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DETERMINING THE STRENGTH OF THE CEMENT-CARRYING WAGON-HOPPER AGAINST OVERTURNING

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A B S T R A C T	K E Y W O R D S
The purpose of the study is to check the stability calculation of the cement-carrying wagon-hopper against overturning. Research work was carried out based on "Standards for calculation and design of new and modernized wagons, 1520 mm gauge (non-self-propelled) norms of the Ministry of Railways". According to the requirement, the calculations were performed based on two calculation cases: because of rolling off the curve and based on rolling on the curve. As a result, in the case of the first calculation, the car has a reserve of stability in case of overturning out of the curve. In the case of the second calculation, it was found that the wagon has a reserve of stability when it rolls into a curve. According to the calculation results, the 19-9596 model hopper wagon for cement transportation meets regulatory requirements for rollover resistance.	compression, wagon-hopper, stability, strength against overturning, standards.

Introduction

Since the first years of independence of our country, effective work has been carried out in the direction of development, modernization and expansion of production capacity of the wagon industry. Large industrial enterprises within the railways of Uzbekistan mastered the construction of freight and passenger wagons, modernized production, began repairing wagons, and began producing import-substituting products and components.

Today, industrial enterprises such as Tashkent Passenger Car Construction and Repair Plant, Quyuv Mechanics Plant and Andijan Mechanics Plant, which are rapidly developing, based on high technologies, of the railway network are rightly the pride of Uzbekistan Railways JSC. It is especially noteworthy that they are equipped with modern equipment and new generation technological networks of European countries, Japan, South Korea and other leading foreign companies.

Of course, modernization, introduction of advanced technologies and their effective use in wagon-making enterprises is the basis of the development of the industry. "Uzbekistan railways" JSC pays special attention to the development of factories, including the financial resources directed to technical and technological re-equipment in production are bearing fruit.

II. Literature review

The preliminary evaluation of the stability of rotation of the 19-9596 cement carrier wagon-hopper with a body volume of 61.6 m3 and an axle load of up to 23.5 ts was carried out in the following calculations as part of the sketch project.

Calculation works are carried out in accordance with "Standards for calculation and design of new and modernized wagons, 1520 mm track (non-self-propelled) norms of the Ministry of Railways".

Calculations according to were performed for two calculation cases:

- overturning outside the curve;
- turning into a curve.

When evaluating the stability against overturning of the curve, the movement at the maximum speed (for a given radius of the curve and the height of the outer rail) is taken into account. It takes into account the centrifugal and wind loads directed outside the curve and the transverse component of the longitudinal quasi-static compressive forces acting on the car through the couplings.

III. Analysis

When evaluating the stability against inward rollover of the curve, the movement of the train in traction mode is taken into account at minimum speed (if centrifugal forces are practically absent).

Table 1 Preliminary data for calculating the hopper wagon model 19-9596 for cement transportation

Naming	Designation	Value
Wagon weight (gross), t	G_b	93,5
Number of bullets	n	4
Tara, t	T	21
Cart weight, t	G_t	4,85
The length of the wagon along the connecting axes of the vehicles, m	$2L_s$	12,02
Height of the center of gravity of the loaded wagon from the level of the rails, m	H_{sk}	2,3
The height of the center of gravity of the empty wagon above the level of the rails, m	H_{sk}	1,22
The height of the center of gravity of the stroller from the level of the rails, m	H_{st}	0,475

Stability assessment is carried out using the safety factor of stability, determined by the following formula

$$K_{us} = P_{st}/P_{din} \geq [K_{us}], \tag{1}$$

where P_{st} is the static vertical force of the wheel pressure on the rail, taking into account the unloading due to the vertical components of the longitudinal forces acting on the wagon through the couplings; P_{din} is the dynamic vertical force of the wheel pressure on the rail, resulting from the effect of

transverse forces, taking into account the displacement of the centers of gravity of the body and trolleys. $[K_{us}]$ is the allowable reserve safety factor of overturning stability.
The forces P_{st} and P_{din} are determined by the formulas:

$$P_{din} = \frac{F_K \cdot h_{SK} + 2 \cdot F_T \cdot h_{ST} + F_{VK} \cdot h_{VK} + 2 \cdot F_{VT} \cdot h_{VT} + 2 \cdot P_N^n \cdot h_a + G_K \cdot \Delta_K + 2 \cdot G_T \cdot \Delta_T}{2 \cdot n \cdot S}, \quad (2)$$

$$P_{st} = (G_в - 2P_N^v)/2n, \quad (3)$$

where $G_в$ is the gravity of the wagon; P_N^v is the vertical component of the longitudinal force acting on the body through the vehicle connector; n - the number of wagon axles; F_K, F_T are lateral forces acting on the body and carriage equal to the difference between the centrifugal forces and the transverse component of the weight forces resulting from the rise of the outer rail track; F_{VK}, F_{VT} - force of wind pressure on body and stroller; P_N^n is the transverse (horizontal) component of the longitudinal force acting on the wagon through the coupling; $G_к$, G_t - the weight of the body and the carriage;
h_{sk}, h_{st} - the height from the level of the rail heads to the center of gravity of the body and trolley; h_{VK}, h_{VT} - the height from the level of the rail heads to the geometric centers of the side projections of the body and the carriage; h_a - height from the heads of the car coupler to the longitudinal axis of the car coupler; $\Delta_к$, Δ_t – total displacements of the center of gravity of the body and trolley parallel to the plane of the rail heads relative to the central position of the longitudinal axis of the car; $2S$ is the distance between the rolling surfaces of the wheels according to, $2S$ = 1.6 m.
The value $F_к$ and F_t is determined by the following formulas:

$$F_к = m_к \cdot a_{nep}, \quad (4)$$

$$F_t = m_t \cdot a_{nep}, \quad (5)$$

where $m_к$ and m_t are body and carriage masses, respectively; a_{nep} is the lateral acceleration of the wagon on the curved track.
This is obtained within the curve of a_{nep}= 0,7 m/s^2 and for all wagons within the curve of a_{nep}= -0,9 m/s^2 when calculating the stability against overturning of all wagons according to.
The values of F_{VK} and F_{VT} are obtained according to and are determined at 500 Pa when overturning outside the curve from the calculation of the specific wind pressure and 400 Pa when overturning the car from the curve inward, taking into account the shape of the load in the side projection.
The values of P_N^v and P_N^n are determined by the following formulas:

$$P_N^n = N * L_s / R; \quad (6)$$

$$P_N^v = N * _\Delta h / 2a, \quad (7)$$

where N is the longitudinal quasi-static force acting on the body through the car coupling (obtained according to [35]). The power value is taken as follows:
for loaded wagons:
– when compressed N_{sj} = 1.0 MN,
– when pulled N_{rast} = 1.4 MN;
for empty wagons:

– when compressed N_{sj} = -0.5 MN,
– when pulled N_{rast} = 0.7 MN;
R is the calculated radius of the curve according to and is equal to the following: 650 m when turning from the curve to the outside, 300 m when turning from the curve to the inside; Δh is the difference between the level of the axles of the car couplings of the checked and side cars, it is accepted according to and is the same on both sides of the car and is equal to 0.08 m; $2a$ is the length of the rigid boom, obtained according to and formed by two joint autocouplers equal to 2 m in compression and 1.8 m in tension.
In the general case, the complete shift of the center of gravity to Dk is caused by:
Δ_1 - one-way transverse movement from the central position of the trolley frames in relation to the hubs of the wheel pairs;
Δ_2 - as well as spring bars in relation to the frames of the carriage;
Δ_3 - as well as in relation to the spring bars of the body frame (pivot structure);
Δ_4 - transverse displacement of the longitudinal axis of the body (technological) relative to the longitudinal axis passing through the centers of the pivot (turning devices);
Δ_5 - displacement of the center of gravity of the body when the body deviates to the side due to the gaps between the carriages and the body shells.
Δ_6 - also due to one-way bending of the springs under the influence of side forces.
According to, it is accepted that:
$\Delta_1 + \Delta_2 + \Delta_3$ = 12 mm.
The value of Δ_4 according to is taken as 10 mm;
The amounts of Δ_5 and Δ_6 are determined by the following formulas:

$$\Delta_5 = \delta^*(h_{sk} - h_n)/S, \qquad (8)$$
$$\Delta_6 = \Delta f^*(h_{sk} - h_{ress})/b, \qquad (9)$$

where δ is the possible gap between the sliders on one side of the car; $\delta = 0$; for sliders with a solid support; Δf - the additional bending of the springs from the side of the loaded wagon and the rise of such springs is due to the side forces from the unloading side; h_n, h_{ress} - height from the level of the rail heads to the plane of the undercarriage and the upper spring sets, respectively; $2S$, $2b$ - transverse distances between the longitudinal axes of the sliders and the centers of the spring suspensions.

IV. Discussion

In the calculations, the forces acting on the body, the reactive moment of the spring suspension, the overturning moment are checked and determined according to the following formula:

$$M_{reak} = \frac{\Delta f \cdot æ_{ugl}}{b}, \qquad (10)$$

where $æ_{ugl} = 2b$ is the angular stiffness of the wagon's spring sets; $æ$ – is the vertical strength of spring sets on one side of the wagon.
The calculation of the strength against overturning under tension in a curve (loaded wagon) is shown in Table 2.

Table 2 **Calculation of strength against overturning of a curved (loaded wagon) under tension**

Naming	**Designation**	**Value**
Static weight, t	P_{st}	10,132
Vertical component of longitudinal force, t	P_N^v	6,222
Lateral force affecting the body, t	$F_к$	7,688
Lateral forces affecting the wheelchair, t	F_t	0,445
Wind pressure affecting the hull, t	F_{vk}	0,776
Wind pressure acting on the stroller, t	F_{vt}	0,088
Transverse component of longitudinal force, t	P_N^n	2,805
The displacement of the center of gravity of the body due to the gap between the scolzones	Δ_5	0,023
Additional bending of springs by loading, m	Δf	0,008
Shift of the center of gravity of the body as a result of bending of spring suspensions, m	Δ_6	0,010
The sum of the displacement of the center of gravity of the body, m	$\Delta_к$	0,055
Dynamic power, t	P_{din}	4,267
Stability reserve coefficient	K_{us}	2,375

Calculation of stability when rolling over inside a curve (empty car) is presented in Table 3.

Table 3 **Calculation of stability against overturning in a curve (empty vehicle) under tension**

Naming	**Designation**	**Value**
Static loading, t	P_{st}	1,847
Vertical organizer of longitudinal forces, t	P_N^v	3,111
Lateral forces affecting the body, t	$F_к$	1,037
Lateral force acting on the wheelchair, t	F_t	0,445
The force of wind pressure affecting the hull, t	$F_{vк}$	0,776
The force of wind pressure acting on the stroller, t	F_{vt}	0,088
Transverse component of longitudinal force	P_N^n	1,402
The displacement of the center of gravity of the body due to the space between the skolzuns, m	Δ_5	0,009
Additional bending of the springs on the loaded side, m	Δf	0,001
As a result of bending of the spring assembly, the center of gravity of the body shifts, m	Δ_6	0,001
The sum of the center of gravity of the body, m	$\Delta_к$	0,032
Dynamic power, t	P_{din}	1,062
Stability reserve coefficient	K_{us}	1,740

Calculation of the stability against overturning of the curve (loaded wagon) under compression is shown in Table 4.

Table 4 **Calculation of stability against an external curve (loaded wagon) under compression**

Naming	Designation	Value
Static loading, t	P_{st}	10,688
Vertical organizer of longitudinal forces, t	P_N^v	4,000
Lateral force acting on the body, t	$F_к$	5,980
Lateral force acting on the body, t	F_t	0,346
The force of wind pressure affecting the hull, t	$F_{vк}$	0,970
The force of wind pressure acting on the stroller, t	F_{vt}	0,110
Transverse component of longitudinal force	P_N^n	0,925
Shift of the center of gravity of the body due to the gap between the skolzun, m	Δ_5	0,023
Additional bending of the springs on the loaded side, m	Δf	0,006
As a result of bending of the spring assembly, the center of gravity of the body shifts, m	Δ_6	0,008
The sum of the center of gravity of the body, m	$\Delta_к$	0,043
Dynamic power, t	P_{din}	3,032
Stability reserve coefficient	K_{us}	3,525

Calculation of stability against overturning of the curve (empty wagon) under compression is shown in Table 5.

Table 5 **Calculation of stability against an overturned external curve (empty vehicle) under compression**

Naming	Designation	Value
Static voltage, t	P_{st}	2,500
Vertical organizer of longitudinal forces, t	P_N^v	2,000
Lateral force acting on the body, t	$F_к$	0,806
Lateral force acting on the wheelchair, t	F_t	0,346
The force of wind pressure affecting the hull, t	$F_{vк}$	0,970
The force of wind pressure acting on the stroller, t	F_{vt}	0,110
Transverse component of longitudinal force	P_N^n	0,462
Displacement of the center of gravity of the body due to the gap between the skolzun, m	Δ_5	0,009
Additional bending of the springs on the loaded side, m	Δf	0,001
As a result of bending of the spring assembly, the center of gravity of the body shifts, m	Δ_6	0,000
The sum of the center of gravity of the body, m	$\Delta_к$	0,031
Dynamic power, t	P_{din}	0,748
Stability reserve coefficient	K_{us}	3,340

The minimum permissible value of the factor of safety for the first calculation case (overturning inside the curve under tension) $[K_{us}] = 1.2$. The actual value of the coefficient of stability reserve inside the curve:

$K_{us} = 2.375$ (for a loaded wagon);

$K_{us} = 1{,}740$ (for an empty wagon);

As a result, in the case of the first calculation, the wagon has a reserve of stability.

V. Conclusion

The minimum permissible value of the factor of safety for the second calculation case (overturning outside the curve under compression) $[K_{us}] = 1.3$. The actual value of the coefficient of stability reserve inside the curve:

$K_{us} = 3{,}525$ (for loaded wagon); $K_{us} = 3{,}340$ (for an empty wagon);

Thus, the wagon has a reserve of stability in the second calculation position.

According to the calculation results, the 19-9596 model hopper wagon for cement transportation meets the normative requirements for rollover resistance.

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