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Transversal Displacement of Freight Wagons Bogies

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Abstract. This paper is focused to the study of the influence of lateral displacement of open box type cargo wagon bogies on their main dynamic indicators and indicators of the interaction of rolling stock with rails. The theoretical study was carried out using the model of spatial oscillations of the five wagons set. During the calculations, the change in the angle of rotation of the central axis of the wagon body was considered, which leads to a mutual transverse displacement of the bogies. As a result of research, the dependences of the indicated indicators of a freight wagon on the magnitude of the angle of rotation of the body and speed of movement were obtained.

INTRODUCTION

Traffic safety is a strict requirement for the railway transport operation. The importance of traffic safety ensuring increases with increasing of train speeds, which is being hold in Ukraine. Especially important is the question of determining the permissible speeds, which is associated with the assessment of the rolling stock dynamic qualities. This assessment is carried out during the acceptance tests, running dynamic tests and influence to the track tests. It is impossible to replace part of the natural tests with computer simulation, because by reduction of the nature tests change in requirements for traffic safety and the quality of rolling stock dynamic indicators threatens. The mathematical model of rolling stock is a certain idealization of the real system, of some parameters that are not well known. When testing a new rolling stock design, the features that were not taken into account in the mathematical model may appear. Experimental studies, although they are sufficiently reliable, require significant skill and long time. They are not able to cover all situations that may arise in operation. Theoretical and experimental studies should be carried out jointly. Experimental studies should provide an opportunity to clarify the main hypotheses, design schemes. If necessary, correct the results of theoretical studies and perform the recalculations [1 - 4].

ANALYSIS OF THE LATEST RESEARCH AND PROBLEM FORMULATION

The process of interaction between the rolling stock and the track is determined by many factors, as presence of superelevation and curved track sections; vehicles weight, length and speed; locomotives power. The greatest horizontal transverse forces are caused by curved track sections passing. Moreover, in curves with a radius of less than 800 - 600 m, the flanges of some wheels are pressed against the outer rail for almost the entire length of the curve. The rail contact-fatigue formation and large number of train derailments caused by lack track and rolling

stock structural strength, as well as the loss of their stability on curved tracks. Derailments can be divided into two types. The first one is attacking of the wheelset to the outer rail with subsequently derailment. The second one is attacking of the wheelset to the rail head or tongue rail with the subsequent derailment of another wheelset to the track axis due an unacceptable transverse hijacking of the rail-sleeper grid [5]. This is most likely for empty wagons in curves, or even in straight sections while wagon squeezing under the action of oncoming unbraked wagons when the train is braked. If there is even a slight tilt of the rail, the wheel flange under action of the transverse force that arises in this case, rolls along the inclined face first onto the rail head, and after a few meters it falls from the rail head on the sleepers and to the ballast prism [5].

When the wagons are situated at an angle one to another, lateral components from the wagon coupling forces appear. When the height of couplings is different, vertical components create additional loading or unloading of the respective wheels. This negatively affects the lateral and vertical wheel forces. Difference in the height between wagon couplings in cause of freight train is allowed no more than 100 mm [6 - 8].

The main reason of the second type derailment is the unacceptable widening of the track gauge, which occurs due to force acting of the flange to one of the rails. In this case, the second wheel just falls inside the track. The probability of such derailment is higher in small radius curves, on wooden sleepers, on rails with low transverse rigidity, when used typical rail-to-sleeper fixtures used in order to ensure rails inclination. On track sections where concrete sleepers are used, the derailment probability is significantly minimized because of increasing stiffness of the rail-sleeper fixtures. [5, 9 - 11].

An increase in the level of interaction forces between railway carriages and rail tracks is caused by rail and wheel profile wear, so a horizontal gap is enlarged. Because of wear, additional gaps in the wheelset guidance is formed which in some cases exceeds 5 mm. In the bogie to vehicle body connection, additional gap of 10 mm can occurs because of the main pivot wear or malfunction.

Transverse forces in the rail-wheel contact are influenced by the action of longitudinal forces in the wagon's couplings [6, 7]. The auto coupler model CA-3 (CA-3M), does not have a stabilizing hinge in the shank. The shank radius of the automatic coupling is 130 mm hence the base plate, to which the longitudinal force of the train is transmitted has a radius of 150 mm. So, the compressive force transmitting point moves across the crew [8].

The lateral displacement of bogies relative to each other need to be considered. It is accepted that the rear bogie is centered along the longitudinal track axis, and the front one has a transverse displacement corresponding to the initial angle of body rotation in terms of the track axis in range up 0.002 to 0.008 rad with a plus sign, which increases the attack angle of the front wheelset to the rail.

PURPOSE AND TASKS OF THE RESEARCH

Aim of this study is to determine the influence of freight wagon bogie transverse displacement, taking into account the magnitude of the ride speed on its main dynamic indicators and indicators of the interaction of rolling stock with rails. The main dynamic parameters, as maximum coefficients of dynamic addition of sprung and non-sprung parts, maximum ratio of the frame force to the static axial load, stability factor of the wheel against derailment, coefficients of the vertical and horizontal track dynamics by the rails - wheels interaction forces, stability coefficient of the rail-sleeper grid from yaw under the action of transverse forces, lateral force acting from the track to the wheel, edge stress in the rail bottom, wear factor of the wheel bandage side face, the values of the rail base and rail heads squeezing are studied.

This study is the development of methods for mathematical modeling of dynamic processes of rolling stock - track interaction. Similar theoretical calculations can be applied when assessing the effect of the lateral displacement of the bogie to the dynamic qualities of rolling stock and indicators of the interaction of rolling stock with gauge, taking into account the wear and nodes of parts and components of the bogie when moving in straight and curved track sections with irregularities. In the process of research, the following results are planned:

- Applying the mathematical model to the five cargo wagons set in order to study the open box type wagon and the track dynamic loading process.
- Estimating the main dynamic indicators and indicators of the rolling stock - rails interaction, considering the bogie transverse displacement when running along curved track sections.

MATERIALS AND METHODS OF RESEARCH

A mathematical model describing the spatial oscillations of wagon as part of a train was proposed in [12]. One wagon is considered according to the full design scheme as "zero", and the design schemes of neighboring wagons are simplified as the distance from the "zero wagon" in both directions. As a design scheme of "zero" wagon is used a mathematical model of freight wagon spatial oscillations in the form of a nonlinear multi-mass mechanical system with 58 degrees of freedom, which moves along an inertial, elastic-dissipative track. The wagons adjacent to the "zero" are represented by a system with 12 degrees of freedom. The last coupled wagons are considered in even more simplified scheme - they are systems with six degrees of freedom.

In this paper, we study the effects of the change in the rotation angle of the wagon body central axis to the main dynamic indicators and indicators of the rolling stock - rails interaction. The choice of this factor as a subject of research is related to the fact that it depends on the size of the total free transverse run-up of the wagon body frame relative to the axis of the track, including the transverse gap in the track. In curves, the relative elastic and inelastic wheel slip relative to the rails and the magnitude of the corresponding forces, as well as the geometric conditions of the rail – wheel contact, depend on the track clearance. Taking into account the fact that the lateral wear in the bogie to body connection can reach 10 mm, the angle of wagon body central axis rotation will be 0.007 rad.

The study was conducted using the model of spatial oscillations of the five wagons set in the train. Movement of a 12-532 open box type wagons with 18-100 bogies is considered. Speed range up to 90 km/h, 600 m track radius curve with outer rail superelevation of 120 mm, rails type R 65, wooden sleepers and sharp gravel ballast were used. The stationary mode of motion was studied in order to establish the influence of only the factor under consideration. The running gears of the wagon, the rail and wheel rolling surface were provided in the normal technical condition.

In order to study the dynamic forces and processes operating in the component parts of the bogie or wagon body, the maximum coefficients of the dynamic addition for suspended and unsuspended parts, the maximum ratio of the frame force to the static axial load and the coefficient of stability from derailment are used.

With an increase in the angle of wagon body rotation, the dynamic coefficients under study remain generally unchanged. Over the entire speed range, the indicators in the case of an increase from 0 to 0.007 rad do not exceed the permissible rate. From the obtained results can be concluded that the angle of wagon body rotation in the case of increasing speed does not cause a significant increase in the vertical dynamics of the primary and secondary suspension, and their values do not exceed the values determined by the normative documentation.

The lateral forces acting from the track to the wheel increase, and have a maximum of 42 kN at 0.006 rad of wagon body rotation. Compared to the maximum allowed value of 90 kN, the condition of stability against wheel flanges creeping on rails is ensured. The angle of wagon body rotation is largely dependent on the interaction forces of the contacting bodies and their relative movements. These forces and displacements are determined by the conditions of interaction between wheels and rails. The conditions of bodies interaction in this system depend on the design, technical condition and modes of rolling stock movement, as well as on the design, condition and parameters of the track. But, it was at body lateral displacement of 52 mm (0.006 rad) when the maximum values of the lateral force were obtained. Compared to 0.005 rad, these values are on average 9.5% more.

With increasing train speeds, the dynamic effect of the rolling stock to the track increases. As a result, stresses increase at the edges of the rail bottom part. The maximum stresses, arising in the edges of the rail bottom, are used as a criterion for establishing the permissible speeds. They should not exceed 200 MPa. But in the case of the considered arrangement of freight wagon bogies along the track, redistribution of vertical and horizontal reactions occurs in the contact zone between the wheel and the rail. According to the calculations results, edge stresses increase with increasing of the speed, but they have a minimum at 0.006 rad in the speed range up to 70 to 90 km/h and maximum at the range up to 50 to 60 km/h. However, the obtained results do not exceed the permissible values.

The permissible value of the vertical track dynamics coefficient is calculated in accordance with the permissible dynamic linear load on the railway from the group of bogie axles, with the load of 168 kN/m. The coefficient of vertical track dynamics does not exceed the permissible value. The coefficient of horizontal track dynamics does not exceed the permissible value and also has a maximum in the entire range of the considered speeds at 0.006 rad.

According to the calculation results, the value of the rail-sleeper grid stability coefficient under the action of transverse forces, in the track with sharp gravel ballast is 0.3–0.56, which is less than the permissible value 0.85.

The presented calculations were carried out with the 21.4 t axle load and in the speed range up to 90 km/h, in 600 m track curve. Comparison of the calculation results shows a satisfactory agreement with the experimental data.

CONCLUSIONS

This paper presents some results of theoretical studies of the main dynamic indicators and rolling stock - track interaction indicators using the example of open box type wagons set. The following conclusions can be drawn:

- The maximum coefficients of the dynamic addition of sprung and non-sprung parts, the maximum ratio of the frame force to the static axle load and the coefficient of the stability against derailment are much more dependent on the vehicle speed than on the wagon body longitudinal axis rotation angle.
- The analyzed indicators of the rolling stock - track interaction do not exceed standard values and have extremes when the body rotation angle is 0.006 rad.
- Simultaneously increase in the wagon body rotation angle and the vehicle speed leads to the increase in the wear factor of wheels and rails.
- With the increase in the running speed, the inner rail – sleeper fixtures and railheads are significantly squeezed out; both indicators have maximum values when the wagon body rotation angle is 0.006 rad.
- Comparison of the calculation results with experimental data obtained for the rail foot lateral force parameter shows a good consistency.

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