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Calculation-empirical estimation of the vertical dynamic coefficient in the event of additional oscillations of the car

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Abstract. According to the results of derailment investigations, the ensuring of train traffic safety was found to be a paramount and urgent task. The growth of freight density and speed of movement on the railway network leads to a significant increase in the number of damaged wheels and rails and, as a result, to an increase in the level of force interaction during load transfer, which provokes the likelihood of a wagon derailment. The operation safety level of rolling stock on railways is determined mainly by the presence of a car derailment stability margin, which depends on the vertical dynamics and the coefficient kdv, which changes throughout the entire time of movement and, accordingly, on the technical condition of the car and the track as a whole. The vertical addition to each element of this "wheel-rail" system depends on many factors, therefore, the level of operational safety of the car, as a mechanical system, is determined mainly by its sprung and non-sprung car parts. The calculation schemes used to study the stability of unperturbed motion are non-linear systems. Non-linearities occur due to the gaps in axle boxes, side bearings and similar connections, non-linearity of the interaction forces of wheels with rails and, at present, also in some types of spring suspension. When moving in a curve, the cart is known to perform a complex movement. The rotation and transverse displacement of the bogie is prevented by frictional forces between the wheels and the rails. The bogie is affected by a part of the centrifugal force, unbalanced by the elevation of the outer rail and depending on the speed of movement, as well as by the guiding force from the side of the outer rail. It is known the process of a car movement results in the appearance of horizontal transverse (relative to the direction of movement) forces, which are determined by the level of interaction between the wheel flanges and the rails. The nature of the perturbed movement of the car leads to the appearance of horizontal transverse with respect to the axis of the path of movement. At relatively low speed, the forces that arise after wearing out of the gaps in the track and when the wheel flanges rest against the rail heads, at a certain level of irregularities, are not dangerous for the wheel-rail system. But with an increase in the speed of the car, these forces increase so that there is a threat to traffic safety as the likelihood of derailment increases.

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1 Introduction

The spring suspension system is one of the main parameters influencing the oscillatory process of the car body. It is supposed to provide the necessary smooth running, dynamic stability of the car when moving at speeds that reach the design specifications. This is achieved by the rational parameters of the spring suspension, the main of them being the static deflection and its distribution over the suspension stages; design margin of deflection and coefficient of distribution (friction) of dampers. In order to provide the necessary properties of the car running, the parameters of the elastic elements and vibration dampers are determined, as well as the strength characteristics of the spring suspension elements to ensure their reliable operation.

It should be noted that the nature of wheel wear and galloping (body oscillation around a transverse axis passing through the center of gravity) is due to the unequal deflection of the spring suspension of the front and rear bogies of the rolling stock and is determined not only by defects in the rolling surface of the wheelsets, but also by the characteristics of the path, the number of curves and transition curves, on which this rolling stock mainly circulated. Since the wheel crawls onto the rail when the lateral forces acting on the wheelset coincide with the unloading of the oncoming wheel due to body vibrations on the springs, it is necessary to set the type of spring suspension:

1) linear spring suspension with reduced stiffness over the entire operating load range;

2) bilinear suspension, in which only a part of the springs operates in the empty mode (outer springs in sets located under the bolster), and in loaded mode (all the springs of the set are included in the operation).

In linear spring suspension, all springs operate, since springs of the same height are used in the kit. However, increased suspension flexibility leads to increased stress in the springs and a decrease in their fatigue strength. In linear suspension, the degree of damping and cohesion of the bogie sidewalls are proportional to the static load, which leads to a decrease in the critical speed at which wobbling occurs, especially in the case of wear of the wedge system.

In bilinear suspension, the stress in the springs under full static load is smaller, and therefore their fatigue strength is higher. The wedge springs have a more rigid linear characteristic; in the empty mode the wedge system accounts for a larger share of the load than in the laden one. This leads to an increase in the relative friction of the vibration damper of the spring suspension and the connectivity of the sidewalls, which makes it possible to increase the critical speed of empty cars.

At the end of the 1990s, individual components and parts of the 18-100 bogie were modernized, aimed at improving its performance. In this regard, it was necessary to consider the issue of removing restrictions on the permissible speed, in connection with which the speed was limited depending not on the state of the chassis, but on a combination of road irregularities. When setting these limits, it was considered that intense wagon vibrations are excited by an unfavorable combination of track deviations, which should decrease to a safe value if the state of the running gear improves as a result of the modernization and increase in suspension flexibility.

2 Methods and materials

When investigating the derailments, a set of experimental studies was carried out in order to identify additional wagon vibrations and deviations in the wheel-rail system. The frame force Hp as a fraction of the axial load Hp/p0 is approximately 0.09 to 0.31 with a rate of 0.4. When driving at speeds of 60-80 km / h in a curve with a radius of 300-650 m and a combination of track irregularities, the skew of deviations in the plan and the drawdown

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increase the indicators of frame forces, the coefficients of vertical dynamics of sprung and non-sprung parts of the car (Table 1).

Table.1 Dynamics values of sprung and non-sprung car parts

	Normative value	Car type	Speed, km/h					
			44	50	55	65	77	78
Frame forces in shares of axial load[Hp/Po], no more than	0,40	boxcar 11-280	0,15					
		gondola car 12-127		0,16				
		boxcar 11-066-04			0,18			
		gondola car 12-132-02				0,21		
		tank 15-869					0,24	
		tank 15-1547						0,25
Vertical dynamics coefficient of non-sprung parts, no more than	0,98	boxcar 11-280	0,67					
		gondola car 12-127		0,76				
		boxcar 11-066-04			0,84			
		gondola car 12-132-02				0,8		
		tank 15-869					**1,14**	
		tank 15-1547						**1,16**
Vertical dynamics coefficient of sprung parts, no more than	0,75	boxcar 11-280	0,6					
		gondola car 12-127		0,67				
		boxcar 11-066-04			0,74			
		gondola car 12-132-02				0,7		
		tank 15-869					**1,04**	
		tank 15-1547						**1,06**

According to the "Norms", the stability of the wheelset against derailment is checked for the most dangerous cases of a combination of a large transverse interaction force between the oncoming wheel and the rail and a small vertical load on the wheel. In this case, it is possible for the crest of the oncoming wheel to crawl onto the rail head with the subsequent derailment of the car off the rails. Forces Pv1, Pv2 for existing car designs are determined by formulas (1) and (2).

$$P_{B1} = p_0 \cdot \left[\frac{b - a_2}{l} \cdot \left(1 - \bar{k}_{\partial в}\right) - \frac{b}{l} \bar{k}_{\partial б} \right] + H_P \frac{r}{l} + P_{кп} \frac{b - a_2}{l}, \quad (1)$$

$$P_{B2} = p_0 \cdot \left[\frac{b - a_1}{l} \cdot \left(1 - \bar{k}_{\partial в}\right) + \frac{b}{l} \bar{k}_{\partial б} \right] - H_P \frac{r}{l} + P_{кп} \frac{b - a_1}{l}, \quad (2)$$

where p_0 is axial static load ;

$P_{кп}$ - self-gravity of the wheelset;

b - half the distance between the centers of the axle journals, for standard axles b = 1,018 м;

l - distance between wheel-rail contact points , l = 1,555 м;

a_1, a_2, distance from the points of contact to the middle of the jornals a_1= 0,217 м; a_2= 0,264 м;

r - radius of the wheel in a rolling circle, r = 0,475 м;

$k_{дб}$- the average value of the roll dynamic coefficient, approximately equal to

$$\bar{k}_{\partial б} = 0{,}25 \cdot \bar{k}_{\partial в}, \quad (3)$$

where $k_{дв}$ is the average value of the coefficient of vertical dynamics, the approximate value of which is calculated

by the formula (3)

$$\bar{k}_{\partial в} = 0{,}75 \cdot k_{\partial в} = 0{,}75 \cdot \lambda_в (A + \frac{B \cdot v}{f_{ст}}), \quad (4)$$

where $\lambda_в$ λ is the value depending on the alignment of the bogie. For a freight four-axle wagon $\lambda_в$ = 1;

A, B are the values depending on the flexibility of the spring suspension and the type of car A = 0,03; B =6*10^4;

v - car speed,

[$k_{ук}$]- normalized wheel stability coefficient, [$k_{ук}$] = 1,4;

$$K_{ук} \geq \left\lfloor k_{ук} \right\rfloor.$$

3 Results and discussion

Based on the data obtained, it can be concluded that the normalized wheel stability coefficient [kuk], which should not be less than 1.4, showed completely different values for all derailments. In the study of these derailments, the coefficient of stability of the wheelset against derailment was calculated. The stability of the wheelset against derailment is due to the ratio of the horizontal and vertical loads acting along its axis. The greatest horizontal forces occur when moving in a curve. When the interaction forces of the wheel with the rail are combined, the crest of the oncoming wheel pair of the bogie can crawl onto the rail head, followed by derailment. The critical value of the wheel set stability factor against derailment corresponds to freight cars and is equal to 1.4. Table 2 presents the calculations of the acting forces and coefficients in the studied sections of the derailments.

Table 2 Calculations of forces and coefficients in the studied sections of derailments

№	Derailment section	V, km/h m/sec	R, m	a, m/sec	$Н_{цб}$, kN	$F_{тяж}$, kN	$Н_б$ t	$К_{дг}$	$К_{дв}$	$Р_{в1}$	$Р_{в2}$	$Р_б$	$К_{ук}$
1	Keshevo-Targiz	55/ 15,3	1940	0,119	10,9	10,58	0,29	0,18	0,157	159,91	120,47	161,67	3,69
2	Nizhneudins Taishet	78/ 21,7	1062	0,44	41,5	42,15	0,67	0,249	0,93	59,487	90,49	147,85	1,5
3	Kasyanovka-Polovina	62/ 17,2	630	0,46	44,2	43,29	0,87	0,201	0,725	72,757	112,45	158,99	1,69
4	Taldan-Gudachi	49/ 13,6	635	0,29	27,4	23,95	3,44	0,162	0,013	84,207	130,3	167,86	1,87
5	Atamanjvka-Kruchina	53/ 14,7	320	0,67	62,9	51,94	10,9	0,174	0,014	81,496	126,21	166,91	1,82
6	Tankhoy-Cedar	77/ 21,4	1070	0,43	37,7	23,31	14,39	0,246	0,023	56,638	85,4	138,75	1,52
7	Kamarchaga-Taiga	46/ 12,8	324	0,50	44,5	50,01	5,49	0,153	0,012	84,059	129,91	164,26	1,91
8	Zalari-Tyret	44/ 12,2	430	0,34	32,6	33,16	0,6	0,147	0,011	88,496	137,4	171,50	1,93
9	Kamarchaga Balai	50/ 13,9	324	0,60	56,1	58,89	2,86	0,165	0,013	83,2	129,02	167,17	1,86
10	Delure-Tyret	65/ 18,1	870	0,37	35,2	36,65	1,42	0,21	0,45	70,29	108,11	156,72	1,81
11	Chernaya-Sredneilimskay	58/ 16,1	270	0,95	90,3	92,12	1,82	0,189	0,016	76,483	118,01	161,76	1,76
12	Slyudyanka-Angasolka	51/ 14,2	320	0,62	59,3	58,9	0,4	0,168	0,014	80,735	124,74	162,85	1,85

Flange overhang, bogie base, and curve radius are factors that affect track slope calculation, hence side wear depends on the roll angle, which in turn depends on wheel flange overhang, curve radius, and wheel tread wear. As a result, a side impact occurs, its strength depending on the length of the transition curve; the shorter the curve, the stronger the impact. It is obvious that a large number of factors affect the occurrence of forces, and so do track malfunctions and body roll. The change in the outstanding centrifugal forces affecting the car in the curves is due to the insufficient elevation of the outer rail h. The railway track turns out to be inclined relative to the horizon at an angle

$$\alpha = \arcsin\left(\frac{h}{2S}\right), \quad (5)$$

where 2S is the track gauge, m.

Due to this, part of the centrifugal force is compensated by gravity, since projections of forces directed in opposite directions now act in the floor plane. It is obvious that the result of the vector addition of these forces will be some outstanding lateral force $F_н = F_{ц1} – F_{т1}$.

From the condition of ensuring traffic safety, the outstanding lateral force must be equal to zero, i.e. the above component of the centrifugal force must be fully compensated for by the component of gravity. The maximum elevation of the outer rail is limited to prevent tipping into the inside of the car's curve.

In the study of derailments, the problem of non-compliance with the conditions for ensuring traffic safety was identified, which is associated with many factors, one of them being the elevation of the outer rail over the inner one, which does not meet the standards. For example, the Kamarchaga-Taiga exit: the elevation should be 90 mm according to the established norms for the maintenance of track sections, but factors affecting the change in values and leading to a change in the dynamics of the rolling stock are not taken into account:

- the gauge in this section does not correspond to the norm, there is a broadening throughout the entire section, ranging from 6 to 20 mm, since the elevation depends directly on the state of the gauge, this state does not give the car the optimal location in this section to maintain a balance between the centripetal force and gravity; with the vector addition of these forces there will be some outstanding lateral force, from the safety conditions, the outstanding force must be equal to zero, i.e. the centrifugal force component must be fully compensated by the gravity component. This condition cannot be met, since the conditions are violated for normal traffic in this section, for example, the elevation was 78 mm instead of 90, a difference of 12 mm leads to serious consequences, especially when the track is widened. In this case, an outstanding acceleration equal to 0.5 m/s2, exceeding the norm of 0.3 m/s2, and arising from a lack of elevation of the outer rail above the inner one, occurred;
- to reduce the influence of centrifugal force, the elevation of the outer rail can be up to 0.15 m. To calculate the magnitude of the elevation, you need to know the speed, radius, and broadening of the track. The angle of inclination of the car body depends on the elevation of the outer rail; When calculating the deraiment on the Kamarchaga-Tayozhny section, the slope angle turned out to be much higher than the norm.

4 Conclusion

The occurrence of oscillations during the passage of unevenness is observed in the case of a decrease in the friction force in the spring sets. The damping deterioration caused by wear in the spring sets, when the rolling stock moves at a speed of 60 to 80 km/h (speed during the investigation of derailments), regardless of the form of subsidence, leads to additional body roll, resulting in large lateral forces that can be determined by ΔH=1 .5 h. It should be noted that a decrease in friction in the bogie spring sets to 50% or less leads to a significant increase in body vibrations, which causes an increase in cd by about 1.5 times. This increase leads to additional vertical and lateral forces, resulting in the overturning of the car body.

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