

THE VERTICAL VIBRATIONS STUDY OF MOTOR BOGIES FOR RAILWAY VEHICLES

V. ŞTEFAN¹ I. SEBEŞAN² C. PLEŞCAN³ L. BLAGA⁴
M. CARABINEANU⁵ G. A. BADEA⁶ G. DUMITRU⁷

Abstract: *A vehicle passing over irregularities or discontinuities tread (joints, wear undulating spaces merge two coupons welded rail, etc.) induces mechanical vibrations that are transmitted tread and vehicle accelerating phenomenon of wear. The dynamic forces occurring with these vibrations by adding them to the static values affect road safety if they exceed certain limits, so it is important to know them. A well-balanced primary suspension makes bogie accelerations received from track to be smoothed largely.*

Key words: *wear undulating, vertical acceleration, bogie, gallop.*

1. Introduction

The running track is a mechanical system whose elements deform elastically under time varying external requests, which take arise interaction with moving vehicles [1]. These elements (rail, sleepers, fastenings, ballast, substructure, etc) resiliently deform [6] and moving relative to each other during movement of vehicles [7], inducing oscillations in the wheel-rail contact that transmit both vehicle structure and track [10]. The priming vibration in the wheel-rail system occurs mainly at passing vehicles over disruptions to the running rail track and showing various defects railhead (rust, wear, wave, etc.) [9].

This paper will study the vertical dynamic overloads that occur in the bogie when travelling on a way to wear wave amplitudes and accelerations are primed using deterministic methods.

2. The Study Bogie Vibrations Bob

The railway bogie is considered a motor vehicle whose equivalent mechanical model is shown in Figure 1. having suspended bogie mass denoted by $m+$ and pass over the path defective levelling with speed v . Assuming that defects longitudinal level of the two threads are symmetrical about the center line of the track and considering that η it is a function harmonic form $\eta = \eta_0 \sin \omega t$,

¹ Railway Territorial State Inspectorate of Braşov - ASFR.

² Rolling Stok Department, *Politehnica* University of Bucharest.

³ Civil Engineering Department, *Transilvania* University of Braşov.

⁴ *Politehnica* University of Bucharest & ASFR.

⁵ *Politehnica* University of Bucharest & Romanian National Railway Safety Authority - ASFR (NSA).

⁶ *Politehnica* University of Bucharest - ASFR.

⁷ Control Traffic Safety Department, National Railway Safety Agency - ASFR.

corresponding equivalent mechanical model shown in Figure 1, the equations of motion for the bob will be given in the form (1), the disturbance to the bouncing has the expression (2).

$$\eta_s = \frac{\eta_1 + \eta_2}{2} \quad (1)$$

$$m_{rs} \ddot{q} + \rho_{rs} (\dot{q} - \dot{\eta}_s) + k_{rs} (q - \eta_s) = 0 \quad (2)$$

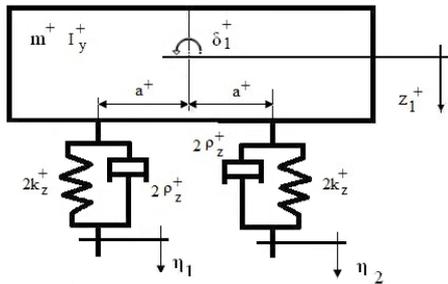


Fig. 1. Model equivalent to study mechanical vibrations of the bogie jumping movement

Given that the present rail vehicles and elastic symmetry plans mass vertical - transverse and vertical - longitudinal study of kick on forced vibration can be used adequately reduced model of Figure 2.

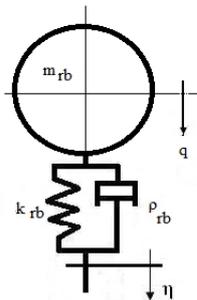


Fig. 2. Model equivalent to study mechanical vibrations of the bogie jumping movement

Correspondingly reduced model equation of motion bob, is given in the form (3).

$$\eta = h_1 \cos \frac{4\pi x}{L} = h_1 \cos 2\omega t \quad (3)$$

In Figure 2, the notation used is the reduced mass of the bogie to kick on: $m_{rs} = m^+$, reduced stiffness of the bogie to the jumped movement: $k_{rs} = 4k_z^+$, low damping coefficient for the bogie jumping motion: $\rho_{rs} = 4\rho_z^+$ and that the movement in the vertical direction of the reduced mass of the bogie: q .

Due to the interaction with the wheels of the railway vehicle tread wear [10] and forms of wear with some periodic gait [2]. The wavelengths range between 30÷300 mm and amplitude to 0,015 mm. If we denote by h_1 the undulations depth on a stretch of rail length L equation profile these irregularities [5] can be described by the relation (4).

$$\bar{\eta}_s = \eta_{0s} e^{j\omega t} \quad \text{and} \quad \bar{q} = q_0 e^{j(\omega t + \alpha)} \quad (4)$$

$$\bar{\eta}_s = \eta_{0s} e^{j\omega t} \quad \text{and} \quad \bar{q} = q_0 e^{j(\omega t + \alpha)} \quad (5)$$

Since the system excitation sinusoidal and vertical movement of the system will be sinusoidal in the complex form of these movements can be written as shown in equation (5), where η_{0s} is the amplitude of the disturbance that produces bouncing, its pulsation, q_0 is the amplitude of vertical displacement of the sprung mass of the bogie α - is the phase shift between the excitation η_s and the moving q [4].

$$q_0 (-m_{rs} \omega^2 + j\rho_{rs} \omega + k_{rs}) = \eta_{0s} (k_{rs} + j\rho_{rs} \omega) \quad (6)$$

After the derivation and replaced them expressions in (3), we obtain the equation (6). Considering the bogie an object oriented transfer function attached to equation (6) is given by the ratio between

the output function \bar{q} and the input function $\bar{\eta}_s$, resulting the relationship (7).

$$\bar{H}_q(\omega) = \frac{\bar{q}}{\bar{\eta}_s} = \frac{k_{rs} + j\rho_{rs}\omega}{k_{rs} - m_{rs}\omega^2 + j\rho_{rs}\omega} \quad (7)$$

If the equation (7) the following symbols are namely pulsation of the own

bogie $\omega_{0s} = \sqrt{\frac{k_{rs}}{m_{rs}}}$, disagreement between excitation and pulsation own

bogie $\lambda_s = \frac{\omega}{\omega_{0s}}$, to the amortization of the

bogie $D_s = \frac{\rho_{rs}}{2m_{rs}\omega_{0s}} = \frac{\rho_{rs}}{2\sqrt{k_{rs}m_{rs}}}$,

obtaining expression of the transfer function module (8), which allows the transition from q to ω .

$$H_{qs}(\omega) = \sqrt{\frac{1 + 4D_s^2\lambda_s^2}{[1 - \lambda_s^2]^2 + 4D_s^2\lambda_s^2}} \quad (8)$$

Accordingly, the phase difference between the disturbance η and the answer q of the vibrant system will be in the form (9).

$$\alpha_s = \tan^{-1} \frac{2D_s\lambda_s^3}{1 + (4D_s^2 - 1)\lambda_s^2} \quad (9)$$

The same way that we obtained relation (8), determine response relationship for acceleration factor \ddot{q} , whose expression will be given in the form (10).

$$H_{\ddot{q}s} = \frac{\ddot{q}_{0s}}{\eta_{0s}} = \omega^2 H_{qs} = \omega^2 \sqrt{\frac{1 + 4D_s^2\lambda_s^2}{[1 - \lambda_s^2]^2 + 4D_s^2\lambda_s^2}} \quad (10)$$

In order to determine the amplitude of the disturbing movement η_{0s} a two-axle bogie, it should be noted that the pulses η_1 ,

and η_2 which acts on the two axles are out of phase [4]. If the bogie wheelbase is $2a^+$, the movement velocity is v , for an harmonic amplitude η_{0k} and pulsation ω we have the expression (11). Substituting the relations (11) in (2) follows the relationship (12) amplitude disturbance η the movement of the jumping.

$$\begin{aligned} \eta_1 &= \eta_{0k} \sin \omega t \text{ and} \\ \eta_2 &= \eta_{0k} \sin \omega(t - \frac{2a^+}{v}) \end{aligned} \quad (11)$$

$$\eta_{0s} = \eta_{0k} \cos\left(\frac{\omega a^+}{v}\right) \quad (12)$$

Corresponding relation (12), amplitude perturbation depends on the jumped movement considered harmonic amplitude ratio between the product and cosine pulse time fault semi-wheelbase the bogie and the speed at which it passes over defect.

2.1. The Study of Bogies Vibration Gallop

Gallop movements of the bogies due to the excitation received from the wheel-track system, may overlap with the flexural vibration of the vehicle box, and if the oscillation frequencies of the two motions are similar in certain conditions can lead to the movement of the resonance [4]. To avoid these situations, we have determined the main parameters that influence the movement of the vehicle and set conditions that gave resonance occurs.

Under the assumption that the bogie is symmetrical (both geometrically and in terms of the distribution of masses, stiffness and depreciation) against plans longitudinal and transverse pass vertically through the centre of mass of the bogie, we have a decoupling of movement saltation vertical movement of the bogie frame and its gallop [2].

Considering further that defects longitudinal level of the two-wire path are symmetrical with respect to the central axis of the track, and that η is a function harmonic form $\eta = \eta_0 \sin \omega t$, reduced accordingly equivalent mechanical model shown in Figure 3 equations of motion will be played gallop form (13).

$$I_y^+ \ddot{\delta}_1^+ + 2\rho_z^+ a^+ (a^+ \dot{\delta}_1^+ - \dot{\eta}_1) + 2\rho_z^+ a^+ (a^+ \delta_1^+ + \eta_2) + 2k_z^+ a^+ (a^+ \delta_1^+ - \eta_1) + 2k_z^+ a^+ (a^+ \delta_1^+ + \eta_2) = 0 \quad (13)$$

Substituting in (13) the moment of inertia I_y , suspended parts of the bogie in relation to the y axis, corresponding to the radius of gyration determined by the relationship (14) can be written as (14).

$$\frac{m^+}{2} \left(\frac{i_y^+}{\alpha}\right)^2 (a\ddot{\delta}) + 2\rho_z^+ [(a\dot{\delta}) - \frac{\dot{\eta}_1 - \dot{\eta}_2}{2}] + 2k_z^+ [(a\delta) - \frac{\eta_1 - \eta_2}{2}] = 0 \quad (14)$$

From equation (14) and to study the vibrations that forced gallop reduced model can be used in Figure 4, where the identification, we will track the disturbance that produces motion gallop [5]: $\eta_g = \frac{\eta_1 - \eta_2}{2}$, the reduced mass of the bogie for movement gallop: $m_{rg} = \frac{m^+}{2} \left(\frac{i_y^+}{\alpha}\right)^2$, reduced stiffness of the bogie: $k_{rg} = 2k_z^+$, the low damping coefficient for the bogie: $\rho_{rg} = 2\rho_z^+$, the displacement in the vertical direction of the reduced mass of the bogie movement gallop: $q = a^+ \delta$.

With these notations, equation (14) can be written as (15) and (16) using the following notations for pulsation own bogie for movement gallop:

$$\omega_{0g} = \sqrt{\frac{k_{rg}}{m_{rg}}}$$

the difference between the pulsating excitement that produces its own motion gallop and pulsation bogie [2]:

$$\lambda_g = \frac{\omega}{\omega_{0g}}$$

and that the degree of damping of the bogie for movement gallop:

$$D_g = \frac{\rho_{rg}}{2m_{rg}\omega_{0g}} = \frac{\rho_{rg}}{2\sqrt{k_{rg}m_{rg}}}$$

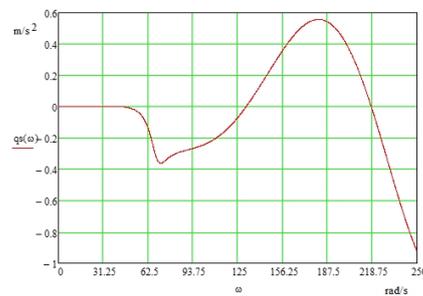


Fig. 3. The variation the vibration acceleration jumping movement $qs(\omega)$, excitation depending on the defect path

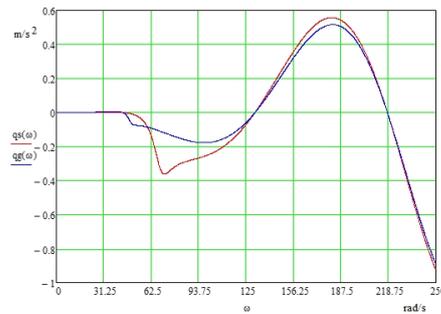


Fig. 4. The variation the vibration acceleration jumping motion $qs(\omega)$ and vibration acceleration gallop $qg(\omega)$, excitation depending on the defect path

The gallop bogie movement is sinusoidal, can be written in complex form (17) where η_{0g} It is the amplitude of the disturbance that produces gallop, pulsation of this movement, q_0 represents the amplitude of vertical displacement of the sprung mass of the bogie α - the phase

shift between the excitation η_g and moving displacement q .

$$\bar{\eta}_g = \eta_{0g} e^{j\omega t} \text{ and } \bar{q} = q_0 e^{j(\omega t + \alpha)} \quad (15)$$

$$\bar{H}_{qg}(\omega) = \frac{\bar{q}}{\bar{\eta}_g} = \frac{1 + j2D_g \lambda_g}{1 - \lambda_g^2 + j2D_g \lambda_g} \quad (16)$$

$$H_{qg}(\omega) = \sqrt{\frac{1 + 4D_g^2 \lambda_g^2}{[1 - \lambda_g^2]^2 + 4D_g^2 \lambda_g^2}} \quad (17)$$

After the derivation expressions (17) and replaced them in equation (16) gives the response factor (18) of the gallop movement. The module of the transfer function, which allows switching from q to ω , is determined by the relation (19).

$$\alpha_g = \tan^{-1} \frac{2D_g \lambda_g^2}{1 + (4D_g^2 - 1)\lambda_g^2} \quad (18)$$

$$H_{\ddot{q}g} = \frac{\ddot{q}_{0g}}{\eta_{0g}} = \omega^2 H_{qg} = \omega^2 \sqrt{\frac{1 + 4D_g^2 \lambda_g^2}{[1 - \lambda_g^2]^2 + 4D_g^2 \lambda_g^2}} \quad (19)$$

Accordingly, the phase difference between the disturbance η_g and the answer q of the system will be played rousing form (20). The same way that we obtained relation (19), determining factor module response time acceleration gallop [5], this will be the expression (21).

$$\begin{aligned} \eta_1 &= \eta_{0k} \sin \omega t & \text{and} \\ \eta_2 &= \eta_{0k} \sin \left(t - \frac{2a^+}{v} \right) \end{aligned} \quad (20)$$

$$\eta_{0g} = \frac{\eta_1 - \eta_2}{2} = \eta_{0k} \sin \left(\frac{\omega a^+}{v} \right) \quad (21)$$

The amplitude η_{0g} of the disturbance that produces gallop, is determined taking into

account the pulsations η_1 and η_2 the two axles are out of phase, the vehicle wheelbase $2a^+$ the forward speed v , for an harmonic amplitude η_{0k} and pulsation ω , can be written as (22). After replacing the disturbance in relation to gallop apparent relationship (23).

$$\ddot{q}_{0g} = \eta_{0g} \omega^2 \sqrt{\frac{1 + 4D_g^2 \lambda_g^2}{[1 - \lambda_g^2]^2 + 4D_g^2 \lambda_g^2}} \quad (22)$$

$$\ddot{q}_{0g} = \omega^2 \left[\sqrt{\frac{1 + 4D_g^2 \lambda_g^2}{[1 - \lambda_g^2]^2 + 4D_g^2 \lambda_g^2}} \right] \eta_{0k} \cos \left(\frac{\omega a^+}{v} \right) \quad (23)$$

As in the case of the jumping movement, perturbation amplitude galloping movement depends amplitudes harmonic sense considered and pulsation ratio of product defect with semi-wheelbase date the bogie and the speed with which it passes over defect [2].

3. The Determination of Parties Suspended Bogie Accelerations, Vibrations Jumping Products and Gallop

The amplitude the acceleration produced by the parties suspended bogie of the bouncing motion with viscous damping can be determined from equation (10), the replacement of variable can play in form (24).

$$\ddot{q}_{0g} = \eta_{0g} \omega^2 \sqrt{\frac{1 + 4D_g^2 \lambda_g^2}{[1 - \lambda_g^2]^2 + 4D_g^2 \lambda_g^2}} \quad (24)$$

The amplitude the acceleration produced by the parties suspended bogie bouncing motion with viscous damping can be determined from equation (10), replacement variables that can play into shape (25).

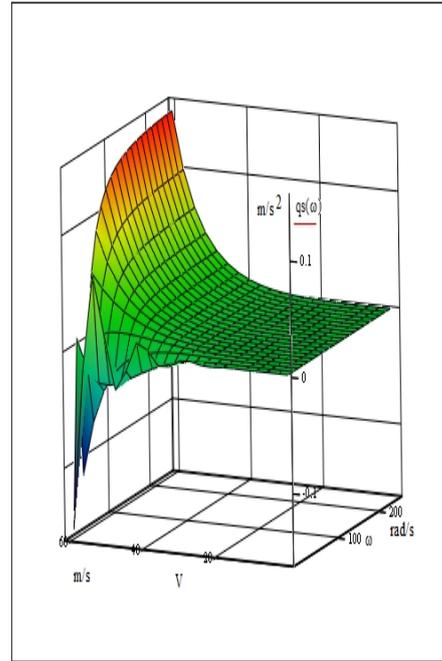
$$\ddot{q}_{0g} = \omega^2 \left[\sqrt{\frac{1+4D_g^2\lambda_g^2}{[1-\lambda_g^2]^2+4D_g^2\lambda_g^2}} \right] \eta_{0k} \sin\left(\frac{\omega a^+}{v}\right) \quad (25)$$

Likewise the results in the relation of the amplitude of acceleration of the bogie suspension parts produced by the movement of galloping shown in equation (23) or replace the amplitude of the corresponding relationship given by equation (23) we obtain the final shape (22) amplitude. From relations (24) and (25) that the vertical accelerations of the bogie jumped and the gallop depend on the elastic parameters of the suspension damping and especially the amortization. The moving at high speed bogies of a vehicle over track defects at the wavelengths small and large amplitudes can produce accelerations unacceptable system [8].

3.1. Practical Application

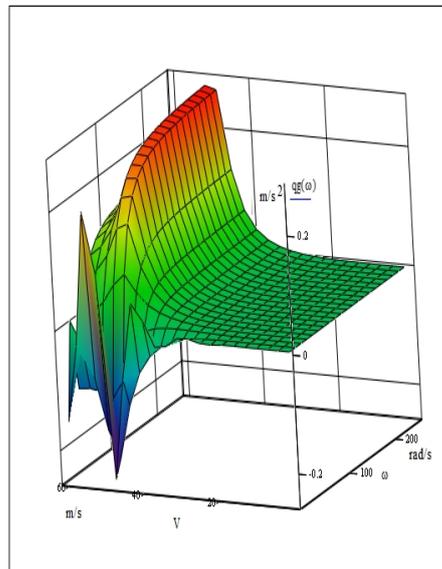
It is considered elastic system consists of a rail bogie motor vehicle, having primary suspension and amortization on each of the two axles, horny similar path wears uneven vertical wave [5]. The study intends to system parameters in Table. 1 to determine accelerations arising from a faulty level crossing whose profile is known equation.

The graph accelerations due to the bogie vibration jumping recorded travelling at speed 27.77 m/s over an area of undulating wear of the rail, (Figure 5) shows that up to a value of the pulsation excited about 45 rad/s, there are no significant values [3].



Qs

Fig. 5. The variation the vibration acceleration jumping motion $q_s(\omega)$ versus excitation path and velocity the defect



Qg

Fig. 6. The variation the vibration acceleration jumping movement $q_s(\omega)$ versus excitation path and velocity the defect

The elastic parameters of the bogie system

Table 1

Parameters Name	Symbols	Values	Unit of measure
unsparing mass of the axle	m_0	1600	Kg
vertical stiffness of the primary suspension	k_z^+	3.164×10^6	N/m
vertical suspension damping coefficient primary	ρ_z^+	7.8×10^3	Ns/m
pulsation suspended bogie own mass to jumping motion	ω_{0z}	68.46	rad/s
frequency of the sprung mass of the bogie to jumping motion	ν_{0z}	10.9	Hz
suspended mass of a bogie	m^+	2700	Kg
reduced mass of the bogie to jumping motion	m_{rs}	1600	Kg
radius of turn of the bogie relative to the axis y	i_y^+	1.4	m
low bogie mass gallop	m_{rg}	2646	Kg
semi-wheelbase the bogie	a^+	1	m
pulsation own bogie for movement sprung mass gallop	ω_{0g}	48.9	rad/s
frequency of the sprung mass of the bogie to gallop	ν_{0g}	7.78	Hz
forward speed of the vehicle	v	27.77	m/s
amplitude undulations	h_1	15×10^{-6}	m
the wavelength of the defect	Λ	2	m
the degree of damping of the suspension jumping motion primary	D_s	0.084	-
the degree of damping of the suspension gallop primary	D_g	0.06	-

Around bogie jump oscillation its own frequency is recorded maximum accelerations then reverses and dangerous increase with increasing speed and / or decreasing wavelength defect.

Comparing graphics acceleration due to both vibration and vibration gallop jumping the bogie recorded travelling at speed 27.77 m/s over an area of undulating wear of the rail (Figure 6) shows that there is a value and of the pulse galloping excited about 45 rad/s, to which the accelerations are not as significant.

The gallop frequency is determined around the bogie is recorded maximum accelerations then reverses and a similar pattern to those of the jumped movement [2]. The movement of railcar thought over this defect can be done safely up to speed 44.44 m/s, this being the maximum allowed speed bus traffic on railway lines from our country. Furthermore, in the graph 3D of Figure 5 and Figure 6 there may be same.

4. Conclusions

The comfort criterion variable vertical loading vehicles is essential, damping coefficients corresponding to the two floors of the suspension being adopted in the majority of cases this criterion.

The study vibrations and the bogie gallop jumping are revealed the dependence vertical accelerations of two important parameters and elastic degree of damping suspension axles and pulsation excitation received from track.

The speed circulation greater than 40 m/s on track with undulating who wear large amplitudes and small wavelengths, can produce unwanted acceleration in the bogie frame, which transmit the box and which affects both comfort and road safety.

Acknowledgements

The work has been funded by the Sectorial Operational Programme Human Resources Development 2007-

2013 of the Ministry of European Funds through the Financial Agreement POSDRU / 159 / 1.5 / S / 132395.

References

1. Dósa, A, Ungureanu, V. V. - *SCFJ – model discret de pierdere a stabilității căii fără joante (SCFJ – discrete model for continuous welded rail buckling)*, “Timisene Academical Days: 10th Edition, Timisoara, Romania, 24th –25th of May 2007 - Symposium: Efficient infrastructure for the terrestrial transport” – Solness Publishing House, Timisoara, ISBN 978-973-729-101-1.
2. Dumitru, G. *Observations on some aspects related to dynamic of railway vehicle engine*, MID-CF Magazine, no. 1/2008.
3. Kalker, J. J., *A strip theory for rolling with slip and spin*, Proceedings Kom. Akad. Wet., Amsterdam, Section B. 70, 1967.
4. Mazilu, T., *Vibrații (Vibrations)*, Matrix Rom Publishing House, Bucharest, 2012.
5. Sebeșan, I., Mazilu, T., *Vibrațiile vehiculelor feroviare (Vibrations of the railway vehicles)*, Matrix Rom Publishing House, Bucharest, 2010.
6. Ungureanu, V. V., Comănici, M. - *Sensitivity study of a model for the stability analysis of continuous welded rail*, Intersections / Intersecții, Vol.4, 2007, No.1, “Transportation Infrastructure Engineering”, ISSN 1582 – 3024.
7. Ungureanu, V. V., Dósa, A. - *Parametrical study of the effect of the torsional resistance of the fastenings on the stability of continuous welded rail*, Intersections / Intersecții, Vol.4, 2007, No.1, “Transportation Infrastructure Engineering”, ISSN 1582 – 3024.
8. Ungureanu, V. V. - *On Elimination of Interior Rail Joints and the Including of Welded Railway Switches in Continuous Welded Rail Track* (2009), SUSTAINABILITY in SCIENCE ENGINEERING, Volume II, Proceedings of the 11th WSEAS International Conference on Sustainability in Science Engineering (SSE '09), Timisoara, Romania, 27th – 29th of May, 2009, ISSN: 1790-2769, ISBN: 978-960-474-080-2, WSEAS Press.
9. Ungureanu, V. V., Dósa, A. - *Algoritm pentru determinarea probabilității de pierdere a stabilității cadrului șine - traverse (Algorithm for determining the probability of continuous welded rail track buckling)*, Proceedings of Scientific Conference Civil Engineering – Building Services CIB 2007, 15th – 16th of November, Brașov, România, Transilvania University Publishing House – Brașov 2007, ISSN 1843-6617.
10. Ungureanu, V. V. - *Cercetări privind simularea pierderii stabilității căii fără joante (Research about simulation of continuous welded rail buckling)*, Ph.D. Thesis, Transilvania University of Brașov, Brașov, Romania, 2007.