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DYNAMIC STUDY ON SAFETY AGAINST THE DERAILEMENT TO THE SIX AXLE LOCOMOTIVES

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Abstract: *This paper treats the problem of dynamic forces produced flow curves that there is a "discontinuous bend" mathematically represented as an angle where there are so-called" shock angle". It is a situation that can occur in curves with seamless lines, the vehicle being applied to a dynamic force shock that may affect traffic safety. Example of calculation was done in the case of a tank wagon which the center of mass due to high, maybe even topple under the effect of shock.*

Key words: *frictional forces, derailment, landslides, longitudinal slides, guiding, attack angle*

1. Introduction

Safety against derailment of a railway vehicle is determined by the capacity of the axle driving guidance, which is the maximum force attack guidance wheel limit the derailment. The capacity of the axle driving guidance of the equilibrium conditions results in the vertical plane - the transverse forces acting on the axle. The spatial orientation of the normal force N_i is determined solely by geometric conditions. Because of the small angles, normal operating, the longitudinal component of the normal force can be neglected, the normal force is considered to act in the vertical plane - cross (*YZ*), where transfers occurr mostly load (Figure 1). The spatial orientation of the friction force T_i is determined first of all on the geometric because it is contained in the plane of the tangent contact, but because according to the general laws of friction, has the same direction as the sliding speed and oriented in the opposite direction thereof, is driven by slip occurs the contact points on the kinematic that. The size who defines the spatial orientation in the tangent plane of contact the friction force T_i, due to the kinematic conditions is a slip angle ξ_i, which the contact points on the tread is approximately equal to the tangent of the ratio of longitudinal and cross slides.

Usually the cross slips are determined by the angle of attack α and as a result, the transverse components of the forces of friction will have the same effect on both wheels. The longitudinal sidings', if the axle is free to run, are determined by differences in actual driving circles radii,

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which depend on the gap y_c, being generally opposite on two wheels. The value of the term $\cos\xi_i$ in this case, regardless of the angle of attack α, is approximately uniform. In the drive or braking, increasing the velocity of longitudinal slip makes value $\cos\xi_i$ to decrease well below the unit, which is influenced by decreasing the angle of attack α. The size the friction force $T_i = \tau_i .N_i$ is dependent on the coefficient of friction τ_i, which has a non-linear sliding speed, i.e. the pseudo-sliding. If driving axles, which also guide other axles in the same chassis, wheel attack no. 1 can run the rail bicontact contested situation that is proper bevel profiles new condition. In this case, the next point of contact A_1, the second point appears $A_1^{'}$, which become points of guidance. In the contact wheel - rail, namely its central act of the rail normal force bearing N_i perpendicularly to the tangent plane of contact and contained in the plane normal to the two - wire path and friction T_i who is perpendicularly to the normal force and thus contained in the plane tangential contact. Spatial orientation of the contact forces as well as the slip velocity depends on the position of the axle in the way, which is characterized by the angle α and the gap offset delay y_c to the middle position. The contact tangent inclination to the horizontal is given by the angle δ_i which is the angle between the lines of intersection of the vertical plane path with tangent plane wires the contact with the horizontal plane passing through the point of contact.

In the case of wheel with two-point, the contact angle flank attack $\gamma_1^{'}$ is small, then the driving force can be considered $P=N_1\sin\gamma_1$. On the unassailable wheel axle, all thanks to lower sidewall angle at the point of contact A_2, the driving force is negligible. The strength force size of the guidance Y_i is arising (results) on each wheel, the vertical summation of the horizontal component of the force normal to the horizontal component - cross friction. The maximum value of Y_i occurs on the attack wheel of a driving axle, where intervenes the driving force P.

2. The limit of the derailment

In the case of bi-attack wheel rail contact, employment growth guidance Y_1 makes guidance point to increase A_1 the reaction N_1 and, therefore, increase the action of the unload component T_{1yz} of the friction force, so the reaction decreases $N_1^{'}$ the fulcrum of anchorage $A_1^{'}$. The situation is reached $N_1^{'}= 0$, therefore when the fulcrum anchorage of $A_1^{'}$ is completely discharged and the mass load Q_1 attackers on the wheel goes full guide wheel lip to the point A_1, derailment limit is reached.

Whether that situation to force Y_1 further increase lip driving wheel will climb the inner side of the rail, causing derailment. In the case of mono-switched contact, derailment limit is reached when the single point of contact A_1 has reached the edge of the wing at an angle of maximum.

Following the derailment process analysis, we noted that *Nadal's* formula was deduced only from the wheel contact forces attackers, without taking into account the dependence that exists between the load on a wheel and guiding force nor influence effect spin the contact point on the rim on the coefficient of friction. As a result of experiments conducted in the Committees ORE B 55 and B 136, recommended the adoption of the value of the friction coefficient μ = 0,36, considered when applying the calculations covering safety against derailment *Nadal's* formula, showing the influence of positive growth flank angle of the lip rim γ_1 the ability guidance.

However, it emerged from the calculations and experiments, the advantage flank angle 70° outer surface of guidance wheel lip to enhance the capacity of the axle guidance. Works Committee ORE B 55 showed that to avoid derailing current vehicle line report Y_1/Q_1 must be below the limit $(Y_1/Q_1)_{\lim}$ = 1,2. At the circulation over switches, experiences made in the Committees C 9 and C 70 are allowed to conclude that the switches $(Y/Q)_{\lim} = 0{,}8$ and in contact with the frog tip lip of the crossings to be compared to provide a $(Y/Q)_{\lim} = 0{,}4$.

3. The safety the derailment under the influence of external forces

The report $(Y_1/Q_1)_{\lim}$ can not be a proper criterion for assessing the safety against derailment but only if the load on the wheel attackers Q_1 is the actual vertical component of the reaction at the limit of derailment of the rail, in view of the fact that the guiding force is dependent on the Y_1. The flank angle γ_2 depends on the shape of the profile of the wheel and the rail and the track path (conical profiles tg γ_2 = 0,05). In generally, the term tg ($\gamma_2 + \delta_2$) is specific to each vehicle and those bodies running, influenced by the angle of attack α. At the limit of the derailment, according to *Nadal's* formula, Y_1 must satisfy the condition $Y_1 = Q_1.\mathrm{tg}(\gamma_1 - \delta_1)$ where tg (γ_1 - δ_1) has well-defined limit values depending on the angle of the flank rim lip.

4. The influence of shock attack on security derailment

In track curves may deviate from nominal dimensions occur in the form of continuous or discontinuous bends, the dynamic forces produced by interaction between the vehicle and the undercarriage in a transverse direction, which damaged the ride quality, and may jeopardize the safety of vehicle guidance. Continuous the path bends are characterized by deviations of curvature continuous variable, overlapping any distortion pathway, leading to variation so cant deficiency and transverse acceleration of the vehicle. In Romania, the continuous bends are limited by the arrows tolerances measured.

Whether the vehicle is traveling at constant speed in a curve without misconduct cant deficiency I, the box his mass mc will be subjected to quasi-static cross transversal accelerations γ_{T0} , the centrifugal force respectively uncompensated F_n. Consequently as a result, each axle will act quasi-static driving force of the chassis H. Simultaneously with the appearance the strength force H the stretch resilient elastic compression occurs and track superstructure elements and the vehicle. Taking the view that their total stiffness is c_y , their static deformation will be $y_c = H / c_y$. Therefore, the vehicle could be considered as a simple harmonic oscillator, i.e. a mass - spring, in which y_c is the static deflection of the spring. When the vehicle reaches the outer wheel of the first axle top of a discontinuous bend, the track will be attacked with a speed of attack $v.\sin\delta \approx v\delta$, having a direction perpendicular to the rail attackers. It produces a dynamic force $H_d = c_y\ y_d$ driving force chassis called shock or force attack, in which yd represents the dynamic deformation of the arc spring stiffness cumulative c_y. The shock does not take part in the whole mass of the vehicle, only a portion of this mass that "low", denoted by m_r and the maximum dynamic the force term expression H_{dmax} can be deduced by applying the theorem of the energy conservation.

The attack speed component $v.\sin\ \delta$ is perpendicular to the rail track and give mass m_r in this direction with a kinetic energy value $(1/2).m_r.(\mathrm{v}.\sin\delta)^2$ which is taken elastic spring stiffness cy between the mass ground and the track rails. Once the maximum compression of the arch spring is y_{dmax}, the kinetic energy becomes zero, turning by converting itself entirely within of potential energy. When will associated of a discontinuously the elbow angled bend side, another side that is continuous, then because of the variation ΔI [mm] of cant deficiency through at the time of the attack, the vehicle will have an additional acceleration $\Delta\gamma_{T0}$ and will conduct mechanical work (mechanical force couple) by the supplementary distance $y_{dmax.}$ The maximum force of impact occurs when the term $sin.(\omega t - \varphi) = 1$, therefore the outer wire path after time reckoned from the moment of reaching the wire tip and inner elbow after the time period namely $t_i = 3t_e$, where t is defined by the relation number two (2). The frictional effects damping system produces vibratory phenomenon until its complete disappearance, whether big forces driving this have caused the vehicle meanwhile derailment. It follows that the maximum forces transmitted path, asking her to displacements track due to vehicle lateral loads are $H_{\max} = H + H_{d\max}$ the outer thread of the tread, and that $H_{\max} = H_{d\max} - H$ the inner thread. Vehicle derailment by overcoming report $(H/Q_0)_{lim}$, takes place usually on the inside of the wire path, which is discharged only at the outside. Since the coefficient of adhesion can be in the range (0,2...0,8) depending on the quality of wheel - rail contact will have values in Table 1.

5. Search results and experimental measurements

The existence of a continuous bend over base curve measurement his highlighted by an arrow f_1, which is lead results to a radius of curvature $R_1=C^2/(8f_1)$. Such an angled bend elbow in the path, the continuous variation of the radius of curvature of the R to R_1, and keeping the cant superelevation h, result into an cant deficiency variation ΔI, where $\Delta I = 11{,}8.V^2.[(1/R_1) - (1/R)]$ [mm] and to an a supplementary transverse acceleration $\Delta\ \gamma_{to} = \Delta I\ /153$ [m/s^2]. In this paper we presented the example calculation made for a two-axle bogie vehicle without central suspension, running speed V [km/h] in a curve bend of radius R [m] with the cant superelevation h [mm]. That the curve, for the rope chord length C [m], it is corresponds with an arrow $f = C^2/(8R)$ [m]. We have also felt deemed thought that the curve radius R_1 an angled bend elbow occurs discontinuously, which was revealed by measuring a difference in arrow $f_d - f_1$, the angle δ are given by the formula number (1) and computational study we considered that the pivot (bolster) is located in the center of mass of the bogie, subsequently causing this low value of the mass of the box m_{rc}.

The quasi-static force H which is acts on the axle, will be given by the formula number (3), where m_0 is represents the adequate unsprung (unsuspended) mass of the bogie axle, $2Q_0$ is the axle load and I is the cant deficiency on the curve radius R. The safety of the vehicle derailment shall be checked for the two wires of the path imposing the condition $H_{max}\ /Q_0\ \leq\ (H\ /\ Q_0)_{lim}$, after previously determined the mass load transfer ΔQ_0. For the freight wagon four-axle tank Z series for the oil tanker, we can use the next formula

$\sum Y_{\max} \leq 0{,}85.[10 + (200/3) = 65{,}17$ [kN],

where the axle load was thought considered $2Q_0$=200 [kN], the gravity acceleration of approximately 10 [m/s2]. The reduced mass m_r of the entire vehicle has been determined by the relation (1), where the m_b terms are represents the sprung mass of the bogie and i_{bx} and i_{bz} are the rays of inertia of the bogie. Knowing the lateral stiffness of the suspension axle c_y, with the formula number (2) shall be determined the maximum dynamic force H_{dmax}. The limitations for the protecting of the running gear of the vehicle are for the force who is acting by the axle axis (bogie wheelset) H, according to the relationship number three (3). As well (likewise), $H_{\max}$ = 80 [kN] and H_{med} = 50 [kN] and axial force H of the tank wagon who is analyzed $H = 9{,}33$ [kN], where, with the E term, the cant excess has been noted that the path was considered in accordance with [1], E=60 [mm]. For to make the verification by the lateral movement (sway - cross) of railway track path, must be taken into account that this is required so the force H_{max}, who is given by the formula number (4), and the inertial force of the axle and the axle expression of inertia the force, which is given as the equation (5). Also, the car looked, we have considered the fact that the guiding force $Y = \sum Y_{\max} = 65.17$ [kN] is attain achieve the maximum possible value $Y/Q_0 = 0{,}65$. It is also observed that $Y/Q_0 \lhd (Y/Q_0)_{\lim} = 1{,}2$ that i.e., the vehicle is traveling safely in the current line and switches (turnouts point rods). At the crossing over the crossbreeds junctions of the inequality $Y/Q_0 \rhd 0{,}4$ ensure safety against derailment is concluded because in reality guiding force is less than $\sum Y_{\max}$. The latter amount is charged and discharged attackers wheel on the inside of the track thread. On the other hand there is the mass load transfer $\Delta Q_H = \lambda.(H.r/2.e) = 3.08\, kN$ and thus resulting axle loads Q_1 and Q_2 are given by the relations (6) and (7). Considering that taking into account that there is a transfer ΔQ_0 uncompensated centrifugal force and $H \cdot h_c = \Delta Q_0 \cdot 2e$ where with h_c has been noted the height of the center of mass C of the wagon about the axis of the axle (Figure 3). For $h_c \approx$ 2050 [mm], $\Delta Q_0 = H.(h_c/2.e) = 16.13\, kN$. The values obtained resulting forces that guidance wheels, wheel attackers Y_1 and Y_2 the thread wheel on the inside of the track, whose values are found in expressions equations (8) and (9).

When traveling with excess E the cant force H shown in Figure 3 will be directed towards the center of the curve thus causing a discharge to the outside the thread wheel on the wheel load and on the inner the thread of the tread. Therefore no danger analyzed wagon derailment except crossing peaks at crossings hearts when we are around the limit allowed. Neither in that situation we consider that there are no problems because the calculation is completely covering the value of failure I the cant adopted. Thus (therefore) for H = 9,33 kN, we have ΔQ_0 = 12,75 [kN] respectively ΔQ_H = 2,43 [kN] and thereby $Q_1 = Q_0 - \Delta Q_0 - \Delta Q_H = 100 - 12.75 - 2.43 = 84.82$ [kN] respectively $Q_2 = Q_0 + \Delta Q_0 + \Delta Q_H$ = 115,18 [kN]. In this case, the guiding forces in this situation would be: $Y_1 = Q_1 tg(\gamma_1 + \delta_1)$ = 38,17 [kN] and $Y_2 = H + Q_1 tg(\gamma_1 + \delta_1)$ = 47.5 [kN]. At the wheel of the inner the thread is consuming game between the wheels rim and railway track $Y_2/Q_2 = 0.412 < 1.2$ and so we have provided in this case safety guidance.

6. Conclusions and Summary

At the locomotives because of the traction force transmission to the chassis, the practice large longitudinal stiffness of the suspension axles and therefore can be considered as axles fixed bogie frame. The formulas established to assess the safety against derailment can be applied to any running speed, provided that transfers the load to be determined properly taking into account the dynamic actions of the vehicle in the most adverse situations. When rotating the bogie (at the turning of bogies) in curves with radii opposing forces generate appreciable amounts of the force H, what compels a reduction transfers the load to avoid derailments. The flank angle of the wheel rim. The capacity guidance the axle load decreases with decreasing wheel attackers, so as the wheel load transfer from the unassailable attack is higher. The safety against derailment is influenced by load transfer as well as wheel radius and maximum. The limit situation (the deadline) for wheel unloading attack can occur at low speed going through curves with maximum cant and maximum twisting path.

The negative charge transfers from the inside of the curve tilting of the box to the vehicle are increased by the flexibility and torsion coefficient of the path, which are taken up primarily by the wheels of the vehicle suspension and the small diameter greater danger of derailment. The increase maximum The flank angle of the lip is favorable safety against derailment, it leads to increase both capacity minimum guidance and maximum axle.

The verifications to avoid complete discharge of the wheels are not challenged for the movement curve at its maximum authorized speed The driving (leading) forces of the chassis has a practical significance because determining safety path for displacements of rail due to vehicle loads , the request of the vehicle running and derailment safety. The maximum permissible mass transfers in this situation must not exceed $\Delta Q_0/Q_0 \leq 0{,}6$ for wheels rim flank angle of 70°, provided that the overhung *H* to be very close to zero. This is usually achieved only on vehicles with steerable axles. The situation is worst at quasi-static movement of the vehicle in low speed (up to 40 [km/h]) curve with a radius of 150 [m] and maximum permissible track any distortion.

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The values of $(Y/Q)_{\lim}$ *versus coefficient of adhesion* μ Table 1

μ	0,2	0,36	0,4	0,6	0,8
$(Y/Q)_{\lim}$	1,65	1,2	1,12	0,81	0,55

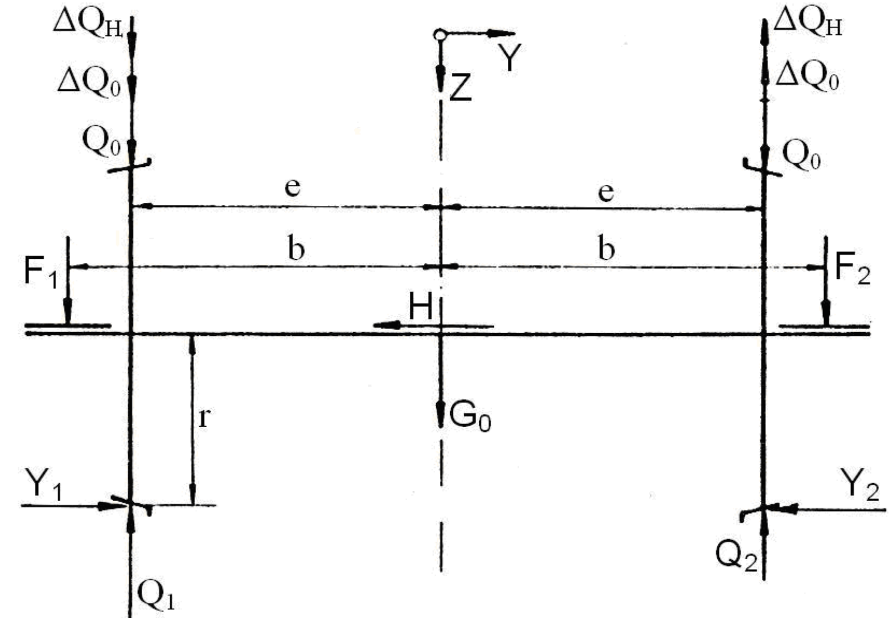

Fig. 1. *The vehicle axle mass load transfers*

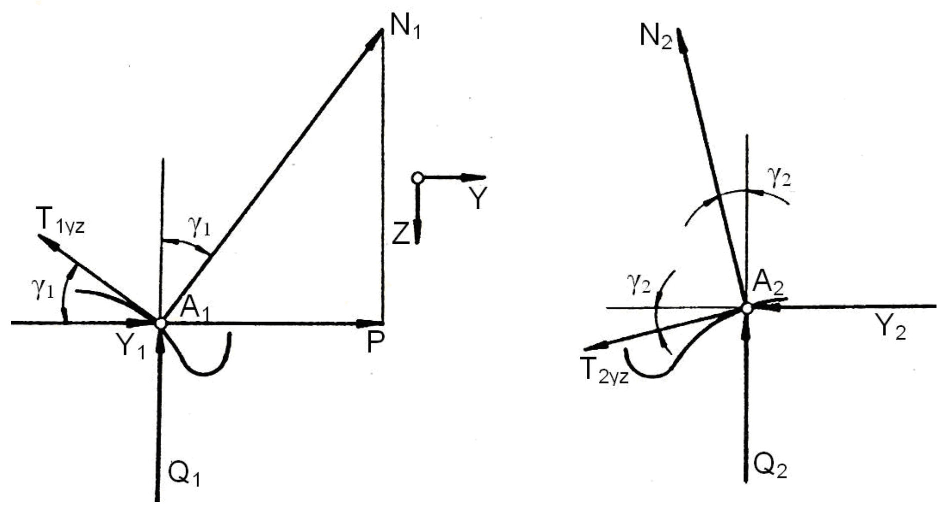

Fig. 2. *The contact forces between wheel and rail in the horizontal and vertical – transverse planes*

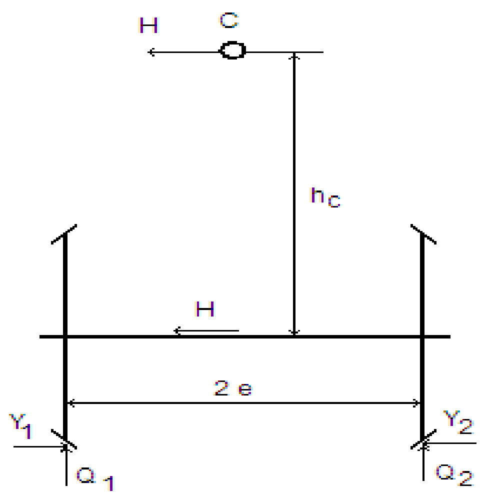

Fig. 3. *The mass loads on the wagon wheels*

$$m_r = \frac{m_{rc} + m_b}{1 + (x_b / i_{bz})^2 + (z_b / i_{bx})^2} \tag{1}$$

$$H_{d\max} = c_y y_{d\max} = m_r \Delta\gamma_{T0} + \sqrt{(m_r \Delta\gamma_{T0})^2 + c_y m_r (\mathrm{v}\sin\delta)^2} \tag{2}$$

$$H = \left(\frac{2Q_0}{g} - m_0\right)\gamma_{\mathrm{T0}} = (2Q_0 - m_0 g)\frac{I}{1500} \tag{3}$$

$$H = \left(\frac{2Q_0}{g} - m_0\right)\gamma_{\mathrm{T0}} = (2Q_0 - m_0 g)\frac{I}{1500} \tag{4}$$

$H_{\max} = H + H_{d\max}$, $H_{\max} = H_{d\max} - H$ (the outer rail thread and the inner rail thread of the rolling track) (5)

$$Q_1 = Q_0 + \Delta Q_0 + \Delta Q_H = 100 + 16.13 + 3.08 = 119.21\ kN \tag{6}$$

$$Q_2 = Q_0 - \Delta Q_0 - \Delta Q_H = 100 - 16.13 - 3.08 = 80.79\ kN \tag{7}$$

$$Y_1 = H + Q_2 tg(\gamma_2 + \delta_2) = 11.8 \cdot 80.79 \cdot 0.45 = 48.16\ kN \tag{8}$$

$$Y_2 = Q_2 tg(\gamma_2 + \delta_2) = 80.79 \cdot 0.45 = 36.36\ kN \tag{9}$$