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KINEMATICAL EXCITATION DUE TO TERRAIN IRREGULARITIES ON THE ROLLING SYSTEM OF THE AGRICULTURAL MACHINES

Lecturer PhD. Eng. Adrian-Sorin AXINTI
"Dunarea de Jos" University of Galati
The research Center of Machines, Mechanics and Technological Equipments

ABSTRACT

This paper presents the kinematical influences of the rolling way on the response of the equipment in motion. The basic model of the machine consists of a single degree of freedom mechanical system. The kinematical excitation was supposed to have a harmonic shape and to induce into the axle of the rolling body a resistant moment characterized by the amplification factor that can acquire very high values. This study tries to analyze the way a kinematical action induces dynamic behaviour into the traction system of an agricultural automotive equipment in motion over a terrain with strong irregularities.

KEYWORDS: kinematical influences, equipment, agricultural automotive, traction

1. INTRODUCTION

The disturbing factor due to the irregularities of the rolling way about the equipment in motion is induced by the profile of the surface on which the machine is rolling.

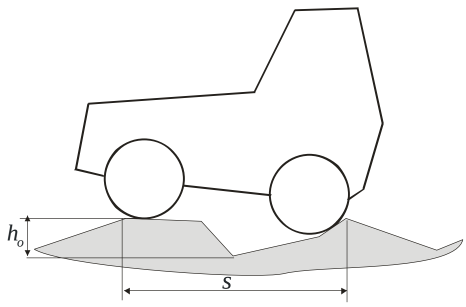


Fig. 1

The most common shape of the rolling way irregularities usually used for the unarranged roads contains break lines that form longitudinal shapes with h_o height and s length. It can be considered that the longitudinal road profile can be simulated by a periodical mathematical law (see Fig. 1).

The bumps of a road with irregularities have certain repeatability in time, which leads to a periodical function that simulates this type of roads with an acceptable error. For simulation of this kind of periodical road bumps it are usually used harmonic functions [5,6] as follows

$$u(x) = u_0 \sin(\omega_0 x);$$
$$u(x) = u_0 (1 - \cos(\omega_0 x)); \qquad (1)$$

where $u_o = h_o/2$ denotes the magnitude of the road bump; $\omega_0 = 2\pi / s = v_0 / r$ denotes the pulsation of the bumps; s means the step of the irregularities; x is the instant displacement of the bump evaluation element, which in a proper way can be the displacement of the moving part of an equipment (wheel or caterpillar).

A most proper way to simulate the rolling longitudinal profile of an unarranged road with realistic bumps inclusion is the sum of harmonic functions with different magnitudes (Fourier series). The mathematical model of such representation is

$$u(t) = u_0 + u_1 \sin(\omega_0 t + \varphi_1) + + u_2 \sin(2\omega_0 t + \varphi_2) + ... \quad ; \qquad (2)$$
$$.. + u_n \sin(n\omega_0 t + \varphi_n);$$

where $u_i (i = 1...n)$ denotes the Fourier coefficients that mean the magnitudes of harmonic components, and u_o is the average value of a function during the period $T = s/v_0$, which simulates the rolling way. From Eqn. (2) it results that for v_o speed value of an equipment, the periodical unharmonical bumps of the road constitute the disturbing factors that act at same time with pulsations

$\omega_1 = \omega_0, \omega_2 = 2\omega_0, ..\omega_n = n\omega_0$, and the magnitudes $u_i\,(i = 1...n)$.

The components of the expression (2) represent the harmonics of the real function that approximates the rolling way, and the first rank term denotes the fundamental harmonic of the kinematical excitation induced by the irregular road. For the forced vibration analysis, the harmonics of the disturbing forces have an important role. In this case, the disturbing forces consist of the kinematical excitation of unarranged road and the periodicity can lead to the resonance phenomenon [8]. For an analytical modeling and for solving this kind of kinematical excitation due to the shape of the longitudinal profile of rolling way, is usually used only the fundamental harmonic defined by the expression as follows

$$u(t) = u_1 \sin(\omega_1 t + \varphi_1); \qquad (3)$$

2. DYNAMIC MODEL INTENDED FOR EQUIPMENT BEHAVIOUR ANALYSIS

The model of the equipment was compiled taking into account the rolling mechanism (wheel or caterpillar) as a single degree of freedom model containing a viscous-elastic element that simulates dissipative and conservative characteristics of the moving system. This model was depicted in Fig. 2, where the two cases denote particular rolling element. The characteristic equation for the model in Fig. 1 results through D'Alembert Principle used for vertical axis (oz) and applied for mass M of the equipment. The rigidity k_z and the damping c_z of the moving system were also considered. Hereby, the differential equation of the model is

$$M.\ddot{z} = c_z(\dot{u} - \dot{z}) + k_z(u - z); \qquad (4)$$

(9)

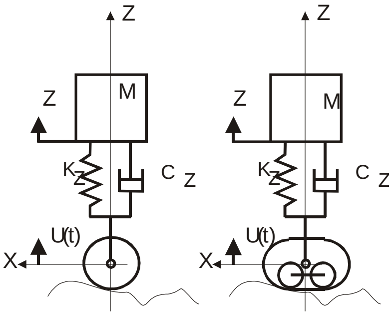


Fig.2. Schematic diagrams of equipment in motion over an irregular road. (a) wheel case; (b) caterpillar case.

Reordering the terms in Eqn. (4) leads to the final characteristic equation of the models in Fig. 1. This model reveals the equipment motion along the z-axis considering the road adherence. It also dignifies the kinematical excitation factor induced by the rolling way (this term is the right term of the equation). The equation is

$$M.\ddot{z} + c_z\dot{z} + k_z z = c_z\dot{u} + k_z u; \qquad (5)$$

3. THE DYNAMIC RESPONSE DUE TO THE HARMONIC KINEMATICAL EXCITATION

First, it considered the kinematical excitation due to the rolling way and simulated as a harmonic function - Eqn. (3). Replace this expression in Eqn. (5) - right term – it results

$$c_z\dot{u} + k_z u = c_z u_0 \omega_0 \cos\omega_0 t + u_0 k_z \sin\omega_0 t = \\ = Z_0 \sin(\omega_0 t + \theta); \qquad ;(6)$$

where

$$Z_0 = u_0\sqrt{k_z^2 + c_z^2\omega_0^2}\,;\theta = arctg[\omega_0 c_z / k_z]; \qquad (7)$$

Replaceing Eqn. (6) by Eqn. (5) and divided by M it results

$$\ddot{z} + 2n\dot{z} + p^2 z = h_0 \sin(\omega_0 t + \theta); \qquad (8)$$

where it were used the following notations

$$2n = c_z / M;\, p^2 = k_z / M;\, h_0 = Z_0 / M;$$

The system behaviour on dynamic regime taking into account the kinematical excitation due to the rolling way is given by the particular solution $z = A\sin(\omega_0 t - \Gamma)$, which must verify Eqn. (8). Hereby it results

$$A = \frac{h_0}{p^2}\frac{1}{\sqrt{(1-\Omega^2)^2 + (2\xi\Omega)^2}};$$

that represents the magnitude of the equipment movement along the z-axis, and

$$\Gamma = arctg\frac{2\xi\Omega}{1-\Omega^2}$$

$$M.\ddot{z} = c_z(\dot{u} - \dot{z}) + k_z(u - z); \qquad (10)$$

that denotes the initial phases of the equipment movement.

In expressions (9) and (10) the notation $\Omega = \omega_0 / p$ means the relative pulsation of the movement, and $\xi = n/p -$ denotes the critical damping ratio. With the notations $A_0 = h_0 / p^2 = Z_0 / k_z = u_0\sqrt{1 + tg^2\theta}$ that means static magnitude of the kinematical excitation and $\eta = A/A_0$ that denotes amplification factor (or dynamicity) results

$$\eta = \frac{1}{\sqrt{(1-\Omega^2)^2 + (2\xi\Omega)^2}}; \qquad (11)$$

Eqns. (10) and (11) highlight the resonance phenomenon with respect to phases, respectively to magnitude, which characterize the equipment motion with velocity $v_0 = const.$ along the x-axis, over the road irregularities.

4. THE RESISTANT MOMENT AT THE EQUIPMENT MOVING SYSTEM

Taking into account the equipment model depicted in Fig. 2, the dynamic reaction acting on the displacement system contains both the elastic and the dissipative components.

Thus, the dynamic reaction of the moving mechanism results from Eqn. (4) considered in the following form

$$Z_m = c_z(\dot{u} - \dot{z}) + k_z(u - z) = M.\ddot{z}; \qquad (12)$$

The variation during the simulation time of the dynamic reaction results from the acceleration $\ddot{z}$ expression inserted into Eqn. (12), followed by regrouping the terms. Hereby the amplification factor of the dynamic reaction due to the rolling way is

$$\Re = \frac{Z_m}{Z_0} = \frac{\Omega^2}{\sqrt{(1-\Omega^2)^2 + (2\xi\Omega)^2}}; \qquad (13)$$

where Z_o results from Eqn. (7) and denotes the static reaction on the rolling mechanism.

Because the resistant moment at the displacement mechanism (based on wheel or caterpillar) depends by the rolling way material type, through the rolling-friction coefficient f, results [1,2]

$$\begin{aligned} M_r &= f(G + Z_m).r = \\ &= f\left[Mg + Z_m \sin(\omega_0 t - \Gamma)\right].r =; \qquad (14) \\ &= M_{RS}\left[1 + \frac{Z_m}{G}\sin(\omega_0 t - \Gamma)\right]; \end{aligned}$$

where $M_{RS} = Mgrf$ denotes the resistant moment at the wheel/caterpillar rolling due to the static charge of the displacement system.

The term in square parenthesis shows the increasing factor of the resistant moment due to the kinematical excitation induced by the rolling way. Thus, it has been defined the amplification factor of the resistant moment at the displacement system of automotive equipment as follows

$$\psi = \frac{Z_m}{G} = \Lambda \frac{\sqrt{1 + (2\xi\Omega)^2}}{\sqrt{(1-\Omega^2)^2 + (2\xi\Omega)^2}}; \qquad (15)$$

where $\Lambda = u_0 v_0^2 / gr^2$ is a coefficient depending on the height (u_o) of the road irregularities, the equipment velocity (v_o), the radius of the moving system ($r = r_d$ that denotes dynamic radius of the wheel, or $r = r_o$ that denotes the gearing radius of the caterpillar system), and the gravity acceleration g.

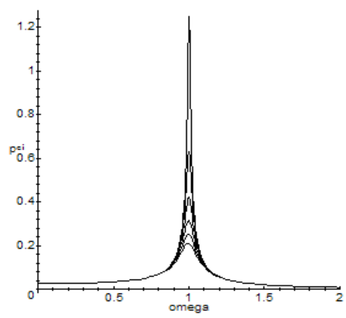
In order to analyze the dynamic behaviour of the moving system of automotive technological equipment (such as considered) in motion over the unarranged road irregularities (supposed to have a harmonic shape), it is considered the resistant moment at the rolling system axis having the following expression

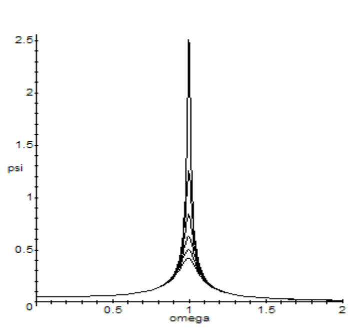
$$M_r = M_{RS}\left[1 + \psi \sin(\omega_0 t - \Gamma)\right]; \qquad (16)$$

5. THE SIMULATION RESULTS

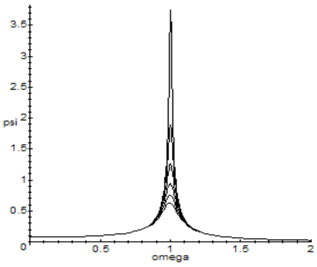
Let us suppose that dynamic amplification of the resistant moment at the rolling system axis is given by the ψ factor - see Eqn. (15). The following results have been obtained for regular values of the terms in the model differential equation and the diagrams in Fig. 3 shows the variation of the essential parameter.

a) $u_0 = 0.1$; $\xi \in (0{,}01 - 0{,}06)$; $v_0 = 2{,}85 km/h$; $r_0 = 0{,}5m$

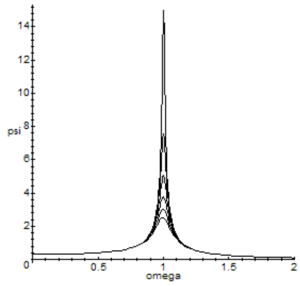




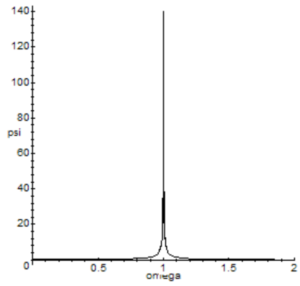
b) $u_0 = 0.2$; $\xi \in (0{,}01 - 0{,}06)$; $v_0 = 2{,}85 km/h$; $r_0 = 0{,}5m$



c) $u_0 = 0.3$; $\xi \in (0{,}01 - 0{,}06)$; $v_0 = 2{,}85 km/h$; $r_0 = 0{,}5m$



d) $u_0 = 0.3$; $\xi \in (0{,}01 - 0{,}06)$; $v_0 = 5{,}65 km/h$; $r_0 = 0{,}5m$



e) $u_0 = 0.3$; $\xi \in (0{,}001 - 0{,}006)$; $v_0 = 5{,}65 km/h$; $r_0 = 0{,}5m$

Fig.3. The variation of the amplification factor of the resistant moment at the wheel system $\psi = f(\xi)$.

6. CONCLUSIONS

From the analysis presented in this paper, it results that at the displacement of the automotive technological equipments over the irregularities of the rolling way appear dynamic loads that induce vibration into the equipment structure and dynamic resistant moments into the rolling systems (based on wheel or caterpillar). These dynamic over-loads result on the vertical axis of equipment and are transmitted from the rolling system to the base machine through the viscous-elastic components that simulate both the conservative and the dissipative characteristics of all mechanical parts composing the rolling and the insulating mechanisms.

It was dignified the amplification factor of the vibration magnitude into the equipment structure, named dynamicity factor - Eqn. (11), and the amplification factor of the dynamic reaction - Eqn. (13). Both these factors have been defined in relation with irregularities essential dimensions, equipment velocity and cumulative damping of rolling system and unarranged road.

It was also dignified the amplification factor of the resistant moment - Eqn. (15) - at the moving system of the automotive equipment, which depends on irregularities height, rolling radius, equipment velocity and cumulative damping of rolling system and unarranged road.

Numerical analysis presented in previous paragraph shows a dynamic increasing of the resistant moment at the wheel axis having one or two additional orders comparative to the same parameter but in static conditions. The amplification factor of the resistant moment grows with the speed increasing, the irregularities height increasing and the damping factor decreasing. The other amplification factors have the same conditioning as the basic parameters.

A direct consequence of these conditioning rules represent the over-loads (or shocks) that have been induced into the equipment structure and into the moving system, being able to produce high peak-to-peak values of dynamic parameters into the equipments additionally attached or even into the regular systems of the base machine (gear system, hydraulic driving system, etc.).

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