EVALUATION THE SAFETY OF THE TRAFFIC RUTTIER FROM THE VIEW STUDIES OF THE VIBRATIONS OF THE VEHICLES

Conf. univ. dr. ing Dumitru BOLCU
University of Craiova, Faculty of Mechanics

Conf. univ. dr. ing. Ilie DUMITRU
University of Craiova, Faculty of Mechanics

Prof. univ. dr. ing. Victor OȚĂT
University of Craiova, Faculty of Mechanics

Mat. Theodor George OPRICA
University of Craiova, Faculty of Mechanics

REZUMAT. Numărul necunoscutelor este mai mare, în ecuațiile care exprimă mişcarea automobilelor, decât numărul ecuațiilor de mişcare. Din acest motiv sunt necesare ipoteze simplificatoare, pentru a determina distribuția forțelor din zona de contact roată-carosabil. Una dintre aceste ipoteze constă în neglijarea deformației corpurilor în comparație cu deformația suspensiilor. Se consideră, în aceste condiții, că forțele verticale sunt proporționale cu deformările şi cu vitezele de deformare ale suspensiilor. În această lucrare se determină forțele verticale în cazul tranziției de la mişcare rectilinie la mişcare circulară. Este prezentată, de asemenea, dependența dintre aceste forțe şi poziția centrului de masă, viteza mişcării, raza traiectoriei, ca şi influența caracteristicilor suspensiilor şi a unghiul dintre viteza centrului de masă şi axa longitudinală a automobilului asupra forțelor verticale. Se analizează aceste rezultate şi se observă creşterea încărcării la exteriorul corpului, cea mai încărcată roată fiind cea din față, la exteriorul curburii. Se observă, de asemenea, că roata din spate de la interiorul curburii suferă o descărcare. În lucrare se determină, totodată, viteza critică, la care forțele verticale din roata din spate de la interiorul curburii devin nule şi apare pericolul derapării.

Cuvinte cheie: ecuații de mişcare, distribuție de forțe, viteză critică.

ABSTRACT. In the cars moving equations, the number of unknown parameters is bigger then the number of moving equations. For this reason, the simplified hypothesis are necessary in order to determinate the distribution of forces from contact area between wheels and the road. One of those hypotheses is referring to neglect the body deformation relative to suspension deformations. In those conditions, it is considerate that the vertical forces are proportional with the deformations and the deformation velocities of the suspensions. In this paper the vertical forces are determinate in case of transition between the rectilinear movement and the circular movement. Also, it is presented the dependence between those forces and the centre of mass position, the movement velocity, the curvature radius of trajectory, and the influence of suspensions characteristics and of angle between the velocity of mass centre and the cars longitudinal axe over the vertical forces. It is analysing those results and for external side of body an increasing loading is observed, the most loading wheel is the front wheel from external curvature. In the same time, it is observed that the rear wheel internal of curvature has an unloading. In paper, it is also determinate the critical velocity, wherefore the vertical forces at the rear wheel internal of curvature is become null and the danger of slide-slip is emergent.

Keywords: moving ecuations, force distribution, critical velocity.

1. THEORETICAL CONSIDERATIONS

The vehicle is related to a proper reference co-ordinate system chooses so that:

– the O origin of the coordinate system belongs to the vehicles centre mass;

– the Ox axe belongs the horizontal plane and it is parallel with the road;

– the Oy axe to be perpendicular to the longitudinal plane of the vehicle;

– the Oz axe to be included in longitudinal plane of the vehicle and it is perpendicular to road

Beside that, by reason of symmetry, it is accepted this axes are principal axes of inertia.

The moving equations of vehicle are induced by principium of mechanics. In order to determinate the interactions forces between wheels and road the derivative impulse axiom and the derivative kinetic moment are used:

$$m \cdot \vec{a}_c = \vec{F} + \sum_{i=1}^{4} \vec{F}_i \quad (1)$$

$$\overset{\Rightarrow}{J}_c \cdot \vec{\varepsilon} + \vec{\omega} \times (\overset{\Rightarrow}{J}_c \vec{\omega}) = \vec{M}_C + \sum_{i=1}^{4} \vec{M}_{Ci} \quad (2)$$

where: m is the vehicle mass; $\vec{a}_c$ – the acceleration of vehicles mass centre; $\vec{F}$ – the resultant of external forces, different by the interactions forces between wheels and the road; $\vec{F}_i$, i = 1...4 are the interactions forces between wheels and the road; $\vec{\vec{J}}_c$ – the inertial tensor of vehicle, related to the proper chosen frame; $\vec{\omega}$ – the angular velocity of the vehicle; $\vec{\varepsilon}$ – the angular acceleration of the vehicle; $\vec{M}_C$ – the resultant moment in mass centre, due to the external forces and external moments, different by those that appear at the interaction between wheels and road; $\vec{M}_{Ci}$, i = 1...4 – the resultant moment in mass centre due to the interaction forces between wheels and the road;

It is obtained an undetermined system of equations by projecting the (1) and (2) equations on the axes of the proper frame. The obtained system of equations is undetermined because the number of unknown parameters is bigger then the number of equations. To eliminate this inconvenient is necessary to taken into account the rolling and the pitching oscillations and the constructive characteristics of suspension.

In this way is considered that the vertical reactions depend on the suspensions deformations, also on the velocity of deformations of suspensions elements.

It will be considered the case of passing from linear movement to circular movement. Also, it will be considered that the suspended mass, at the initial moment, does effectuate neither rolling oscillations nor pitching oscillations. This oscillation will appear as a result of the circular movement for the mass centre of vehicle. In this way, the pitching of suspended mass unto the horizontal plane will produced deformations of the suspension, and those deformations will act on the vertical reaction forces.

In those conditions, it is obtained the next system of equations regarding a vehicle during a circular movement with constant velocity and constant radius:

$$\begin{aligned} F_{1z} = &\frac{m\cdot g\cdot b}{2(a+b)} - \frac{m\cdot h\cdot b}{e(a+b)}\cdot\frac{v^2\cos\gamma}{R}\left[1-e^{-\alpha_1 t}\left(\frac{\alpha_1}{\beta_1}\sin(\beta_1 t)+\cos(\beta_1 t)\right)\right]+ \\ &\frac{m\cdot h}{2(a+b)}\cdot\frac{v^2\sin\gamma}{R}\left[1-e^{-\alpha_2 t}\left(\frac{\alpha_2}{\beta_2}\sin(\beta_2 t)+\cos(\beta_2 t)\right)\right]- \\ &-\xi\cdot\frac{m\cdot h\cdot b}{e(a+b)}\cdot\frac{v^2\cos\gamma}{R}\cdot\frac{\alpha_1^2+\beta_1^2}{\beta_1}\cdot e^{-\alpha_1 t}\cdot\sin(\beta_1 t) \\ &+\xi\cdot\frac{m\cdot h}{2(a+b)}\cdot\frac{v^2\sin\gamma}{R}\cdot\frac{\alpha_2^2+\beta_2^2}{\beta_2}\cdot e^{-\alpha_2 t}\cdot\sin(\beta_2 t) \end{aligned} \tag{3}$$

$$\begin{aligned} F_{2z} = &\frac{m\cdot g\cdot b}{2(a+b)} + \frac{m\cdot h\cdot b}{e(a+b)}\cdot\frac{v^2\cos\gamma}{R}\left[1-e^{-\alpha_1 t}\left(\frac{\alpha_1}{\beta_1}\sin(\beta_1 t)+\cos(\beta_1 t)\right)\right]+ \\ &\frac{m\cdot h}{2(a+b)}\cdot\frac{v^2\sin\gamma}{R}\left[1-e^{-\alpha_2 t}\left(\frac{\alpha_2}{\beta_2}\sin(\beta_2 t)+\cos(\beta_2 t)\right)\right]+ \\ &+\xi\cdot\frac{m\cdot h\cdot b}{e(a+b)}\cdot\frac{v^2\cos\gamma}{R}\cdot\frac{\alpha_1^2+\beta_1^2}{\beta_1}\cdot e^{-\alpha_1 t}\cdot\sin(\beta_1 t) \\ &+\xi\cdot\frac{m\cdot h}{2(a+b)}\cdot\frac{v^2\sin\gamma}{R}\cdot\frac{\alpha_2^2+\beta_2^2}{\beta_2}\cdot e^{-\alpha_2 t}\cdot\sin(\beta_2 t) \end{aligned} \tag{4}$$

$$\begin{aligned} F_{3z} = &\frac{m\cdot g\cdot a}{2(a+b)} - \frac{m\cdot h\cdot a}{e(a+b)}\cdot\frac{v^2\cos\gamma}{R}\left[1-e^{-\alpha_1 t}\left(\frac{\alpha_1}{\beta_1}\sin(\beta_1 t)+\cos(\beta_1 t)\right)\right]- \\ &\frac{m\cdot h}{2(a+b)}\cdot\frac{v^2\sin\gamma}{R}\left[1-e^{-\alpha_2 t}\left(\frac{\alpha_2}{\beta_2}\sin(\beta_2 t)+\cos(\beta_2 t)\right)\right] \\ &-\xi\cdot\frac{m\cdot h\cdot a}{e(a+b)}\cdot\frac{v^2\cos\gamma}{R}\cdot\frac{\alpha_1^2+\beta_1^2}{\beta_1}\cdot e^{-\alpha_1 t}\cdot\sin(\beta_1 t) \\ &-\xi\cdot\frac{m\cdot h}{2(a+b)}\cdot\frac{v^2\sin\gamma}{R}\cdot\frac{\alpha_2^2+\beta_2^2}{\beta_2}\cdot e^{-\alpha_2 t}\cdot\sin(\beta_2 t) \end{aligned} \tag{5}$$

$$\begin{aligned} F_{4z} = &\frac{m\cdot g\cdot a}{2(a+b)} + \frac{m\cdot h\cdot a}{e(a+b)}\cdot\frac{v^2\cos\gamma}{R}\left[1-e^{-\alpha_1 t}\left(\frac{\alpha_1}{\beta_1}\sin(\beta_1 t)+\cos(\beta_1 t)\right)\right]- \\ &-\frac{m\cdot h}{2(a+b)}\cdot\frac{v^2\sin\gamma}{R}\left[1-e^{-\alpha_2 t}\left(\frac{\alpha_2}{\beta_2}\sin(\beta_2 t)+\cos(\beta_2 t)\right)\right]+ \\ &+\xi\cdot\frac{m\cdot h\cdot b}{e(a+b)}\cdot\frac{v^2\cos\gamma}{R}\cdot\frac{\alpha_1^2+\beta_1^2}{\beta_1}\cdot e^{-\alpha_1 t}\cdot\sin(\beta_1 t) \\ &-\xi\cdot\frac{m\cdot h}{2(a+b)}\cdot\frac{v^2\sin\gamma}{R}\cdot\frac{\alpha_2^2+\beta_2^2}{\beta_2}\cdot e^{-\alpha_2 t}\cdot\sin(\beta_2 t) \end{aligned} \tag{6}$$

with the next notations: g is the acceleration of gravity; a – the distances between the mass centre and the front axle; b – the distances between the mass centre and the rear axle; e – the vehicle gauge; h – the centre mass height; R – the curvatures radius of the centre masses trajectory; v – the velocity of the vehicle; γ – the angle between the vehicles velocity and the vehicle longitudinal axe; α_1, β_1 – the real part, respectively the imaginary part of the characteristic equations roots of the rolling oscillations; α_2, β_2 – the real part, respectively the imaginary part of the characteristic equations roots of the pitching oscillations; ξ – coefficient who taking into account the springiness constants and the damping coefficients of the vehicles suspension;

The anterior relations permit us to determinate the vertical forces and it is observed that the vertical forces of the rear wheel internal of curvature is decreasing.

This vertical force may become null and the due velocity is given by:

$$v = R\cdot\sqrt{\frac{eg}{h\left(2\sqrt{R^2-b^2}+e\right)}} \tag{7}$$

Numerical application: J_x = 250 kgm^2; J_y = = 900 kgm^2; m=1300 kg; a = 1,15 m; b = 1,35 m; e = 1,4 m; $k_1 = k_2$ =12000 N/m; $k_3 = k_4$ =14000 N/m; $c_1 = c_2$ =1200 Ns/m; $c_3 = c_4$ =1200 Ns/m; $\xi = 0{,}1$.

The radius of curvature of circular movement is R = = 30 m, and the velocity of vehicle is 10 m/s, 12 m/s and the third case is for the 15 m/s velocity.

In the figure 1 it is presented the vertical force variation upon the time, for the front wheel internal to curvature.

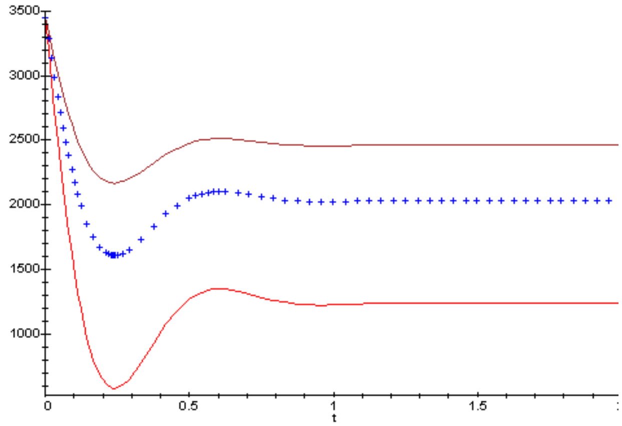

Fig. 1

In the figure 2 it is presented the vertical force variation upon the time, for the front wheel external to curvature.

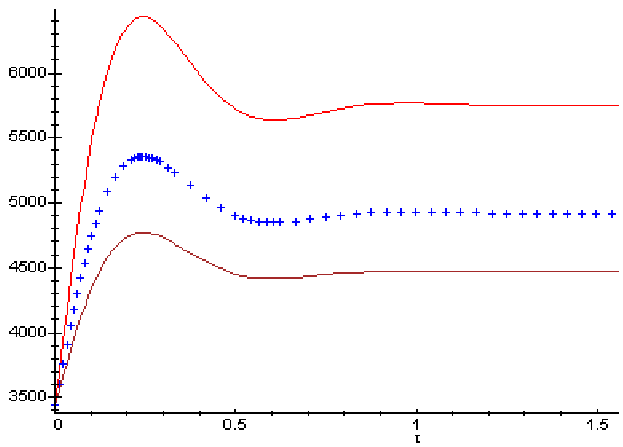

Fig. 2

In the figure 3 it is presented the vertical force variation upon the time, for the rear wheel internal to curvature.

In the figure 4 it is presented the vertical force variation upon the time, for the rear wheel external to curvature.

CONCLUSIONS

The next conclusions are deduced by analysing the anterior figures:

– during the circular movement, the wheels from external curvature are additional loading, the maximum value of loading is obtained for the front wheel.

– the wheels from internal curvature are unloading, the maximum value is obtained for the rear wheel. For velocity bigger then the critical velocity given by (7) relation the vertical force from the rear wheel internal to curvature becomes null, and the danger of side-slip appears.

– it is possible to appear the losing of contact between wheel and the road for a short of time and for a velocity smaller then critical velocity, due to oscillations of the vehicle.

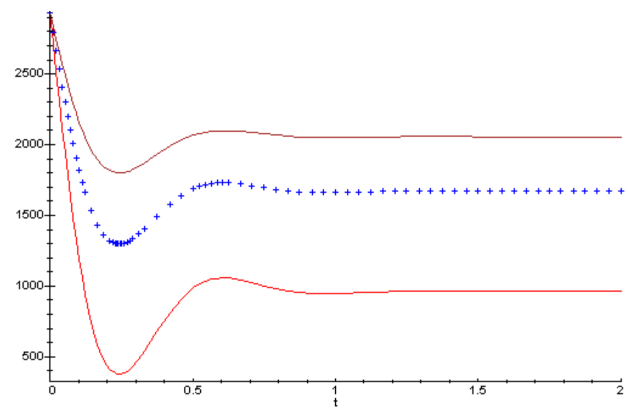

Fig. 3

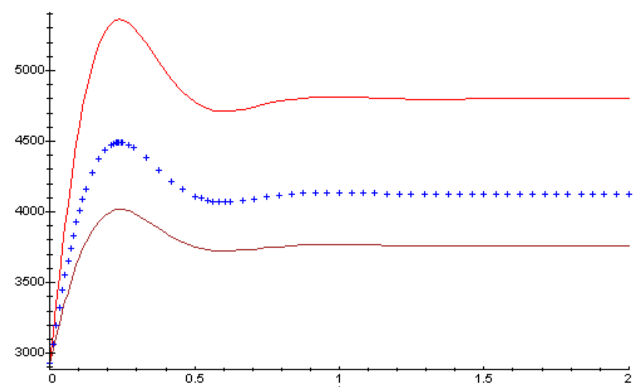

Fig. 4

REFERENCES

[1] Bolcu, Dumitru., Tică, B., Oțăt, V., *Study of variation steering angle for an auto vehicle*, World Automotive Congress, Helsinki, June, 2-7, 2002, Finland;

[2] Ghiulai, C., Vasiliu, C., *Dinamica autovehiculelor*, Editura Didactică şi Pedagogică, Bucureşti, 1975;

[3] Oțăt, V. ş.a., *Dinamica autovehiculelor*, Editura Universitaria, Craiova, 2005;

[4] Stoicescu, A., *Dinamica autovehiculelor*, vol. 1,2,3, Reprografia Institutului Politehnic, Bucureşti, 1980, 1982, 1986;

[5] Untaru, M., Pereş, Gh. ş.a., *Dinamica autovehiculelor pe roți*, Editura Didactică şi pedagogică, Bucureşti, 1981.