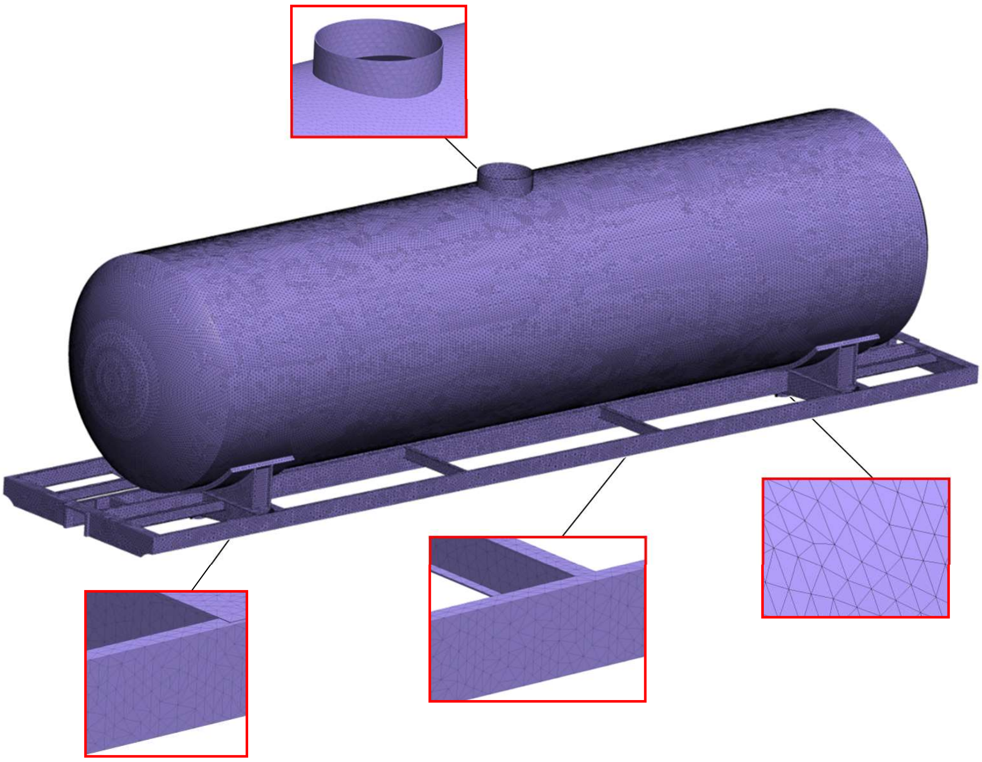

Figure 6 – Finite element model of the load-carrying structure of a tank car

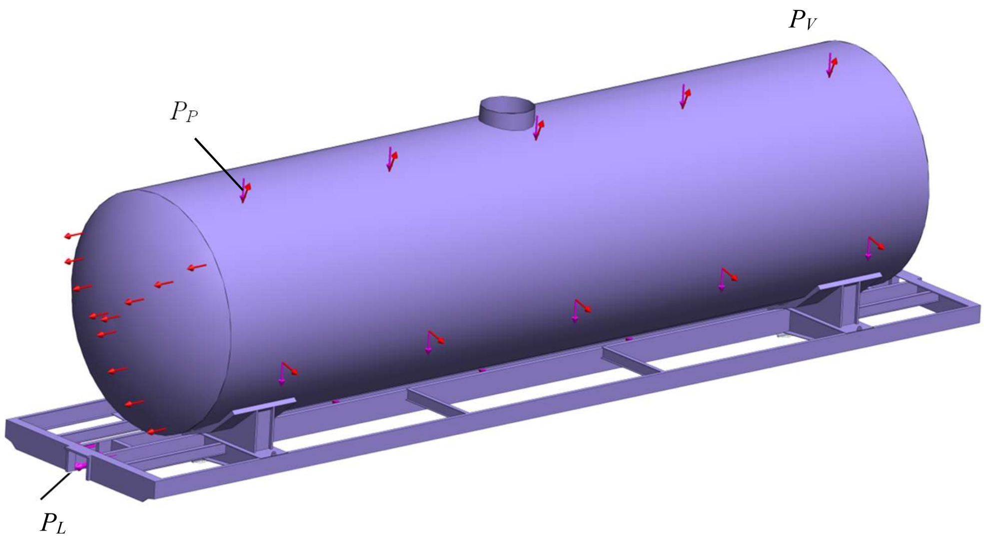

Figure 7 – The design model of the load-carrying structure of a tank car

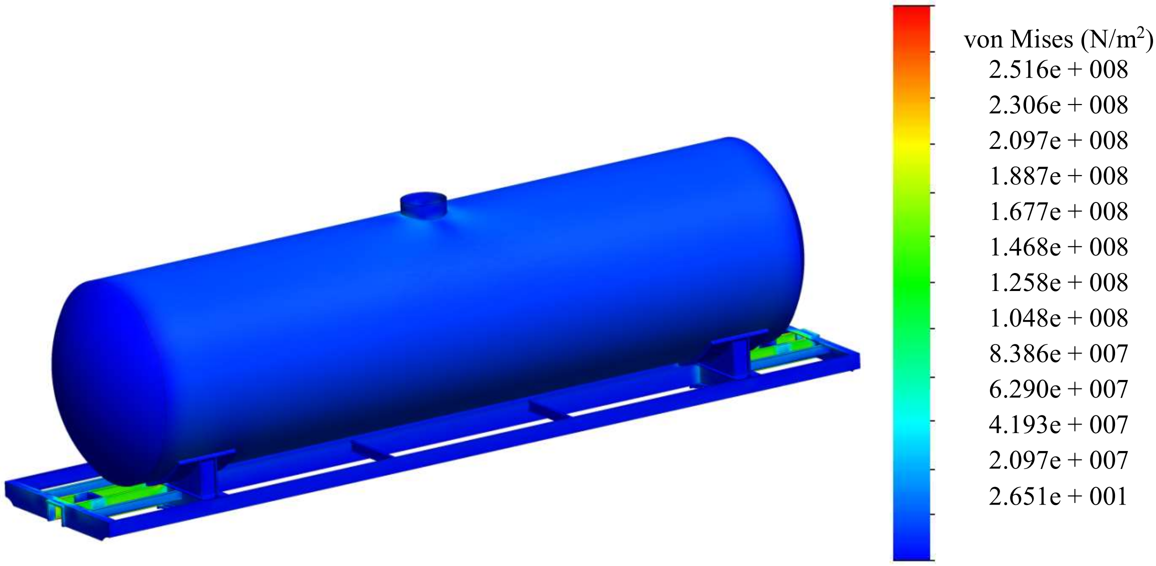

Figure 8 – Stress state of the load-carrying structure of a tank car in the first design mode ("jerk")

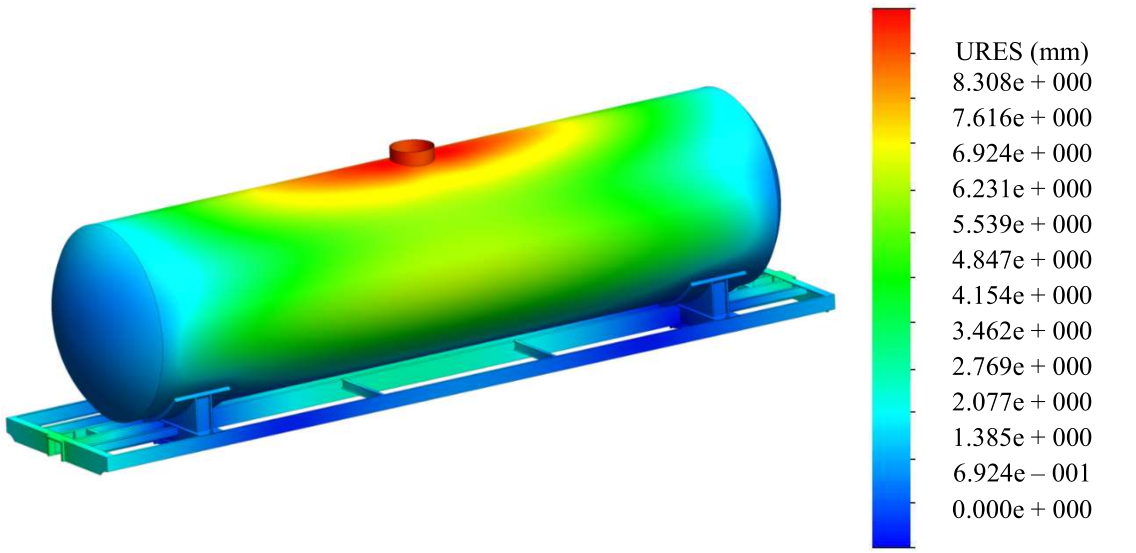

Figure 9 – Displacements in the joints of the load-carrying structure of a tank car in the first design mode ("jerk")

To determine the operational life of the tank car we used the method, described in [20]:

$$T_n = \frac{\left(\frac{\sigma_{-1D}}{[n]}\right)^m \cdot N_0}{B \cdot f_e \cdot \sigma_{sw}\left(k_{dv} + \frac{\psi_\sigma}{K_\sigma}\right)^m} \tag{7}$$

where σ_{-1D} is the average value of the endurance limit;
n is the allowable assurance coefficient;
m is the degree of fatigue curve;
N_0 – test base;
B – coefficient that characterises the time of the object's continuity of service in seconds;
$f_{в}$ – effective frequency of dynamic stresses;
σ_{sw} – stress from static weight load;
k_{dv} – coefficient of vertical dynamics;
ψ_σ – sensitivity factor;
K_σ is the total coefficient of reduction of fatigue strength.

When determining the amplitude of equivalent dynamic stresses, we took into account the coefficient of influence of lateral forces equal to 1.1.

The following input parameters are taken into account in the calculations: σ_{-1D} = 245 MPa; n = 2; m = 8; $N_0 = 10^7$; $B = 3.0 \cdot 10^6$ s; $f_{в}$ = 2,7 Hz; ψ_σ/K_σ = 0,2.

The strength calculation was performed to determine the stresses from the static weight load of the load-carrying structure of a tank car. It was established that the maximum equivalent stresses in the load-carrying structure of a tank car are 87.5 MPa.

In this case, the design lifetime of the load-carrying structure of a tank car is more than 20% higher than the design lifetime of the prototype car. It is important to say that the obtained value of the design lifetime should be specified taking into account additional studies of the longitudinal load of the load-carrying structure of a tank car and experimental (field or benchmark tests) studies.

Also in the framework of the study, the fatigue resistance of the load-carrying structure of a tank car was calculated.

The calculation of fatigue resistance was carried out taking into account the assurance coefficient n by the formula [21]:

$$n = \frac{\sigma_{-1D}}{\sigma_{a,e}} \geq [n] \ , \qquad (8)$$

where $\sigma_{a,e}$ is the calculated value of the amplitude of dynamic stress of the conditional symmetric cycle, reduced to the base N_0, equivalent in damaging action to the value of the amplitudes in the real mode of random operating tensions during the design lifetime, MPa;
$[n]$ – allowable factor of safety against fatigue failure.

The calculation results showed that there is a probability of occurrence of stresses with a level of σ_1, that is 0.95 values of $\sigma_{a,e}$ = 58.3 MPa. Hence the coefficient of fatigue resistance is 4.2. However, due to the lack of experimental data, the allowable value of the coefficient of fatigue resistance is assumed to be 2.2. Therefore, condition (8) is met and the fatigue strength of the load-carrying structure of a tank car is provided.

Conclusions

1. Determination of the dynamic load of the load-carrying structure of a tank car with elastic-friction connections in the boiler bearings and between the bearings and the boiler bearer was carried out. It was established that the maximum vertical acceleration of the load-carrying structure of an empty tank car is 1.35 m/s^2 (0.14 g), and carts – 8.91 m/s^2 (0.91 g). At the same time, the use of elastic-friction connections allows to reduce the dynamic load of the tank car by almost 36% in comparison with the prototype.

2. The stress calculation of the load-carrying structure of a tank car was carried out. The maximum equivalent stresses occur in the area of interaction of the center sill with the draw-bar and are about 250 MPa and do not exceed the allowable stress values. The maximum displacements were 8.3 mm and were concentrated in the manhole area.

3. The calculation of the design lifetime was made, as well as of the coefficient of fatigue resistance of the load-carrying structure of a tank car. It was established that the design lifetime of the load-carrying structure of a tank car is more than 20% higher than the design lifetime of the prototype car. The coefficient of fatigue resistance was 4.2, which is twice as allowable.

The conducted research will allow to increase the operational efficiency of tank cars, and will also promote the creation of their innovative designs.

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