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Modeling of the Cryogenic Section's Dynamics of an Experimental GAS-Diesel Locomotive

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Summary

The article is devoted to solving the problem of using liquefied natural gas as a motor fuel in transport. With regard to rail transport, this problem can be solved by creating two-section gas-diesel locomotives and developing of the special cryogenic section, which is supposed to be placed between the traction sections of these locomotives. There are special tanks for storing liquefied natural gas in the cryogenic section, as well as equipment for its regasification and gas supply for diesel locomotives. The presence of equipment in the cryogenic section previously not operated on railway transport requires classifying this section as a fundamentally new mobile unit. Comprehensive experimental and theoretical studies of the section must preced the development of its pilot batch. The article deals with the issues related to the simulation of stationary dynamic processes corresponding to the motion of the cryogenic section in the straight sections of the track. Basing on the numerical implementation of the developed mathematical model, the quasi-static loads of the equipment in the cryogenic section were calculated, the negative properties of its spring suspension were identified, and recommendations for improving the dynamic characteristics were given.

Keywords: gas-diesel locomotive, liquefied natural gas, cryogenic section, mathematical model, dynamic characteristics.

INTRODUCTION

The rise in prices for diesel fuel forces the railways to seek rational options for its replacement. First of all, the possibility of transferring diesel locomotives to liquefied natural gas, which is a relatively inexpensive and environmentally friendly energy source, is being considered. The works carried out on the railways of North America [1, 2] and Europe [3] are devoted to the solution of this problem. One of the options for this solution is the creation of three-section gas-diesel main diesel locomotives [4], in the middle non-tensional (cryogenic) section of which there is a gasifier and auxiliary equipment providing storage and supply of diesel locomotives with gas. The gasifier contains two tanks for storing liquefied natural gas, which are arranged coaxially (one above the other) in the vertical longitudinal plane of symmetry of the cryogenic section. The lower tank is installed on the frame of the experimental cryogenic section with the help of intermediate supports, the design of which allows to compensate for the nonflatness of the supporting surfaces of the tank when it is mounted on the frame, as well as the angular deformation of the frame of the cryogenic section during operation. The upper tank of the gasifier is installed directly on the lower tank. To do this, there are special box-type support belts on the lower tank capable of withstanding loads from the top tank filled with liquefied natural gas, taking into account inertial loads acting on it. Each tank is a horizontal double-walled device consisting of an inner vessel with thermal insulation and a casing. The working pressure in the internal vessels of the gasifier tanks is 1.6 MPa at a cryogenic liquid temperature of 160 K. The space between the inner vessel and the sealed casing (heat-insulating cavity) is under vacuum. The gasifier equipment and its attachment points are designed for quasistatic loads, which in the fractions of gravity acceleration g are: in the vertical direction – 1.3; in the transverse direction – 1.5; in the direction of travel - 3.0.

The body of the cryogenic section is of a bearing type. Its main parts are a frame, side and end walls, roof sections, transitional platforms. The main elements of the frame that receive loads include the main beams, tie drawers, pivot girders and beams of tank supports.

The running part of the cryogenic section of the experimental gas-diesel locomotive is based on two-axle jawless non-motorized bogies of the KVZ-I2 type used in freight cars of five-car refrigerator sections of Russian production [5]. Figure 1 shows the model of one of the modifications of such a bogie in the software package "Universal Mechanism" [6].

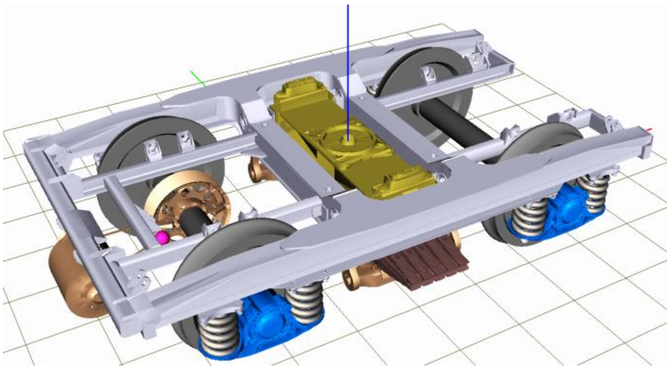

Figure 1: Model of the KVZ-I2 bogie in the "Universal Mechanism" software package

The bogie KVZ-I2 has a two-stage spring suspension. The main features of the spring suspension system are the absence of dampers in the first stage and the presence of a leaf elliptical spring in the second stage [5].

The presence of equipment previously not operated on railway transport in the cryogenic section requires classifying it as a fundamentally new mobile unit, the development of a pilot batch of which must be preceded by comprehensive experimental and theoretical studies.

Mathematical model

Let us compose the differential equations of oscillations of the cryogenic section in the vertical longitudinal plane. For this, we use the known methods of mechanics [7-10]. Let us assume that the kinematic perturbation acts on the wheelsets of the bogies from the rail track side. The vibrations are transferred to the sprung masses of the bogies and the body through the system of spring suspension. In this case, we consider the bolster structure of the section of the upper gasifier tank having incomplete filling with the liquefied natural gas as a two body system: the body of the section without taking into account the oscillating fluid and the reduced mass of the first form of fluid oscillations.

In accordance with Figure 2, we introduce the inertial coordinate system $O_k XZ$ and the coordinate system $O_1 x_1 Z_1$ associated with the body, starting at the point O_1 that coincides with the center of mass of the first form of oscillation of the cryogenic liquid in a state of rest. To consider the oscillation of bouncing and galloping of bogies, we introduce generalized coordinates $z_1, z_2, \varphi_1, \varphi_2$, and for oscillations of bouncing and twitching of the body, as well as longitudinal oscillations of the fluid, coordinates Z, X, x_1. For simplicity, we arbitrarily refer to angular displacements relatively to the coordinate Ψ as the oscillations of the body galloping.

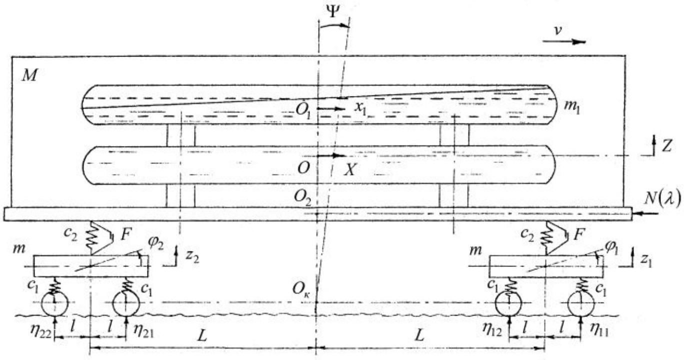

Figure 2: Calculation scheme of the cryogenic section

We also introduce the following notations: M is the mass of the body without an oscillating fluid; m_1 is the reduced mass of the first form of fluid oscillations; m_X is the mass of each bogie participating in longitudinal oscillations; m is the curb weight of each bogie; I_0 is the moment of the body inertia with respect to the transverse axis passing through the center of mass; I is the moment of the bogie inertia relatively the transverse axis; c_{cr} is the equivalent longitudinal stiffness of the first form of fluid voscilliation; ε is the coefficient of internal resistance of the oscillating fluid; c_1 is vertical stiffness of the first stage of the bogie suspension, falling on the wheelset; c_2, F is rigidity and frictional force in an elliptical spring located in the second stage of suspension; $2L$ is the base of suspension of the body; h_0, h_1 is the distance in a state of rest from the center of oscillations taken at the level of the bogies centers of mass, respectively to the body center of mass without an oscillating fluid and to the center of mass of the first form of fluid oscillations; $N(\lambda)$ is a nonlinear longitudinal force arising in the spring-friction automatic coupler; h is the distance from the body center of mass O to the line of action of the longitudinal force $N(\lambda)$.

For calculating the force $N(\lambda)$, the method of determining the forces in a spring-friction device was used, depending on the absolute value λ of elongation or shortening of the apparatus elements, and also taking into account the gap in the joint, which was described in detail in [4]. Scattering of oscillation energy in an elliptical spring under the influence of frictional force F will be taken into account by using the "signum" sign change function [4, 10].

On the basis of Lagrange equations of the second kind and in accordance with the parameters of the design scheme of the cryogenic section, the following system of nonlinear ordinary differential equations describing the oscillations of the object under study in time t as a mechanical system with eight degrees of freedom was obtained:

$$
\begin{aligned}
&(M + m_1 + m_X)\ddot{X} - (Mh_0 + m_1 h_1)\ddot{\Psi} + m_1 \ddot{x}_1 + N(X - h\Psi) = 0;\\
&\left(I_0 + Mh_0^2 - m_1 h_1^2\right)\ddot{\Psi} + (Mh_0 + m_1 h_1)\ddot{X} + m_1 h_1 \ddot{x}_1 + FL \cdot sign\left(\dot{z}_1 - \dot{z}_2 - 2L\dot{\Psi}\right) -\\
&- hN(X - h\Psi) - g(m_1 h_1 + Mh_0)\Psi - m_1 g x_1 + c_2 L(z_1 - z_2 - 2L\Psi) = 0;\\
&m_1 \ddot{x}_1 + m_1 \ddot{X} - m_1 h_1 \ddot{\Psi} + 2\varepsilon m_1 \dot{x}_1 - m_1 g \Psi + c_{cr} x_1 = 0;\\
&(M + m_1)\ddot{Z} + F \cdot sign\left(2\dot{Z} - \dot{z}_1 - \dot{z}_2\right) + c_2(2Z - z_1 - z_2) = 0;\\
&m\ddot{z}_i + (2c_1 + c_2)z_i + c_2\left((-1)^i L\Psi - Z\right) + F \cdot sign\left(\dot{z}_i + (-1)^i L\dot{\Psi} - \dot{Z}\right) = c_1(\eta_{i1} + \eta_{i2});\\
&I\ddot{\varphi}_i + 2c_1 l^2 \varphi_i = c_1 l(\eta_{i1} - \eta_{i2}); \qquad\qquad i = 1,2.
\end{aligned}
$$

The presented system of differential equations was reduced to the normal form of Cauchy and investigated by numerical integration. The problem was solved both in a deterministic and random setting.

The unevenness proposed by Charte [4] was used as a deterministic disturbance from the rail track:

$$\eta_{ij}(t) = \left(a\sin\left(\frac{\pi}{L_r}\left(vt - \Delta_{ij}\right)\right) + b\sin\left(\frac{3\pi}{L_r}\left(vt - \Delta_{ij}\right)\right)\right) \cdot sign\left(\sin\left(\frac{\pi}{L_r}\left(vt - \Delta_{ij}\right)\right)\right), \; j = 1,2,$$

where *a, b* are amplitudes of the first and third harmonic components;

L_r is the length of the rail link;

v is the speed of the cryogenic section;

Δ_j is the transport lag, which is determined through the parameters of the design scheme as follows:

$$\Delta_{11} = 0;\ \Delta_{12} = 2l;\ \Delta_{21} = 2L;\ \Delta_{22} = 2(L + l).$$

As a random perturbation, the spectral density of disturbances from the track is used, taking into account periodically repeated butt jaggies, unevenness of the rail links, micro- and macro unevenness of the track (without taking into account the joints) and unevenness of the wheels. The use of such perturbations in the calculations for linearized dynamic models of rail vehicles is described in detail in [10].

Analysis of simulation results

Analysis of the dynamic model shows that the equation includes 19 basic parameters that directly characterize the state of the system under investigation. In addition, there are more than 10 auxiliary parameters that are used to determine the main parameters and are not explicitly present in the equations.

When carrying out computational experiments, the most characteristic stationary modes of oscillations, typical for the cryogenic section, corresponding to the actual operating conditions, were considered. These regimes correspond to a sufficiently large number of states of the object of investigation, which is explained by the known spread of a number of parameters of the real system. Among the most significant changing parameters are: 1) coefficient of friction between leaves of bogie elliptical springs, the values of which, according to the normative documents, can vary from 0.3 to 0.8 depending on the state of the friction surfaces; 2) level of filling of gasifier tanks with liquefied natural gas; 3) state of the track; 4) section speed.

Free and forced stationary oscillations of a section with nonlinear and linearized bonds were investigated. The study of free vibrations made it possible to refine the natural frequencies of oscillations of the elements of the system, as well as the mutual influence of the subsystems on the oscillatory processes in the complete system. So, for example, for the basic version of the calculation, which corresponds to the mass of the supersorbable structure of 73963 kg, the natural frequencies of the body oscillations were equal: for bouncing 1.58 Hz; for galloping 1.67 Hz and for jerking 3.12 Hz. The same variant was matched by low-frequency oscillations of a cryogenic liquid with a frequency of about 0.1 Hz. In view of the smallness of the bogie masses, it were these oscillatory processes that mainly determine the dynamic characteristics of the cryogenic section. The mutual influence of subsystems was analyzed by sequentially setting the initial conditions for the investigated generalized coordinates and the subsequent consideration of damped oscillations in the system. So, for example, the greatest influence on oscillations of a liquid in tanks is rendered by the coordinate Ψ corresponding to angular fluctuations of the body. At $\Psi = 0{,}2\ rad$, the "throws" of the displacements of the liquid center of mass up to 0.17 m were recorded. Longitudinal oscillations of the body have less impact: with the initial value of the movement of the body twitch $X = 0.01$ *m*, the value of x_1 did not exceed 0.015 *m*. The inverse effect of fluid oscillations on generalized coordinates Ψ and X is insignificant, which is also confirmed by the analysis of the interrelation of subsystems according to L.I. Mandelshtam [4].

Forced oscillations of the cryogenic section were considered in the speed range 5.5 ... 33.3 *m/s* for zero and nonzero initial conditions applied to the Ψ, X and x_1 coordinates. The parameters of the disturbing effect corresponded to a satisfactory and good state of the track.

The analysis of forced oscillations showed that regardless of the speed of motion and combinations of initial conditions applied to generalized coordinates, in 1.5 ... 2.0 *s* the system realizes oscillations close to those that have been established with frequency

$$f_r = 2\frac{v}{L_r}.$$

This generalization is applicable to a lesser extent to oscillations of a cryogenic liquid for which the initial conditions applied to the Ψ, X and x_1 coordinates, due to the small values of the damping and the natural frequency, generally have a more significant effect than the perturbing effect from the track. For oscillations with zero initial conditions, the generalized coordinate x_1 is also rapidly "drawn" into a regime close to the single-frequency one.

The analysis of the calculation results shows that the introduction of nonlinear bonds significantly changes the dynamic characteristics of the cryogenic section. Thus, for example, at v = 23.6 *m/s*, the introduction to the longitudinal nonlinear coupling system $N(\lambda)$ reduces the amplitude of the steady forced vibration of the body twitch by about 4 times. At the same time, the frequency of body twitching practically does not change. Calculations also show that the introduction of such a nonlinear coupling does not significantly affect the galloping of the body.

The calculations made it possible to reveal the negative properties of the suspension of a body containing leaf springs. This is manifested in a much weaker damping effect on the oscillation of the body galloping compared with the fluctuations of bouncing.

In the process of investigating of the nonlinear oscillations of the cryogenic section, two parameters varied: the speed of motion at a constant dry friction force in suspension and the frictional force at a fixed speed.

Figure 3 shows the amplitudes of the steady body oscillations, depending on the speed of the section with frictional force realized in the springs, F = 11.75 *kN*. The nature of the curves reflects the main features characteristic of the linear model: in the velocity interval 19.4 ... 20.8 *m/s*, there is a resonant peak corresponding to the bouncing of the body, and in the interval 22.2 ... 23.6 *m/s* there is a galloping resonance bodywork. It should be noted that the amplitude of galloping of the body in the nonlinear system, which is associated with the weak and uneven damping ability of the elliptical springs, is substantially increased - by more than 60%, as noted earlier.

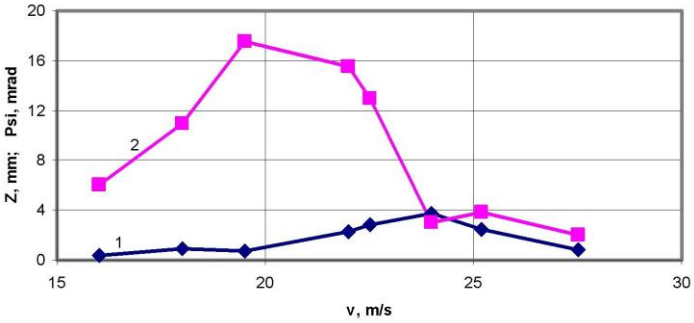

Figure 3: Amplitudes of stabilized body oscillations depending on the speed of movement

1 - galloping; 2 - bouncing

Figure 4 shows the amplitudes of the forced body oscillations depending on the magnitude of dry friction forces in the suspension of the body at a speed of 23.6 *m/s*. This option corresponds to the state of the system, which is close to the body's resonant galloping.

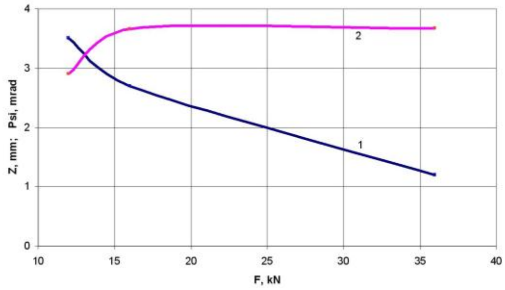

Figure 4: Amplitudes of forced body oscillations in dependence on the magnitude of the dry friction force in the suspension of the body

1 - galloping; 2 - bouncing

Figure 4 shows that an increase in the frictional force in the suspension of the body leads to a redistribution of the oscillation energy in the system, causing a gradual decrease in the galloping amplitude of the body, but at the same time leading to a slight increase in its vertical displacements. Reducing the amplitude of body galloping leads to an increase of the acceleration of this mode of oscillations up to 40%. Such a redistribution of energy is also observed with resonant oscillations of the body bouncing.

Particular attention given in the numerical analysis of the model to the body oscillations is explained by the desire to predict the possibility of non-standard dynamic regimes in the system. The hydrodynamic impact is one of such regimes for fluid systems.

We reduce the oscillations of the cryogenic liquid in the inner vessel of the gasifier to the equivalent plane problem for a rectangular reservoir [4]. We will assume that the initial phase corresponding to the occurrence of hydrodynamic impact in the vessel is the moment when the upper generatrix of the cylindrical reservoir or the upper face of the parallelepiped of the equivalent rectangular tank touches the free surface of the oscillating liquid. Then the following inequality becomes the condition for the hydrodynamic impact [4]:

$$\left|x_1 - X - \Psi h^*\right| \geq \frac{L_p \cdot \Delta h}{6h^*}, \quad \text{at} \ \ \Delta h \leq h^*,$$

where h^* is the reduced depth of the liquid [4]; L_p is the length of the tank;

Δh is incomplete filling with liquid.

The analysis of the oscillations of liquefied natural gas in gasifier tanks showed that in the stationary dynamic regimes the condition of the hydrodynamic impact is not met. In the steady-state modes, the longitudinal displacements of the center of mass of the liquid do not exceed small values of 0.004 ... 0.006 *m*. However, there is a qualitative difference between the oscillatory processes of a fluid and the dynamic processes of other elements of the system under consideration. This difference is manifested in a much longer duration of the transient process, which is due to the smallness of the damping values and the natural oscillation frequencies of the liquid. Due to this, in non-stationary dynamic modes (transient processes of train movement, a passage of the sequences of short unevennesses, etc.) there is accumulation in the coordinate x_1 of the non-stationary perturbing influences, which is also facilitated by strong interconnection of fluid oscillations with angular body oscillations, the damping efficiency of which by dry friction dampers is low.

The use of the principles of system analysis, which is the basis for the numerical implementation of the mathematical model of the cryogenic section, provides not only the use of models of rational complexity depending on the structure of the research object, and a reasonable choice of software and computer facilities in accordance with the level of complexity of models, but also provides undeniable advantages in terms of expanding the range of estimates of the dynamic models correctness. This is manifested in the possibility of mutual comparison of models of different levels of complexity, which are based on different modeling principles or different types of disturbances. An illustrative example here is a comparison of the indicators of dynamic qualities predicted on the basis of deterministic linear and nonlinear models, as well as a stochastic linearized model of the cryogenic section.

Figure 5 shows the curves of the change in the mean square values of the dynamics coefficient of the second stage of suspension of a refrigerator car depending on the speed of movement. The running gear of this car, based on two-axle bogies KVZ-I2, and the main parameters of the mechanical part are approximately the same as those of the cryogenic section. Curve 4 corresponds to the data of full-scale tests carried out by the Research Institute of Railway Car Building [4], in the cold period of time.

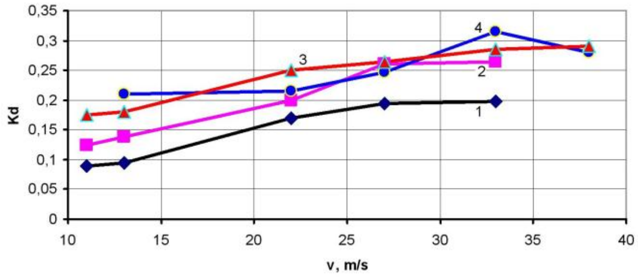

Figure 5: Comparison of calculated and experimental mean square values of the coefficient of dynamics of the second stage of suspension

1 - linearized model, summer conditions of the track;

2 - linearized model, winter conditions of the track;

3 - nonlinear model, winter conditions of the track;

4 - experiment, winter track conditions

The analysis of the calculation results shows that in the speed range from 19.4 to 38.9 *m/s*, the discrepancies between the experimental and theoretical values for the nonlinear model do not exceed 11%, and for the stochastic model, depending on the number of the bogie in the course of movement, vary from 5 to 20 %. The mean values of the last two dependencies are close to the data of the nonlinear model.

CONCLUSION

Theoretical forecasting of dynamic characteristics of the cryogenic section is performed basing on the developed mathematical model of its oscillations. This made it possible to identify two critical speeds in the field of operational speeds corresponding to the resonances of the body according to the galloping and bouncing oscillations. The calculated values of the coefficient of vertical dynamics to 0.34 ... 0.36 and the acceleration of the body to 0.34g are obtained, which indicates that the vertical quasistatic loads have been exceeded; which were taken into account while choosing gasifier equipment and its support units. The negative properties of the spring suspension of the cryogenic section are proved; they are explained by the absence of dampers in the pedestrian suspension stage and the presence in the central stage of elliptical springs with unstable scattering characteristics. The ability of liquefied natural gas, due to the smallness of the damping, to accumulate during the movement of the section relatively often repeated non-stationary effects, the superposition of which can lead to the occurrence of a hydrodynamic impact in the internal vessel of the gasifier.

REFERENCES

[1] Lenz, M., 2014, “Diezel fuel: liquefied or compressed natural gas”. International Railway Journal, #9, pp. 69-72.

[2] Vantuono, W., Smith, W., 2013, “Liquefied natural gas – fuel of the future”. International Railway Journal, #12, pp. 32-35.

[3] Kossov, V.S., Babkov, Yu.V., Sazonov, I.V., 2016, “Locomotives fueled by liquefied natural gas”. Russian Journal of Heavy Machinery, #9, pp. 34-39.

[4] Volkov, I.V., 2000, “Prediction of dynamic characteristics of the rolling stock on the basis of mathematical modeling”, monograph, Rostov on Don: Publishing house of the North-Caucasian Scientific Center of Higher School, Russia. 136 p.

[5] Bykov, V.B., 2004, “Construction of trucks for freight and passenger wagons”, Moscow: Marshrut, Russia. 36 p.

[6] Volkov, I.V., Voron, O.A., Bulavin, Yu.P., 2016, “Simulation of railway carriage with KVZ-I2 bogie and undercarriage v-belt generator drive”. Vestnik RGUPS, #3, pp. 14-22.

[7] Shimanovsky, A.O., 2016, “Recent research of dynamics and strength of tank vehicles”. Mechanics of Machines, Mechanisms and Materials, 2016, #3 (36), pp. 59-70.

[8] Bogomaz, G.I., 2004, “Dynamics of railway tank-wagons”, monograph, Kiev: Naukova dumka, Ukraine. 224 p.

[9] Babakov, I.M., 2004, “Theory of oscillations’, Moscow: Drofa, Russia. 591 p.

[10] [10] Biryukov, I.V., Savoskin, A.N., Burchak, G.P., etc., 2013, “Mechanical part of traction rolling stock”, Moscow: Alians, Russia. 440 p.