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Impulsive Action Equipments Simulation - a Needful Stage in Vibrations Control

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Abstract: The dynamic equipments impact evaluation over the environment became a priority regarding the keeping of the social security rules. This evaluation can be achieved by experimental determination, in situ, but unfortunately this method is just to ascertain and it can show the adequate measures for the decrease of the vibrations negative impact on the surroundings. Thus, it is imposed that the design stage of a dynamic actioning installation is attended by the appropriate dynamic and mathematical simulation in order to estimate the impact that the functioning equipment may have over the surroundings. This work suggests a dynamic simulation of a certain type of equipment that is considered the one of the most polluted industrial equipments in vibration transmission.

Keywords: simulation, impulsive action, vibration control, security.

1 Introduction

The essence of the impulsive excitation demands analyses of the foundations for machines consists actually in establishing the actual link between the parameters of the excitation force and the vibration parameters, assuming that the mechanical characteristics of the oscillatory system. In this paper will be analyze the dynamical behaviour of the press with eccentric, which is a technological equipment as must produce a greatest mechanical work in a short time, so with intensive stretch. This mode of action product shocks and vibration which are propagated both to environment and to technological equipment. This technological equipment by its specific mode of operating is propagating potentially noxious vibration during the working process, both over the environment as well as over itself, that is why the equipment becomes a pollution source by vibration. For this reason, the application of the anti-vibrating visco-elastic type systems is needful for the purpose of damping and insolating the generated vibration (fig. 1).

2 Model of the system

The simplest method to evaluate the negative impact of the vibration generated by the eccentric press consists in the system's theoretical simulation that contains two stages: the physical simulation and the numerical simulation.

For this study it is necessary to elaborate a physical and mathematical model able to approximate the real case. We consider the technological equipment was mounted on viscous-elastic elements and the block of foundation is installed directly on viscous-elastic soil. The simplified model of the press with eccentric is presented in fig. 2, where: M_1 – upper part of the equipment mass; M_2 – under part of the equipment mass; M_3 – mass of the foundation block; c_1 – damping of the element between mass M_1 and M_2; k_1 – rigidity of the element between the mass M_1 and M_2; k_2 – rigidity of the viscous-elastic elements between equipment and foundation; c_2 – damping of the elements between equipment and foundation; k_3 – rigidity of soil that is installed foundation; c_3 – damping of soil that is installed foundation.

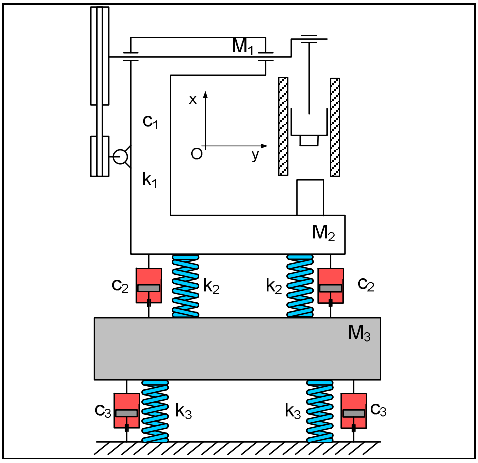

Figure 1. The model of displacement of the press on soil

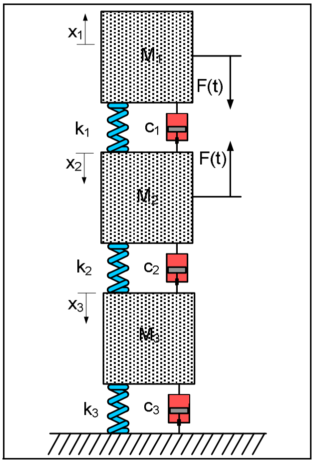

Figure 2. Physical model

For this model the following sequences hypothesis:

- the mass will considerate concentrate in the centre of mass
- will be neglected the turn round of the frame of press
- will be take into consideration only moves on Ox direction
- will be neglected the friction forces in time of press functioning
- the cutting force is the shock tip
- the frame of the press will be two-part considerate M_1 and M_2
- the elastic and damping forces are linear expression.

The mathematical model can be written like differential equations system:

$$\begin{aligned}
&M_1\cdot\ddot{x}_1+c_1\cdot(\dot{x}_1+\dot{x}_2)+k_1\cdot(x_1+x_2)=F(t);\\
&M_2\cdot\ddot{x}_2+c_1\cdot(\dot{x}_1+\dot{x}_2)+c_2\cdot(\dot{x}_2-\dot{x}_3)+k_1\cdot(x_1+x_2)+k_2\cdot(x_2-x_3)=F(t);\\
&M_3\cdot\ddot{x}_3+c_2\cdot(\dot{x}_3-\dot{x}_2)+c_3\cdot\dot{x}_3+k_2\cdot(x_3-x_2)+k_3\cdot x_3=0;
\end{aligned}\tag{1}$$

where: x_1 – displacement of the mass M_1; $\dot{x}_1$ – speed of the mass M_1; $\ddot{x}_1$ – acceleration of the mass M_1; x_2 – displacement of the mass M_2; $\dot{x}_2$ – speed of the mass M_2; $\ddot{x}_2$ – acceleration of the mass M_2; x_3 – displacement of the mass M_3; $\dot{x}_3$ – speed of the mass M_3; $\ddot{x}_3$ – acceleration of the mass M_3.

The numerical solution of the system equation (1) is computed by Runge – Kutta method, the stability solution and the precision of the results depend only the value of the time – increment who has to be little than 0.001 seconds. The solving system has made in hypothesis of the next numerical value: $k_1=9\cdot10^9$ [N/m]; $c_1=2\cdot10^5$ [Ns/m]; $m_1=16\cdot10^3$ kg; $k_2=10\cdot10^8$ [N/m]; $c_2=5\cdot10^6$ [Ns/m]; $m_2=25\cdot10^3$ kg; $k_3=25\cdot10^8$ [N/m]; $c_3=5\cdot10^6$ [Ns/m]; $m_3=80\cdot10^3$ kg; $P=10\cdot10^6$ N.

3 Perturbation forces

In the technical literature, the most used functions for impulsive actions simulation developed during the operation of the eccentric press are: the semi-sinusoidal form wave, the rectangular wave and the triangle wave. In this paper will by study only half-sine and rectangle type of shocks cases, with 0.03 s applied time (fig. 3, 4).

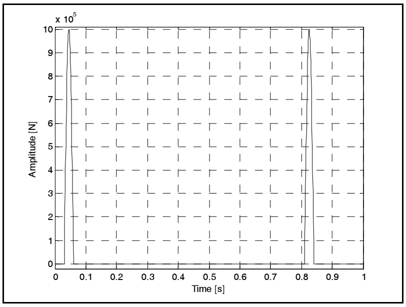

Figure 3. Half-sine shock

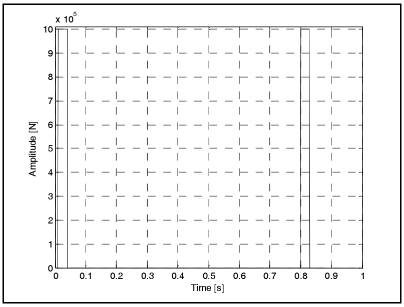

Figure 4. Rectangle shock

3.1 The half-sine shock case

The dynamical behavior of the technological equipment and the characterization of vibration transmitted to environment will make by three kinematical parameters displacement, speed and acceleration, as well as spectral analysis of the foundation displacement (fig. 5–12).

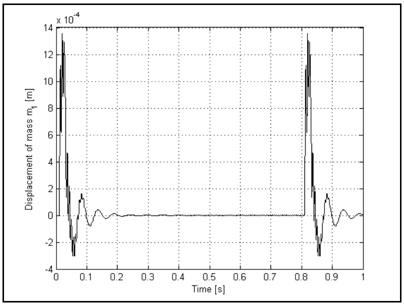

Figure 5. Displacement of mass m_1

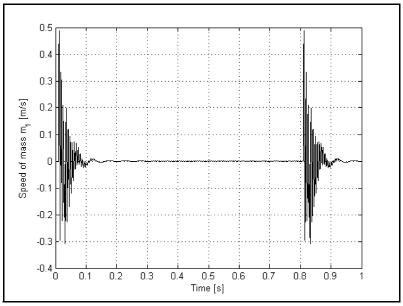

Figure 6. Speed of mass m_1

Because the maximum speed of the transmitted vibrations is 35mm/s, by rule of evaluation criteria from speedily, the neighbor building will not suffer significance damage.

In the figure 12 is represented the cumulative movement of the mass m_1 and m_2 in order to underline the impact of the shocks generated during the technological process on the equipment itself. This parameter has a real significance in the preventive maintenance activity, hereby anticipating the equipment damage caused by the fatigue solicitation. Fourier spectra vibration displacement of the m_3 mass, fig. 11 indicative the dominant range of the press with eccentric:

25–30Hz. The maximum displacement of the foundation is $2.25\cdot10^{-5}$m $=22.5\cdot10^{-6}$m=22.5μm, value whereon fissured appear on neighbor buildings.

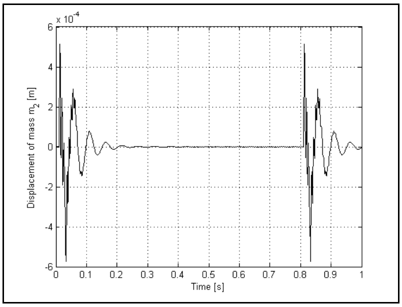

Figure 7. Displacement of mass m_2

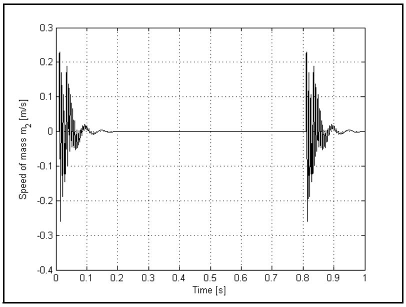

Figure 8. Speed of mass m_2

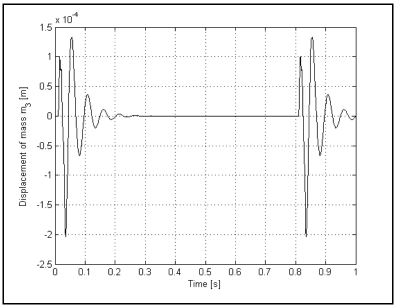

Figure 9. Displacement of mass m_3

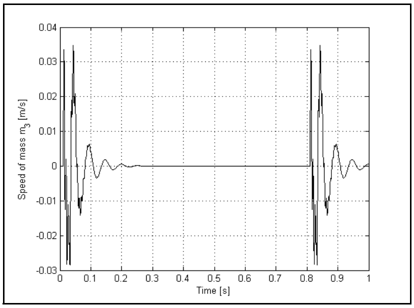

Figure 10. Speed of mass m_3

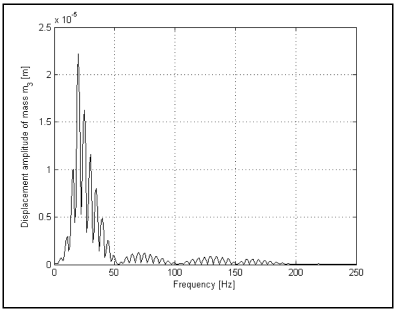

Figure 11. Fourier spectra vibration displacement m_3

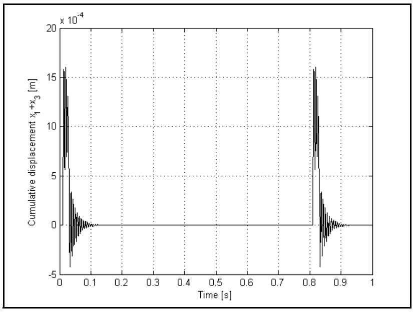

Figure 12. Cumulative displacement x_1+x_2

3.2 The rectangular shock case

Like in precedent case, the variation of the kinematics parameters and spectral analysis of the foundation displacement are depicted in fig. 13–20.

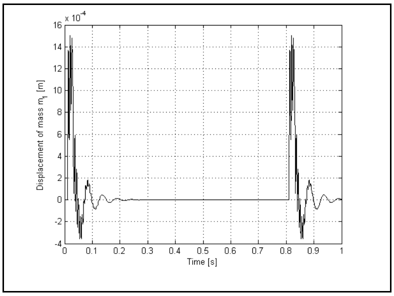

Figure 13. Displacement of mass m_1

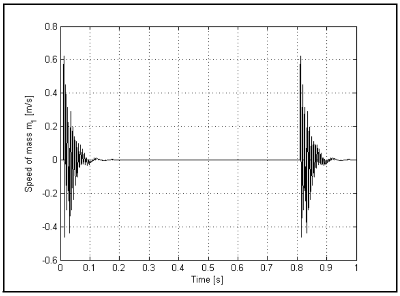

Figure 14. Speed of mass m_1

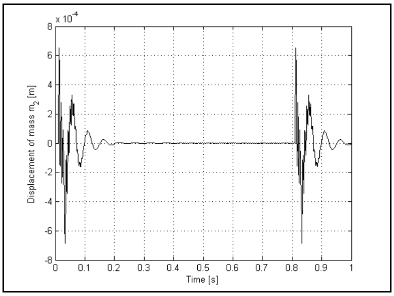

Figure 15. Displacement of mass m_2

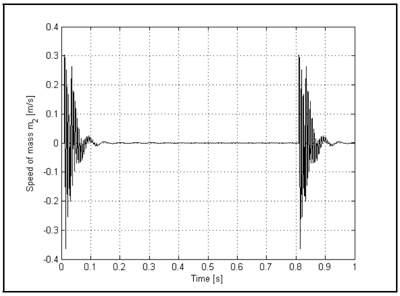

Figure 16. Speed of mass m_2

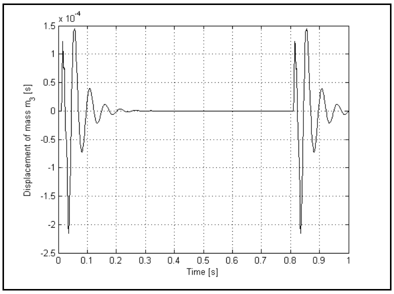

Figure 17. Displacement of mass m_3

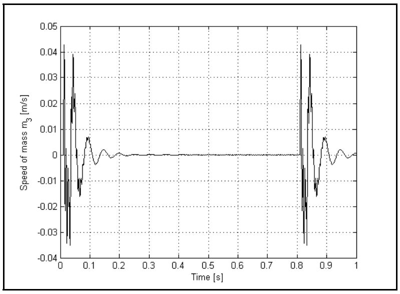

Figure 18. Speed of mass m

3

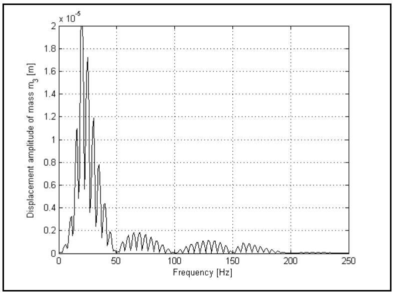

Figure 19. Fourier spectra vibration displacement m_3

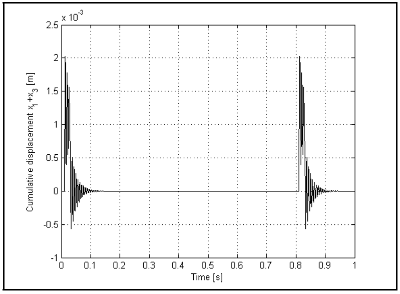

Figure 20. Cumulative displacement x_1+x_2

In case of collision excitation of the technological equipment by rectangle shock, we observe the same behavior like in collision excitation by half-sine shock case, because the period of loading is small.

4 Conclusion

This simulation of the press whit eccentric, represent an estimative possibility for dynamical characterization of the behavior in time of production process, as well as for estimate the impact of propagated vibration on to environment. This comparative dynamic analysis is meant to evaluate the kinematical parameters of the vibration propagated through the technological equipment fundament into the environment in order to compare the numerical values obtained for them within the standard allowable limits in the field. The evaluation of the movement in time of the system's fundament, known also as the „*equipment signature*", as well as its spectral composition known as the „*equipment imprint*" allows the detection and reading of the equipment involvement into a complex signal achieved during the operation of many machines. The representation in frequency of the equipment fundament movement shows that the dominant frequency spectrum is in the range of 0–50 Hz.

The innovative approaches proposed and presented in this work dignify new, relevant and suitable aspects for vibration control. This fact framed the researches into the well knowledge-based technologies for a sustainable development and creates the new opportunities for a large sustainability through a high reliability of technological systems.

References

Axinti, G. 2008. *Compendiu de Mecanica (Mechanics Digest)*. Ed. Tehnica-Info, Chisinau.

Babitsky, V. I.; Krupenin, V. L. 2001. *Vibration of Strongly Nonlinear Systems*. Springer.

Benaroya, H. 2004. *Mechanical Vibration: Analysis, Uncertainties and Control*. Marcel Dekker.

Bratu, P. 1999. *Vibratiile sistemelor elastice*. Editura Tehnica, Bucuresti.

Buzdugan, Gh. 1993. *Izolarea antivibratorie.* Ed. Academiei Române, Bucuresti.

Crocker, M. J. 2007. *Handbook of Noise and Vibration Control.* John Wiley&Sons Inc.

Debeleac, C. 2008. Vibratory Diagnosis of the Earthmoving Machines for the Additional Necessary Power Level Evaluation, in *9th WSEAS International Conference on ACOUSTICS & MUSIC: THEORY & APPLICATIONS (AMTA '08),* Bucharest, Romania, June 24–26, 83–88. ISBN 978-960-6766-74-9, ISSN 1790-5095.

Leopa, A.; Oproescu, Gh. 2001. A dynamical model of the press with eccentric, in 3^{rd} *International Conference of PhD Students*, 13–19 august University of Miskolc, Hungary, 273–278. ISBN 963 661 480 6.

Nastac, S. 1997. On the Increase of the Isolation Efficacy for the Passive Elastic Devices by the Structural Configuration Optimization, in *9th WSEAS International Conference on ACOUSTICS & MUSIC: THEORY & APPLICATIONS (AMTA '08),* Bucharest, Romania, June 24–26, 2008, 83–88. ISBN 978-960-6766-74-9. ISSN 1790-5095.