STUDY CONCERNING THE INFLUENCE OF THE FORCES CAUSED BY THE BUFFING GEAR AND DRAW GEAR IN THE CURVE RUNNING TO THE SAFETY GUIDANCE OF THE RAILWAY VEHICLES

Laurențiu Cornel DUMITRU[1]

Studiul analizează influența forțelor produse la circulația în curbă de aparatele de tracțiune-legare-ciocnire asupra siguranței ghidării unui vehicul feroviar, respectiv asupra raportului Y/Q (coeficientul de siguranță contra deraierii). S-a analizat cazul profilului de uzură „S78", luând în considerare transferul de sarcină de pe o roată pe alta. Ca exemplu de calcul, a fost analizat cazul unui vagon platformă pe două osii tip Ks, circulând pe o curbă cu raza de 150 m, pentru viteze variind de la 0 la viteza maximă, fiind reprezentate grafic curbele de variație ale forței de ghidare Y şi ale coeficientului de siguranță împotriva deraierii Y/Q.

The study analyses the influence of the forces caused by the buffing gear and draw gear in the curve running, on the safety guidance of a railway vehicle, respectively on the factor Y/Q (safety coefficient against the derailment. The case wear profile "S78" was analised taking into account the load transfer from one wheel to another. As a calculation example, the case of a type Ks platform wagon on two axles, running to a curve with a radius of 150 m, for speeds ranging from 0 to maximum speed was analysed. The curves of variation of the guiding force Y and of the safety factor against derailment Y/Q are presented.

Key words: leading force, guiding force, safety coefficient against the derailment.

1. General introduction

The aim of this study is to determine the influence of the forces caused by the buffing gear and draw gear in the curve running, on the safety guidance of a railway vehicle, respectively on the factor Y/Q (safety coefficient against the derailment), based on the studies and calculation methods presented in [1], [2], [3].

For a quick, efficient and correct solving of the necessary calculus, a computer program was used. It allowse an automatic calculation of the phenomena and characteristics involved, linear or nonlinear.

Thus, the formulae succession and the correspondent explanations, are presented exactly as in the used software.

[1] PhD Eng., Romanian Railway Authority, Romania, e-mail: laur_dum@yahoo.com

2. Calculus of the influence of the forces caused by the buffing gear and draw gear in the curve running, on the safety guidance of a railway vehicle

Calculation hypothesis:

- one takes the wear profile case "S78", introduced through points, for which the radius values depending on the movement against the medium axle of the track y_c are obtained through the Lagrange interpolation with three points (that has the same error as the square interpolation),
- one does not take into account the vertical elements of the friction forces at the wheel - rail contact,
- one takes into account the load transfer from one wheel to another through the different values of the friction forces at the wheel - rail contact.

The over-widening values S and the correspondent radius of the curve are presented in Table 1.

Table 1

Over-widening

R[m]	100-150	151-250	251-350
S[mm]	25	20	10

Over – raising

- for R < 350 m , $h[mm] = \frac{R[m]-50}{2}$; (1)

- for. R $\geq$ 350 m, $h = 150mm$;

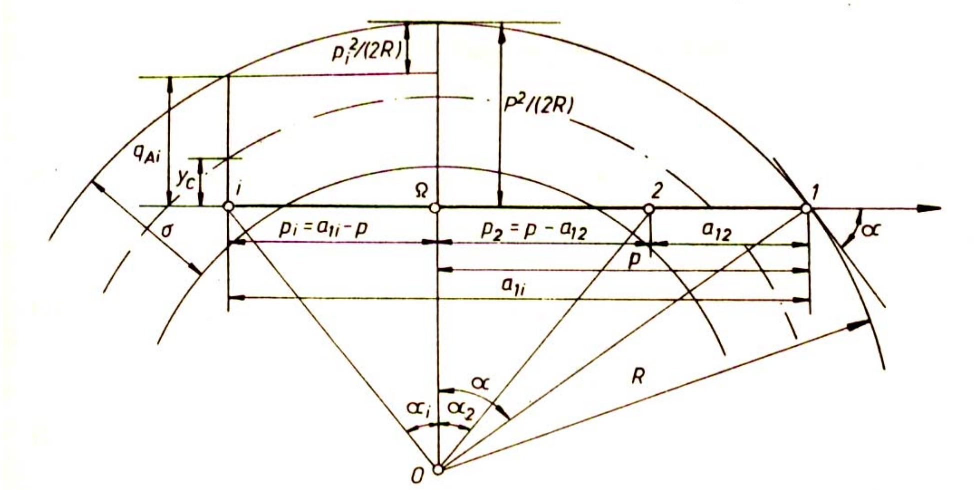

Fig. 1. Impact of the clearance on the geometrical placing of the vehicle

In paper [1] was presented the analytic study of the vehicles position in curves, according to Fig. 1. In paper [4] problems of the wheels running profiles, as well as aspects about the geometrical conditions of the wheel - rail contact were treated.

The Ω base of the perpendicular line drawn from the curve centre of the track on the longitudinal axle of the vehicle is named pole, and the perpendicular line on this is named polar axis.

The distance from the first axle to the pole Ω is named polar distance.

For the study of vehicle placed in curves one determines first the distance y between the pole Ω and the exterior track of rail of radius R.

$$y = \frac{p^2}{(2R)} \; ; \tag{2}$$

q_{Ai} – the axle distance against its exterior rail of the railway curve, that establish its position on track (q_{Amax}=σ), that is the distance between the point A_{10} of the exterior wheel lip and the interior flank of the rail from the exterior rail of the railway curve.

One takes a vehicle with n axles, with the pitch a, placed in the curve with radius R, in free position.

Approaching to the exterior rail of railway curve of an intermediary axle (Figure 1) will be:

$$\mathbf{q_{Ai}} = \left(\mathbf{p}^2 - \mathbf{p_i}^2\right)/(2\mathbf{R}) = \left[\mathbf{p}^2 - (\mathbf{a_{1i}} - \mathbf{p})^2\right]/(2\mathbf{R}) = (\mathbf{a_{1i}}/\mathbf{R}) \cdot (\mathbf{p} - \mathbf{a_{1i}}/2); \tag{3}$$

where y_{ci} – transversal displacement against the medium centre of the track at the axle i, with positive sign at the movement to the exterior of the curve;

From here the polar distance depending on the q_{AI} results as:

$$p = \frac{a_{1i}}{2} + \frac{R \cdot q_{Ai}}{a_{1i}} \; ; \tag{4}$$

The over-raising insufficiency maximum accepted in our country is I=90 mm.

σ - clearance of the axle in the track;

$\sigma = 10 + S$;

a_{1i} – distance between the axle 1 and the axle i;

$a = a_{1n}$ – vehicle pitch;

p_i – polar distance of the axle i, that is the distance from the axle i to the pole Ω. The sign is positive if the pole is back of the axle in the direction of traffic and negative if it is in front of the axle;

$p = p_1$ – polar distance of the axle 1, that is the polar distance of the vehicle;

$p_i = p_1 - a_{1i}$;

$p_c = \frac{a_{1n}}{2}$ - polar distance in the case of rope position;

$p_s = \frac{a_{1n}}{2} + \frac{R \cdot \sigma}{a_{1n}}$ - polar distance in the case of secant position;

Sliding speeds between the wheels and the rails

$\Delta r_{e,i}$ - radius difference because of the movement with y_c on the exterior track of rail, respectively interior.

The signs convention is the following:

- on the exterior track of rail, at a positive movement of y_c (to the exterior of the curve), a positive movement Δr_e (to the wheel tyre) is obtained;
- on the interior rail of the railway curve, at a positive movement of y_c (to the exterior of the curve) a negative movement Δr_i (contrary to the wheel tyre) is obtained. At the movement y_c on the interior track of rail the over – widening S is also added.

$\Delta r_{e,i}$ is obtained through the with the Lagrange polynomial interpolation.

$$\mathbf{w_{ex}} = \mathbf{V}\left[(1-\mathbf{K}) + \left(\frac{\mathbf{e}}{\mathbf{r}} - \mathbf{K}\frac{\Delta \mathbf{r_e}}{\mathbf{r}}\right)\right]; \quad (5)$$

$$\mathbf{w_{ix}} = \mathbf{V}\left[(1-\mathbf{K}) + \left(\frac{\mathbf{e}}{\mathbf{r}} - \mathbf{K}\frac{\Delta \mathbf{r_i}}{\mathbf{r}}\right)\right]; \quad (6)$$

K – regime value;

$\omega_y = \frac{V}{r}$ - for free axle case;

$\omega_y = K\frac{V}{r}$ - for free axle in traction or braking condition.

$K = 1$ – for the free axle case;

$0 < K < 1$ – for the axle in braking conditions;

$1 < K < \infty$ - for axle in traction conditions;

η - transversal distance against the transversal axis of the axle, where the running cone meets the turning cone in hauling or braking conditions and taking into account the load transfer between the two wheels of the same axle (there is in fact an offset due to the regim of traction or braking, respectively due to taking into account the load transfer).

s – the height of the running cone;

$s = \dfrac{2(e+S)}{\Delta r_e - \Delta r_i}$ This formula is valid for the wear profile and results from the similitude of the triangles formed by the running cone. The minus sign from the denominator is due to the signs convention adopted for $\Delta r_{e,j}$

$$\eta = \frac{\Delta Q_0}{Q_0} e \quad ; \qquad K = \frac{s}{R} \frac{R+\eta}{s+\eta} \quad ; \tag{7}$$

$$\mathbf{w_{ey}} = \mathbf{w_{ix}} = -(\mathbf{p}_1 - \mathbf{a}_{1i}) \frac{\mathbf{V}}{\mathbf{R}} \quad ; \tag{8}$$

Sliding coefficients

$\mathbf{\nu_{e,ix}} = \dfrac{\mathbf{w_{e,ix}}}{\mathbf{V}}$ - sliding coefficients on the x direction, on the exterior respectively interior rail of railway curve;

$\mathbf{\nu_{e,iy}} = \dfrac{\mathbf{w_{e,iy}}}{\mathbf{V}}$ - sliding coefficients on the direction y, on the exterior respectively interior rail of railway curve;

$\mathbf{\nu_{e,i}} = \sqrt{\mathbf{\nu^2_{e,ix}} + \mathbf{\nu^2_{e,iy}}}$ - resulted sliding coefficients;

Hertz coefficients

$$(A+B)_{e,i} = \frac{r_{e,i} + \rho_s}{r_{e,i} - \rho_s}; \; (A-B)_{e,i} = \frac{\rho_s - r_{e,i}}{r_{e,i} - \rho_s}; \; \cos\beta_{e,i} = \frac{\left|(A-B)_{e,i}\right|}{(A+B)_{e,i}}; \tag{9}$$

where:
r - the radius of the nominal running tread;
ρ_s - the radius of the rail running surface; ρ_s=0,3 m.

Table 2

Values of a and b coefficients corresponding to angle $\beta[^\circ]$

$\beta[^o]$	90	80	70	60	50	40	30	20	10	0
m	1	1,128	1,284	1,486	1,754	2,136	2,731	3,778	6,612	∞
n	1	0,893	0,802	0,717	0,641	0,567	0,493	0,408	0,319	0

We interpolate with the Lagrange polynominal through three points, having the β angle value, the coefficients m and n (on the on the exterior respectively interior rail of railway curve) are thus obtained.

$$g_{e,i} = \frac{a_{e,i}}{b_{e,i}} = \frac{n_{e,i}}{m_{e,i}} \quad ; \tag{10}$$

$a_{e,i}$, $b_{e,i}$ - semiaxis of contact ellipses on the on the exterior respectively interior rail of railway curve.

Table 3

Values of c_{11} and c_{22} coefficients corresponding to g value

g	1	0,9	0,8	0,7	0,6	0,5	0,4	0,3	0,2	0,1
c_{11}	4,3	4,4	4,54	4,72	4,96	5,27	5,74	6,52	7,97	11,92
c_{22}	3,72	3,87	4,06	4,29	4,6	5,02	5,63	6,59	8,43	13,35

Also, through the interpolation with the Lagrange polynomial through three points, the Kalker coefficients c_{11} and c_{22} , calculated and listed in paper [5] are obtained.

Interpolation with the polynomial Lagrange

Generally one can write:

$$y = \sum_{i=1}^{n} \prod_{j=1}^{n} \frac{(x - x_j)}{(x_i - x_j)} y_i \quad ; \tag{11}$$

For the Lagrange interpolation through 3 points one obtains:

$$y = \frac{(x - x_2)(x - x_3)}{(x_1 - x_2)(x_1 - x_3)} y_1 + \frac{(x - x_1)(x - x_3)}{(x_2 - x_1)(x_2 - x_3)} y_2 + \frac{(x - x_1)(x - x_2)}{(x_3 - x_1)(x_3 - x_2)} y_3 \tag{12}$$

Profile constant value [6]

$$\mathbf{K}_{\mathbf{e},\mathbf{i}} = \mathbf{m}_{\mathbf{e},\mathbf{i}} \cdot \mathbf{n}_{\mathbf{e},\mathbf{i}} \left[\frac{3\sqrt{\mathbf{G}}(1 - \vartheta)}{2(\mathbf{A} + \mathbf{B})_{\mathbf{e},\mathbf{i}}} \right]^{2/3} \quad ; \tag{13}$$

unde:

$\vartheta = 0,3$ - Poisson value;

$G = 840$ tf/cm^2 - transversal flexibility mode;

$G \cdot (a \cdot b)_{e,i} = KN_{e,i}^{2/3}$;

$\mathbf{N}_{\mathbf{e},\mathbf{i}} \cong \mathbf{Q}_{0\mathbf{e},\mathbf{i}}$ - wheel load;

Decreased coefficient of pseudo-sliding χ, according to [6]:

$$\chi_{e,i} = \frac{G \cdot (a \cdot b)_{e,i}}{N} \cdot \frac{c_{11} + c_{22}}{2} \quad ; \tag{14}$$

Friction coefficient μ

$$\mu_{e,i} = 0,35 - 0,02425 Q_{0_{e,i}} + 0,001 Q_{0_{e,i}}^2 \quad ; \tag{15}$$

Friction coefficients with pseudo-sliding

$$\tau_{e,ix} = \frac{\chi_{e,i}\nu_{e,ix}}{\sqrt{1+\left(\frac{\chi_{e,i}\nu_{e,i}}{\mu_{e,i}}\right)^2}}$$ - on the exterior respectively interior rail of railway curve; (16)

$$\tau_{e,iy} = \frac{\chi_{e,i}\nu_{e,iy}}{\sqrt{1+\left(\frac{\chi_{e,i}\nu_{e,i}}{\mu_{e,i}}\right)^2}}$$ - on the exterior respectively interior rail of railway curve; (17)

$$\tau_{e,i} = \sqrt{\tau_{e,ix}^2 + \tau_{e,iy}^2}$$ - resulting.

Friction forces at the wheel-rail contact

$$\mathbf{T}_{e,ix} = -\tau_{e,ix} \cdot \mathbf{N}_{e,i} \quad ; \qquad \mathbf{T}_{e,iy} = -\tau_{e,iy} \cdot \mathbf{N}_{e,i} \quad ; \tag{18}$$

Moment due to buffing gear and draw gear in the curve running

Due to railway vehicles running in the curve [7], in the draw gear a traction force appears, equal and opposite to the compression force from the buffing gear of the coupled vehicles, posted on the interior side of the curve, and it is noted F_{TLC} .

These traction, respectively compression forces, opposites, but on the same direction, are applied in two points at the distance BG one to another (distance form the vehicle axle to the buffing gear axle, as shown in Figure 2). In these conditions, the force arm decreases with the decrease of the curve radius and can be considered, in the woe\rst possible loading case, equal to the distance BG, taking into account the small values of the corresponding angle.

Thus, the moment due to buffing gear and draw gear in the curve running is:

$$M_{TLC} = BG \cdot F_{TLC}\,; \tag{19}$$

2.1. General equilibrium equations on the vehicle generally in case of a two axle vehicle, and calculus algorithm

Starting from the forces and moments equations from [1], [8], [9], according to Figure 2, we can write them taking into account the load transfer, respectively the different values from one axle wheel to another of the friction forces at the wheel - rail contact, for the three possible cases of the vehicle positions in curve: free, secant and rope.

The signs convention is this classic: axis x has the direction of the track centre and the positive sign in the vehicle direction of traffic, axis y has the transversal direction of the track and the positive sign to the exterior of the curve, and the axis z has vertical direction and positive sign downwards.

We can observe that through the force decomposition of the F_{TLC} , equals and opposite, on the same direction, applied in two points at the distance BG one to another (distance form the vehicle axle to the buffing gear axle), due to the formed force arm, so we will not take into account in the forces equations.

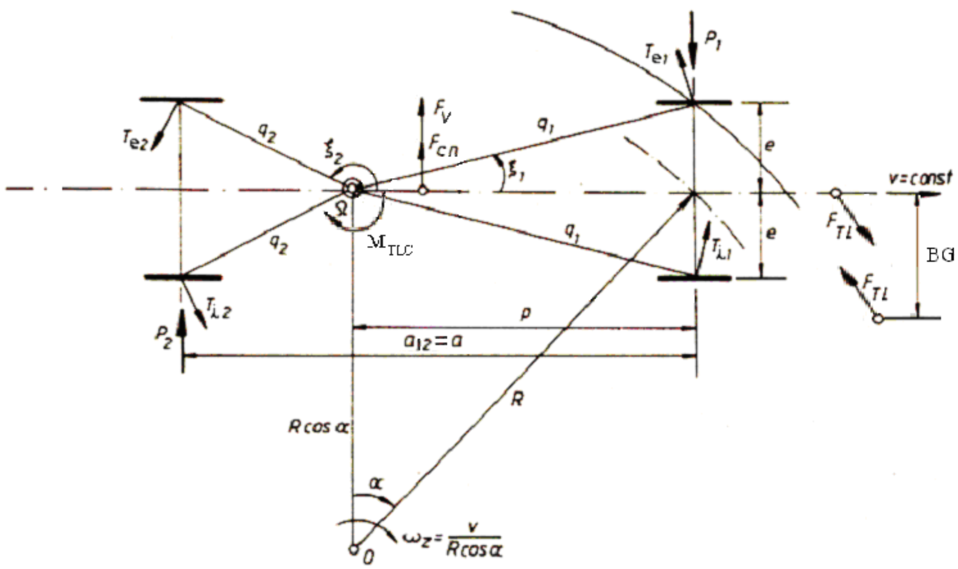

Fig. 2. Model for the curve submition study of the of the vehicle

Free position

$$-\mathbf{P}_1+\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ey_i}}+\sum_{i=1}^{2}\mathbf{T}_{\mathbf{iy_i}}+\mathbf{F}_{\mathbf{cn}}+\mathbf{F}_{\mathbf{v}}=0;$$

$$-\mathbf{M}_{\mathbf{TLC}}-\mathbf{P}_1\mathbf{p}_1-\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ey_i}}\mathbf{p_i}-\sum_{i=1}^{2}\mathbf{T}_{\mathbf{iy_i}}\mathbf{p_i}+(\mathbf{e}-\eta)\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ex_i}}\mathbf{p_i}+(\mathbf{e}+\eta)\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ix_i}}\mathbf{p_i}+ \qquad (20)$$

$$(\mathbf{F}_{\mathbf{cn}}+\mathbf{F}_{\mathbf{v}})\left(\mathbf{p}_1-\frac{\mathbf{a}_{12}}{2}\right)=0,$$

Secant position

$$-\mathbf{P}_1+\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ey_i}}+\sum_{i=1}^{2}\mathbf{T}_{\mathbf{iy_i}}+\mathbf{F}_{\mathbf{cn}}+\mathbf{F}_{\mathbf{v}}+\mathbf{P}_2=0;$$

$$-\mathbf{M}_{\mathbf{TLC}}-\mathbf{P}_1\mathbf{p}_1+\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ey_i}}\mathbf{p_i}+\sum_{i=1}^{2}\mathbf{T}_{\mathbf{iy_i}}\mathbf{p_i}+(\mathbf{e}-\eta)\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ex_i}}\mathbf{p_i}+(\mathbf{e}+\eta)\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ix_i}}\mathbf{p_i}+ \tag{21}$$

$$(\mathbf{F}_{\mathbf{cn}}+\mathbf{F}_{\mathbf{v}})\left(\mathbf{p}_1-\frac{\mathbf{a}_{12}}{2}\right)-\mathbf{P}_2(\mathbf{a}_{12}-\mathbf{p}_1)=0;$$

Rope position

$$-\mathbf{P}_1+\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ey_i}}+\sum_{i=1}^{2}\mathbf{T}_{\mathbf{iy_i}}+\mathbf{F}_{\mathbf{cn}}+\mathbf{F}_{\mathbf{v}}-\mathbf{P}_2=0$$

$$-\mathbf{M}_{\mathbf{TLC}}-\mathbf{P}_1\mathbf{p}_1+\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ey_i}}\mathbf{p_i}+\sum_{i=1}^{2}\mathbf{T}_{\mathbf{iy_i}}\mathbf{p_i}+(\mathbf{e}-\eta)\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ex_i}}\mathbf{p_i}+(\mathbf{e}+\eta)\sum_{i=1}^{2}\mathbf{T}_{\mathbf{ix_i}}\mathbf{p_i}+ \tag{22}$$

$$(\mathbf{F}_{\mathbf{cn}}+\mathbf{F}_{\mathbf{v}})\left(\mathbf{p}_1-\frac{\mathbf{a}_{12}}{2}\right)+\mathbf{P}_2(\mathbf{a}_{12}-\mathbf{p}_1)=0$$

The unknowns are the polar distance p_1 and the guiding forces P_1, respectively P_n for the secant and rope positions. One has only two equations, so the system can not be directly solved in the cases of secant and rope positions. But in the free position one has no guiding force at the last axle, so the system can be solved in this case. This system has the particularity that although the friction forces are known for each position occupied by the vehicle through the polar distance that comes into the calculation formula, they have no linear variation, being used the interpolation during the calculation of the Hertz and Kalker coefficients. That means that only one rough solving method can be found. A way to solve it is the graphic one, with the curves M D Z, according to famouse Heumannn theorem of minimum and to the procedure that he proposed in [10]. Another way is given by the numerical methods for calculation, rough way at which is established the desired precision, and then, through successive iterations the solution is determined. In this application, the bisection method or the interval reducing to one half method was used.

So, solving the system of free position one obtains the value of the polar distance.

After that, one can discuss the following:

- if $p_c<p<p_s$ it results that the vehicle is in free position, so one can establish directly P_1 and p_1.

- if $p<p_c$, it results that the vehicle is in rope position and replacing in the system of the rope position the value of the polar distance $p=p_c$, it result the guiding forces P_1 and P_n.

- if $p>p_s$, results that the vehicle is in secant position and replacing in the system of the secant position the value of the polar distance $p=p_s$, it result the guiding forces P_1 and P_n

Guiding force: $Y = P_1 - T_{ey_1} - T_{iy_1}$;

Maximum running speed in curve: $V_{max} = \sqrt{\frac{R}{11,8}(h+I)}$;

Total uncompensated centrifugal force distributed on the vehicle is:

$$F_{cn} = \left(\frac{V^2}{3,6^2 \cdot R} - g\frac{h}{2e}\right)[G_{cs}(1+S)] \quad ; \qquad (23)$$

S - vehicle flexibility coefficient. We consider S=0,3.

The force given by the pressure of the wind, distributed on the vehicle, is:

$$F_v = F_{cv} = S_c W \quad ; \qquad (24)$$

W - specific pressure of the wind: W=50 daN/m^2 ;

Load transfer: $\Delta Q = \frac{F_{cn} h_c + F_v h_v}{2e}$; (25)

2.2. Calculation example

In this study we consider as a calculation example a platform wagon on two axles, type Ks, with the following characteristics:

- length over the buffers L = 13,86 m;
- vehicle pitch 2a = 9 m;
- the type of the fastening elements of the utilized draw gear: ZCM05/12;
- the type of the fastening elements of the utilized buffing gear: TA-07/1.5.0;
- clearance of the lateral gauge J=(1465-d)/2+q+w [mm], where:
  - d – the minimum value of the track gauge of the wheel set at the highest wearing [mm];
  - q – lateral movement of the wheel set from the centre line of the wagon, at the limit of wearing of the running gear [mm];
  - w – lateral movement in the centre casting (w=0 in the case of the two axle vehicles) [mm];

  It results J = 0,0475 mm;
- tightening the screw of the coupling device: $2S^M$=825-750=75mm,
- the length of the half-coupling in alignment, with buffers in contact:

$$L_D = 530 + 376 + \frac{75}{2} + \frac{2S^M}{2} + \frac{35}{2} = 998,5\text{mm};$$

- the length of the half-coupling in curve, with the screw completely tightened :

$$L_C = 998{,}5 - \frac{2S^M}{2} = 961\text{mm};$$

- the radius of the buffer spring cup PB [mm];
- the distance between the axles of the draw gears and the buffing gears from Figure 2, BG [mm];
- C_A – compression of the draw gear [mm];
- C_T – compression of the buffing gear [mm];
- radius of the track curve R [m].
- radius of the nominal running circle of the wheel r = 0,46 m;
- wagon tare =13,2 t;
- load on wheel Q_0=4,3 t;
- axle mass M_{axle}= 1,1 t;
- lateral surface of the wagon 4,17 m^2;

The calculation was done with the computer program that solves the generally equilibrium equations on the vehicle (20), (21), (22), according to the numerical solving algorithm described, in these 2 cases (with and without taking into account the moment due to the couple of forces F_{TLC}), for the radius of the curve R = 150 m, in the worst case, which is the one of the unloaded wagon.

After we make the initial calculations for determining the compression of the buffing gear and the corresponding forces that occur, according to the provisions of ERRI report B36 RP 32, it result:

a) with maximum initial tightening of the screw:
$C_{T1} = C_{T2}$= 59 mm, result F = 200 kN;
b) with the screw initially half tightened:
$C_{T1} = C_{T2}$ = 48 mm, result F = 140 kN;
c) with no initial screw tightening:
$C_{T1} = C_{T2}$ = 36 mm result F = 80 kN;

We performed the calculations in the three cases described before, with and without taking into account the moment due to the couple of forces F_{TLC} , at the circulation speed form 0 to the maximum speed allowed in th radius of the curve R = 150 m. The results are shown in Table 4, the values being represented in Figures 3 and 4.

The calculation of the safety coefficient against derailment value was made on the wheel from the exterior track of the rail, for negative results of the guiding force Y_1 , opposite to axle y positive sign, with the value $Q_e = Q_0 + \Delta Q_0$, and for positive results of the guiding force Y_1 , oriented to the axle y positive sign , corresponding to a guiding force on the wheel from the interior track of the rail, with the value $Q_i = Q_0 + \Delta Q_0$.

The diagrams that presents the variation of the safety coefficient against derailment always have positive sign.

Moreover, the force caused by the wind pressure have the orientation to the exterior of the curve for negative guiding forces Y_1 , respectively to the interior of the curve for positive guiding forces Y_1 .

The curves presented on the diagrams from Figures 3 and 4 will be noted as follows:

- without taking into account the moment of forces due to draw gear and buffer gear (1)
- taking into account the moment of forces due to draw gear and buffer gear, with no initial screw tightening (2)
- taking into account the moment of forces due to draw gear and buffer gear, with the screw initially half tightened (3)
- taking into account the moment of forces due to draw gear and buffer gear, with maximum initial tightening of the screw (4)

Table 4

Values of the characteristics related to safety guidance of a wagon, type Ks, on a curve with the radius R=150 m, depending of taking into account or not the moment of forces due to draw gear and buffer gear

Velocity [km/h]	F_{cn} [kN]	Fv [kN] (with the condition to have the same orientation with F_{cn} meaning the worst case)	Without taking into account the moment of forces due to draw gear and buffer gear (1)				Taking into account the moment of forces due to draw gear and buffer gear, with no initial screw tightening (2)			
			Y_1 [kN]	$Q_e = Q_0 + \Delta Q_0$ [kN]	$Q_i = Q_0 + \Delta Q_0$ [kN]	$Y/Q_{e,i}$ (condition is $Y_1<0 \rightarrow Q_e$ $Y_1>0 \rightarrow Q_i$)	Y_1 [kN]	$Q_e = Q_0 + \Delta Q_0$ [kN]	$Q_i = Q_0 + \Delta Q_0$ [kN]	$Y/Q_{e,i}$ (condition is $Y_1<0 \rightarrow Q_e$ $Y_1>0 \rightarrow Q_i$)
2	-4.647	-1.043	4.720	39.624	46.376	0.102	9.417	39.624	46.376	0.203
4	-4.558	-1.043	4.587	39.689	46.311	0.099	9.284	39.689	46.311	0.200
6	-4.411	-1.043	4.367	39.797	46.203	0.095	9.064	39.797	46.203	0.196
8	-4.205	-1.043	4.058	39.948	46.052	0.088	8.755	39.948	46.052	0.190
10	-3.941	-1.043	3.661	40.142	45.858	0.080	8.358	40.142	45.858	0.182
12	-3.617	-1.043	3.175	40.379	45.621	0.070	7.872	40.379	45.621	0.173
14	-3.234	-1.043	2.601	40.660	45.340	0.057	7.298	40.660	45.340	0.161
16	-2.793	-1.043	1.939	40.983	45.027	0.043	6.636	40.983	45.027	0.147
18	-2.293	-1.043	1.189	41.350	44.650	0.027	5.886	41.350	44.650	0.132
20	-1.734	-1.043	0.350	41.760	44.240	0.008	5.047	41.760	44.240	0.114
22	-1.116	-1.043	-0.577	42.213	43.787	0.014	4.120	42.213	43.787	0.094
24	-0.439	-1.043	-1.592	42.709	43.291	0.037	3.105	42.709	43.291	0.072
26	0.297	1.043	-2.695	43.982	42.018	0.061	2.002	43.982	42.018	0.048
28	1.010	1.043	-3.887	44.565	41.435	0.087	0.810	44.565	41.435	0.020
30	1.944	1.043	-5.167	45.190	40.810	0.114	-0.470	45.190	40.810	0.010
32	2.856	1.043	-6.535	45.859	40.141	0.143	-1.838	45.859	40.141	0.040
34	3.827	1.043	-7.991	46.571	39.429	0.172	-3.294	46.571	39.429	0.071

36	4.857	1.043	-9.535	47.326	38.674	0.201	-4.838	47.326	38.674	0.102
38	5.946	1.043	-11.168	48.125	37.875	0.232	-6.471	48.125	37.875	0.134
40	7.094	1.043	-12.888	48.966	37.034	0.263	-8.191	48.966	37.034	0.167
42	8.300	1.043	-14.697	49.851	36.149	0.295	-10.000	49.851	36.149	0.201
42.186= V_{max}	8.415	1.043	-14.870	49.936	36.064	0.298	-10.173	49.936	36.064	0.204

Velocity [km/h]	F_{cn} [kN]	Fv [kN] (with the condition to have the same orientation with F_{cn} meaning the worst case)	Taking into account the moment of forces due to draw gear and buffer gear, with the screw initially half tightened (3)				Taking into account the moment of forces due to draw gear and buffer gear, with maximum initial tighten of the screw (4)			
			Y_1 [kN]	$Q_e = Q_0 + \Delta Q_0$ [kN]	$Q_i = Q_0 + \Delta Q_0$ [kN]	$Y/Q_{e,i}$ (condition is $Y_1<0 \rightarrow Q_e$ $Y_1>0 \rightarrow Q_i$)	Y_1 [kN]	$Q_e = Q_0 + \Delta Q_0$ [kN]	$Q_i = Q_0 + \Delta Q_0$ [kN]	$Y/Q_{e,i}$ (condition is $Y_1<0 \rightarrow Q_e$ $Y_1>0 \rightarrow Q_i$)
2	-4.647	-1.043	14.114	39.624	46.376	0.304	18.811	39.624	46.376	0.406
4	-4.558	-1.043	13.981	39.689	46.311	0.302	18.678	39.689	46.311	0.403
6	-4.411	-1.043	13.761	39.797	46.203	0.298	18.458	39.797	46.203	0.399
8	-4.205	-1.043	13.452	39.948	46.052	0.292	18.149	39.948	46.052	0.394
10	-3.941	-1.043	13.055	40.142	45.858	0.285	17.752	40.142	45.858	0.387
12	-3.617	-1.043	12.569	40.379	45.621	0.276	17.266	40.379	45.621	0.378
14	-3.234	-1.043	11.995	40.660	45.340	0.265	16.692	40.660	45.340	0.368
16	-2.793	-1.043	11.333	40.983	45.027	0.252	16.030	40.983	45.027	0.356
18	-2.293	-1.043	10.583	41.350	44.650	0.237	15.280	41.350	44.650	0.342
20	-1.734	-1.043	9.744	41.760	44.240	0.220	14.441	41.760	44.240	0.326
22	-1.116	-1.043	8.817	42.213	43.787	0.201	13.514	42.213	43.787	0.309
24	-0.439	-1.043	7.802	42.709	43.291	0.180	12.499	42.709	43.291	0.289
26	0.297	1.043	6.699	43.982	42.018	0.159	11.396	43.982	42.018	0.271
28	1.010	1.043	5.507	44.565	41.435	0.133	10.204	44.565	41.435	0.246
30	1.944	1.043	4.227	45.190	40.810	0.104	8.924	45.190	40.810	0.219
32	2.856	1.043	2.859	45.859	40.141	0.071	7.556	45.859	40.141	0.188
34	3.827	1.043	1.403	46.571	39.429	-0.030	6.100	46.571	39.429	0.155
36	4.857	1.043	-0.141	47.326	38.674	0.003	4.556	47.326	38.674	0.118
38	5.946	1.043	-1.774	48.125	37.875	0.037	2.923	48.125	37.875	0.077
40	7.094	1.043	-3.494	48.966	37.034	0.071	1.203	48.966	37.034	0.032
42	8.300	1.043	-5.303	49.851	36.149	0.106	-0.606	49.851	36.149	0.012
42.186= V_{max}	8.415	1.043	-5.476	49.936	36.064	0.110	-0.779	49.936	36.064	0.016

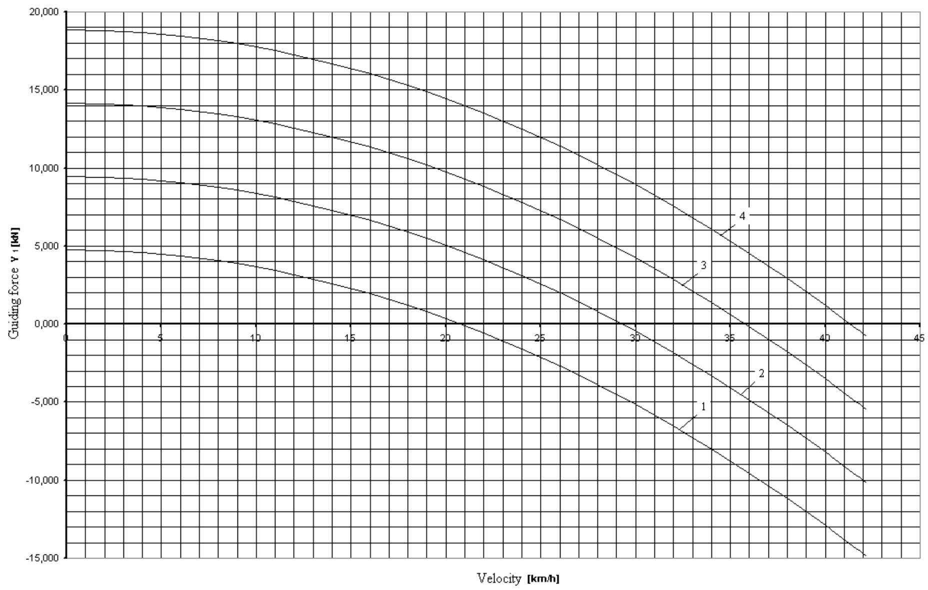

Fig. 3 Guiding force variation with the velocity on a curve with radius R = 150 m

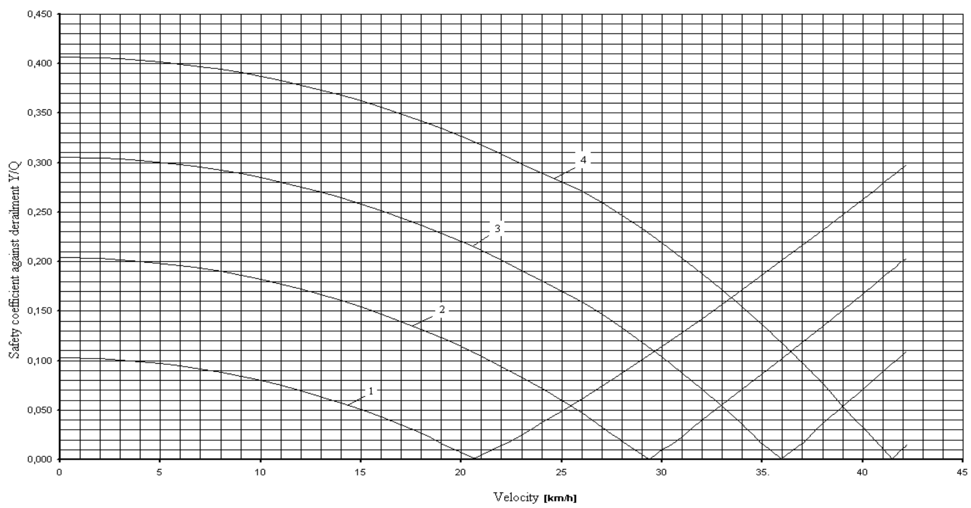

Fig. 4 variation of the safety coefficient against derailment with the velocity on a curve with radius R = 150 m

3. Conclusions

The purpose of this study was that by using the software program based on numerical calculation methods, to describe as exactly the dynamic phenomena that occur to the railway vehicles circulation in curves.

The initial calculation for determining the compression of the buffer gear and the related compression forces, was made according to the algorithm presented in ERRI report B36 RP 32.

After that, to determine the influence of the forces caused by the draw gear and buffer gear on the safety guidance, the case of the wear profile “S78”, introduced by points was considered, for which the radius values of the wheel depending on the displacement over the railway axis were determined by three points Lagrange interpolation.

Also, the load transfer from one wheel to another was considered, allowing to write the forces and moments equations on the vehicle, taking into account the load transfer, and the differences between the interior and exterior track of rail, in the three possible cases of settlement of the vehicle in curve: free, secant and rope.

Thus, it have been used for solving the forces and moments equations on the vehicle, the wheel-rail friction forces, separate on the exterior and interior rail of the railway curve. Moreover, it have been used for solving the over-raising related to the radius of the curve.

The resulting system of equations has the particularity that the wheel-rail friction forces, even if we know them for every case of settlement of vehicle in the curve, have a nonlinear variation, Lagrange interpolation was used to determine the Hertz coefficients.

From the tables and diagrams resulted, we can conclude the following:

- at low velocity, due to the over-raising, the guidance is made on the interior track of rail, then, with the increase of velocity, the guiding force decreases and also the Y/Q factor, until the sign of the guiding force changes. Thus the guidance is made on the exterior track of rail, as a consequence of increase of the centrifugal force with the velocity;
- the variation of safety coefficient against derailment shows that when the effect of forces due to the moment due to buffing gear and draw gear is not taken into account, the safety coefficient against derailment is greater in case of the maximum speed of movement than in the case of over-raising;
- when we take into account the effect of forces due to buffing gear and draw gear, the influence on the safety coefficient against derailment increase significantly if the screw has maximum initial tightening;
- in case of maximum initial tightening of the screw, higher values of safety coefficient against derailment are obtained, ranging from 0.406 to 0.016 with increase of speed, it results that this influence does not create danger of derailment, but can cause faster wear;
- thus it results the influence coefficient of safety against deraierii for small radius curves and low-speed traffic, which cannot lead to derailment as a single factor, but may cause more faster wear;

- finally, it is clear that it is necessary the fulfill of the conditions for binding of vehicles in curves with small radius.

R E F E R E N C E S

[1] *I. Sebeşan*, Dinamica vehiculelor de cale ferată. (Dynamics of the railway vehicles). Bucureşti, Editura Tehnică, 1995, pp 12-216. (in Romanian)

[2] *I. Sebeşan, I. Copaci,* Teoria sistemelor elastice la vehiculele feroviare. (Elastic systems theory on railway vehicles). Bucureşti, Editura Matrix Rom, 2008, pp 7-28. (in Romanian)

[3] *I. Sebeşan, D. Hanganu,* Proiectarea suspensiilor pentru vehicule pe şine. (Suspension design for railway vehicles) Bucureşti, Editura Tehnică, 1993 (in Romanian)

[4] *M. Bouteloup,* Les conditions geométriques du contact rail-roue. (Geometric conditions of wheel-rail contact). In: Bulletin de l'Association Internationale du Congrés des Chemins de Fer, 9, 1952. (in French)

[5] *J. J. Kalker,* A strip theory for rolling with slip and spin. In: Proc. Kon. Akad. Wet., Amsterdam B. 70, 1967

[6] *M. Chartet,* La théorie statique du déraillement d'un essieu. (The static theory of axle derailment). In: Revue Générale des Chemins de Fer, s. 365, 1950 (in French)

[7] *C., Burada, M. Buga, Al. Crăsneanu,* Elemente de structuri portante ale vehiculelor de cale ferată. (Railway vehicles carrying capacity elements) Bucureşti, Editura Tehnică, 1980, pp. 235-332. . (in Romanian)

[8] *L. Dumitru,* Studiu privind comportarea din punct de vedere al siguranței contra deraierii la circulația în curbă a vehiculelor cu roți elastice. (Study concerning the behavior from the safety viewpoint against the derailment depending on the centre casting torque of the railway vehicles with elastic wheels) Simpozionul Național de Material Rulant, Universitatea Timişoara, facultatea de Mecanică, secția Material Rulant de Cale Ferată, 1995 (in Romanian)

[9] *I. Sebeşan, L. Dumitru,* Study concerning the behavior from the safety viewpoint against the derailment depending on the centre casting torque of the railway vehicles (lubricated plane centre casting, respectively non lubricated), XVIIIth International Scientific Conference Transport 2008" Todor Kableshkov Higher School of Transport, Sofia, Bulgaria, 07-08.11.2008