ADVANCED MECHANICAL MODEL FOR DYNAMIC BEHAVIOURAL ANALYSIS OF THE FRONT LOADER TYPE EQUIPMENT

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Abstract: In this paper the author present a new physico-mathematic model which describe the movement of the frontal changer. The technological equipment is considered as a dynamic system with two masses (the base machine and the equipment loaded with change in the bucket) and five degrees of freedom. The motion of the machine is described by the set of differential equations, based on the second order *Lagrange Principle*, with proposed hypothesis of energy dissipative systems. To computing the solution for these equations, it was used the classical numerical methods, implemented on the MATLAB™ software numerical application. The model and the method presented in this work constitute a starting original approach for analysis both of a motions of each constitutive elements of the machine, and of a dynamic stresses and strains of the changer equipment metallic structure. The main purpose in this study consist by underlie a complex and complete numerical model, which leads the simulations and the analysis of the effects of the vibrations, produced in the metallic structure of the technological equipment, for the different phases of the working cycle.

Keywords: changer, dynamic overloads, vibratory system.

1. INTRODUCTORY CONSIDERATIONS

Nowadays, to obtained the technological and functional performances more and more higher, earthmoving machines are conceives to have high working velocity. This fact leads to the necessity of knowing the phenomenon which generate variable dynamics strain into equipment, and, finally optimisation of theses effects. From the effectuated studies until today, it was seen that the transitory work state through the equipment passed are the source of the dynamic considerable solicitations. The review of these includes

- transitory working state produced by alternation of the working manipulation of the earthmoving machine;
- transitory working state produced by resistant force variation at the working tool;
- transitory working state produced by passing through the resonance;
- transitory working state produced by machine passing over the irregular profile roads.

In this paper, that is one of the first component of the largest study about the frontal loader equipments dynamic behaviour analysis, the author propose a new complex approach of a mechanical model for the mentioned equipment type. In this sense, it is presented the physical model, then it is deduced the mathematical expressions for the movement equations, and finally, it is presents a few diagrams that was resulted from a numerical computation of the mathematical model.

2. THE DESCRIPTION OF THE MODEL

For the beginning, it is considered a dynamic system composed by two masses, here by the base machine *(m_1)* that is visco-elastic sustained on four points, and the working body mass *(m_2)*, like as shown in the Figure 1. This study was realised with taking into account the frontal loader model, and with the working tool articulated joint at the equipment, with torsional rigidity k_φ and transversal rigidity k_T.

The five degree of freedom of the model are

- the base machine jumping arround the *OZ* axis, defined by the *z* coordinate;
- the rotational movement of the base machine arround the *OZ* axis, defined by the φ_1 angle;
- the rotational movement of the base machine arround the *OX* axis, defined by the θ_1 angle;
- the rotational movement of the working tool arround the joint with the *OZ* axis, defined by the φ_2 angle;
- the rotational movement of the working tool arround the joint with the *OX* axis, defined by the θ_2 angle.

The geometrical coordinates of the bucket weight point, namely *B* on the figure 1, vis a vis one the same point of the base machine, is computed with following expressions

$$\begin{aligned} x_B &= L\cos(\alpha_1+\varphi_1)+r\cos(\alpha_2-\varphi_2-\alpha_1-\varphi_1); \\ y_B &= \left[L\sin(\alpha_1+\varphi_1)-r\sin(\alpha_2-\varphi_2-\alpha_1-\varphi_1)\cos\theta_2\right]\sin\theta_1; \\ z_B &= \left[L\sin(\alpha_1+\varphi_1)-r\sin(\alpha_2-\varphi_2-\alpha_1-\varphi_1)\cos\theta_2\right]\cos\theta_1 - z. \end{aligned} \tag{1}$$

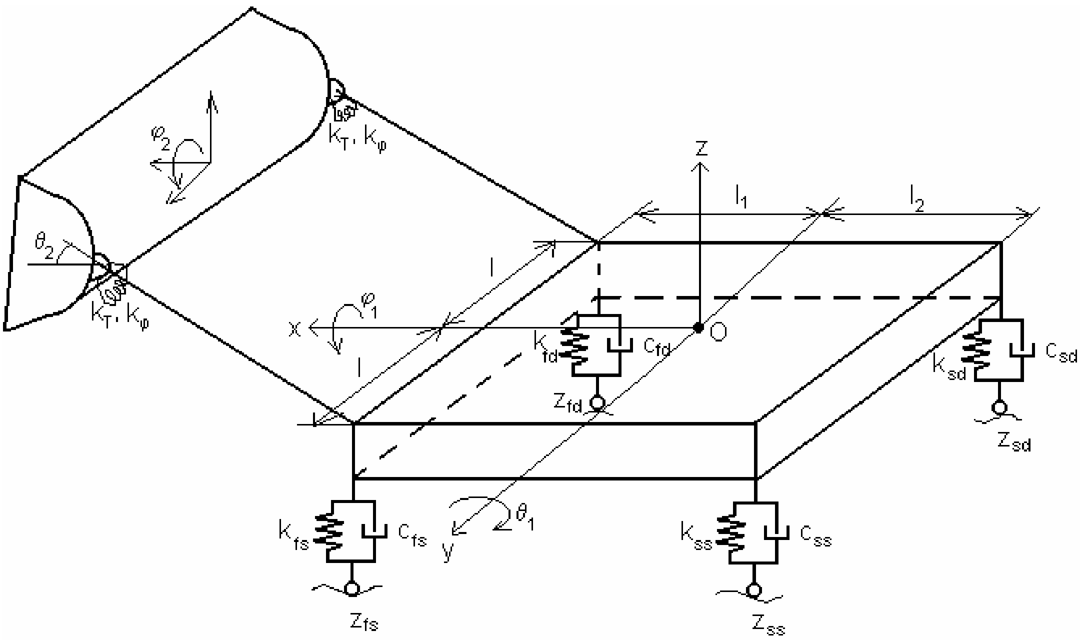

Fig. 1. Front loader physical model with 5DOF

The linear velocity of the m_2 mass is

$$v_B^2 = x_B^2 + y_B^2 + z_B^2 \tag{2}$$

$$\begin{aligned} v_B^2 &= \dot{z}^2 + \dot{\theta}_1^2 a_1^2 + \dot{\varphi}_1^2 a_2^2 + \dot{\varphi}_2^2 r^2 + \dot{\varphi}_1 \dot{\varphi}_2 a_3^2 + \dot{\theta}_1 \dot{z} \theta_1 a_4 - \dot{\theta}_2 \dot{z} \theta_2 a_5 + \dot{\theta}_2 \dot{\varphi}_1 \theta_2 a_6^2 + \\ &+ \dot{\theta}_2 \theta_2 \dot{\varphi}_2 a_7^2 - \dot{\varphi}_1 \dot{z} a_8 - \dot{\varphi}_2 \dot{z} a_9 , \end{aligned} \tag{3}$$

where

$$\begin{aligned} & a_1^2 = [L \sin\alpha_1 - r \sin(\alpha_2 - \alpha_1)]^2 ;\ a_2^2 = L^2 + r^2 + 2Lr \cos\alpha_2 ;\ a_3^2 = 2r^2 + 2rL \cos\alpha_2 ; \\ & a_4 = 2L \sin\alpha_1 - 2r \sin(\alpha_2 - \alpha_1);\ a_5 = 2r \sin(\alpha_2 - \alpha_1); \\ & a_6^2 = 2Lr \cos\alpha_1 \sin(\alpha_2 - \alpha_1) + 2r^2 \cos(\alpha_2 - \alpha_1) \sin(\alpha_2 - \alpha_1) ; \\ & a_7^2 = 2r^2 \cos(\alpha_2 - \alpha_1) \sin(\alpha_2 - \alpha_1);\ a_8 = 2L \cos\alpha_1 + 2r \cos(\alpha_2 - \alpha_1); \\ & a_9 = 2r \cos(\alpha_2 - \alpha_1) . \end{aligned}$$

For simplification, it will be considering that the generalized coordinates values are neglected comparative with the generalized velocities, which allowed the simplifications of the velocity formula for the *B* point. The kinetic energy for the system have the next form

$$E_c = \frac{1}{2} m_1 \dot{z}^2 + \frac{1}{2} J_{1A} \dot{\varphi}_1^2 + \frac{1}{2} J_{1AA} \dot{\theta}_1^2 + \frac{1}{2} m_2 v_B^2 + \frac{1}{2} J_{2B} (\dot{\varphi}_1 + \dot{\varphi}_2)^2 + \frac{1}{2} J_{2BB} \left(\dot{\theta}_1^2 + \dot{\theta}_2^2\right)^2 ; \tag{4}$$

After some substitutions and simplification, the relation (4) become

$$\begin{aligned} E_c &= \frac{1}{2} m \dot{z}^2 + \frac{1}{2} J \dot{\varphi}_1^2 + \frac{1}{2} J_2 \dot{\varphi}_2^2 + \frac{1}{2} J^* \dot{\theta}_1^2 + \frac{1}{2} J_2^* \dot{\theta}_2^2 + \frac{1}{2} \dot{\varphi}_1 \dot{\varphi}_2 \left(J_{2B} + m_2 a_3^2\right) + \dot{\theta}_1 \dot{\theta}_2 J_2^* + \frac{1}{2} m_2 \dot{\theta}_1 \dot{z} \theta_1 a_4 - \\ &- \frac{1}{2} m_2 \dot{\theta}_2 \dot{z} \theta_2 a_5 + \frac{1}{2} m_2 \dot{\theta}_2 \dot{\varphi}_1 \theta_2 a_6^2 + \frac{1}{2} m_2 \dot{\theta}_2 \dot{\varphi}_2 \theta_2 a_7^2 - \frac{1}{2} m_2 \dot{\varphi}_1 \dot{z} a_8 - \frac{1}{2} m_2 \dot{\varphi}_2 \dot{z} a_9 , \end{aligned} \tag{5}$$

where

$$\begin{gathered} m = m_1 + m_2 ;\ J = J_{1A} + J_{2B} + m_2 a_2^2 ; \\ J^* = J_{1AA} + J_{2BB} + m_2 a_1^2 ;\ J_{2E} = J_{2B} + m_2 r^2 ;\ J_2^* = J_{2BB}; \end{gathered}$$

m – total mass of the system; *J* – moment of inertia for the bucket around the joint *A* in the longitudinal plan; J^*- moment of inertia for the bucket around the joint *A* in the transversal plan; J_{2E} - moment of inertia for the bucket around the elastic joint *E* in the longitudinal plan; J_{2BB} - moment of inertia for the bucket around the elastic joint *E* in the transversal plan.

Relative displacements for the four machine's wheels are

$$\begin{aligned} & \delta_{fd} = z + \theta_1 l + \varphi_1 l_1 - z_{fd} ; \delta_{sd} = z + \theta_1 l - \varphi_1 l_1 - z_{sd} \\ & \delta_{fs} = z - \theta_1 l + \varphi_1 l_2 - z_{fs} ; \delta_{ss} = z - \theta_1 l - \varphi_1 l_2 - z_{ss} \end{aligned} \tag{6}$$

The potential energy of the system is

$$V = \frac{1}{2}k_1\left(\delta_{fd}\right)^2 + \frac{1}{2}k_2\left(\delta_{sd}\right)^2 + \frac{1}{2}k_3\left(\delta_{fs}\right)^2 + \frac{1}{2}k_4\left(\delta_{ss}\right)^2 + \frac{1}{2}k_\varphi\varphi_2^2 + \frac{1}{2}k_T\theta_2^2 . \tag{7}$$

$$\begin{aligned} V = {} & \frac{1}{2}k_1\left(z + \varphi_1 l_1 + \theta_1 l - z_{fd}\right)^2 + \frac{1}{2}k_2\left(z - \varphi_1 l_1 + \theta_1 l - z_{sd}\right)^2 + \\ & + \frac{1}{2}k_3\left(z + \varphi_1 l_1 - \theta_1 l - z_{fs}\right)^2 + \frac{1}{2}k_4\left(z - \varphi_1 l_1 - \theta_1 l - z_{ss}\right)^2 + \frac{1}{2}k_\varphi\varphi_2^2 + \frac{1}{2}k_T\theta_2^2 , \end{aligned} \tag{8}$$

and the dissipative function D is

$$\begin{aligned} D = {} & \frac{1}{2}c_1\left(\dot{z} + \dot{\varphi}_1 l_1 + \dot{\theta}_1 l - \dot{z}_{fd}\right)^2 + \frac{1}{2}c_2\left(\dot{z} - \dot{\varphi}_1 l_1 + \dot{\theta}_1 l - \dot{z}_{sd}\right)^2 + \\ & + \frac{1}{2}c_3\left(\dot{z} + \dot{\varphi}_1 l_1 - \dot{\theta}_1 l - \dot{z}_{fs}\right)^2 + \frac{1}{2}c_4\left(\dot{z} - \dot{\varphi}_1 l_1 - \dot{\theta}_1 l - \dot{z}_{ss}\right)^2 . \end{aligned} \tag{9}$$

The equations of the dynamic equilibrium of the two masses was written bassed on Lagrange second order equations. Thus

$$\frac{d}{dt}\left(\frac{\partial E}{\partial \dot{q}_j}\right) - \frac{\partial E}{\partial q_j} = -\frac{\partial V}{\partial q_j} - \frac{\partial D}{\partial \dot{q}_j}, \quad j = \overline{1,5} , \tag{10}$$

where $q_j = \{z, \varphi_1, \varphi_2, \theta_1, \theta_2\}$.

It was taken the hypothesis pursuant to the machine wheels have the same characteristics. Thus

$$k_1 = k_2 = k_3 = k_4 = k; c_1 = c_2 = c_3 = c_4 = c.$$

After substitutions, the differential equations system (11) become

$$\begin{aligned} & m\ddot{z} + 4c\dot{z} + 4kz - 0.5m_2\ddot{\varphi}_1 a_8 - 0.5m_2\ddot{\varphi}_2 a_9 + 0.5m_2\ddot{\theta}_1\theta_1 a_4 + 0.5m_2\dot{\theta}_1^2 a_4 + 4c\dot{\theta}_1 l + 4k\theta_1 l - \\ & - 0.5m_2\ddot{\theta}_2\theta_2 a_5 - 0.5m_2\dot{\theta}_2^2 a_5 = k\left(z_{fd} + z_{sd} + z_{fs} + z_{ss}\right) + c\left(\dot{z}_{fd} + \dot{z}_{sd} + \dot{z}_{fs} + \dot{z}_{ss}\right) \end{aligned}$$

$$\begin{aligned} & -0.5m_2\ddot{z}a_8 + 2c\dot{z}(l_1 + l_2) + 2kz(l_1 + l_2) + J\ddot{\varphi}_1 + 2c\dot{\varphi}_1\left(l_1^2 + l_2^2\right) + 2\varphi_1 k\left(l_1^2 + l_2^2\right) + 0.5\ddot{\varphi}_2\left(J_{2B} + m_2 a_3^2\right) + \\ & + 0.5m_2\ddot{\theta}_2\theta_2 a_6^2 + 0.5m_2\dot{\theta}_2^2 a_6^2 = k\left[l_1\left(z_{fd} + z_{sd}\right) + l_2\left(z_{fs} + z_{ss}\right)\right] + c\left[l_1\left(\dot{z}_{fd} + \dot{z}_{sd}\right) + l_2\left(\dot{z}_{fs} + \dot{z}_{ss}\right)\right] \end{aligned}$$

$$-0.5m_2\ddot{z}a_9 + 0.5\ddot{\varphi}_1\left(J_{2B} + m_2 a_3^2\right) + J_2\ddot{\varphi}_2 + k_\varphi\varphi_2 + 0.5m_2\ddot{\theta}_2\theta_2 a_7^2 + 0.5m_2\dot{\theta}_2^2 a_7^2 = 0$$

$$\begin{aligned} & 0.5m_2\ddot{z}\theta_1 a_4 + 0.5m_2\dot{z}\dot{\theta}_1 a_4 + 4c\dot{z}l + 4zkl + J^*\ddot{\theta}_1 + m_2\dot{\theta}_1\dot{z}a_4 + J_2^*\ddot{\theta}_2 = \\ & = kl\left(z_{fd} + z_{sd} + z_{fs} + z_{ss}\right) + cl\left(\dot{z}_{fd} + \dot{z}_{sd} + \dot{z}_{fs} + \dot{z}_{ss}\right) \end{aligned}$$

$$-0.5m_2\ddot{z}\theta_2 a_5 - 0.5m_2\dot{z}\dot{\theta}_2 a_4 + 0.5m_2\dot{\varphi}_1\dot{\theta}_2 a_6^2 + 0.5m_2\ddot{\varphi}_1\theta_2 a_6^2 + 0.5m_2\dot{\varphi}_2\dot{\theta}_2 a_7^2 + 0.5m_2\ddot{\varphi}_2\theta_2 a_7^2 + \\ + J_2^*\ddot{\theta}_1 + J_2^*\ddot{\theta}_2 + k_T\theta_2 - m_2\dot{\theta}_2\dot{z}a_5 + m_2\dot{\theta}_2\dot{\varphi}_1 a_6^2 + m_2\dot{\theta}_2\dot{\varphi}_2 a_7^2 = 0 \qquad (11)$$

The resulting nonlinear differential equations (11) were complex necessitating the assistance of a computer to aid in their development. Steps were taken to simulate the dynamical equation using a numerical integration routine in the MATLAB™ technical computing environment. The reponse of the loaded model with 5DOF is shows in Figure 2.

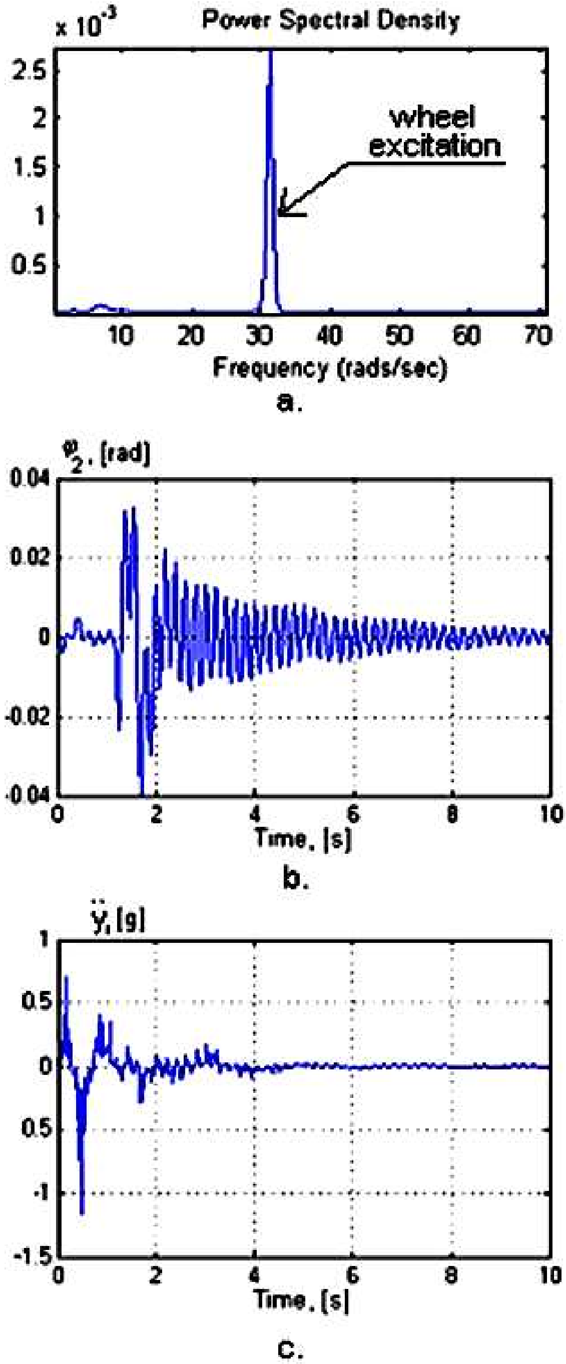

Fig. 2. Numerical simulation results

3. CONCLUDING REMARKS

As it could seen on the presented diagrams, the proposed numerical model come through putting into evidence the dynamic phenomenon that appears on the front charger equipment. It was used the simple excitation signals, as the impulse or the semi-sinus signals, that are simulate the system common inputs on a real working situations.

The simulation outputs evidentiates the transitory states and, the most important fact, the stabilization of the system in the short time, when the input signal disappear. The entire set of the output diagrams indicates a stable system, regarding the initial hypothesis for numerical model settlement of the frontal charger equipment.

On the future, this study will be developed on two ways: first, the numerical analysis will be performed with consideration of a complex inputs, and, second, the numerical model will be tuning by the experimental measurements made on the real system.

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