Bulletin of the *Transilvania* University of Braşov • Vol. 10 (59) Special Issue No. 1 - 2017
Series I: Engineering Sciences

STUDY ON SAFETY AGAINST DERAILMENT SPECIFIC TO DIESEL LOCOMOTIVES CLASS 64, SERIES OF 621 EGM

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Abstract: *The following is the result of the shock attack in the case of the movement of the Class 64 diesel electric locomotive in a curve in which there is a discontinuous elbow, mathematically described by an angular point, the angle of shock at this point producing a sudden change of direction of the vehicle and a dynamic shock force. It also presents the mathematical model on the basis of which the dynamic force of shock is determined and its influence on the safety of the locomotive guidance. The study was exemplified for the case of the DE 621 EGM diesel locomotive on two three-axle bogies, as three-axle bogie-driven rail vehicles generally in curves, the registration is more demanding compared to two-axle bogies and because if accidentally bent elbows appear, the risk of the occurrence of the phenomenon of escalation of the fungus of the track by the striking wheel is more pronounced. Also, in the paper it is also shown that the dynamic force can lead to the inadequate increase of the guiding force Y of the driving axle of the vehicle and thus exceeds the limiting ratio (Y / Q) defining the limit to derailment, Q being the vertical force on the attacking wheel.*

Key words: *derailment, skidding, sliding, guiding, attack angle.*

1. Introduction

In the contact area of the wheel - rail or the center of it, it acts on the side of the rail, the normal bearing force N_i perpendicular to the tangent contact plane and contained in the normal vertical plane on the two track threads and the frictional force T_i perpendicular to the normal force and thus contained in the tangential contact plane.

The spatial orientation of the contact forces, as well as the sliding speeds, depends on the position of the axle in the path, which is characterized by the angle of attack α and through the gap y_c relative to its median position. The inclination of the tangent contact

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plane to the horizontal plane is given by the angle δ_i which is the angle between the intersection planes of the vertical plane on the path lines with the tangent contact plane and the horizontal plane passing through the contact point.

Spatial orientation of normal force N_i is determined exclusively by geometric conditions [7], [8]. Because of the small attack angles in use, the longitudinal component of the normal force can be neglected, the normal force being considered to act in the vertical - transverse plane (*YZ*), where the load transfers (Figure 1). Spatial orientation of the friction force T_i is primarily determined by the geometric conditions because it is contained in the tangent contact plane but because, according to the general laws of friction, it has the same direction as the slipping speed and oriented in the opposite direction, it is also determined by the slides produced in the contact points, ie the kinematic conditions.

The dimension that defines the spatial orientation in the tangent contact plane of the frictional force T_i, due to kinematic conditions is a slip angle ξ_i, which at the contact points on the tread surface has a tangent approximately equal to the ratio of the longitudinal and transverse slides. As a rule, transverse landslides are determined by the angle of attack α and as a result, the transverse parts of the friction forces will have the same meaning on both wheels.

Longitudinal slides reaction, when the axle is running freely, are determined by the difference in radius of the actual rolling circles that depend on the gap y_c, usually being opposite to the two wheels (Figure 2).

The value of the term cos ξ_i in this case, regardless of the angle of attack α, is approximately unitary. In traction or braking, the increase in longitudinal sliding speeds makes the value cos ξ_i to drop well below the unit value, which is also influenced by the attack angle α. The size of friction force $T_i = \tau_i N_i$ is dependent on the coefficient of friction τ_i, which has a nonlinear variation with slipping and pseudo-slipping respectively.

In the case of the driving axles, which also guides the other axles in the same chassis, the striker wheel 1 can run in bicontact with the attacked of rail (Figure 2), which is the situation in the new conical profile. In this case, in addition to the contact point A_1 , the second point appears A_1', which becomes a guiding point.

In the case of the bicontact the flange angle is applied to the wheel γ_1' is small and then the driving force can be considered $P=N_1\sin\gamma_1$ On the unattainable wheel of the axle, also due to the small flank angle at the point of contact A_2 , the driving force is negligible (Figure 3). Size of the guiding force Y_i results on each wheel from the vertical summation of the horizontal component of the normal force with the horizontal - transverse component of the friction force. The maximum value of the force Y_i appears on the striking wheel of a leading axle where the driving force also intervenes *P*.

2. Determination of the Derailment Limit for the DE 621 EGM Locomotive

The derailment safety of an electric diesel locomotive class 64, the DE 621 EGM railway series is determined by the guide axle of the drive axle, which is the maximum driving force on the approaching wheel to the derailment limit (Figure 4) [1].

Driving axle guidance capacity results from the vertical-transverse equilibrium equilibrium conditions of forces acting on the axle. In the case of bicontact of the wheel with the rail, the increase of the guiding force Y_1 it increases at the guiding point A_1 the reaction force N_1 and, consequently, the upward action of the component will increase

T_{1yz} of the frictional force, which is why the reaction is diminished N_1' from the support point A_1'. The situation is reached $N_1' = 0$, so when the point of support A_1' is completely discharged and the load Q_1 from the striking wheel passes entirely on the guide lip of the wheel at the point A_1, the derailment limit is reached.

If against this situation the force Y_1 will continue to rise, the leading wheel's lip will climb on the inner flank of the rail, causing derailment. In case of single contact, the derailment limit is reached when the single point of contact A_1 has reached the lip at a maximum flank angle. Following the analysis of the derailment process, we noticed that *Nadal*'s formula was deducted only from the contact forces on the striker wheel, regardless of the dependence between the load on the wheel and the steering force, or the influence of the effect the spin on the lip contact point on the friction coefficient (Figure 5). As a result of the experiments carried out in the BOR 55 and the BER B 136 Committees, it was recommended to adopt the coefficient of friction $\mu = 0.36$, considered as a cover when applying the Nadal formula to the derailment calculations showing also the favorable influence of the increase in the flank angle of the wheel lips γ_1 on the guiding capacity. At the same time, it emerged from the calculations and experiments, the advantage of the flank angle 70° of the outer lip guiding surface to increase the axle guidance capacity. The work of the BER Committee 55 showed that, in order to avoid derailment of the vehicle in the current line, the ratio Y_1/Q_1 must be lower than the limit value $(Y_1/Q_1)_{\lim} = 1.2$.

In traffic over track-side devices, the experiences of the C9 and C70 Committees have led to the conclusion that $(Y_1/Q_1)_{\lim} = 0.8$ and at the tip of the lip with the tip of the heart at the crosses, a report must be provided $(Y/Q)_{\lim} = 0.4$.

3. Safety Against Derailment under the Influence of External Forces

The report $(Y_1/Q_1)_{\lim}$ can not constitute a correct criterion for assessing the safety against derailment, except when the load on the steering wheel Q_1 represents the vertical component of the effective rail reaction at the derailment limit (Figure 6), taking into account the fact that it is dependent on the guiding force Y_1.

The flank angle γ_2 depends on the shape of the wheel and rail profile, as well as the track gauge (for conical profiles $tg\ \gamma_2=0.05$). In general, the term $tg\ (\gamma_2+\delta_2)$ is proper to each vehicle and its respective rolling parts, being influenced by the angle of attack α. At the limit of derailment, according to *Nadal*'s formula, Y_1 must satisfy the condition imposed by equality $Y_1 = Q_1 . \mathrm{tg}(\gamma_1 - \delta_1)$, in which tg $(\gamma_1 - \delta_1)$ has well-defined limit values based on the flank angle of the wheel lips.

4. Influence of Attack Shock on Derailment Safety

Railway curves may result in deviations from nominal dimensions in the form of continuous or discontinuous bends that produce dynamic forces of interaction between the vehicle and the track in the transverse direction, which aggravate the quality of the ride and jeopardize the safety the vehicles guidance [1]. The continuous path bends are characterized by continuous curvature deviations which, overlapping the track twists, lead to variations in both the cant deficiency and the transverse acceleration of the vehicle. In

Romania, continuous bends are limited by the tolerances admitted to the measured arrows.

If the vehicle travels at a constant speed in a non-deviating curve with cant deficiency I, his weight box m_c will be acting to a quasi-static transverse acceleration γ_{T0}, or an uncompensated centrifugal force F_n. As a result, a quasi-static chassis driving force will operate on each axle H. Simultaneously with the emergence of force H there is also an elastic compression of the superstructure of the track and the vehicle. Considering that their total stiffness is c_y, their static deformation will be $y_c=H/c_y$. Therefore, the vehicle can be considered as a simple harmonic oscillator, ie a mass-arc system, in which y_c represents the static deformation of the spring. When the vehicle reaches the discontinuous elbow tip with the outermost wheel of the first axle, the rail will be attacked with an attack speed $v.\sin\delta \approx v\delta$, having a direction perpendicular to the striking track. This creates a dynamic force $H_d=c_y.y_d$ driving a chassis called a shock force or a force of attack, in which y_d represents the dynamic deformation of the arc with the cumulative stiffness c_y. In the shock process, the entire mass of the vehicle does not take part, but only a part of it, namely a "reduced" mass, marked with m_r and the expression of maximum dynamic force H_{dmax} can be deduced applying the energy conservation theorem.

The attack speed component $v.\sin\delta$ is perpendicular to the rail and will give the mass m_r in this direction, the kinetic energy having the value $(1/2)\cdot m_r\cdot(v\cdot\sin¢)^2$ which is elastically picked up by the stiffness spring c_y between the table and the rails. When spring compression is maximum y_{dmax}, the kinetic energy becomes null, turning entirely into potential energy. If it is associated with a discontinuous elbow, another elbow that is continuous, then due to the variation ΔI [mm] of the cant deficiency, the vehicle will have an additional acceleration at the time of the attack Δa_{TO} and will perform additional mechanical work on the distance $y_{d\max}$. The maximum shock force occurs when the term $\sin.(\omega t-\varphi)=1$, so on the outer thread of the path, after the time counted from the moment of reaching the elbow and the inner thread after $t_i = 3t_e$, where time t is defined by the relation (2). Friction from the vibratory system causes the phenomenon to be damped until its complete disappearance, if large driving forces have not caused the derailment of the vehicle in the meantime [4], [5]. It follows that the maximum values of the forces being transmitted to the paths, by requesting it to derive, are $H_{\max} = H_{d\max} + H$ for the outer thread of the tread and respectively $H_{\max} = H_{d\max} - H$ on the inside thread. Derailment of the locomotive by overcoming the ratio $(H/Q_0)_{lim}$, usually occurs on the inner line of the path, which is more discharged than the outside. Since the value of the coefficient of adhesion can be in the beach (0.2...0.8) Depending on the quality of the wheel - rail contact we will have the values in Table 1.

5. Experimental Results and Measurements

In this paper I presented the calculation example performed for a vehicle on two-axle bogies without a central suspension which circulates at a speed V [km/h] in a radius curve R [m] with cant h [mm]. This curve, for the long string C [m], an arrow corresponds to it $f = C^2/(8R)$ [m]. The existence of a continuous elbow over the base curve was emphasized by measuring an arrow f_1, which leads to a radius of curvature $R_1 = C^2/(8f_1)$. Such an elbow in the path, with the continuous variation of the radius of curvature from R to R_1, and keeping cant h leads to variation cant deficiency:

$\Delta I = 11.8 \cdot V^2 \cdot [(1/R_1) - (1 - R)]$ [mm] and to an additional transverse acceleration $\Delta\gamma_{TO}=\Delta I/153$ [m/s^2]. It was also thought to be on the radius curve R_1 a discontinuous elbow appears, which was highlighted by measuring a difference in arrows f_d–f_1, the δ angle given by the relation (1) and it was considered in the calculation that the pivot (crapodine) is located at the bogie center of the bogie, determining later at this point the value of the reduced mass m_{rc} of the locomotive box [4], [5].

The reduced weight m_r of the entire vehicle was determined with the relation (1), where m_b is the suspended mass of the bogie, i_{bx} and i_{bz} are the inertia rays of the bogie. Knowing the transverse stiffness of the axle suspension c_y, with relation (2) the maximum dynamic force is determined H_{dmax}.

The quasi-static force H acting on the axle will be given by relation (3), where m_0 is the unsustained mass corresponding to one axle, $2Q_0$ - axle load and I cant deficiency on the radius curve R [2]. In order to make the check of the path, it must be taken into account that it is so demanded by the force H_{max}, given by the relation (4), and by the inertia force of the axle, as well as by the expression of the inertia force of the axle, which is shown in the form of the equation (5).

The derailment of the vehicle is checked for the two wires of the track, requiring the condition $H_{max} / Q_0 \leq (H / Q_0)_{lim}$, after having previously the vertical load transfer was determined ΔQ_0. For the diesel electric locomotive class 64, DE 621 EGM series, the relation is valid:

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

where the axle load was considered $2Q_0 = 20 \ x \ 10^3 \ x \ 10 = 200$ [kN], gravitational acceleration 10 [m/s^2] [1]. The limitations for the vehicle's running gear are for force in the axle of the H axle, according to relationship (3). Likewise:

$$\begin{aligned} H_{d\max} &= c_y y_{d\max} = m_r \Delta ¥_{T0} + \\ &+ \sqrt{(m_r \Delta ¥_{T0})^2 + c_y m_r (v \sin ¢)^2} \Rightarrow \end{aligned} \tag{2}$$

$$\begin{aligned} \Rightarrow H &= \left(\frac{2Q_0}{g} - m_0 \right) ¥_{T0} = \\ &= (2Q_0 - m_0 g) \frac{I}{1500} \end{aligned} \tag{3}$$

and the axial force H of the locomotive being analyzed is:

$$\begin{aligned} H_{\max} &= H + H_{d\max}, \\ H_{\max} &= H_{d\max} - H \end{aligned} \tag{4}$$

(for the outer thread and the inner thread of the tread), where E has been noted the excess cant of the track which was considered in accordance with [1], E = 60 [mm]. Also, at the analyzed locomotive, we also considered the guiding force $Y = \sum Y_{max} = 65.17$ [kN] reaches the maximum possible value $Y / Q_0 = 65.17 / 100 = 0.65$. At the same time, it is also noted that $Y / Q_0 \lhd (Y / Q_0)_{lim} = 1.2$ that mean, the vehicle travels safely in a current line and switches. When crossing over the crosses, from the inequation $Y / Q_0 \rhd 0.4$ the safety against derailment is concluded because in reality the guiding force is inferior to the value $\sum Y_{max}$.

Given that there is a vertical load transfer ΔQ_0 due to uncompensated centrifugal force, and $H \cdot h_c = \Delta Q_0 \cdot 2e$ where h_c the height of the center of gravity C of the locomotive relative to the axis of the axle (Figure 3) was noted, for $h_c \approx 2050$ mm. With the value obtained with relation (5), the casting wheel is loaded and discharged from the inner thread of the track. On the other hand, load transfer occurs:

$$\Delta Q_H = \text{ª}(H_r / 2e) = 3.08 kN \tag{5}$$

and thus results on axle loads Q_1 and Q_2 care sunt date de relaţiile (6) and (7). The resulting values result in the wheel guides for the forward wheel respectively Y_1 and Y_2 for the wheel on the inner thread of the path, whose values are found in the expressions of equations (8) and (9) [6].

Therefore, at the DE 621 EGM locomotive analyzed, there is no danger of derailment except passing over the peaks of the hearts from the crossings when we are around the admitted limit [3], [5]. Even in this situation, we do not consider that there are no problems because the calculation is absolutely covered by the value of the cant deficiency I adopted. If it travels with excess E, the force H will be oriented towards the center of the curve, thereby producing a wheel discharge from the outer yarn and loading the wheel from the inner thread of the tread. So, for H = 9.33 [kN], it is obtain a value of 16.3 [kN]. The wheel on the inner thread consumed the play between the rim and the rail:

$$Y_2 / Q_2 = 47.5 / 115.18 = 0.412 < 1.2$$

and so we have provided in this case safety guidance.

$$\Delta Q_0 = H(h_c / 2e) + 11.8 \cdot 2050 / 1500 = 16.13 kN \tag{6}$$

$$\Delta Q_0 = 9.33 \cdot 2050 / 1500 = 12.75 kN \quad \text{respectively} \tag{7}$$

$$\Delta Q_0 = 0.85 \cdot 9.33 \cdot 460 / 1500 = 2.43 kN \quad \text{and therefore:} \tag{8}$$

$$Q_1 = Q_0 - \Delta Q_0 - \Delta Q_H = 100 - 12.75 - 2.43 = 84.82 kN \quad \text{respectively:} \tag{9}$$

$$Q_2 = Q_0 + \Delta Q_0 + \Delta Q_H = 100 + 12.75 + 2.43 = 115.18 kN \tag{10}$$

In this case, the guiding forces in this situation will be:

$$Y_2 = Q_0 tg(\text{¥}_1 + \text{¢}_1) = 84.82 \cdot 0.45 = 38.17 kN \tag{11}$$

$$Y_2 = H + Q_0 tg(¥_1 + ¢_1) = 9.33 + 38.7 = 47.5kN \quad (12)$$

$$Q_1 = Q_0 + \Delta Q_0 + \Delta Q_H = 100 + 16.13 + 3.08 = 119.21kN \quad (13)$$

$$Q_2 = Q_0 - \Delta Q_0 - \Delta Q_H = 100 - 16.13 - 3.08 = 80.79kN \quad (14)$$

$$Y_1 = H + Q_0 tg(¥_2 + ¢_2) = 11.8 \cdot 80 \cdot 79 \cdot 0.45 = 48.16kN \quad (15)$$

$$Y_2 = Q_2 tg(¥_2 + ¢_2) = 80.79 \cdot 0.45 = 36.36kN \quad (16)$$

Tensions in the wheel - rail contact area due to variation of dynamic overloads Table 1

Load force N	MPa	MPa	MPa
119.3	220.9	-611.1	159.8
226.4	274.3	-678.8	206.4
495.1	284.4	-692.7	215.1
660.0	334.4	-828.5	251.6
873.0	414.8	-1358.1	279.0

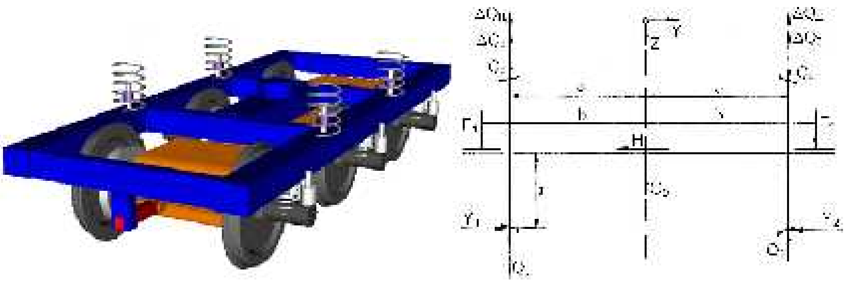

Fig. 1. *Load displacements on the DE 621 EGM locomotive*

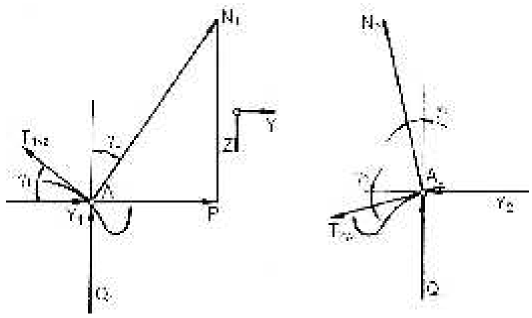

Fig. 2. *Contact forces - wheel in the horizontal and vertical - transverse plane*

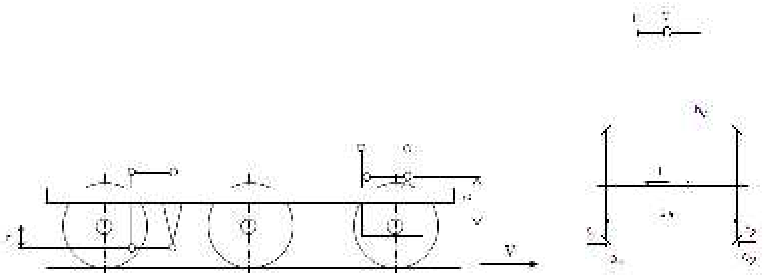

Fig. 3. *Driving forces on DE 621 EGM locomotive wheels*

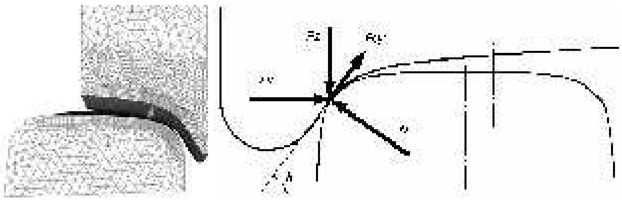

Fig. 4. *Simulation of the rail exclamation phenomenon by the attacking wheel*

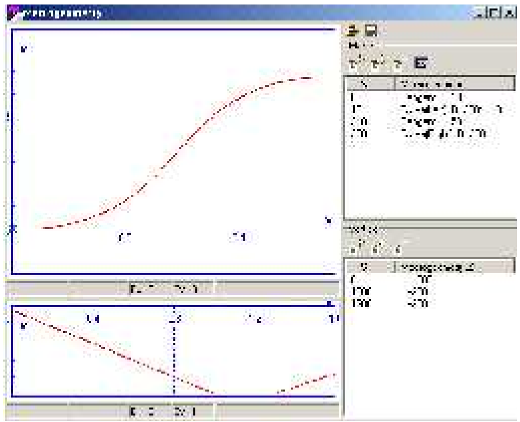

Fig. 5. *Macrogeometry of the dynamic behavior of the attacking wheel - analysis of motion measurements*

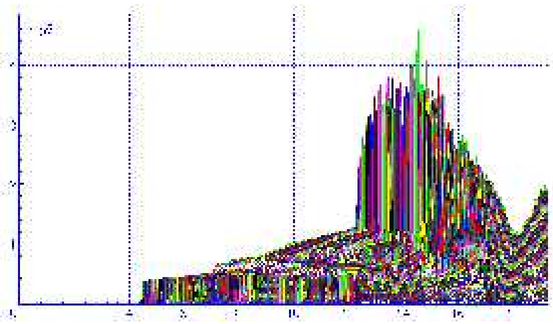

Fig. 6. *Stress strain diagram in the contact area in case of derailment simulation*

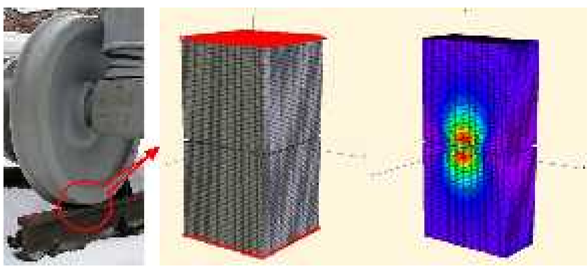

Fig. 7. *Von Mises diagram of isocline curves for dynamic stress simulation and stresses in the wheel-rail contact area due to the variation of the dynamic dynamic overloads on the axle of the locomotive DE 621 EGM*

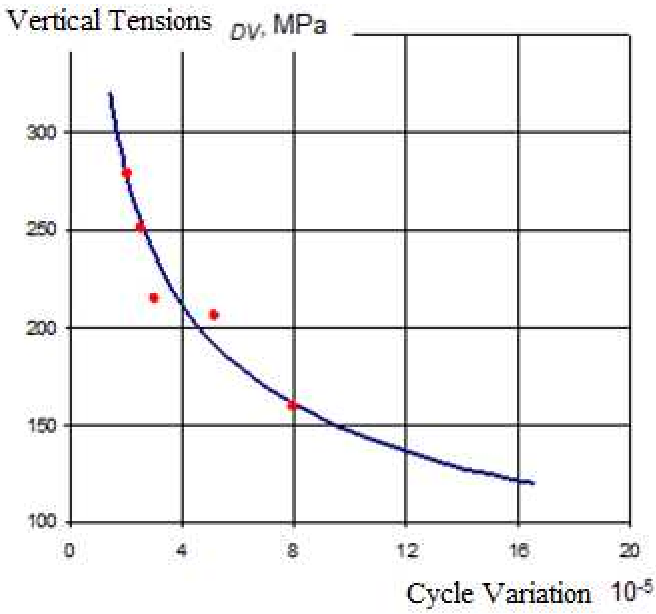

Fig. 8. *Wear Cycle Variation Chart Depending on Vertical Tensions in Wheel-Rail Contact Area*

4. Conclusions

Derailment safety is influenced by both wheel load and wheel radius and maximum wheel flank angle. The axle guidance capacity decreases with the reduction of the load on the forward wheel, so the load transfer from the forward wheel to the unattractive wheel is higher. The limit situation for the unloading of the attacking wheel may occur when the curves with the maximum cant and the maximum torsional traverse.

The negative load transfers originate from the inclination towards the inside of the curve of the vehicle box, being amplified by the coefficient of torque and torsion of the track, which are mainly taken over by the suspension of the vehicle, and in the case of small diameter wheels the danger of derailment is greater. Increasing the maximum rim flange angle is favorable to derailment safety as it increases both the minimum and maximum axle guidance.

The worst case scenario occurs when the vehicle is driven under quasi-static conditions at low speeds (up to 40 [km/h]) in curves with a radius of 150 [m] and maximum torsional tolerances.

Maximum allowable load transfers in this situation should be at most $\Delta Q_0 / Q_0 \leq 0.6$ for wheels with the flange angle of the lip 70°, provided that the transverse force H is very close to zero. This is usually done only on vehicles with adjustable axles.

Defined formulas for the assessment of derailment safety can be applied for any walking speed, provided that the load transfers are correctly determined, taking into account the dynamic actions of the vehicle in the most unfavorable situations. In locomotives, because of the transmission of traction force to the chassis, great longitudinal rigidity of the axle suspension is practiced and consequently the axles can be considered fixed in the bogie chassis. When turning the bogie into small-beam curves, the antagonistic forces generate appreciable values of force H, which forces a reduction in load transfers to avoid derailments. Checks to avoid complete unloading of non-tracing wheels are made for the curve of the curve at its maximum authorized speed.

The driving chassisis forces of particular practical significance as it determines the safety of the road, the demand for the vehicle's running gear and the safety of the derailment.

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