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THE INFLUENCE OF ELASTIC SYSTEMS ON THE TRAVEL SAFETY OF RAILWAY VEHICLES

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Abstract
The paper presents experimental studies on the determination of the torsional rigidity of the bearing structures (carbody, bogies). The methods and the experimental technology of determining the torsional rigidity are revealed for the bearing structures of the chassis and bogies of the freight cars, as determining elements of travel safety.
Finally, a defining computational methodology is presented for establishing the derailment conditions (Nadal's formula) with the consequent conclusions.
Keywords:
torsional rigidity, force as a function of displacement variation, Y/Q ratio

1. INTRODUCTION

Travel safety is areas that belongs to the dynamics of railway vehicles and constitutes, together with bearing structure resistance and travel dynamics, preoccupations and lines of research that define the capacity of a vehicle to travel on the railway.

The possibility and chance of a vehicle to derail is in strict connection with the torsional capacity of the vehicle as a whole, wheel load, railway geometry and irregularities [3].

2. VERIFICATION OF WHEEL LOAD REPARTITION

The purpose of the test is determining the repartition and deviations of wheel loads. The trials were conducted in accordance with the recommendations of the testing program and the methodology of ERRI B55 Rp8, [8], [9], [10], [11].

The car was tested by positioning it successively with the two bogies on the stand with the tensometric transductors, rail type.

The car was tested for torsion by superimposing the effects of torsion on the basis of the car axle base and the bogie axle base. One of the points was put in a lifting-lowering motion, during which time at that point the displacement Δh and the sustainment force ΔF were measured. An analog process was used for the bogie frames.

With the values of the wheel loads determined during the torsion testing, hysteresis diagrams of the unloading were drawn. From these diagrams the decreases of the wheel loads ΔQ_{ij} were determined, which were compared to the admissible limits $\lim \Delta Q_i$, according to the norms of ERRI B55 Rp8.

With the measured values of ΔF and Δh, hysteresis diagrams were drawn. The values from the diagrams and the specific technical and constructive characteristics of the car were used in the computation of safety against derailment.

The torsion to which the car is exposed during testing simulates the crossing of the car over the irregularities of the rail [2], [3], [4], [5], [6], [7].

The torsion corresponding to the car (g^*), and the bogie (g^+) is computed using (1):

$$g^* = \frac{15}{2a^*} + 2\,,\ [‰]; \tag{1}$$

$$g^+ = 7 - \frac{5}{2a^+}\,,[‰] \tag{2}$$

where: $2a^*$ is the car axle base;

$2a^+$ is the bogie axle base.

The values of the maximum admissible unloadings $\lim\Delta Q_j$ are computed for each axle, according to the following equations:

$$\lim\Delta q_j = \frac{\lim\left(\frac{Y}{Q}\right)_a - \frac{Y_{aoj}}{Q_{oj}}}{\lim\left(\frac{Y}{Q}\right)_a}, \quad \lim\Delta Q_j = \lim\Delta q \cdot Q_{oj} \tag{3}$$

where:
$\lim\Delta Q_j$ – the maximum admissible unloading of the wheel in order to ensure against derailment;
$\lim\Delta q_j$ – relative limit value of the unloading;
Q_{oj} – average wheel load at axle j;
$\lim(Y/Q)_a = 1{,}2$;
Y_{aoj} – transverse guidance effort at axle j on the exterior wheel of the curvature.

The condition that the car must satisfy in order to traverse the rail torsions without any risk of derailment is that the wheel unloadings observed during testing do not exceed the admissible limit values, computed with equation (2).

The unloadings ΔQ_j of the wheels are determined by measuring with the above mentioned installation. The measurement is done progressively, starting from the situation of the car resting on a straight railway, the carbody is torsioned, then the bogies, at the values g^* and g^+ computed with equations (1) imposed by ERRI B55 Rp8.

Finally, the following inequality is checked:

$$\Delta Q_{jk} < \lim\Delta Q_j. \tag{3}$$

In case the inequality is respected, the safety against derailment of the car is certified from the point of view of the torsional rigidity of the tested car.

3. DETERMINING THE TORSIONAL RIGIDITY

The torsional rigidity C_t^* of a carbody with own weight and the characteristics of the elastic elements of the suspension, are criteria of appreciating the travel safety of the car [1]. [12].

The torsional rigidity C_t^* is a characteristic of the carbody related to the axle base, expressed by a torsional moment $\left(\Delta F \times 2b_z^*\right)$ applied to the carbody at an angular displacement $\left(\frac{\varphi}{2a^*}\right)$ resulting in:

$$C_t^* = \frac{2a^* \cdot \Delta F \cdot 2b_z^*}{\varphi} \quad \left[\frac{KN \cdot mm^2}{rad}\right] \tag{4}$$

where: $2b_z^*$ - distance between the suspension supports on the axle [mm];
$2a^*$ - car axle base [mm];
φ – angular displacement [rad];
ΔF – variation of the vertical force [KN].

The angular displacement φ [rad] can be expressed as:

$$\varphi = \frac{h}{2b_z^*} \tag{5}$$

And the expression of the torsional rigidity:

$$C_t^* = 2a^* \cdot \left(2b_z^*\right)^2 \cdot \frac{\Delta F}{h} \quad \left[\frac{KN \cdot mm^2}{rad}\right] \tag{6}$$

It is sufficient for the $\frac{\Delta F}{h}$ ratio to be determined experimentally in order to determine the torsional rigidity C_t^*.

The experimental measurements in order to determine the $\frac{\Delta F}{h}$ or $\frac{\Delta F'}{h'}$ ratios can be conducted both for the carbody and the bogie frame. For the carbody, the measurements can be conducted both in the presence of the bogies or in their absence, and in the case of cars on two axles, both in the presence of the axles and in their absence.

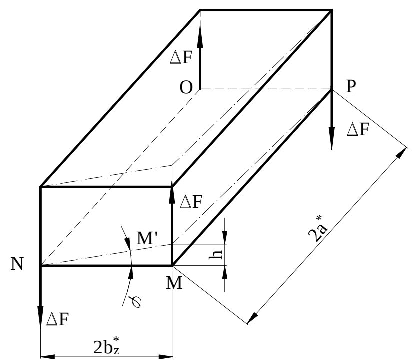
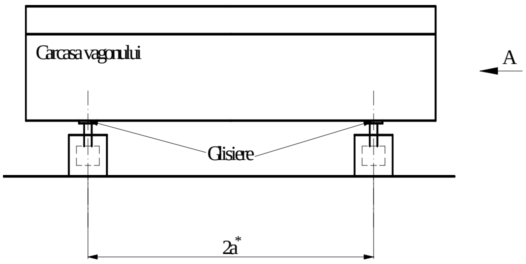


Figure 1. Geometrical characteristics of the car

Figure 2 shows the arrangements necessary in order to conduct the tests to experimentally determine the $\frac{\Delta F'}{h'}$ in the case of the vehicle without the bogies.



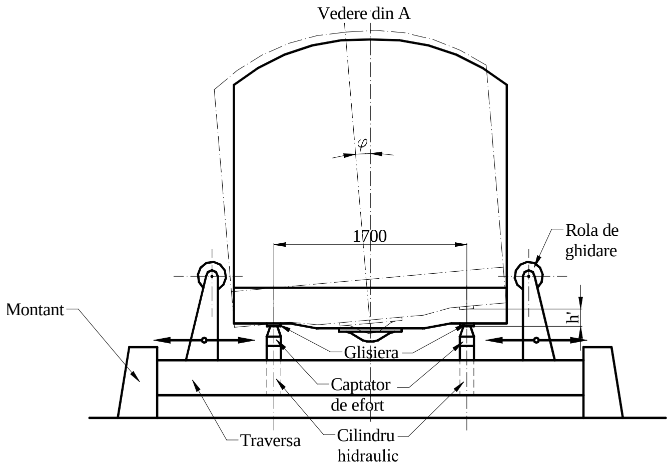
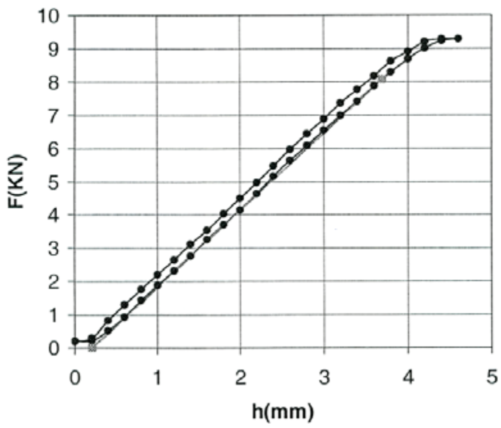


Figure 2. The necessary arrangements for the experimental determination of the $\frac{\Delta F'}{h'}$ ratio for the vehicle without the bogies

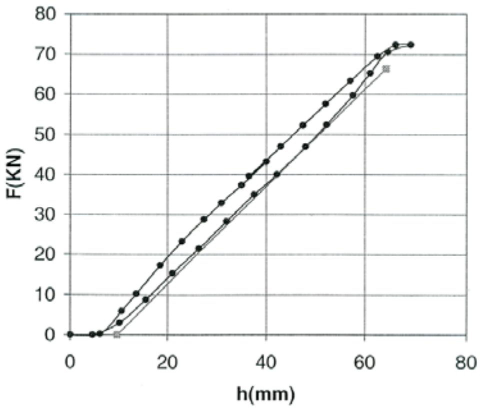
4. EXPERIMENTAL DETERMINATIONS OF THE WHEEL LOAD AND TORSIONAL RIGIDITY

Further on, experimental determinations of the wheel load and the torsional rigidity will be presented for the cistern car with 40m^3 capacity [1], [2].



$C_t^+ = 1{,}68138 \cdot 10^{10}$ KN·mm^2/rad

Figure 3.Force-displacement diagram at the torsion of the bogie frame



$C_t^* = 5{,}440012 \cdot 10^{10}$ KN·mm^2/rad

Figure 4.Force-displacement diagram at the torsion of the chassis

Car technical data

- weight T=24040 kg
- Average wheel load Q_o=29,479 kN
- Load corresponding to the two stage primary suspension inflexion point (ERRI B 12 Rp 49) F_{cz}=26,2 kN
- Mounted axle weight (ERRI B 12 Rp 49) Gr =12,9 kN
- Car axle base $2a^*$= 9360 mm
- Bogie axle base $2a^+$ = 1800 mm
- Wheel base 2e=1435 mm
- Distance between rolling circles $2b_A$ =1500 mm
- Distance between suspension springs $2b_Z$ = 2000 mm
- Distance between gliders $2b_G$= 1700 mm
- Rigidity – measured value C_t^*= 5,440012·10^{10} KN·mm^2/rad
- Bogie frame rigidity – measured value C_t^+ = 1,68138·10^{10} KN·mm^2/rad
- Primary suspension rigidity (ERRI B 12 Rp 49) $C_{z\,1(2)}^+$ =1,004 KN/‰
- Glider springs rigidity ERRI B 12 Rp 49) c_G = 0,57 KN/‰

- Maximum relative deviation of wheel load $\Delta q_0 = 0{,}2$
- Wheel radius r =460 mm
- Radius of the railway used in computation R = 150 m
- Gravitational acceleration $g = 9{,}81\ m/s^2$
- Derailment safety criterion $\lim(Y/Q)_a = 1{,}2$

Computation of the $(Y/Q)_a$ ratio is done according to ERRI B 12 Rp 49

- Torsioning at the car test $g^* = 15 / 2a^* + 2 = 3{,}603$ [‰]
- Torsioning at the bogie test $g^+ = 7\text{-}5 / 2a^+ = 4{,}222$ [‰]
- Exterior leading force $Y_a = 0{,}5319 \cdot Q_0 + 1{,}9062 = Y_a = 17{,}586$ [kN]
- Interior leading force $Y_i = -0{,}4923 \cdot Q_0 - 0{,}1512 = -14{,}664$ [kN]
- Transverse force in the axle box $H_y = -(Y_a + Y_i) = -2{,}922$ [kN]
- Absolute wheel load decrease due to the force

  H_y $\Delta QH_y = H_y \cdot r / 2b_A = -0{,}896$ [kN]
- Absolute maximum wheel load deviation $\Delta Q_{Fz0} = \Delta q_0 \cdot Q_0 = 5{,}896$ [kN]
- Absolute total diminuation of the wheel load due to rail twisting on the basis of the bogie axle base

$$\frac{1}{C_{tA(2a^*)}} = \frac{10^3 \cdot (2b_A)^2}{C_t^+} + \frac{10^3 \cdot b_A^2 \cdot 4}{2a^+ \cdot b_z^{+2} \cdot c_{z1(2)}^+} \qquad \Delta Q_t^+ = g^+ \cdot C_t^+ = 3{,}062 \text{ [kN]}$$

- Absolute total diminuation of the wheel load due to rail twisting on the basis of the car axle base

$$\frac{1}{C_{tA(2a^*)}} = \frac{10^3 \cdot (2b_A)^2 \cdot 2}{C_t^*} + \frac{10^3 \cdot b_A^2 \cdot 2 \cdot 4}{2a^* \cdot b_z^{+2} \cdot 2 \cdot c_{z1(2)}^+} + \frac{10^3 \cdot b_A^2 \cdot 2 \cdot 4}{2a^* \cdot b_z^{+2} \cdot c_G}; \ \Delta Q_t^* = g^* \cdot C_t^* = 2{,}419 \text{ [kN]}$$

- $(Y/Q)_a$ ratio $\left(\frac{Y}{Q}\right)_a = \frac{Y_a}{Q_0 - (\Delta Q_t + \max \Delta Q_{fz0} + \Delta Q_{Hy})} = 0{,}926$

Table 1

Tests	Imposed values by the testing program and procedures	Values obtained from the testing
Verifiying wheel load repartitions, with Y25 Ls(s)d1 bogies and elastic gliders with equal 12 mm clearance	$\lim\Delta Q_{fz1} = 6{,}1149$ KN	$Q_{11} = 29{,}9658$ KN, $Q_{12} = 31{,}1832$ KN $\Delta Q_1 = 0{,}6087$ KN, $\Delta Q_{\mu I} = 1{,}6376$ KN $\Delta Q_{fz1} = \Delta Q_1 + \Delta Q_{\mu I}$, $\Delta Q_{fz1} = 2{,}2462$ KN
	$\lim\Delta Q_{fz2} = 5{,}9229$ KN	$Q_{21} = 29{,}2961$ KN, $Q_{22} = 29{,}9332$ KN $\Delta Q_2 = 0{,}3186$ KN, $\Delta Q_{\mu I} = 1{,}6376$ KN $\Delta Q_{fz2} = 1{,}9562$ KN
	$\lim\Delta Q_{fz3} = 5{,}9288$ KN	$Q_{31} = 28{,}0350$ KN, $Q_{32} = 31{,}2525$ KN $\Delta Q_3 = 1{,}6087$ KN, $\Delta Q_{\mu II} = 1{,}2720$ KN $\Delta Q_{fz3} = 2{,}8807$ KN
	$\lim\Delta Q_{fz4} = 5{,}6175$ KN	$Q_{41} = 27{,}0843$ KN, $Q_{42} = 29{,}0906$ KN $\Delta Q_3 = 1{,}0031$ KN, $\Delta Q_{\mu II} = 1{,}2720$ KN $\Delta Q_{fz4} = 2{,}2751$ KN

Table 2

Tests	Values imposed through the testing program	Values obtained from testing
Verifying the torsional rigidity, with Y 25 Ls(s)d1 bogies and elastic gliders with equal 12 mm clearance		$C_t^* = 1{,}68138 \cdot 10^{10}$ KN·mm²/rad $C_t^+ = 5{,}440012 \cdot 10^{10}$ KN·mm²/rad
	$\lim(Y/Q)_a = 1{,}2$	$(Y/Q)_a = 0{,}926$
	$\lim\Delta Q_1 = 15{,}9084$ KN	$\Delta Q_{11} = 4{,}950$ KN $\Delta Q_{12} = 4{,}320$ KN
	$\lim\Delta Q_2 = 15{,}5465$ KN	$\Delta Q_{21} = 6{,}880$ KN $\Delta Q_{22} = 5{,}540$ KN
	$\lim\Delta Q_3 = 15{,}5575$ KN	$\Delta Q_{31} = 7{,}120$ KN $\Delta Q_{32} = 4{,}160$ KN
	$\lim\Delta Q_4 = 14{,}9709$ KN	$\Delta Q_{41} = 6{,}280$ KN $\Delta Q_{42} = 4{,}010$ KN

In order to perform the theoretical calculation regarding safety against derailment (Y/Q), for the tested car, the following were considered:
a. With the empty car the Ist suspension level (I) steps into action;
b. In quasistatic conditions, it is acceptable in calculations, for the travel velocity, transverse acceleration and overheightening the use of the values: v = 0, a = 0, u = 0.

5. CONCLUSIONS

In conclusion, it can be considered that the value of the Y/Q ratio of 0,926 is situated under the UIC admissible limit of 1,2 and thus there is a certitude of the elimination of derailment.

Torsional rigidities C_t^* and C_t^+ of the car and bogie respectively, significantly influence travel safety since they can cause large values of the unloading ΔQ_t^* and ΔQ_t^+ when the bearing structures of the car and the bogie have large torsional rigidities and a low elasticity.

The existence of an adequate elasticity of the car and bogie structure thus leads to an improvement of the vehicle behaviour in regards to the derailment risk.

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