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THE INFLUENCE OF DAMPING CHARACTERISTIC ON THE STABILIZATION CONTROL OF HUNTING MOTION OF A RAILWAY VEHICLE WHEELSET

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In this paper is studied the influence of passive linear and non-linear dissipative horizontal forces on the hunting motion stability of a wheelset with elastic joints. The results of numerical simulations show that use of conventional hydraulic dampers with linear or nonlinear viscous damping characteristics can lead to a certain increase of critical speed. The dependence of critical speed on pseudosliding (creep) coefficients, associated with these type of damping, is a serious drawback since the values of these coefficients cannot be accurately predicted. By using a special type of passive damping characteristics, dependent on both relative velocity and displacement between wheelset and bogy frame, the hunting motion can be stabilized for very large values of vehicle speed. Moreover, in this case the wheel-track interaction forces are significantly lower than in the case when conventional viscous dampers are used for hunting motion control.

Key words: Hunting motion, Passive stabilization, Pseudosliding, Damping.

1. INTRODUCTION

The hunting motion occurring in case of the railway vehicles is a consequence of the combined effect of reversed conic shape of the wheel rolling surfaces and the non-conservative contact force [1-5]. Below a certain vehicle riding speed, called the critical speed, the hunting motion appears as a damped sinusoidal oscillation along the track centreline. Above this speed the motion can be violent, damaging track and wheels, and potentially causing derailment. As the speed increases, the wheels- track contact force becomes large enough to cause rail damage, discomfort and eventually can lead to derailment. Therefore, the stability of hunting motion is an important dynamic problem that determines the maximum operating speed of railway vehicles.

Particularly, for the high-speed passenger trains, the problem of achieving high-speed operation without the hunting instability has always been of interest to vehicle designers [6]. The effect of primary suspension or the effect of lateral linear stiffness on the hunting stability of a rail wheelset has been investigated in [7, 8]. An analytical investigation of Hopf bifurcation and hunting behavior of a rail wheelset with nonlinear primary yaw dampers and wheel-rail contact forces is presented in [2]. Passive stabilization of the amplitude of self-oscillations or elimination of self-oscillations by appropriately selecting the parameters of the tread contour has been studied in [9, 10]. A stabilization method for the hunting motion with a gyroscopic damper is proposed in [11] to increase the critical speed. Semi-active control strategies with magnetorheological dampers where applied to control the hunting motion of the locomotive body [12].

In this paper is analyzed comparatively the effect of passive control of hunting motion stabilization, by using hydraulic shock absorbers with linear and nonlinear damping characteristics, working in parallel with the springs of a wheelset with elastic joints [13, 14]. The numerical simulation results, obtained for a simple wheelset model with two-degrees-of-freedom with respect to the lateral and yawing motions, outline the advantages of using displacement-dependent damping characteristics to control hunting motion by passive methods.

2. MECHANICAL AND ANALYTICAL MODELS OF WHEELSET HUNTING MOTION

The wheelset is modelled by an oscillating system with two degrees of freedom with respect to the lateral and yawing motions. The hunting motion is studied with respect to an inertial system of reference, which moves with a constant velocity along the track centreline. The mechanical models of wheelset having elastic joints without and with dampers in parallel connections are shown in Figs. 1 and 2. The bogy frame is assumed to be fixed with respect to the reference system.

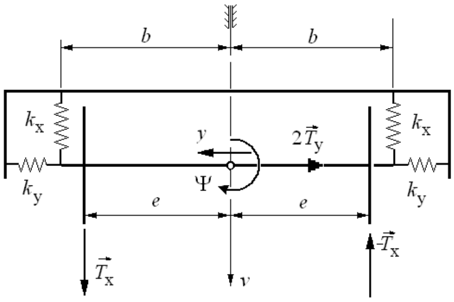

Fig. 1 – The mechanical model of wheelset with elastic joints.

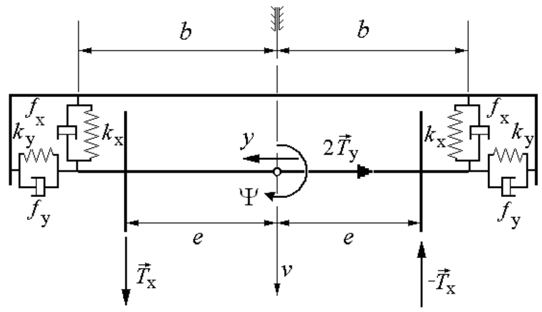

Fig. 2 – The mechanical model of wheelset with elastic joints and dampers in parallel connections.

In these figures, y is the lateral deviation of the wheelset center of inertia from the track centerline, Ψ is the axle yaw angle relative to track.

For the wheelset model shown in Fig. 1, the creep linear forces in lateral and longitudinal direction T_x, T_y are given by [1]

$$T_y = \chi_y Q\left[\left(\frac{\dot{y}}{v}\right) - \Psi\right],\ T_x = \chi_x Q\left[\left(\frac{\lambda}{r}\right)y + \left(\frac{e}{v}\right)\dot{\Psi}\right] \tag{1}$$

where χ_y and χ_x are the lateral and longitudinal creep force coefficients (pseudosliding coefficients), λ is the wheel conicity and Q is the axle load. Assuming equal values for lateral and longitudinal pseudosliding coefficients ($\chi_y = \chi_x = \chi$) and neglecting the spin creep force, the equations of motion for the oscillating system shown in fig.1 can be written as [1].

$$m_0\ddot{y} + \frac{2\chi Q}{v}\dot{y} + 2k_y y - 2\chi Q\Psi = 0, \qquad I_{0z}\ddot{\Psi} + 2\chi Q\frac{e^2}{v}\dot{\Psi} + 2b^2 k_x \Psi + 2e\chi Q\frac{\lambda}{r}y = 0, \tag{2}$$

where $I_{0z} = m_0 e^2$ is the wheelset central moment of inertia and r is the nominal wheelset rolling radius.

For the wheelset model shown in Fig. 2, the equations of motion are of the form

$$m_0\ddot{y} + 2F_y + \frac{2\chi Q}{v}\dot{y} + 2k_y y - 2\chi Q\Psi = 0, \quad I_{0z}\ddot{\Psi} + 2bF_x + 2\chi Q\frac{e^2}{v}\dot{\Psi} + 2b^2 k_x\Psi + 2e\chi Q\frac{\lambda}{r}y = 0. \tag{3}$$

The damping characteristics, considered in this paper, have the general analytical form

$$\begin{aligned} F_x(x,\dot{x}) &= c_x\dot{x} + a_x|x|^{\alpha}[1+\varepsilon\cdot\operatorname{sgn}(x\dot{x})]\cdot|\dot{x}|^{\beta}\operatorname{sgn}(\dot{x}), \\ F_y(y,\dot{y}) &= c_y\dot{y} + a_y|y|^{\alpha}[1+\varepsilon\cdot\operatorname{sgn}(y\dot{y})]\cdot|\dot{y}|^{\beta}\operatorname{sgn}(\dot{y}), \end{aligned} \tag{4}$$

which can portray both conventional viscous damping (dependent only on the relative velocity) and a class of physically realizable dissipative characteristics, dependent on both relative displacement and velocity. Thus, for $a_x = a_y = 0$, (5) yields linear damping characteristics:

$$F_x(\dot{x}) = c_x \dot{x}, \quad F_y(\dot{y}) = c_y \dot{y}, \tag{5}$$

and for $c_x = 0,\ c_y = 0,\ \alpha = 0,\ \varepsilon = 0$ one obtains viscous damping governed by a nonlinear viscous power law:

$$F_x(\dot{x}) = a_x |\dot{x}|^{\beta} \operatorname{sgn}(\dot{x}),\ F_y(y, \dot{y}) = a_y |\dot{y}|^{\beta} \operatorname{sgn}(\dot{y}). \tag{6}$$

For $\varepsilon = 0$, equation (4) yields damping characteristics dependent on both relative displacement and velocity. Figure 3 depicts schematically a device, which can provide displacement-dependent dissipative characteristics.

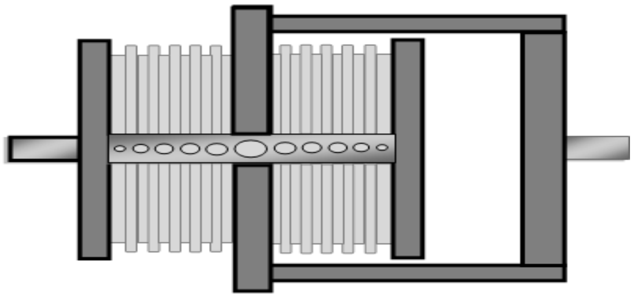

Fig. 3 – Hydraulic nonlinear devices with metal bellows.

This device is a monotubular hydraulic damper with two metallic bellows filled with fluid and separated by a rigid plate, which can slide along a fixed tube with calibrated orifices. The fluid can flow from one chamber to the other through the calibrated orifices, due to equal absolute variation of their volume for tension or compression strokes. For elastic bellows with appropriate length, the strains produced by relatively small strokes of the stable haunting motion (generally, less than 10^{-2} m) can be kept within the safety limits of fatigue life. Their diameter must be sufficiently large to provide the fluid flow through the calibrated orifices such as the required damping forces to be developed at safe pressure values.

The dissipated energy per cycle by a device with a damping characteristic of the form (4), for an imposed harmonic displacement $y(t) = y_0 \sin \omega t$, is given by

$$E_{\mathrm{d}} = \oint F_y(y, \dot{y})\,\mathrm{d}y = \pi c_y \omega y_0^2 + (2 - \varepsilon) a_y y_0^{\alpha+\beta+1} \omega^{\beta} \frac{\beta \cdot \Gamma((\alpha+1)/2) \cdot \Gamma(\beta/2)}{(\alpha+\beta+1) \cdot \Gamma((\alpha+\beta+1)/2)}. \tag{7}$$

3. NUMERICAL SIMULATION RESULTS

The numerical simulations were conducted to compare the effect of different types of damping characteristics on the stabilization of hunting motion of a wheelset with elastic joints.

The simulation parameters are:

- Geometric parameters: $b = 1\text{m},\ e = 0.75\text{m},\ r = 0.46\text{m},\ \lambda = 0.13$
- Inertial parameters: $m_0 = 1{,}500\,\text{kg}$, $I_{z0} = 874\text{Nm}^2$
- Material parameters: $E = 210\ \text{kN/mm}^2$, $\vartheta = 0{,}3$
- Axle load: $Q = 75{,}000$ kN
- The range of all possible values of pseudosliding coefficient: $50 \le \chi \le 400$
- Stiffness parameters: $k_x = 9 \cdot 10^5$ N/m, $k_y = 5.43 \cdot 10^5$ N/m
- Damping parameters for linear viscous damping: $c_x = c_y = 10{,}000\,\text{Ns/m}$, $a_x = a_y = 0$

- Damping parameters for viscous damping governed by a nonlinear viscous power law: $c_x = c_y = 0,\ a_x = a_y = 10^5\ \mathrm{Ns^2/m^3},\ \alpha = 1,\ \beta = 2,\ \varepsilon = 0$
- Damping parameters for displacement-dependent dissipative characteristics: $c_x = c_y = 500,\ a_x = a_y = 6\times 10^8\ \mathrm{Ns^2/m^3},\ \alpha = 1.1,\ \beta = 2,\ \varepsilon = 1$
- Initial conditions for systems of differential equations (2) and (3): $y(0) = 0.005\mathrm{m},\ \dot{y}(0) = 0;\ \Psi(0) = \dot{\Psi}(0) = 0.$

From equations (2), one can obtain the critical pulsation and the critical speed of the hunting motion [1] for the wheelset with elastic joints without dampers:

$$\omega_c = \sqrt{\frac{k_x}{m_0}\frac{b^2}{e^2} + \frac{k_y}{m_0}},\ v_c = \omega_c \left(\sqrt{\frac{\lambda}{er} - \omega_c^4 \left(\frac{m_0}{2\chi Q} \right)^2} \right)^{-1} \cong \omega_c \sqrt{\frac{er}{\lambda}}\ . \tag{8}$$

In this paper, the critical speed for hunting motion of wheelset with elastic joints without damping or with viscous linear damping was determined by numerical integration of system (3) as the riding speed v_c for which the motion amplitude $y(t)$ is constant when $t \to \infty$ ($\lim_{t\to\infty} y(t) = y_c < \delta$). For lower speed values $v < v_c$ the equilibrium position is asymptotic stable ($\lim_{t\to\infty} y(t) \to 0$) while for higher speed values $v > v_c$ the motion is unstable ($\lim_{t\to\infty} y(t) \to \infty$). In the case of nonlinear damping governed by power law, the equilibrium position is a centre, but the amplitudes of stabilized motion can become too large if the riding speed is higher than a certain value, which in this work is called also the critical speed. Figures 4-6 illustrate this dynamic behaviour for $\chi = 100$.

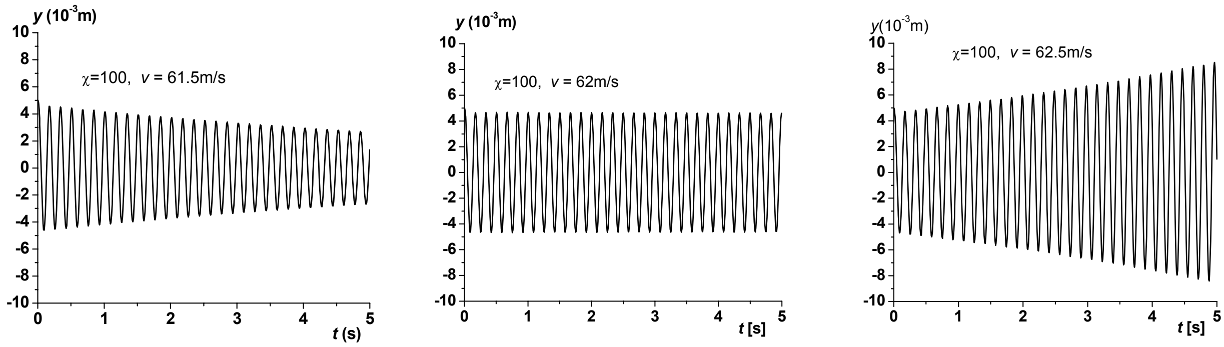

Fig. 4 – The dynamic behaviour of the simple wheelset without damping.

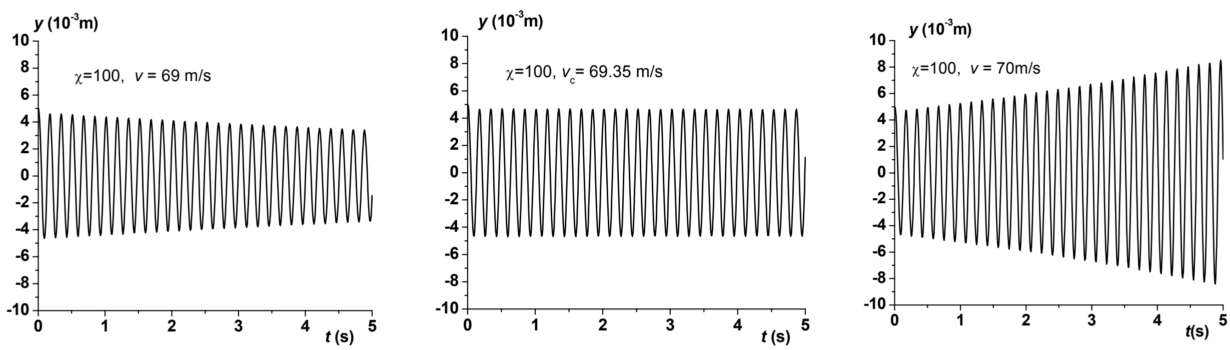

Fig. 5 – The dynamic behaviour of the wheelset for linear viscous damping.

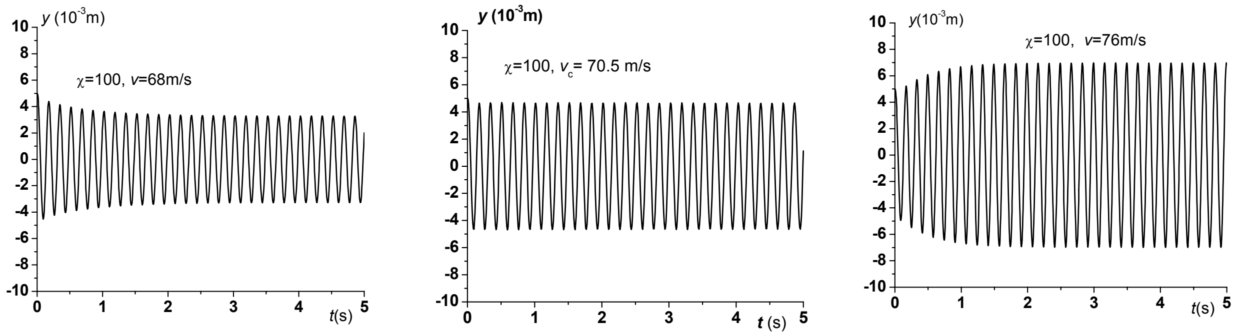

Fig. 6 – The dynamic behaviour of the wheelset for viscous damping governed by a nonlinear viscous power law.

A substantial improvement of hunting motion stability of a wheelset with elastic joints is obtained by using dampers with displacement-dependent hydraulic characteristics. In this case, the instability of hunting is unlikely to occur even for very high riding speeds. This assertion is illustrated by the time histories of hunting motion determined for a high value of vehicle speed ($v = 83.3\text{m/s} = 300\text{km/h}$) and three values of pseudosliding coefficient covering the interval of its possible values (Fig. 7).

The amplitude spectra plots, presented in Figs. 8–11, show the influence of damping characteristic on the hunting motion frequency for $\chi = 50,\ 100,\ 400$.

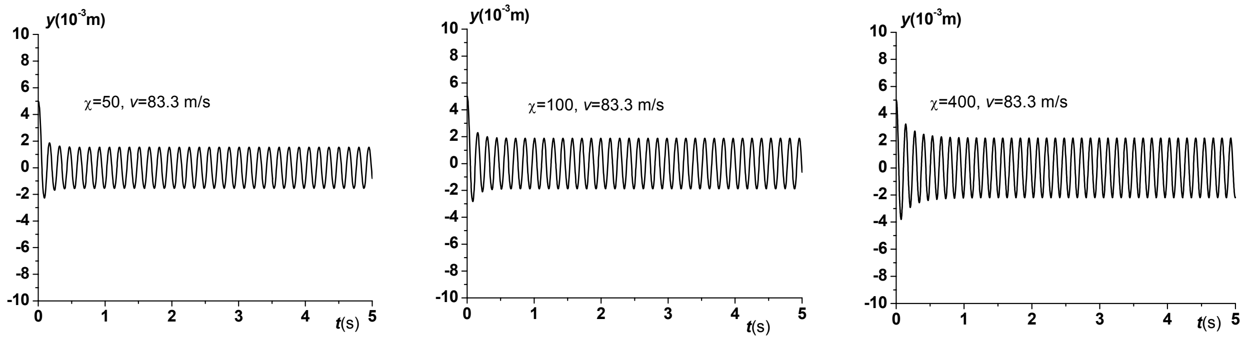

Fig. 7– The dynamic behaviour of the wheelset for displacement-dependent dissipative characteristics.

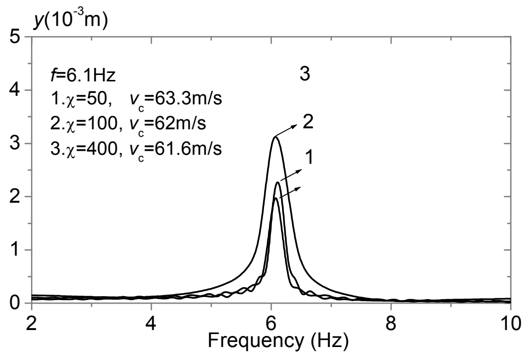

Fig. 8 – Amplitude spectra of hunting motion for wheelset without damping.

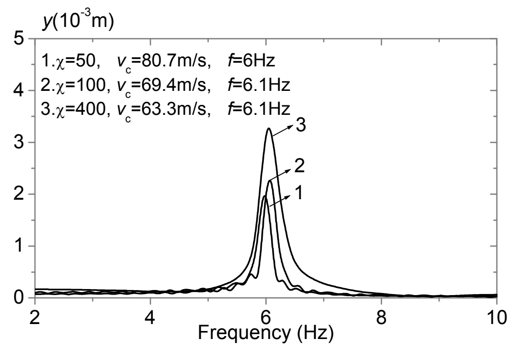

Fig. 9 – Amplitude spectra of hunting motion with linear viscous damping.

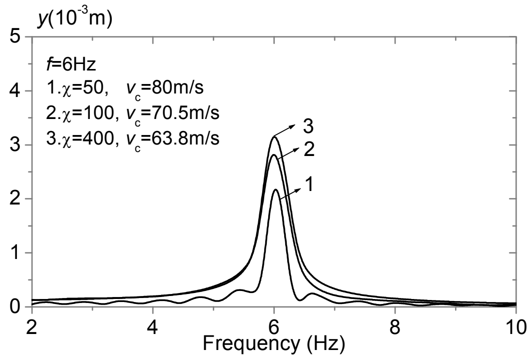

Fig. 10 – Amplitude spectra of hunting motion for wheelset with viscous damping governed by a nonlinear viscous power law.

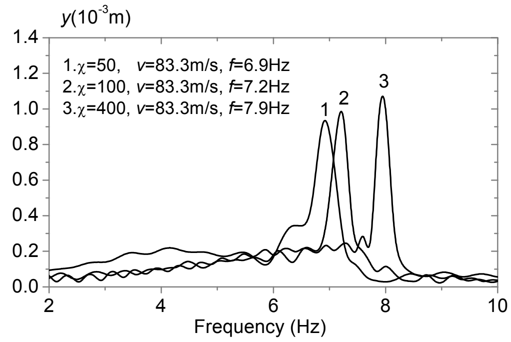

Fig. 11 – Amplitude spectra of hunting motion for wheelset with displacement-dependent dissipative characteristics.

The elastic effect of the displacement-dependent damping characteristic is obvious from the fact that the hunting motion frequencies are higher and more dependent on the sliding coefficient values then in the case of conventional viscous damping characteristics. The phase plane orbits, plotted in Fig. 12 for same initial conditions ($y(0)=0.005\text{m},\ \dot{y}(0)=0.05\text{m/s}, \Psi(0)=0.005\text{rad},\ \Psi(0)=0.05\text{rad/s}$), outline once again the superiority of the stabilization potential of displacement-dependent viscous damping characteristic.

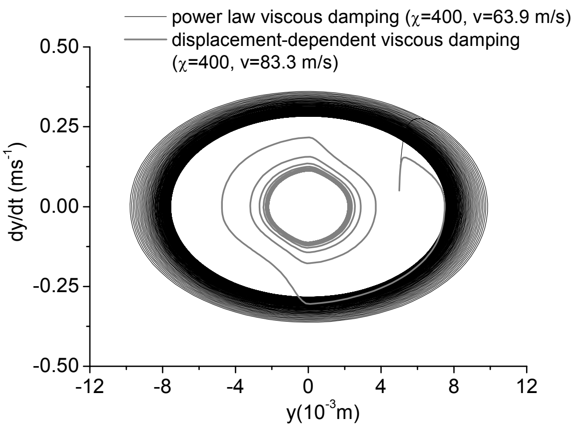

Fig. 12 – The phase plane orbits for linear viscous damping and for displacement depending viscous damping.

Figure 13 show the force–displacement plots of the considered damping laws obtained for the stable regimes of the hunting motion for $\chi = 50, 100, 400$.

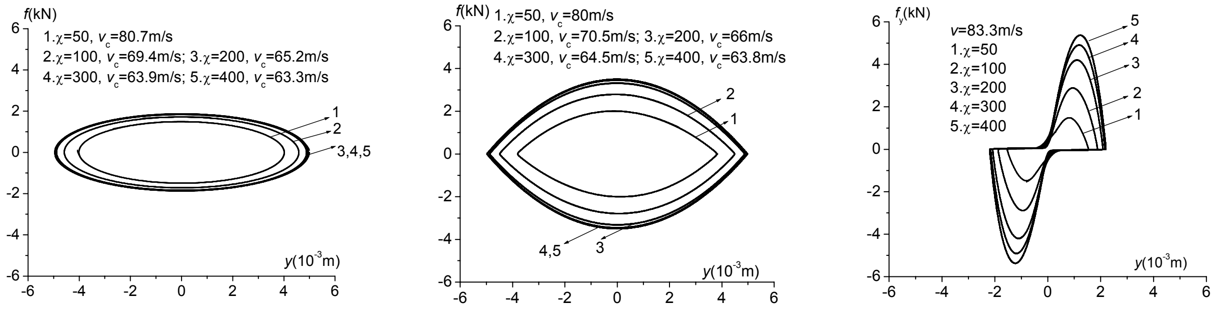

Fig. 13 – The force-displacement plots of the considered damping laws for the stable regimes of the hunting motion: a) linear viscous damping; b) viscous damping governed by a nonlinear viscous power law; c) displacement-dependent dissipative characteristics.

The variation of dissipated energy per cycle versus pseudosliding coefficient, obtained from (7) for the stabilized hunting motion (i.e. the areas of the above hysteretic loops), is presented in Fig. 14.

The variation of critical speed v_c and of maximum amplitudes of transversal force Y *versus* pseudosliding coefficient χ, are shown comparatively in Figs. 14 and 15.

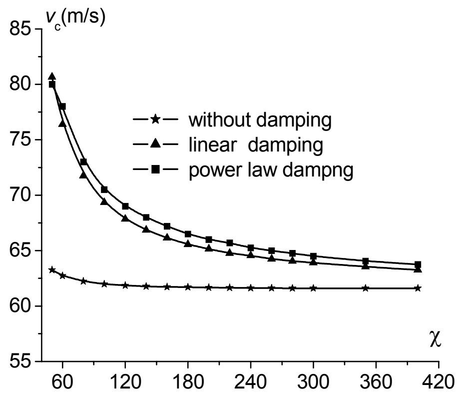

Fig. 14 – The variation of critical speed for stabile regimes of the hunting motion.

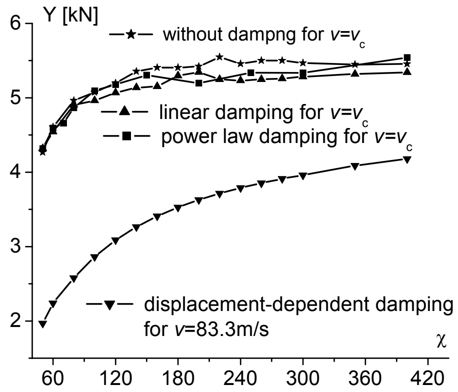

Fig. 15 – The variation of the maximum amplitude of transversal force for stabile regimes of the hunting motion.

These plots show that the critical speed is practically independent of pseudosliding coefficient for wheelset without dampers. However, in this case the critical speed is too low for normal operating conditions of a high speed railway vehicle. When dampers with hydraulic characteristics governed by linear or nonlinear viscous power law are used, the critical speed can be increased, but becomes strongly dependent on the pseudosliding coefficient, particularly for $\chi \leq 200$. Because the value of sliding coefficient is practically impossible to be accurately predicted, this dependence is a serious drawback of using conventional viscous dampers.

A substantial improvement of hunting motion stability of wheelset with elastic joints is obtained by using dampers having displacement-dependent hydraulic characteristics. In this case, as said before, the instability of hunting is unlikely to occur even for very high riding speeds and all possible values of pseudosliding coefficient. Therefore, from the stability point of view, the riding speed is not limited by a critical value but only by the requirements of keeping the transversal displacement $y(t)$ and force $Y(t)$ within the safety limits. The results plotted in Figs. 5-18 highlight the stabilization potential of the displacement-dependent damping characteristics. This type of damping allows higher riding speed with lower amplitudes of hunting motion and creep forces for less dissipated energy per cycle.

It should be pointed out that the improvement of hunting motion stability was obtained by a passive control system, which is much cheaper and easier to be practically implemented than semi-active or active control systems.

4. CONCLUSIONS

In this study, a passive stabilization control method is proposed for the hunting motion in a simple wheelset model, i.e. a fundamental two-degrees-of-freedom model with respect to the lateral and yawing motions of the wheelset with elastic joints. By numerical simulations, it is shown that a non–conventional viscous damper with displacement-dependent dissipative characteristic is very effective in increasing the railway vehicles speed by mitigation of hunting motion. The benefits of using this type of damping are highlighted by a comparative analysis with the effect of using conventional hydraulic dampers in order to control the hunting motion stabilization. The main benefits of the proposed passive control are:

- the instability of hunting motion is unlikely to occur up to very high riding speeds;
- the vehicle speed is not limited by a critical speed but only by the safety requirements imposed to the stabilized hunting motion in terms of the transversal displacements and forces;
- the dependence of maximum allowed riding speed on the values of pseudosliding coefficients is unimportant from practical point of view.

Further work should consider more realistic models of suspension system of railway vehicles, equipped by anti-hunting dampers with displacement-dependent dissipative characteristics.

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