THE ANNALS OF “DUNAREA DE JOS” UNIVERSITY OF GALATI
FASCICLE XIV MECHANICAL ENGINEERING, ISSN 1224-5615
2006

DYNAMICAL CHARACTERIZATION BEHAVIOR OF THE MACHINE FOUNDATION WHIT PERCUSSION STRESSES APPLIED

S.l.ing. LEOPA ADRIAN
Universitatea "Dunarea de Jos" din Galati

ABSTRACT

This paper presents a dynamical analyze on the foundation of machine when are apply stresses such the beat (percussion). In this way can be evaluate dynamical response of the structure, in our case foundation of machines, as in time both in frequency.

1. Introduction

The knowledge and evaluation of the shock and vibration influences on environment become a priority in European integration conditions of our country. In this way European Directive 44/2002 establish minimal requirements, in purpose limiting of the expose level vibration transmitted on the human or environment.

In this study well be analyze dynamically behavior from foundation of technological equipment stresses from shocks. Exist a multitude of technological equipment which utilize in the production process the shocks, but the present study refer on forging hammer. Through peculiarity of production process, these equipment are shocks and vibration, motive wherefore is impose implement of the ant vibration protection system, able to decrease the effect impact on the environment.

2. Mathematical and physical modeling

Well consider a rigid in the inertial system OXYZ, considered fixed and a system of reference attached on rigid, with the origin is placed in he's centre of mass Cxyz (fig. 1).

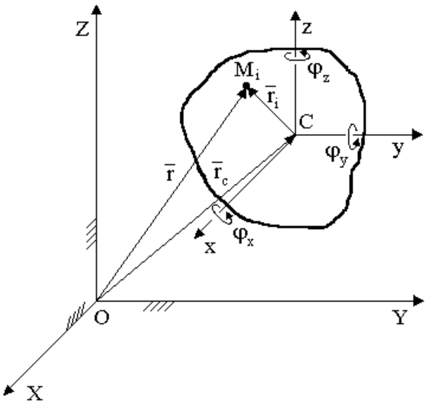

Fig. 1

The linear movements of the centre of mass C are function on X, Y, Z coordinate toward fixed system OXYZ, and the rotary movements are function on angular movements φ_x, φ_y and φ_z of the Oxyz system.

For calculate the movement of the point of the rigid toward Cxyz system when the rigid make an instantaneous, count the case from fig. 2. The movement analyzed point is A, which after rigid rotation with $\Delta\varphi$ can be result of sum by infinitesimal rotation.

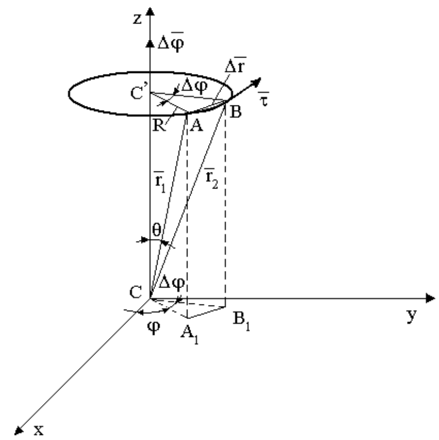

Fig. 2

Using the second kind Lagrange equation, obtain the differential equation system for the movement. The general form of the second kind Lagrange equation is:

$$\frac{d}{dt}\left(\frac{\partial E}{\partial \dot{q}_i}\right) - \frac{\partial E}{\partial q_i} = Q_i^P + Q_i^F + Q_i^R \text{ , i=1..6} \qquad (1)$$

where:

$Q_i^P = -\dfrac{\partial V}{\partial q_i}$ generalized forces on potential kind;

$Q_i^R = -\dfrac{\partial D}{\partial \dot{q}_i}$ generalized forces on viscous kind;

$Q_i^F = \dfrac{\partial L_{q_i}}{\partial q_i}$ generalized forces on perturbation;

δL_{q_i} - virtual mechanical work on perturbation which corresponding q_i coordinate.

The metrical expression on the second kind Lagrange equation is like:

$$\underline{\underline{A}}\ddot{q} + \underline{\underline{B}}\dot{q} + \underline{\underline{C}}q = \underline{\underline{f}} \qquad (2)$$

Where:

$\underline{\underline{q}} = [q_1, q_2, q_3, q_4, q_5, q_6]^T = [X, Y, Z, \varphi_x, \varphi_y, \varphi_z]^T$ - the vector of generalized coordinates;

$\underline{\underline{\dot{q}}} = [\dot{q}_1, \dot{q}_2, \dot{q}_3, \dot{q}_4, \dot{q}_5, \dot{q}_6]^T = [\dot{X}, \dot{Y}, \dot{Z}, \dot{\varphi}_x, \dot{\varphi}_y, \dot{\varphi}_z]^T$ - the vector of generalized velocities;

$\underline{\underline{\ddot{q}}} = [\ddot{q}_1, \ddot{q}_2, \ddot{q}_3, \ddot{q}_4, \ddot{q}_5, \ddot{q}_6]^T = [\ddot{X}, \ddot{Y}, \ddot{Z}, \ddot{\varphi}_x, \ddot{\varphi}_y, \ddot{\varphi}_z]^T$ - the vector of generalized accelerations;

f - the vector of generalized forces;

$$\underline{\underline{f}} = \begin{Bmatrix} f_1 \\ f_2 \\ f_3 \\ f_4 \\ f_5 \\ f_6 \end{Bmatrix} = \begin{Bmatrix} Q_X^F \\ Q_Y^F \\ Q_Z^F \\ Q_{\varphi_x}^F \\ Q_{\varphi_y}^F \\ Q_{\varphi_z}^F \end{Bmatrix} = \begin{Bmatrix} \sum_{k=1}^{p} F_{kx} \\ \sum_{k=1}^{p} F_{ky} \\ \sum_{k=1}^{p} F_{kz} \\ \sum_{k=1}^{p} \left(y_k F_{kz} - z_k F_{ky} \right) + \sum_{l=1}^{q} M_{lx} \\ \sum_{k=1}^{p} \left(z_k F_{kx} - x_k F_{kz} \right) + \sum_{l=1}^{q} M_{ly} \\ \sum_{k=1}^{p} \left(x_k F_{ky} - y_k F_{kx} \right) + \sum_{l=1}^{q} M_{lz} \end{Bmatrix} \qquad (3)$$

$\underline{A}$ - inertial matrix;

$$\underline{A} = \begin{bmatrix} m & 0 & 0 & 0 & 0 & 0 \\ 0 & m & 0 & 0 & 0 & 0 \\ 0 & 0 & m & 0 & 0 & 0 \\ 0 & 0 & 0 & J_x & 0 & 0 \\ 0 & 0 & 0 & 0 & J_y & 0 \\ 0 & 0 & 0 & 0 & 0 & J_z \end{bmatrix} \qquad (4)$$

$\underline{B}$ - damping matrix;

$$\underline{B} = \begin{bmatrix} \sum c_{jx} & 0 & 0 & 0 & \sum c_{jx}z_j & -\sum c_{jx}y_j \\ 0 & \sum c_{jy} & 0 & -\sum c_{jy}z_j & 0 & \sum c_{jy}x_j \\ 0 & 0 & \sum c_{jz} & \sum c_{jz}y_j & -\sum c_{jz}x_j & 0 \\ 0 & -\sum c_{jy}z_j & \sum c_{jz}y_j & \sum \left(c_{jy}z_j^2 + c_{jz}y_j^2\right) & -\sum c_{jz}x_jy_j & -\sum c_{jy}z_jx_j \\ \sum c_{jx}z_j & 0 & -\sum c_{jx}x_j & -\sum c_{jz}x_jy_j & \sum \left(c_{jz}x_j^2 + c_{jx}z_j^2\right) & -\sum c_{jy}y_jz_j \\ -\sum c_{jx}y_j & \sum c_{jy}x_j & 0 & -\sum c_{jy}z_jx_j & -\sum c_{jy}y_jz_j & \sum \left(c_{jx}y_j^2 + c_{jy}x_j^2\right) \end{bmatrix} \qquad (5)$$

$\underline{C}$ - rigidity matrix;

$$\underline{C} = \begin{bmatrix} \sum k_{jx} & 0 & 0 & 0 & \sum k_{jx}z_j & -\sum k_{jx}y_j \\ 0 & \sum k_{jy} & 0 & -\sum k_{jy}z_j & 0 & \sum k_{jy}x_j \\ 0 & 0 & \sum k_{jz} & \sum k_{jz}y_j & -\sum k_{jz}x_j & 0 \\ 0 & -\sum k_{jy}z_j & \sum k_{jz}y_j & \sum \left(k_{jy}z_j^2 + k_{jz}y_j^2\right) & -\sum k_{jz}x_jy_j & -\sum k_{jy}z_jx_j \\ \sum k_{jx}z_j & 0 & -\sum k_{jx}x_j & -\sum k_{jz}x_jy_j & \sum \left(k_{jz}x_j^2 + k_{jx}z_j^2\right) & -\sum k_{jy}y_jz_j \\ -\sum k_{jx}y_j & \sum k_{jy}x_j & 0 & -\sum k_{jy}z_jx_j & -\sum k_{jy}y_jz_j & \sum \left(k_{jx}y_j^2 + k_{jy}x_j^2\right) \end{bmatrix} \qquad (6)$$

In this way will consider that foundation is place on the four identically viscoelastic elements, and she have one plane of symmetry (fig. 3).

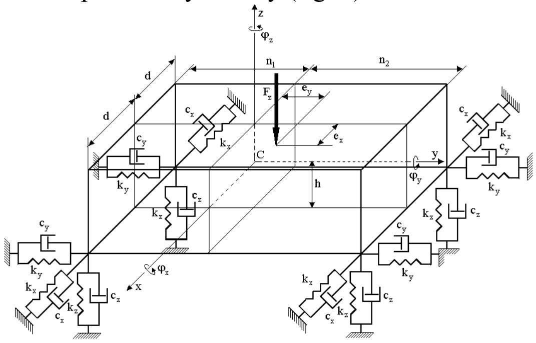

Fig. 3

This model presents a general character, the possible rigid’s movement is:

X – forcing lateral vibration;

Y - forcing longitudinal vibration;

Z - forcing vertical vibration

φ_x - forcing pitching vibration;

φ_y - forcing rolling vibration;

φ_z - forcing turning vibration;

The principal axis of the elastic supports are parallel with the references axis. In this case, the movement on the six degree of freedom is decoupled in two possibilities: Y, Z and φ_x and coupled movement who are representing through the coordinate variations Y, φ_y and φ_z.

Forwards, will be analyzed the coupled model Y, Z, φ_x.

The mathematical model who characterized this system dynamically is:

$$\begin{aligned} & m\ddot{X} + 4c_x\dot{X} - 4hc_x\dot{\varphi}_y - 2c_x(n_2 - n_1)\dot{\varphi}_z + 4k_xX - 4hk_x\varphi_y - 2k_x(n_2 - n_1)\varphi_z = 0 \\ & J_y\ddot{\varphi}_y - 4hc_x\dot{X} + 4\left(c_zd^2 + c_xh^2\right)\dot{\varphi}_y + 2c_xh(n_2 - n_1)\dot{\varphi}_z - \\ & \qquad - 4hk_xX + 4\left(k_zd^2 + k_xh^2\right)\varphi_y + 2k_xh(n_2 - n_1)\varphi_z = 0 \\ & J_z\ddot{\varphi}_z - 2c_x(n_2 - n_1)\dot{X} + 2hc_x(n_2 - n_1)\dot{\varphi}_y + 2\left[c_x\left(n_2^2 + n_1^2\right) + 2c_yd^2\right]\dot{\varphi}_z - \\ & \qquad - 2k_x(n_2 - n_1)X + 2hk_x(n_2 - n_1)\varphi_y + 2\left[k_x\left(n_2^2 + n_1^2\right) + 2k_yd^2\right]\varphi_z = 0 \end{aligned} \qquad (7)$$

where:

m – foundation mass;

k – rigidity of the viscous-elastic element;

c – damping of the viscous-elastic elements;

J – inertia moments of the foundation block.

The analyze of this system will be make evaluating three cinematically measure: - acceleration, velocity, movement, and frequency response.

The excitation force is on OZ direction, applied point being eccentrically toward mass centre fig. 3. The excitation of the system is half-sine shock pulse (fig. 4), the applied being T=0.005 s.

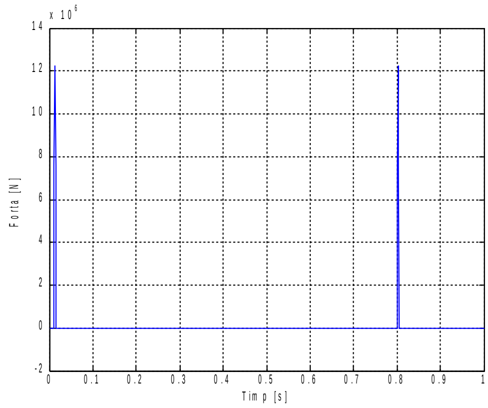

Fig. 4

The equation system was resolve with Runge – Kutta method with absolute error 10^{-5}.
Rate of curves from the three cinematically measure are presents in the next figure 5, 6 and 7.

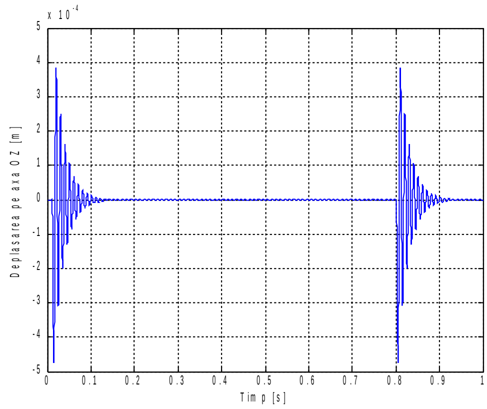

Fig. 5

The solving system has made in hypothesis of the next’s numerical value:
$P=1288\cdot10^4$N; $k_y=2\cdot10^9$ N/m; $c_y=2.5\cdot10^6$ Ns/m; $m=100\cdot10^3$ kg; $k_z=9505\cdot10^7$ N/m; $c_z=2.1\cdot10^6$ Ns/m; $J=77\cdot10^4$ kgm^2; e=0.05 m; n_1=3m; n_2=3m; h=1.5m.

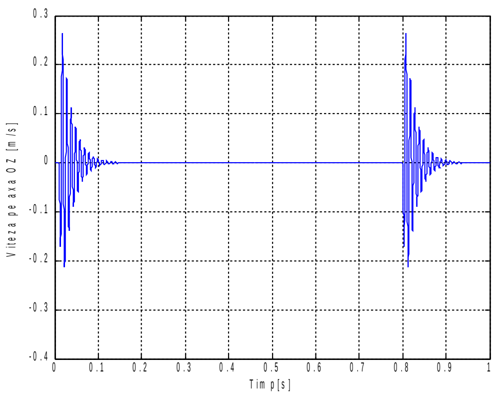

Fig. 6

These three cinematically measure are quantitative criteria for evaluating the vibration effects on the human structure or on environment.

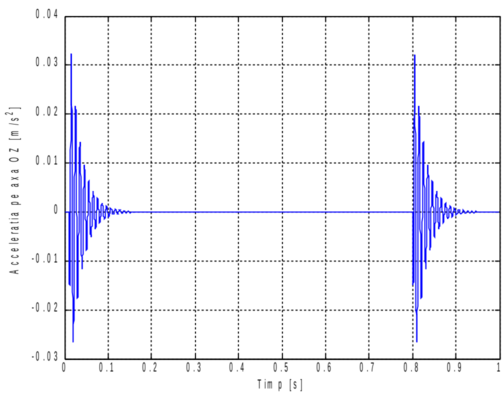

Fig. 7

Eliminating the time from movement and velocity expression, obtain the characteristically curve or movement trajectory (fig. 8).

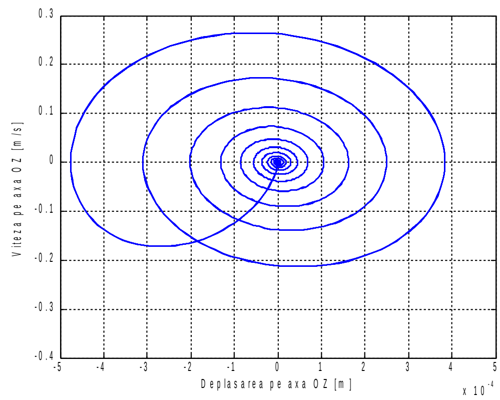

Fig. 8

From the figure 8 observe that movement is damping and stable because the amplitude of movement don’t grow on infinite value.

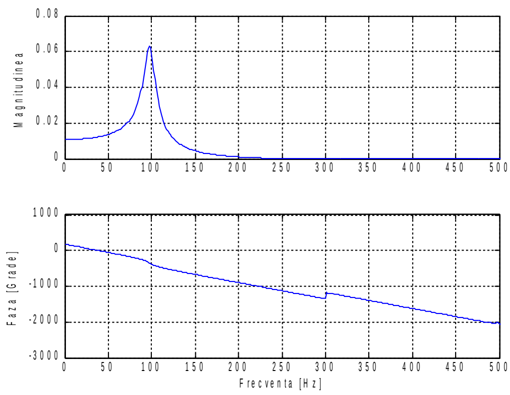

Fig. 9

In the figure 9 is presented the system response in the frequency representation.

From this representation we observe that dominant frequency domain is belonging around on 97Hz value.

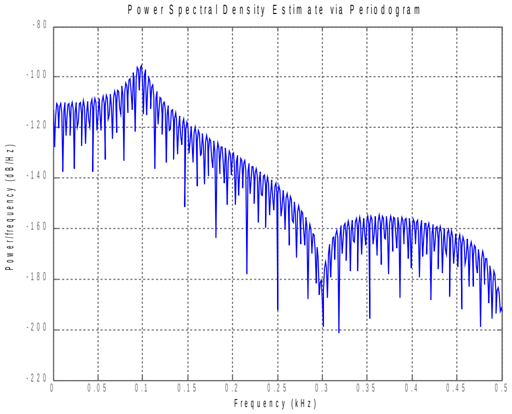

Fig. 10

Another analyze in frequency response is power spectral density (fig. 10). Because the elements on is place foundation are viscous-elastic characteristic, these elements dissipate energy – hysteretic W=1206.6W.

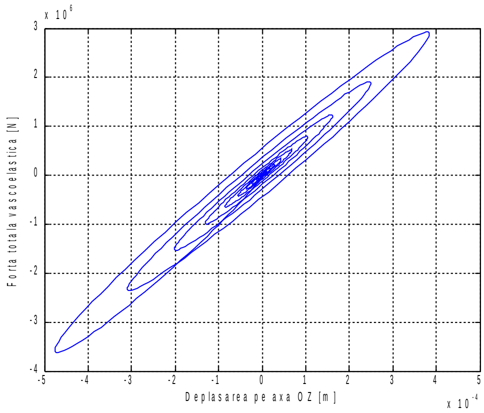

Fig. 11

3. Conclusions:

This paper presents a theoretical able his model characterize from dynamic viewpoint, gama very a diverse of real situation technological which equipments utilize the in the production process shocks and vibration.

Thus through differed the digital values of the elastic constant the si of damping, is can estimated response dynamically of systems the thus this impact about human factors the and builded area.

The characterization response dynamically of systems through three one kinematic measure he permits the appreciation vibration result from the viewpoint of admissible limits established vibration through standard in vigor.

Also, through the characterization response dynamically of systems frequent identify spectral components wherewith is disipate energy elders.

4. Bibliography:

[1] **Bratu, P.**, - *Sisteme elastice de rezemare pentru masini si utilaje*, Editura Tehnica, Bucuresti, 1990;

[2] **Bratu, P.**, - *Vibratiile sistemelor elastice*, Editura Tehnica, Bucuresti, 1999;

[3] **Bratu, P.**, - *Izolarea si amortizarea vibratiilor la utilaje de constructii*, Editura INCERC, Bucuresti, 1982;

[4] **Bratu, P.**, - *Vibratii neliniare* , Editura IMPULS, Bucuresti, 2001;

[5] **Buzdugan, Gh.**, - *Izolarea antivibratorie*, Ed. Academiei Romane, Bucuresti, 1993;

[6] **Buzdugan, Gh., Hamburgher. H., Wermescher. V.**, - *Fundatii de masini*, Ed. Tehnica, Bucuresri, 1958;

[7] **Buzdugan, Gh.**, - *Izolarea antivibratorie a masinilor*, Ed. Academiei Romane, Bucuresti, 1980;

[8] **Buzdugan, Gh.,** - *Vibratiile sistemelor mecanice*, Ed. Academiei Romane, Bucuresti, 1975.

[9] **Leopa, A., Dragan, N., Anghelache, D., Debeleac, C.** - *The influence of nonlinear damping on dynamic action equipment* - 23-25 Noiembrie 2005, trans & MOTAUTO'05+, Veliko Tarnovo, Bulgaria, ISBN 954-9322-11-4, pg. 135-138;

[10] ***Leopa, A.*** - *The influences of nonlinear viscoelastic systems behaviour about dynamic's of the foundation of technological equipment* - Noiembrie 2005, Disiparea energiei procese acustice, vibratorii si seismice, ICECON, Bucuresti, Romania, Editura Impuls, ISBN 973-8132-53-3, pg. 121-124.

[11] **Leopa Adrian** – *Influenta elementelor de amortizare vascoelastice asupra utilajului tehnologic* – A XXVIII- a Conferinta Nationala de Mecanica Solidelor, 28 -29 mai Targoviste, 2004, ISBN 973 – 86834 – 2 - 4, pg. 18-21;

[12] **Leopa Adrian** - *The model of forging hammers like system with three degree of freedom* - Analele Universitatii "Dunarea de Jos" din Galati, 2004, pg. 31-34, ISSN 1224-5615;