IOP Conference Series: Materials Science and Engineering

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To cite this article: O M Pshinko *et al* 2020 *IOP Conf. Ser.: Mater. Sci. Eng.* **985** 012014

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IOP Conf. Series: Materials Science and Engineering **985** (2020) 012014 doi:10.1088/1757-899X/985/1/012014

To the problem of train running safety

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Abstract. The paper is presenting the procedure used for establishing the cause of the wagon derailment. To do this, the computer simulations and the computational software, developed in the Dnipro National University of Railway Transport (DIIT) were used. The level of longitudinal forces and the wagons dynamic performance have been evaluated using the mathematical models of longitudinal oscillations of a train and the spatial vibrations of wagons, in particular of tank wagons. As a result of modeling we obtained oscillograms of longitudinal forces in each inter-wagon connection, the dependence of the largest longitudinal forces on travel time and distance traveled, the distribution of the maximum longitudinal forces along the train length, the speed dependence on travel time and track coordinates. We also obtained the dynamic performance of wagons: the vertical dynamics coefficients of the axle-box and central suspension, the horizontal dynamics coefficients and the derailment stability coefficient. The influence of the movable load in the tank wagons and the characteristics of rail irregularities on the stability coefficient against wheel climbing onto the rail is also considered. The presented methodology was used to determine the cause of the tank wagon derailment in a non-homogenous freight train consisting of 50 wagon tanks on an existing track section of the Lithuanian railways. When simulating the train movement, it was assumed that the train was equipped with elastic-friction absorbing devices and air distributors, turned on to the average operation mode. As a result of numerical experiments, an assumption was made about the cause of the train derailment.

1. Introduction

Despite the large number of solved problems related to analysis of the dynamic loading of wagons, which affects the traffic safety, situations leading to derailment still take place at the railways of the world [1-4]. As a result of such accidents, not only rolling stock but also infrastructure can be damaged. It is important to determine the causes of such accidents in order to prevent their repeating. As one knows, driving non-homogenous tanker trains is connected usually with some difficulties [5, 6]. In a traditional train the wagons' vibrations take place under the action of longitudinal forces, originated from the driver's control actions as well as because of the forces caused by track profile and its curvilinearity in plan. In a train with tank wagons, those vibrations take place additionally under the action of the fluid, moving in a tank wagon relatively to the shell.

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2. Methodology

This article provides a methodology for determining the causes of derailment using computer simulation of train movement.

Determining a cause of wagon derailment is based on the materials provided by the official investigation:

- the consignor list;
- modes of train driving;
- the data on the tank shells calibration;
- the certificate of the train's speedometer tape transcript;
- the certificate on the wagons' brakes;
- the data on the longitudinal track profile and on its curvilinearity in a horizontal plane;
- freight wagons' technical passports;
- the scheme of the derailment place.

In general, the wagon derailment is possible for the following causes:

- technical condition of wagons;
- improper driver's actions;
- deviations in the track maintenance.

If, according to the technical passports of wagons and the certificate on brakes, all wagon parameters comply with regulatory requirements, then the first derailment cause can be excluded. To evaluate the driver's actions that can lead to wagon derailment, especially as a result of braking, it is first advisable to implement provided parameter chart (train driving modes that preceded wagon derailment) using computer simulation and compare the obtained trajectory using the train's speedometer tape provided for transcript. During simulation, the train is considered as a one-dimensional array of bodies (carriages) interconnected by non-linear deformable elements, taking into account the presence of gaps in the inter-wagon connections. If the train includes tank wagons, then it is taken into account the fact that in case of tank ullage, part of the fluid moves in the longitudinal and transverse (horizontal) directions. In this case, in the design scheme, the fluid is represented as one moving mass and one stationary mass rigidly connected to the body. It is assumed that the moving mass is located above the stationary mass and is "articulated" with it and the tank by a "spring." Fluid oscillations caused by perturbations are described by differential equations having a certain specificity [8]. Braking force, resisting force from track profile and plan, the force of the main resistance to the translational movement of the train, the traction force or the electric braking force of the locomotive and the longitudinal forces arising in the inter-wagon connections were taken as disturbances acting on each carriage [5, 6]. As a result of simulation, oscillograms of longitudinal forces in each inter-wagon connection, the dependences of their maximum values on the travel time and the distance traveled, the distribution of the maximum values of longitudinal forces along the train length, the movement speed dependence on the travel time and the track coordinates are obtained. If the obtained movement speed curve satisfactorily coincides with the results of transcript of speedometer tape, then we can analyze the level of longitudinal compressive forces. As it is known [9], for lift-off of a 4-axle fully loaded open wagon with 18-100 bogies, the dangerous level of quasistatic longitudinal compressive forces (acting during 1.5-2 sec) is 1000 kN, and for empty wagon – 500 kN. If the level of longitudinal compressive forces obtained as a result of simulation exceeded the indicated values, the wagon lift-off is possible. In order to make sure that such a control mode has led to the emergence of dangerous longitudinal forces, another control option should be implemented (change the track coordinate in which braking started or perform braking with less discharge of the brake line). If the level of

longitudinal forces does not exceed the permissible value, it can be argued that the control mode implemented by the driver has led to the wagon derailment.

Next, you need to evaluate the dynamic indicators of the wagons, characterizing the movement stability. To do this, we simulate the spatial vibrations of the wagon, arising not only from external forces, but also from rail irregularities.

The controlled simulation parameter is the derailment stability coefficient.

If the derailment stability coefficient does not exceed the permissible value, then it can be argued that the driver`s actions could not lead to the wagon derailment.

Next, you need to evaluate the derailment stability coefficient taking into account the rail irregularities. If the stability coefficient has exceeded the permissible value, then the cause of derailment is a deviation in the track maintenance.

To simulate the spatial vibrations of the wagon, a spatial model of the carriage is used. The wagon is divided into separate objects with the corresponding connections between them.

In this work, for the first time, when simulating the spatial vibrations of tank wagons, a movable load was also taken into account. Therefore, the wagon (tank) was divided into the following objects:

- movable load;
- body + fixed load;
- two truck bolsters
- four side frames;
- four wheel sets.

These bodies are regarded as solid and non-deformable. In addition, the inertial and viscoelastic properties of the track in vertical and horizontal transverse directions were taken into account. Therefore, the above list of objects also includes the reduced mass of the rails.

All objects have at least one inertial parameter – mass (for example, movable load and rail). In addition, objects can also have other inertial parameters – mass moments of inertia relative to their main central axes (if the angular displacements of objects are taken into account). Position of any object at any time moment is determined by six coordinates. The movement equations of the object under consideration in Cartesian coordinates have the following form:

$$\begin{cases} m \cdot \ddot{x} = F_x + \sum_{k=1}^{n} R_{xk} \\ m \cdot \ddot{y} = F_y + \sum_{k=1}^{n} R_{yk} \\ m \cdot \ddot{z} = F_z + \sum_{k=1}^{n} R_{zk} + W \\ J_x \cdot \ddot{\theta} = M_x + \sum_{k=1}^{n} MR_{xk} \\ J_y \cdot \ddot{\varphi} = M_y + \sum_{k=1}^{n} MR_{yk} \\ J_z \cdot \ddot{\psi} = M_z + \sum_{k=1}^{n} MR_{zk} \end{cases} \quad \textbf{(1)} \qquad (1)$$

where F_x, F_y, F_z – projections of external forces on the x, y, z axes, respectively, R_{xk}, R_{yk}, R_{zk} – projections of connection reactions on the x, y, z axes, respectively, M_x, M_x, M_x – moments of external forces F relative to the x, y, z axes, respectively, MR_{xk}, MR_{yk}, MR_{zk} – reaction moments R_k relative to the x, y, z axes, respectively, $\ddot{x}$, $\ddot{y}$, $\ddot{z}$ – objects acceleration along the x, y

, z axes, $\ddot{\theta}$, $\ddot{\varphi}$, $\ddot{\psi}$ – angular accelerations of objects when rotating them around the x, y, z axes; J_x, J_y, J_z – inertia moments relative to the main central axes x, y, z respectively, m – object mass, W – object weight. As external forces F_x, F_y, F_z, acting on the tank shell, the components of the longitudinal forces in the inter-wagon connections are taken. Vertical components are caused by the difference in height between the longitudinal axes of automatic couplers of neighboring wagons; the horizontal ones are caused by the relative rotation of automatic couplers of two neighboring wagons. External forces acting on wheel sets are caused by rail irregularities. In equations (1), n is the number of connections applied to the object. As can be seen from the movement equations of body (1), the right parts include the connection reactions R_{xk}, R_{yk}, R_{zk}, which depend on the type of connection (linear elastic-viscous, bilinear elastic-viscous, "dry friction"). Between wheels and rails, friction and interaction forces arise due to the elastic movements of the rails. The friction forces (creep) and their components along the axes x and y are determined by Carter's theory. When creating a carriage model, the type of connection and its numerical values are indicated for each pair of objects [10].

In our computations the coefficient of stability against derailment (sometimes called coefficient of resistance to derailment or stability factor) has been computed by the following formula: $C_S = \dfrac{tg\alpha - \mu}{1 + \mu tg\alpha}\dfrac{S_p}{S_y}$, where: α – is the flange inclination angle, μ – friction coefficient, S_p – vertical interaction force between the leading wheel and rail, S_y – lateral interaction force between the leading wheel and rail [4, 10-12].

Based on mathematical models of the longitudinal train dynamics, fluid oscillations in a tank, and spatial vibrations of wagon [5–8], a computer program was developed that was used for calculations.

The above methodology was used to determine the derailment cause of the first tank wagon, in the direction of travel, in non-homogenous train consisting of 50 tank wagons with one 2M62m diesel locomotive, which is located in the head of the train. The train was formed according to the following scheme: 19 loaded tank wagons with tank ullage of 0.59 m and a mass of 86 tons, 24 empty tank wagons and 7 loaded tank wagons with tank ullage of 0.49 m and a mass of 88 tons.

Figure 1 presents the site diagram, in the lower part of which the distances to the control points relative to the point of computation beginning are given. The same distances are shown in the graphs of the computation results. In the above figure, L_1 and L_2 are the lengths of the transition curve at the entrance to the corresponding curve, R_1 и R_2 are the radii of the circular curves, respectively. During computations, an element 800 m long with a slope of 0.008 was added before the first element of the track profile, on which the train was located at the time of brake test.

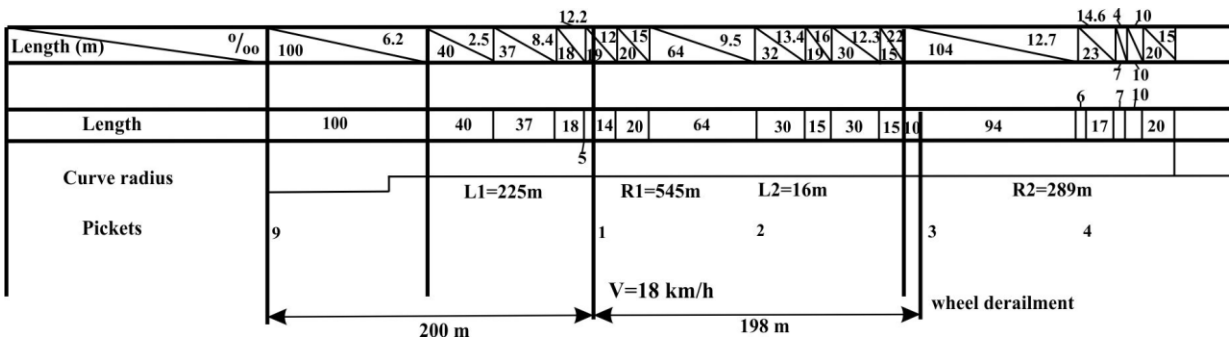

Figure 1. The track section scheme.

As appears from the official investigation materials, immediately before the tank wagon derailment the driver run brake test (on 38 km/h speed) on a descending grade and in a curve. Then, after brake release, he allowed the train to speed-up till 24 km/h and performed the emergency braking till the train's stop. The longitudinal compressive forces arising in the train during such a maneuver can cause lift-off of wagons. To determine the cause of the derailment, the computer modelling was performed. Its results are presented below.

3. Study case

3.1. Determination of longitudinal forces in a train

The proposed methodology was used to determine the cause of the tank wagon derailment in a non-homogenous freight train consisting of 50 wagon tanks on an existing track section of the Lithuanian railways. The motion trajectory (dependences of movement speed on running time and track coordinate) was evaluated first (see Figures 2 and 3). It was performed at the set modes of train driving, starting from the brake test at 38 km/h, the following brake release, speeding-up from 18 km/h at coasting retardation to 24 km/h due to the longitudinal profile configuration and then emergency braking. In these Figures the track coordinates are shown for the locomotive.

As a result of the computations, the satisfactory coincidence between the train motion trajectory and the data presented in the speedometer tape transcript has been obtained. In particular, according to this source of information, speeding up of the train from 18 km/h to 24 km/h after the brake release, took 200 m of the track, and the modelling gave the value of 185 m. Braking distance after applying the emergency brakes at 24 km/h was 80 m according to the investigation materials that fully coincides with the simulation results (Figures 2 and 3). Therefore, we can proceed to the analysis of the longitudinal forces obtained as a result of the computation.

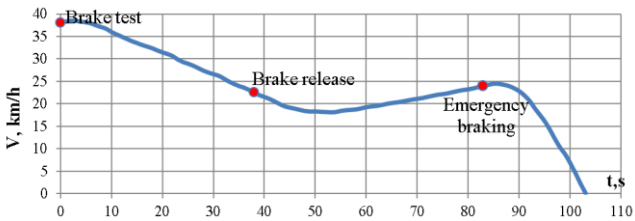

Figure 2. Dependence of movement speed on travel time.

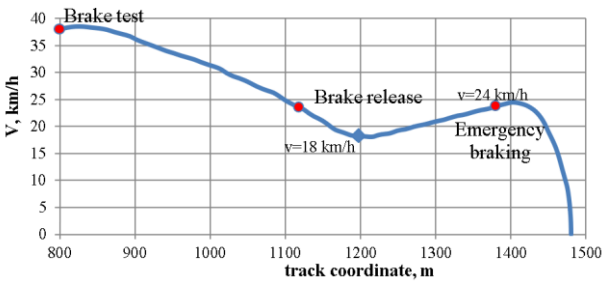

Figure 3. Dependence of speed on the coordinate of the path.

During computations, oscillograms of the longitudinal forces in the front and rear automatic couplers of the derailed tank-wagon (Figure 4) and the distribution of the maximal longitudinal forces $\max S$ along the train length (Figure 5) were obtained.

The level of quasistatic longitudinal compressive forces acting on the 1st tank wagon after brake release was approximately 500 kN (Figure 4), and the maximum shock compressive forces did not exceed 600 kN. This result does not give reason to make assumptions about wagon lift-off due to braking.

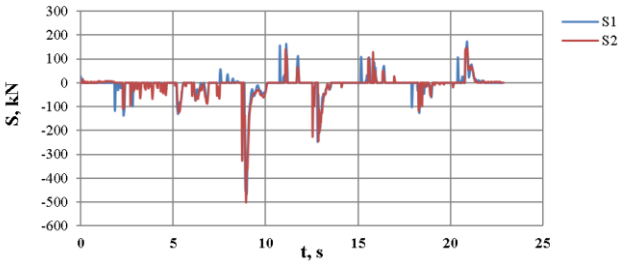

Figure 4. Longitudinal forces acting on the derailed tank wagon.

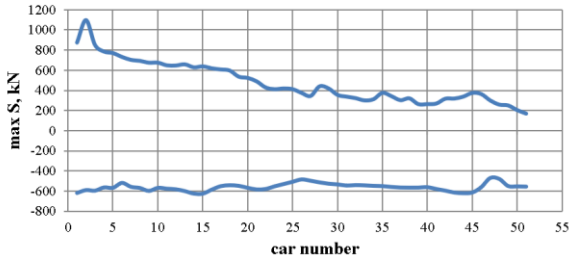

Figure 5. Distribution of the forces' maxima along the trainset length.

3.2. Determination of the dynamic indicators of the wagon tank without taking into account the irregularities of the railway track

Now we evaluate the derailment stability coefficient. As a result of modeling the spatial oscillations of the first (derailed) tank wagon after the locomotive, the following was obtained:

- vertical dynamics coefficients of the axle-box Kdv1 and central Kdv2 suspension stages,
- derailment stability coefficient,
- horizontal dynamics coefficients Kdh .

As an example, the values of the coefficients of vertical and lateral dynamics as well as the derailment stability coefficient are shown in Figures 6 and 7. The track coordinates are given for the first tank wagon, also on the track section preceding the derailment site.

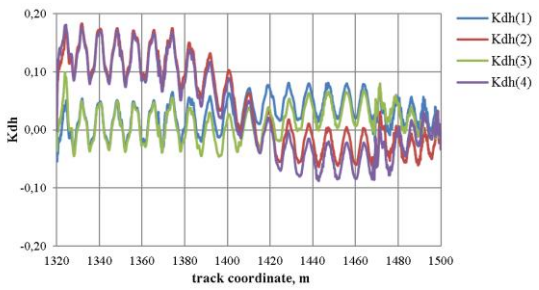

Figure 6. Coefficients horizontal dynamics 4 wheelsets.

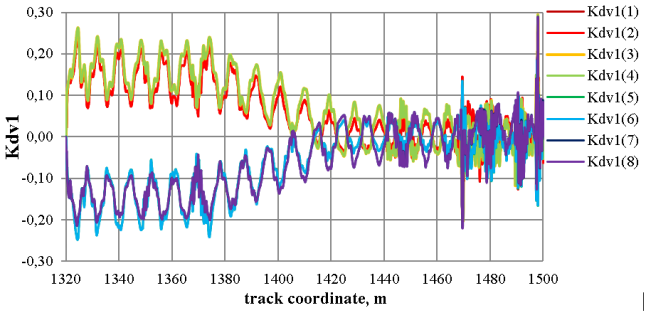

Figure 7. Coefficients of the vertical dynamics of the axle box.

In Figure 6, the notation $Kdh(i)$ correspond i -th to the wheel pair number. In Figure 7, the graphs $Kdv1(1) - Kdv1(4)$ – correspond to the change in the coefficients of the vertical dynamics in the left axle boxes of all wheel sets, $Kdv1(5) - Kdv1(8)$ – in the right axle boxes of all wheel sets. The nature of the obtained coefficients of the horizontal dynamics of the wheel sets is explained by the curve running in the track plan. As can be seen from the above Figures, the values of the coefficients of horizontal and vertical dynamics do not exceed the permissible values of 0.38 and 0.6, respectively [13, 14].

The performed modelling gave the value of 1.8 for the coefficient of stability for the first tank wagon. That is not less than the permissible value of 1.4 [10, 11].

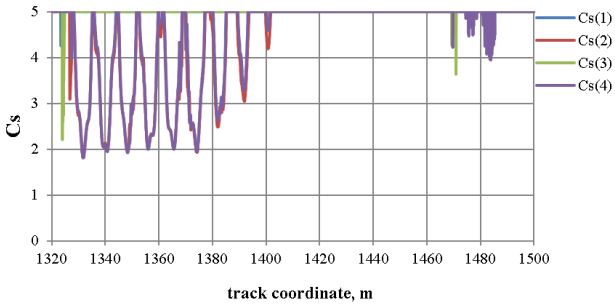

Figure 8. Derailment stability coefficient.

Thus, the obtained results allow to say that the train driving mode and the drivers' actions could not lead to the derailment.

3.3. Determination of the dynamic indicators of the wagon tank taking into account the irregularities of the railway track

Let us now evaluate the influence of the track condition on the derailment stability coefficient. Based on the given information the displacement of the left and the right rails in relation to their axes in the plan and profile were determined. These data were used as the rail irregularities for determination of the tank wagon motion stability (Figure 9, 10).

Below in Figure 9, deviations in the vertical plane of each rail from its longitudinal axis are shown. Positive values of Z correspond to downward deviation from the longitudinal axis of the track. Figure 10 presents deviations in the horizontal plane of each rail from its longitudinal axis. In this Figure,

positive Y values correspond to the rail deviation to the right of its longitudinal axis in the direction of travel.

The results of calculations for determining the tank wagon running safety taking into account the above mentioned rail irregularities in the vertical and horizontal planes, are shown in Figure 11.

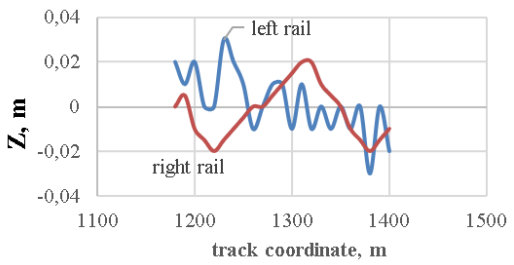

Figure 9. Vertical track irregularities.

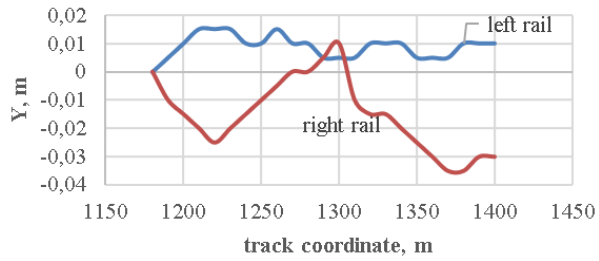

Figure 10. Lateral track irregularities.

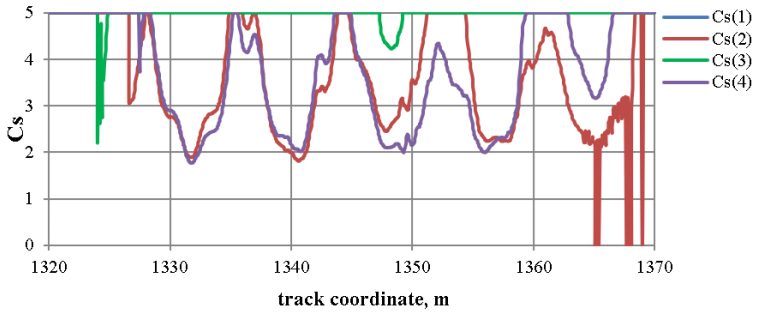

Figure 11. Derailment stability coefficient.

The coefficients of vertical dynamics in the axle-box Kdv-1 and in the central suspension Kdv-2 correspondingly, as well as the coefficient of horizontal dynamics Kdh are shown in Figures 12-14.

Taking into account the rail irregularities significantly changed the form of the dependences of the dynamics coefficients on the track coordinate (Figure 12-14), although their maximum values are within the normal range. It is seen in Figure 11, that the stability coefficient has fallen to zero. Thus, one can claim with confidence, that the studied derailment case was caused by non-compliance of the track with the Norms of Track Maintenance as a result of badly performed repair works.

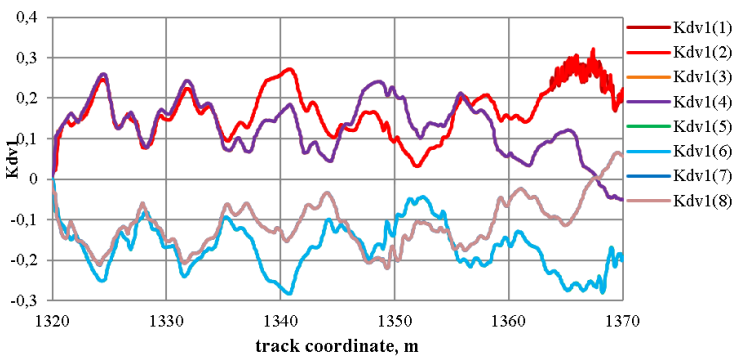

Figure 12. Coefficients of vertical dynamics in the axle-box unit.

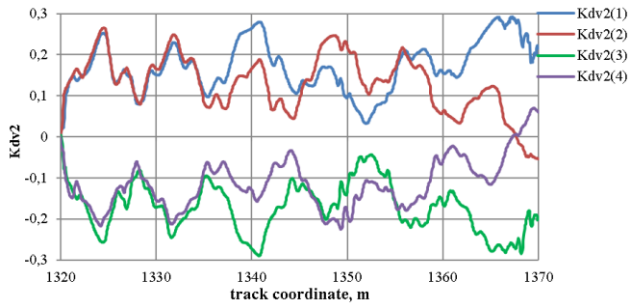

Figure 13. Coefficients of vertical dynamics in the central suspension.

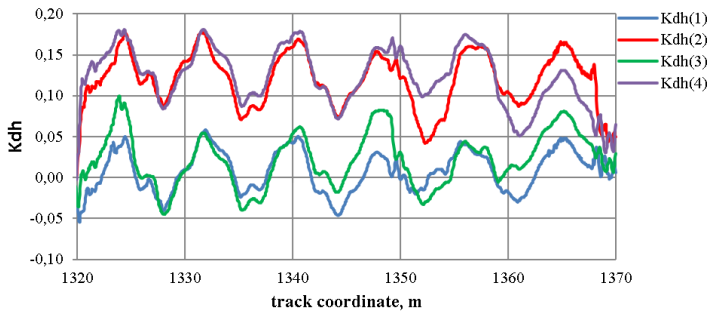

Figure 14. Coefficients of lateral dynamics.

To evaluate innovations in the spatial model of the tank wagon (load movability) for the stability of wagon movement, a similar simulation of the movement of a freight non-homogenous train was carried out, in which load movability was not taken into account. The results of the calculations for determining the stability coefficient of the leading tank wagon taking into account given rail irregularities in the vertical and horizontal planes are shown in Figure 15.

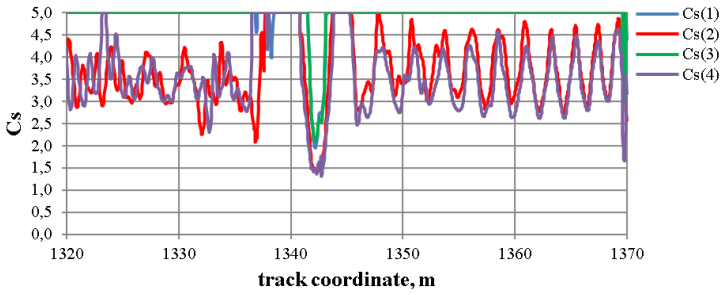

Figure 15. Derailment stability coefficients for open wagon freight train.

As one can see from the plot, the derailment stability coefficients in this case are greater than 1.4. From this, it is possible to say that not only deviations in the track maintenance, but also fluid vibration in the tank wagons were the cause of derailment.

4. Conclusions

When analysing the results of modelling, one can come to the following conclusions:

- the given methodology can be used to establish the causes of wagon derailments, after providing all materials of the official investigation, using computer simulation;
- for determining the derailment causes for the first time its spatial model, which takes into account the load movability was used;
- as a result of simulation of train movement and the spatial oscillations of the tank wagon, the oscillograms of the longitudinal forces, the distribution of their maximum values along the train length, the dependence of speed on the travel time and the track coordinate, as well as the wagon dynamic indicators and the derailment stability coefficient should be obtained and analyzed;
- the above methodology was used to determine the derailment cause in a real non-homogenous freight train formed from 4-axle tank wagons;
- based on the analysis of the results, it was concluded that the cause of the tank wagon derailment was not only the deviation from the track maintenance norms, but also the fluid fluctuations in the tank wagons.

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