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Determination of loading of a hopper car with an improved design of the spine beam

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Abstract

The substantiation of use of a closed structure of the spine beam of a hopper car filled with a filler possessing viscoelastic properties is carried out. The dynamic loading of the load-bearing structure of a hopper car is determined taking into account the proposed solutions. It is established that the value of acceleration of the improved load-bearing structure of a hopper car during shunting collision is 3.7% lower than that obtained for the load-bearing structure without a filler. The main indicators of strength of the load-bearing structure of a hopper car were determined by the finite element method, which was implemented in the SolidWorks Simulation software package. The maximum equivalent stresses in this case were recorded in the area of interaction of the spine beam with the pivot one. The obtained value of the maximum equivalent stresses is 6% lower than that obtained for the structure without a filler. Modal analysis of the load-bearing structure of a hopper car was performed. The conducted research will help to ensure the strength of the load-bearing structures of hopper cars in operation, reduce maintenance costs, as well as create recommendations for the design of their modern structures.

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Keywords: Transport mechanics, hopper car, load-bearing structure, dynamic loading, strength, project life cycle, modal analysis

1. Introduction

The efficiency of the transport industry functioning necessitates the introduction of modern vehicles. Since the main segment of the transportation process is devoted to railway transport, special conditions must be imposed on the creation of modern wagon designs. In particular, this applies to their load-bearing structures (Soloviova et al. (2020), Strelko et al. (2019) and Bondarenko et al. (2020)).

Hopper cars are used to transport bulk cargo by railway with the possibility of using gravitational properties during their unloading. The peculiarity of such wagons is that the end walls are placed at a certain angle. At present, there is a great variety in the design features and purpose of hopper cars. One of the most laden types of hopper cars are those designed to transport hot pellets and sinter.

The spine beam of the wagon experiences significant loads during operational modes (Chen Chao et al. (2019) and Shi (2017)). Due to the action of cyclic loads on it during operation, such damage as cracks, deformations, etc. may occur (Sepe et al. (2015) and Antipin et al. (2016)). This circumstance poses a significant threat to the safety of operation of the wagon as part of the train. Therefore, to ensure the strength of the spine beam of the hopper car frame, traffic safety, reduce maintenance costs, increase operational efficiency, it is important to implement measures to improve their designs.

Paper by Kebal et al. (2019) highlighted the results of improving the design of a hopper car. When optimizing the load-bearing structure of a hopper car, the experience of operation of individual body components with subsequent integration into the new structure is taken into account.

Peculiarities of optimization of load-bearing structures of freight wagons were considered in publication by Bain, (2011). An algorithm for compatible structural and parametric optimization of the side wall and the frame of a wagon were developed.

It is important to say that the issues of optimization of hopper cars for the transportation of pellets and hot sinter were not paid attention to in these works.

In work by Kuczek and Szachniewicz (2014) the results of topological optimization of the load-bearing structure of a wagon were considered. Calculations were performed by the finite element method. The efficiency of using the proposed methodology for optimizing the load-bearing structures of wagons was substantiated. However, the paper presented the results of application of this methodology in relation to the load-bearing structure of a passenger wagon. That is to say, studies on the possibility of its application to the load-bearing structure of a hopper car were not conducted.

Improvement of the load-bearing structures of wagons to reduce their loading in operation was carried out in work by Hyun-Ah Lee et al. (2016). This goal was achieved by using aluminum panels of the "sandwich" type in the load-bearing structure of a wagon. The substantiation of the offered improvement of a wagon was given. At the same time, no attention was paid to reducing the loading of the wagon frame.

Interest is attracted to work by Mrzyglod and Kuczek (2014), which highlighted the features of a unified concept of impact strength of vehicles. The load-bearing structure was considered in the form of a rigid frame one, and the optimization problem was solved according to the criterion of minimum material consumption. However, the authors did not take into account the possibility of reducing the loading of the load-bearing structure of the vehicle during its optimization.

In paper by Płaczek et. al. (2016), measures were proposed to modernize the bodies of freight wagons in order to extend their service life. Improvement of the system of diagnostics of technical condition of modernized wagons was also offered in the work. At the same time, the offered modernization does not contribute to the reduction of loading of the wagon frame at operational modes.

Measures to reduce the dynamic loading and ensure strength of the load-bearing structures of wagons were proposed in works by Fomin et al. (2021), Lovska and Fomin (2020). The authors proposed the use of malleable links in the load-bearing structures of wagons, as well as in the nodes of their interaction with the means of combined transport. At the same time, no attention was paid to the issue of reducing the loading on the load-bearing structure of a hopper car.

In work by Sokolov et. al. (2019), the use of a new profile for the spine beam of a freight wagon was proposed and substantiated. The results of the calculation of strength of the load-bearing structures of wagons taking into account the proposed solutions confirmed their feasibility. However, these works did not consider measures to reduce the loading of the load-bearing structures of wagons at operational modes.

Peculiarities of application of the theory of optimal design of the load-bearing structure of a closed bottom gondola car body were covered in publication by Chepurchenko et. al. (2018). The results of the research made it possible to determine the optimal configuration of the unloading bunkers of a gondola car. It is important to say that the optimization of the load-bearing structure of the wagon did not take into account measures to reduce the dynamic loading in operation.

The analysis of the literature allows us to conclude that it is advisable to conduct research to improve the frame of a hopper car in order to reduce its loading at operational modes.

Nomenclature

P_l	longitudinal loading on the rear support of the automatic coupling which is taken equal to 3.5 MN [12, 13]
P_v^{st}	vertical static loading
P'_r	reaction in body centre plates
P_{fr}	friction force between the body centre plate and the thrust bearing
c	rigidity of the material filling the spine beam
β	viscosity of the material filling the spine beam
M_{gm}	gross mass of a hopper car
M_{hc}	mass of the load-bearing structure of a hopper car
I_{hc}	moment of inertia of a hopper car
l	half of the base of a hopper car
F_{fr}	value of dry friction force in the spring set
k_1, k_2	rigidity of springs of the spring suspension of the hopper car trolleys
x, φ, z	coordinates corresponding to the longitudinal, angular about the transverse axis and vertical movement of a hopper car, respectively.
σ_{-1E}	average value of endurance limit
n	allowable coefficient of strength reserve
m	indicator of the degree of fatigue curve
N_0	test base
B	coefficient characterizing the time of continuous operation of the object in seconds
f_e	effective frequency of dynamic stresses
σ_{sw}	stress from static weight load
k_{vd}	coefficient of vertical dynamics
ψ_σ	sensitivity coefficient
K_σ	overall coefficient of the reduction of fatigue strength

2. Methodology

The purpose of the article is to highlight the results of determining the loading of a hopper car with an improved design of the spine beam by using a closed profile filled with a filler. To achieve this goal, the following research methodology was used. To achieve this goal, the following research methodology was used. At the initial stage, the dynamic loading of the supporting structure of the hopper car with a closed center beam filled with filler was determined. After that, the main indicators of the strength of the supporting structure of the hopper car were determined. At the next stage of the study, the calculation of natural frequencies and modes of vibration of the supporting structure of the hopper car was carried out.

3. Results and discussion

To reduce the loading of the load-bearing structure of a hopper car, it is proposed to improve its frame by using a closed profile of the spine beam filled with a filler possessing viscoelastic properties (Fig. 1).

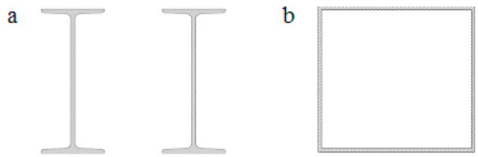

Fig. 1. Cross section of the spine beam of the hopper car frame (a) typical; (b) improved.

The geometric parameters of the spine beam were determined by the method of optimization for strength reserves. The spatial model of the improved design of the hopper car frame is shown in Fig. 2.

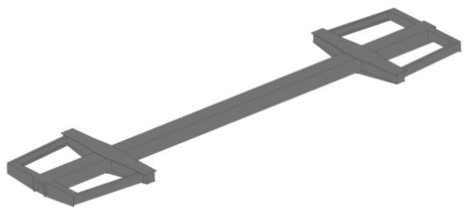

Fig.2. Spatial model of the hopper car frame

In this case, the material with viscoelastic properties is placed between the rear supports of the automatic couplings (Fig. 3). The damping of the kinetic energy of impact P_l on the rear support of the automatic coupling is carried out due to the viscoelastic resistance of the material with the characteristic of elasticity c and viscosity β.

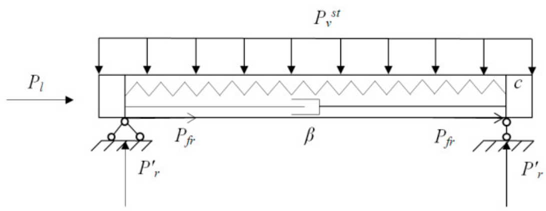

Fig. 3. Scheme of loading of the hopper car frame

To determine the dynamic loading of a hopper car, taking into account the proposed solutions, mathematical modelling was performed. In this case, the mathematical model developed by Bogomaz et. al. (1999) was used. In this study, the model was adapted to determine the dynamic loading of a hopper car.

The equations of motion have the form:

$$M_{gm} \cdot \ddot{x} + \left(M_{hc} \cdot h\right) \cdot \ddot{\varphi} = P_n - 2P_{fr} - \beta \cdot \dot{x} - c \cdot x, \tag{1}$$

$$I_{hc} \cdot \ddot{\varphi} + \left(M_{hc} \cdot h\right) \cdot \ddot{x} - g \cdot \varphi \cdot \left(M_{hc} \cdot h\right) = l \cdot F_{fr}\left(sign\dot{\Delta}_1 - sign\dot{\Delta}_2\right) + l\left(k_1 \cdot \Delta_1 - k_2 \cdot \Delta_2\right), \tag{2}$$

$$M_{hc} \cdot \ddot{z} = k_1 \cdot \Delta_1 + k_2 \cdot \Delta_2 - F_{fr}\left(sign\dot{\Delta}_1 - sign\dot{\Delta}_2\right), \tag{3}$$

in this case

$$\Delta_1 = z - l \cdot \varphi; \quad \Delta_2 = z + l \cdot \varphi,$$

The calculation was made in relation to the hopper car for transportation of pellets of model 20-9749 (Fig. 4) constructed by SE "Ukrspetsvagon" (Ukraine). The solution of differential equations of motion was carried out by the Runge-Kutta method in the MathCad software package (Lovska (2015), Fomin (2015), Dudnyk et. al (2020), Pievtsov et. al (2020)). Initial displacements and velocities were taken to be zero (Krol and Sokolov (2020), Sokolov et. al (2020)). The results of the calculations showed that the maximum accelerations acting on the load-bearing structure of the hopper car were 36.2 m/s^2. This value of acceleration is 3.7% lower than that obtained for the load-bearing structure without a filler. The rigidity of the material filling the spine beam should be about 80 kN/m, and the coefficient of viscous resistance – about 118 kN·s/m.

Fig. 4. Hopper car of model 20-9749

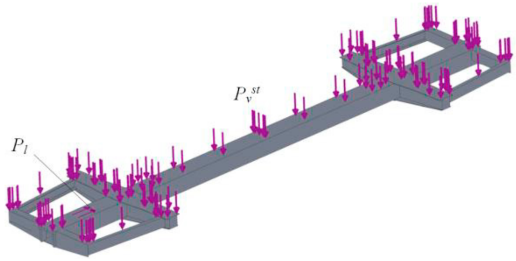

Fig. 5. Calculation scheme of the hopper car frame

To determine the strength of the hopper car frame taking into account the proposed solutions, calculation was performed. In this case, the finite element method was used, which was implemented in the SolidWorks Simulation software package (Lovskaya (2015), Fomin et. al. (2017) and Goolak et. al. (2019)).

The optimal number of elements of a finite-element model of the hopper car frame was determined using the graph-analytical method (Vatulia et. al. (2018), Vatulia et. al. (2019), Fomin and Lovska (2020)). Isoparametric tetrahedra were used as finite elements (Píštek et. al. (2020)). The number of grid nodes was 14858, and the number of elements was 42531. In this case, the maximum size of the element was 100 mm, the minimum one – 20 mm. The number of elements in the circle was 9. The ratio of increasing the size of the element was 1.7. The presence of a filler in the spine beam was modelled by making appropriate connections using the software package options (Vatulia et al. (2017)).

When compiling the calculation scheme, the following loads were taken into account: vertical static load P_v^{st}, as well as the longitudinal load P_l acting on the frame from the automatic coupling device (Fig. 5). The maximum equivalent stresses in this case were recorded in the zone of interaction of the spine beam with the pivot one and amounted to about 311 MPa (Fig. 6), i.e. do not exceed the yield strength of the material (DSTU 7598:2014 and GOST 33211-2014). The obtained value of the maximum equivalent stresses is 6% lower than that obtained for the structure without a filler. The distribution of the maximum equivalent stresses along the length of the spine beam is shown in Fig. 7. In the middle part of the spine beam, the maximum equivalent stresses were about 200 MPa. The lowest value of stresses was observed in the cantilever parts of the spine beam. The maximum displacements were recorded in the middle part of the frame and amounted to 6.8 mm. The results of the calculation of strength of the hopper car frame were carried out in relation to other operational modes as well. It is established that the strength of the frame is provided.

To determine the project life cycle of the hopper car frame, the method described in Ustich (1999) was used:

$$T_n = \frac{\left(\sigma_{-1E} / [n]\right)^m \cdot N_0}{B \cdot f_e \cdot \sigma_{sw} (k_{vd} + \psi_\sigma / K_\sigma)^m}, \tag{4}$$

The following input parameters were taken in the calculations: σ_{-1E}=245 MPa; n=2; m=8; N_0=10^7; B=3.07·10^6 s;

f_e=2.7 Hz; k_{vd}=0.35; ψ_σ/K_σ=0.2.

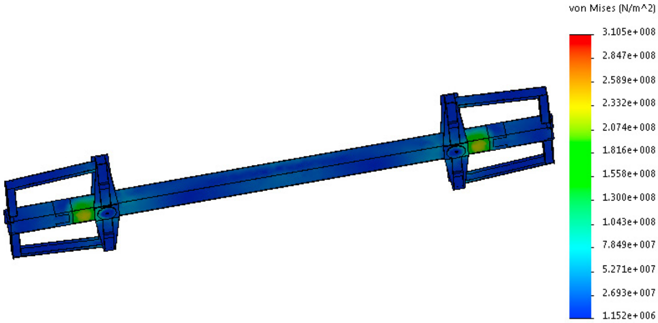

Fig. 6. The stress state of the hopper car frame

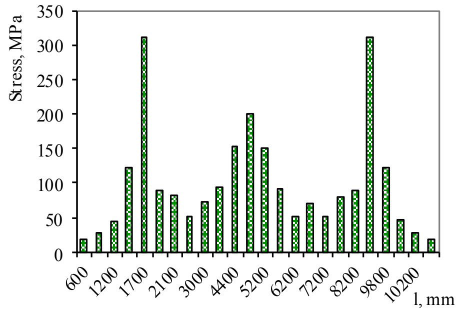

Fig. 7. Distribution of stresses along the length of the spine beam

The calculations showed that the project life cycle of the hopper car frame was 16% higher than the life cycle of the prototype wagon. However, it is important to say that the obtained value of the project life cycle should be specified taking into account the additional studies of the vertical loading of the frame.

Modal analysis of the hopper car frame was also conducted in the study. The calculation was performed according to the scheme shown in Fig. 6 in the SolidWorks Simulation software package. The values of the natural frequencies of oscillations of the hopper car frame are given in Table 1.

Table 1. Values of the natural frequencies of oscillations of the hopper car frame.

Mode	Frequency, Hz	Mode	Frequency, Hz
1	59.7	6	99.2
2	74.2	7	122.6
3	74.6	8	124.0
4	82.4	9	173.8
5	97.4	10	194.5

Some forms of oscillations of the hopper car frame are shown in Fig. 8.

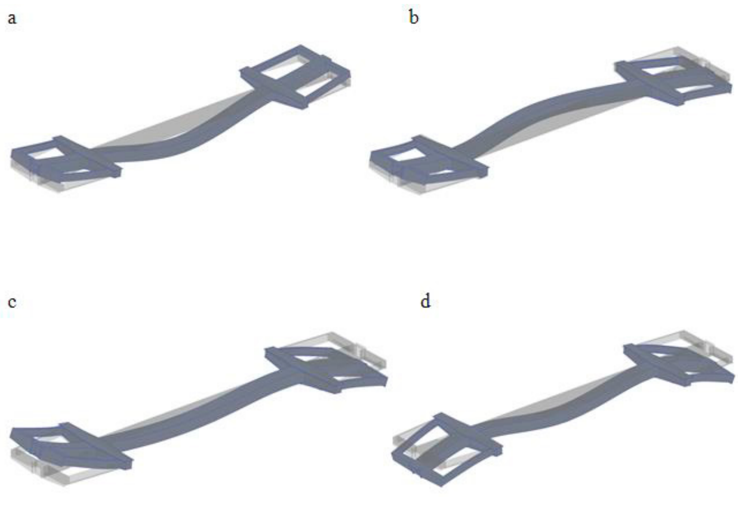

Fig. 8. Forms of oscillations of the hopper car frame (deformation scale 10:1) (a) the first mode (b) the second mode (c) the third mode (d) the fourth mode

According to the data given in Table 1, we can conclude that the value of natural frequencies of oscillations is within acceptable limits, because the first natural frequency has a value greater than 8 Hz (DSTU 7598:2014 and GOST 33211-2014).

4 Conclusions

The dynamic loading of the load-bearing structure of a hopper car with a closed spine beam filled with a filler is determined. The maximum accelerations acting on the load-bearing structure of the hopper car were 36.2 m/s^2. This value of acceleration is 3.7% lower than that obtained for the load-bearing structure without a filler. The rigidity of the material filling the spine beam should be about 80 kN/m, and the coefficient of viscous resistance - about 118 kN·s/m.

The main indicators of strength of the load-bearing structure of a hopper car are determined. The maximum equivalent stresses in this case were recorded in the zone of interaction of the spine beam with the pivot one and amounted to about 311 MPa. The obtained value of the maximum equivalent stresses is 6% lower than that obtained for the structure without a filler. The project life cycle of the improved structure of the hopper car frame is 16% higher than the life cycle of the prototype wagon.

The natural frequencies and forms of oscillations of the load-bearing structure of a hopper car are determined. It is established that the value of natural frequencies of oscillations is within acceptable limits, because the first natural frequency has a value greater than 8 Hz. The conducted research will help to ensure the strength of the load-bearing structures of hopper cars in operation, reduce maintenance costs, as well as create recommendations for the design of their modern structures.

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