EVALUATION OF QUALITY DYNAMIC PERFORMANCES IN CASE OF VIBRATING ROLLERS USED FOR ROAD WORKS*

Polidor BRATU

The paper is focused on the quality concept using dynamic performances expressed by vibration transmissibility. Thus, the dynamic effect of vibrating roller is analyzed to the compacting surface and, simultaneously, to the cab chassis. The optimum domain for vibrating regime is established based on the calculus dynamic model, so that the force transmitted to the compacting surface is maximum and the force transmitted to the cab is minimum.

1. FUNDAMENTALS OF THE VIBRATING ROLLERS

Dynamic action machines intended for compaction of road layers consisting on soil, aggregates, asphalt mixtures are designated for compaction technologies using translation and rolling motions of the vibrating rollers. The most important problems are located at the self-propelled vibrating rollers and ride-on control post level owing to the following:

– the vibrations transmitted to the control post should be framed in permissible limits in order to provide the operator's safety and

– the technological vibrations transmitted to the medium to be compacted should have values high enough in order to assure the designed compacting degree.

In this situation the functional and construction structure should be adopted so that the two mentioned desiderata are simultaneously attained. Aiming to parametrically express the imposed requirements, the following parameters must be evaluated:

– transmissibility and insulation of vibrations transmitted by the vibrating roller to the cabin;

– transmissibility of the vibration force transmitted by the vibrating roller to the working medium (soil, asphalt mixture).

*Presented at SISOM2007 and Homagial Session of the Commission of Acoustics, Bucharest, 29–31 May 2007.

Research Institute for Construction Equipment and Technology – ICECON SA, Bucharest, E-mail: icecon@icecon.ro

Though the two requirements are contradictory, meaning that in situ “hard” vibrations are imposed and in the cabin “soft” vibrations having low intensity for the vibrating machines in this class, the performance could be obtained by an appropriate dynamic analysis.

In this paper the dynamic performance for a self-propelled vibrating compaction machine is analysed. This consists on a rigid chassis on which the cabin is mounted and two rollers. The vibrating roller is mounted in front and the rear roller assures the traction. Both rollers perform the static compaction process. More, the vibrating roller performs the dynamic compaction also.

2. DYNAMIC PARAMETERS ANALYSIS

The dynamic model for a vibrating compaction machine is illustrated in Fig. 1:

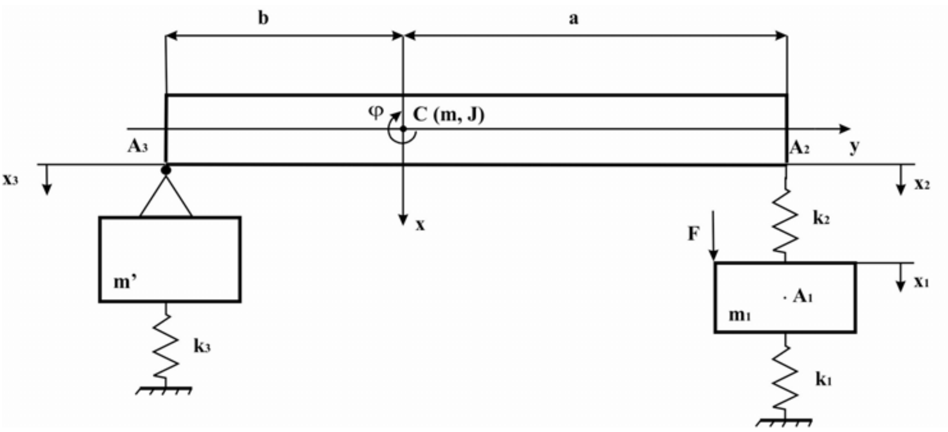

Fig. 1 – Dynamic model of the compacting machine.

where we denoted as follows:

A_1	–	the punctiform model of the front vibrating roller;
A_2	–	the connection point of the chassis edge to the elastic support with the vibrating roller;
A_3	–	the connection point between the chassis and the rear static roller;
m_1	–	the vibrating roller mass;
m	–	the chassis mass;
J	–	the chassis inertia moment related to the mass centre C;
M'	–	the static roller mass;
K_1, k_3	–	the rigidity of the medium to be compacted;
k_2	–	the rigidity of the elastic system intended for vibration insulation;
a, b	–	the mass centre position related to the chassis edge, with $a + b = l$, where l represents the chassis equivalent length;
x, φ, x_1	–	the generalised co-ordinates for the material system;

x_1, x_2, x_3 – the absolute co-ordinates for the system representing the absolute instantaneous displacements related to the fixed co-ordination system with respect to the ground.

The motion differential equations could be stated basing on three methods, namely:

a) using the generalised co-ordinations x_1, x and φ by means of the *rotation anti-symmetric matrix*;
b) using elastic elements deformations basing on the *influence matrix of displacements* upon the deformations;
c) using *absolute displacements* of the points A_1, A_2 and A_3.

a) **The generalised co-ordination method.** The instantaneous displacement in point A_i, $i = 1, 2, 3$ is given by the matrix relation [1] of the form:

$$u_i = \begin{Bmatrix} x \\ y \\ z \end{Bmatrix} + \begin{bmatrix} 0 & -\varphi_z & \varphi_y \\ \varphi_z & 0 & -\varphi_x \\ -\varphi_y & \varphi_x & 0 \end{bmatrix} \begin{Bmatrix} x_i \\ y_i \\ z_i \end{Bmatrix} \tag{1}$$

where: x, y, z are linear co-ordinates of the point in the fixed system axis direction, representing possible displacements;
x_i, y_i, z_i – the linear co-ordinates of the point related to the mass centre of the rigid body in rotation and translation instantaneous motion;
φ_x, φ_y, φ_z – the instantaneous angular co-ordinates of the rigid body related to the three-orthogonal axes having the origin in the mass centre.

For the given model it results in the following relations:

$$u_{A_1} = \begin{Bmatrix} x_1 \\ 0 \\ 0 \end{Bmatrix} + \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} 0 \\ 0 \\ 0 \end{Bmatrix} = x_1,$$

$$u_{A_2} = \begin{Bmatrix} x \\ 0 \\ 0 \end{Bmatrix} + \begin{bmatrix} 0 & -\varphi_z & 0 \\ \varphi_z & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} 0 \\ a \\ 0 \end{Bmatrix} = x - a\varphi_z,$$

$$u_{A_3} = \begin{Bmatrix} x \\ 0 \\ 0 \end{Bmatrix} + \begin{bmatrix} 0 & -\varphi_z & 0 \\ \varphi_z & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} 0 \\ -b \\ 0 \end{Bmatrix} = x + b\varphi_z,$$

and for $\varphi_z = \varphi$, we have:

$$u_{A_1} = x_1,$$
$$u_{A_2} = x - a\varphi, \tag{2}$$
$$u_{A_3} = x + b\varphi.$$

The deformation potential energy for the elastic system is given by the relation

$$2V = k_1 u_{A_1}^2 + k_2\left(u_{A_2} - x_1\right)^2 + k_3 u_{A_3}^2 \tag{3}$$

or

$$2V = k_1 x_1^2 + k_2\left(x - a\varphi - x_1\right)^2 + k_3\left(x + b\varphi\right)^2 . \tag{4}$$

The restoration potential forces could be written under the form

$$Q_1 = -\frac{\partial V}{\partial x_1};\ Q_1 = -\left[k_1 x_1 - k_2\left(x - a\varphi - x_1\right)\right], \tag{5}$$

$$Q_2 = -\frac{\partial V}{\partial x};\ Q_2 = -\left[k_2\left(x - a\varphi - x_1\right) + k_3\left(x + b\varphi\right)\right], \tag{6}$$

$$Q_3 = -\frac{\partial V}{\partial \varphi};\ Q_3 = -\left[-k_2 a\left(x - a\varphi - x_1\right) + k_3 b\left(x + b\varphi\right)\right]. \tag{7}$$

The kinetic energy of the system is determined as:

$$2E = m_1\dot{x}_1^2 + m_1\dot{x}^2 + J\dot{\varphi}^2 + m'\dot{x}_3^2 , \tag{8}$$

where introducing $\dot{x}_3 = \dot{u}_{A_3}^2 = \dot{x} + b\dot{\varphi}$ we have:

$$2E = m_1\dot{x}_1^2 + m\dot{x}^2 + J\dot{\varphi}^2 + m'\left(\dot{x} + b\dot{\varphi}\right)^2$$

or

$$2E = m_1\dot{x}_1^2 + \left(m + m'\right)\dot{x}^2 + \left(J + m'b^2\right)\dot{\varphi}^2 + 2m'b\dot{x}\dot{\varphi} .$$

Using the following notations for the inertia coefficients:

$$\widetilde{m}_1 = m_1; \quad \widetilde{m}_2 = m + m'; \quad \widetilde{m}_3 = J + mb^2; \quad \widetilde{m}_{23} = m'b;$$

the kinetic energy can be written as

$$2E = \widetilde{m}_1\dot{x}_1^2 + \widetilde{m}_2\dot{x}^2 + \widetilde{m}_3\dot{\varphi}^2 + 2\widetilde{m}_{23}\dot{x}\dot{\varphi} . \tag{9}$$

Applying the second order Lagrange equations it results in the following differential equation system:

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial E}{\partial \dot{x}_j}\right)-\frac{\partial E}{\partial x_j}=Q_j^V+Q_j^F\text{ },j=1,2,3, \tag{10}$$

where: $Q_j^V = -\dfrac{\partial V}{\partial x_j}$ represent the restoration generalised forces, previously determined according to relations (5), (6) and (7); $Q_j^F = \dfrac{\delta L_j^F}{\delta x_j}$ – the generalised forces corresponding to the perturbing forces and for the given system we have $Q_1 = \dfrac{F\delta x_1}{\delta x_1} = F = F_0 \sin \omega t$.

In accordance with the above mentioned statements and taking into account relations (4), (5), (6), (7) and (8), Lagrange equations (10) become:

$$\begin{aligned}
&m_1\ddot{x}_1+\left(k_1+k_2\right)x_1-k_2x+ak_2\varphi=F_0\sin\omega t,\\
&m\ddot{x}+m_{23}\ddot{\varphi}-k_2x_1+\left(k_2+k_3\right)x+(-ak_2+bk_3)\varphi=0,\\
&m_{23}\ddot{x}+m_3\ddot{\varphi}+ak_2x_1+(-ak_2+bk_3)x+\left(a^2k_2+b^2k_3\right)\varphi=0.
\end{aligned} \tag{11}$$

b) The matrix of displacements influence upon deformation. We denote the elastic elements deformations v_1, v_2, v_3 representing the components of the **deformation vector *v***. Thus, its components are:

$$\begin{aligned}
v_1 &= x_1\\
v_2 &= x - a\varphi - x_1\\
v_3 &= x + b\varphi
\end{aligned}$$

where $v^{\mathrm{T}} = \left[v_1,\ v_2,\ v_3\right]$.

The generalised co-ordinates vector **q** consists on three components of the linear (x_1, x) and angular (φ) instantaneous displacements, of the form $q^{\mathrm{T}} = \left[x_1,\ x,\ \varphi\right]$.

The linear correlation between v and q could be written as:

$$\mathbf{v} = \mathbf{A}\mathbf{q}, \tag{12}$$

where **A** stands for the matrix of displacements influence upon deformations and is structured under the form:

$$\mathbf{A} = \begin{bmatrix} 1 & 0 & 0 \\ -1 & 1 & -a \\ 0 & 1 & b \end{bmatrix}. \tag{13}$$

The kinetic and potential energy in the matrix formulation are:

● **the kinetic energy**

$$2E_c = \left\langle \dot{\mathbf{q}}, \widetilde{\mathbf{M}}\dot{\mathbf{q}} \right\rangle \tag{14}$$

with $\dot{\mathbf{q}}^{\mathbf{T}} = \left[\dot{x}_1,\ \dot{x},\ \dot{\varphi}\right]$ – the generalised velocity vector; $\widetilde{\mathbf{M}} = \begin{bmatrix} \widetilde{m}_1 & 0 & 0 \\ 0 & \widetilde{m}_2 & \widetilde{m}_{23} \\ 0 & \widetilde{m}_{23} & \widetilde{m}_3 \end{bmatrix}$ – the mass matrix.

● **the potential energy**

$$2V = <\mathbf{v},\ \mathbf{K_0 v}> = <\mathbf{Aq},\ \mathbf{K_0 Aq}> = <\mathbf{q},\ \mathbf{A^T K_0 Aq}>$$

or

$$2V = <q,\ Kq> \tag{15}$$

with: $q^{\mathrm{T}} = [x_1,\ x,\ \varphi]$ the generalised co-ordinates vector; $K_0 = \mathrm{diag}[k_1,\ k_2,\ k_3]$ – the diagonal matrix of rigidities; $K = A^{\mathrm{T}} K_0 A = \begin{bmatrix} k_1 + k_2 & k_2 & ak_2 \\ -k_2 & k_2 + k_3 & -ak_2 + bk_3 \\ ak_2 & -ak_2 + bk_3 & a^2 k_2 + b^2 k_3 \end{bmatrix}$ – the rigidity matrix of the system having “coupling” elements.

The differential equation has the matrix form:

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{Kq} = \mathbf{f}\,, \tag{16}$$

where $\mathbf{f}^{\mathbf{T}} = [F_0 \sin \omega t, 0, 0]$.

Developing equation (16) it results in the analytic expression of the motion differential equations having the same form as relations (11).

c) Absolute displacements for the points A_1, A_2 and A_3 method. In the kinetic and potential expressions, the generalised co-ordinates x and φ are replaced by the absolute displacements in points A_2 and A_3, by x_2 and x_3 respectively. Thus, by means of the transformation matrix **T**, vector $\mathbf{q_{23}}$ determines the vector $\mathbf{x_{23}}$ under the form:

$$\begin{Bmatrix} x_2 \\ x_3 \end{Bmatrix} = \begin{bmatrix} 1 & a \\ 1 & -b \end{bmatrix} \begin{Bmatrix} x \\ \varphi \end{Bmatrix}$$

or

$$\mathbf{x_{23}} = \mathbf{Tq_{23}} \tag{17}$$

and further on $\mathbf{q_{23}} = T^{-1}\mathbf{x_{23}}$, where

$$T^{-1} = \frac{T^{adj}}{\det T} = \frac{1}{a+b}\begin{bmatrix} b & a \\ -1 & -1 \end{bmatrix} \tag{18}$$

finally we obtain:

$$x = \frac{1}{a+b}\left(ax_3 + bx_2\right),\ \varphi = \frac{1}{a+b}\left(x_2 - x_3\right).$$

In this case, the kinetic energy given by relation (8) becomes:

$$2E = m_1\dot{x}_1^2 + m\frac{1}{(a+b)^2}\left(a\dot{x}_3 + b\dot{x}_2\right)^2 + \frac{1}{(a+b)^2}J\left(\dot{x}_2 - \dot{x}_3\right)^2 + m'\dot{x}_3^2 \tag{19}$$

or after developing and reorganising we have:

$$2E = m_1\dot{x}_1^2 + m_2\dot{x}_2^2 + m_3\dot{x}_3^2 + 2m_{23}\dot{x}_2\dot{x}_3, \tag{20}$$

with the inertia coefficients:

$$m_1 = m_1;\ m_2 = \frac{1}{(a+b)^2}\left(mb^2 - J\right);\ m_3 = \frac{1}{(a+b)^2}\left[ma^2 + J + m'(a+b)^2\right];$$

$$m_{23} = \frac{1}{(a+b)^2}\left(mab - J\right).$$

The potential energy will be of the form:

$$2V = k_1x_1^2 + k_2\left(x_1 - x_2\right)^2 + k_3x_3^2, \tag{21}$$

where the elastic elements displacements are $\Delta_1 = x_1$, $\Delta_2 = x_1 - x_2$, $\Delta_3 = x_3$, and the deformation vector is $\mathbf{\Delta}^{\mathbf{T}} = \left[\Delta_{1,}\ \Delta_{2,}\ \Delta_3\right]$.

As a matrix formulation we have:

- **the kinetic energy**

$$2E = <\dot{\mathbf{x}}, \mathbf{M}\dot{\mathbf{x}}>, \tag{22}$$

with **M** representing the inertia matrix positive defined symmetric and non-singular under the form:

$$\mathbf{M} = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & m_{23} \\ 0 & m_{23} & m_3 \end{bmatrix}.$$

- **the potential energy**

$$2\mathrm{V} = <\mathbf{\Delta}, \mathbf{K_0}\mathbf{\Delta}>, \tag{23}$$

with $\mathbf{K_0}$ the diagonal matrix for the rigidity coefficients defined as $\mathbf{K_0} = \mathrm{diag}[k_1,\ k_2,\ k_3]$ and $\mathbf{\Delta}$ the deformation vector.

- the linear transformation correlating the deformations and displacements

$$\mathbf{\Delta} = \mathbf{Bx}, \tag{24}$$

where **B** represents the matrix of displacements influence upon deformations expressed as:

$$\mathbf{B} = \begin{bmatrix} 1 & 0 & 0 \\ 1 & -1 & 0 \\ 0 & 0 & 1 \end{bmatrix}. \tag{25}$$

By linear transformation (24) one could express the potential energy as a function of absolute displacements, relating to the vector $\mathbf{x}^{\mathrm{T}} = [x_1,\ x_2,\ x_3]$, thus:

$$2V = \langle \mathbf{Bx}, \mathbf{K_0Bx} \rangle. \tag{26}$$

Considering the matrix **B** fulfils conditions of a linear operator continuous on Hilbert space, the adjoint operator can be adopted as matrix $\mathbf{B}^{\mathrm{T}}$, where “T” means the transformation operation for matrix **B**.

This propriety can be put into value in the scalar product:

$$2V = \langle \mathbf{Bx}, \mathbf{K_0Bx} \rangle = \langle \mathbf{x}, \mathbf{B}^{\mathrm{T}}\mathbf{K_0Bx} \rangle,$$

where $\mathbf{K} = \mathbf{B}^{\mathrm{T}}\mathbf{K_0B}$ is the system rigidity matrix.

Thus we have

$$2V = \langle \mathbf{x}, \mathbf{Kx} \rangle \tag{27}$$

and the whole system rigidity matrix **K** results in:

$$\mathbf{K} = \mathbf{B}^{\mathrm{T}}\mathbf{K_0B} = \begin{bmatrix} 1 & 1 & 0 \\ 0 & -1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} k_1 & 0 & 0 \\ 0 & k_2 & 0 \\ 0 & 0 & k_3 \end{bmatrix} \begin{bmatrix} 1 & 0 & 0 \\ 1 & -1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

and

$$\mathbf{K} = \begin{bmatrix} k_1 + k_2 & -k_2 & 0 \\ -k_2 & k_2 & 0 \\ 0 & 0 & k_3 \end{bmatrix}. \tag{28}$$

For the linear-elastic range, the matrix formulation of the motion equation has the form:

$$\mathbf{M\ddot{x}} + \mathbf{Kx} = \mathbf{f}\ , \tag{29}$$

where **f** is the perturbing forces vector expressed as:

$$\mathbf{f}^{T} = [F_0 \sin\omega t, 0, 0] \tag{30}$$

with $F_0 = m_0 r\omega_2$ is the static moment of the dynamic unbalanced masses for the vibrator located in the vibrating roller mass centre and ω is the exciting pulsation (angular velocity of the dynamic unbalanced masses).

As an analytical formulation, developing products in the matrix equation (29), the linear differential equation system is obtained under the form:

$$\begin{aligned} &m_1\ddot{x}_1 + (k_1 + k_2)x_1 - k_2 x_2 = m_0 r\omega^2 \sin\omega t, \\ &m_2\ddot{x}_2 + m_{23}\ddot{x}_3 + k_2 x_2 - k_2 x_1 = 0, \\ &m_3\ddot{x}_3 + m_{23}\ddot{x}_2 + k_3 x_3 = 0. \end{aligned} \tag{31}$$

Solutions of the form $x_j = A_j \sin\omega t$, $j = 1, 2, 3$ lead to the forced vibration amplitudes for the linear-elastic system representing the dynamic model for the compaction machine. Thus, the forced vibration amplitudes under the steady regime are the following:

$$A_1 = H\,[(k_2 - m_2\omega^2)(k_3 - m_3\omega^2) - m_{23}^2\,\omega^4] \tag{32}$$

$$A_2 = H\,(k_3 - m_3\omega^2)k_2 \tag{33}$$

$$A_3 = H\,(\omega^2 k_2 m_{23}) \tag{34}$$

where $H = \dfrac{m_0 r\omega^2}{D}$ is the machine dynamic structure factor; $D = D(\omega)$ is the structure function for the linear-elastic system.

The structure function $D = D(\omega)$ is expressed as:

$$D = \omega^6 d_6 + \omega^4 d_4 + \omega^2 d_2 + d_0, \tag{35}$$

with the structure coefficients are given by the following relations: $d_6 = m_1(m_{23}^2 - m_2 m_3)$; $d_4 = (k_1 + k_2)(m_2 m_3 - m_{23}^2) + m_1(k_2 m_3 + k_3 m_2)$; $d_2 = -(k_1 + k_2)(k_2 m_3 + k_3 m_2) + k_2(k_2 m_3 - k_3 m_1)$; $d_0 = k_1 k_2 k_3$.

3. SYNTHESIS DYNAMIC PERFORMANCES FOR THE VIBRATING ROLLERS

Aiming compactors complete characterisation in the frame of the *performance concept* the two fundamental objectives should be defined in order to satisfy them either on normative base or requirements expressed by the machine user. Thus, in case of vibrating compaction machines, two requirements could be stated, namely:

a) assuring the technological capability in the vibration regime suitable for the vibrating roller so that the medium to be compacted attains the maximum compaction degree provided by the referenced technical documents. Aiming this, parameters such as amplitude A_1 and transmissibility T_{10} for the maximum dynamic force $F_0 = m_0 r\omega^2$ in situ expressed as transmitted force $F_T = F_0 T_{10}$ are followed;

b) assuring the safety for the working personnel at the operator's post inside the cabin, so that the vibrations transmitted have minimum values, limited in accordance with the health and safety norms. In this case, the concerned parameters defining the machine performance are the amplitudes A_2, A_3 and vibration transmissibility T_{12} from the vibrating roller to the chassis and the insulation degree $I_{12} = |1 - T_{12}|$, respectively.

The calculus relations for both performance categories are grouped in order to correspond to the previous requirements:

3.1. CAPABILITY TECHNOLOGICAL PERFORMANCES

Capability requirements are quantified by parametric performances expressed as follows:

– the amplitude $A_1 = A_1(\omega)$ is determined by relation (32) where the current variable ω allows function graphic. Three resonance points right on the vertical asymptotes are remarkable. Also the constant value A_∞ is attained for a post-resonance steady regime. This is illustrated by the horizontal asymptote corresponding to $A_\infty = \lim_{\omega\to\infty} A(\omega) = \frac{m_0 r}{m_1}$ representing the technological vibration amplitude (Fig. 2);

– the dynamic force transmissibility defined by relation $T_{10} = \frac{F_t^{\max}}{F_0}$, where $F_t^{\max}$ is the maximum force transmitted to the compacted medium. Thus, the transmitted force parameter is expressed as:

$$T_{10}(\omega) = \left|\frac{k_1 A_1(\omega)}{m_0 r\omega^2}\right| \tag{36}$$

and represented as a function of the current variation of ω (Fig. 3).

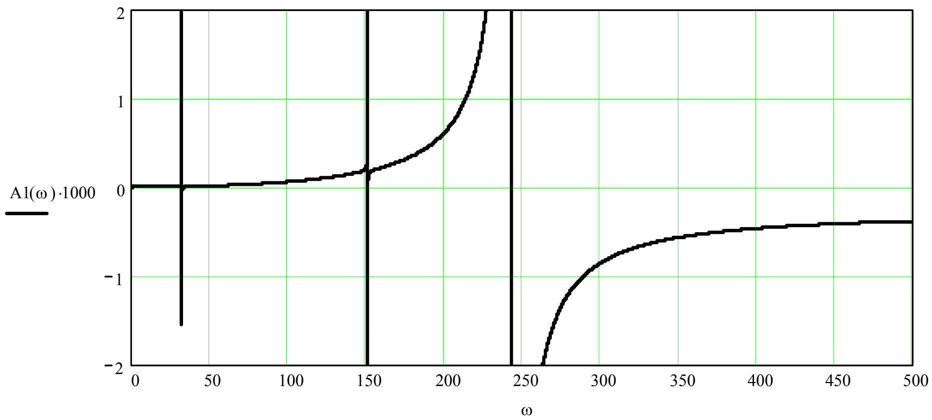

Fig. 2 – Variation of the technological vibration amplitude.

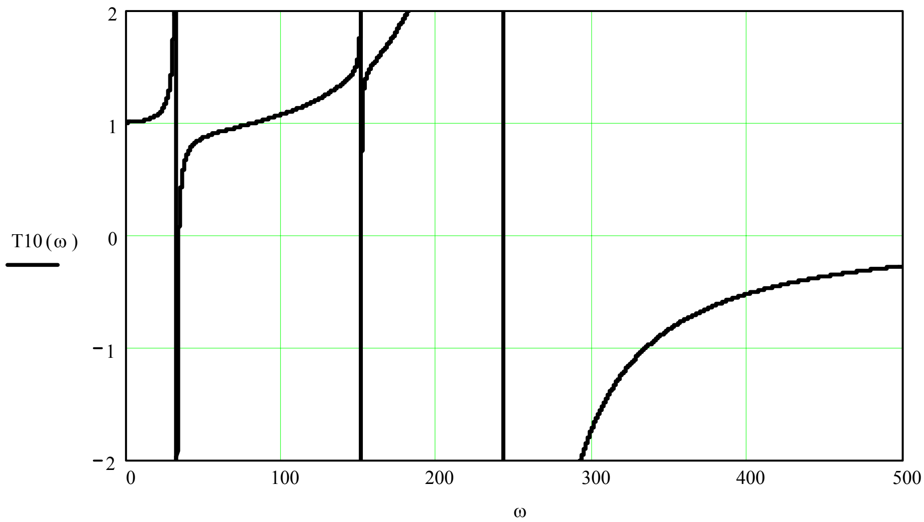

Fig. 3 – Variation of the transmitted force.

The determined parameters correspond to the technological vibration pulsation, namely for ω = 314 rad/s.

3.2. SAFETY PERFORMANCES

In order to attain the above mentioned objectives the performances parameters must be represented as follows:

– amplitudes $A_2(\omega)$, $A_3(\omega)$ are determined basing on relations (33) and (34) and for ω = 314 rad/s diminished values for the vibrations transmitted to the machine chassis are found (Figs. 3 and 4);

– synthesis parameters for the transmitted vibrations such as:

- transmissibility $T_{12} = \dfrac{F_{12}^{\max}}{F_0}$ expressed under the analytic form as:

$$T_{12}(\omega) = \left| \frac{k_2[A_1(\omega) - A_2(\omega)]}{m_0 r \omega^2} \right| \tag{37}$$

and illustrated in Fig. 6.

- vibration insulation degree expressed by relation:

$$I_{12}(\omega) = \left| 1 - T_{12}(\omega) \right| \tag{38}$$

and represented in Fig. 7.

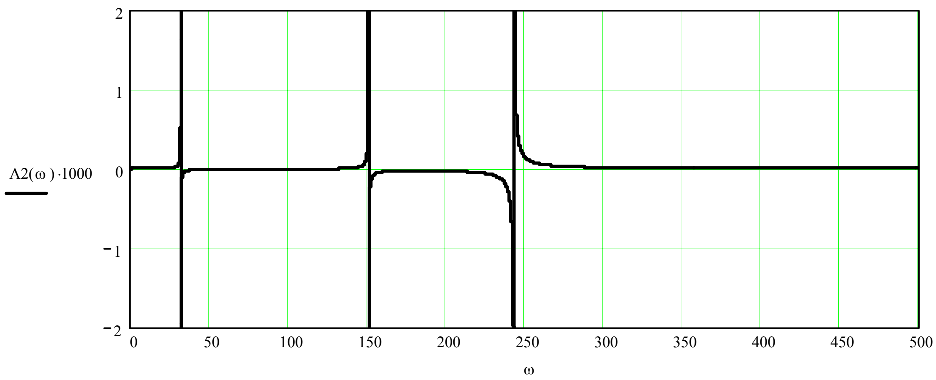

Fig. 4 – Amplitude $A_2(\omega)$.

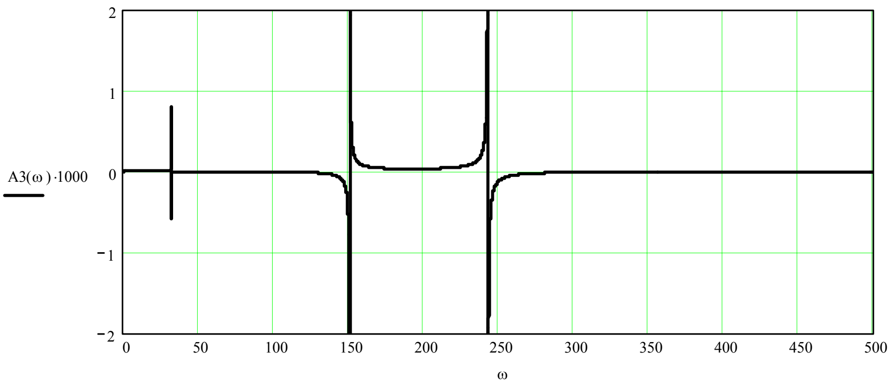

Fig. 5 – Amplitude $A_3(\omega)$.

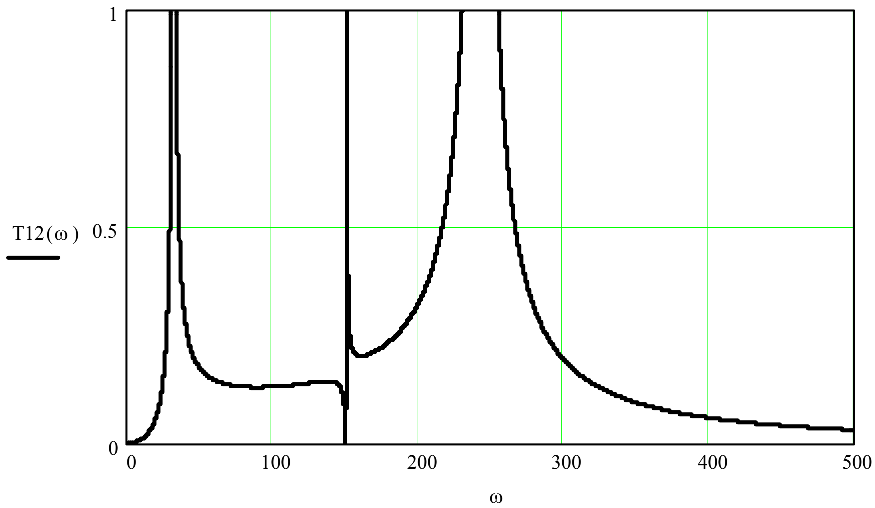

Fig. 6 – Transmissibility.

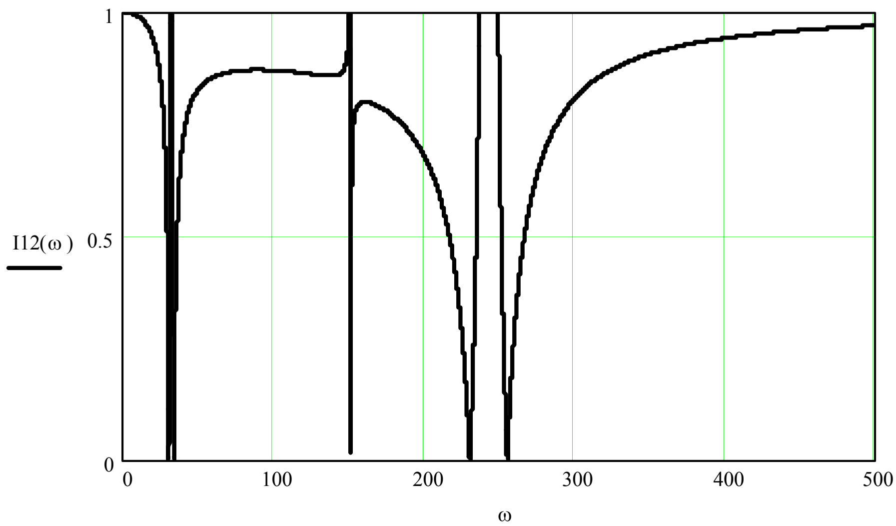

Fig. 7 – Vibration insulation degree.

4. CONCLUSIONS

In case of vibrating compaction machines the dynamic analysis is based on the performance concept stating the necessary requirements in order to attain the parametric objective such as:

a) assuring the technological vibrations parameters aiming to attain the compaction degree provided by norms or beneficiary's technical specification;

b) elastic system intended for vibration passive insulation so that the parameters of vibrations transmitted to the operator's position, in the cabin, provides the operator's health and safety according to the requirements stipulated in norms.

Aiming this, the main requirements converted in performance parameters have been transposed in the present study as:

● variation curves for the amplitude $A_1 = A_1(\omega)$ in case of the vibrating roller and variation of the transmitted force by means of the synthesis parameter $T_{10} = T_{10}(\omega)$. These dynamic quantities reflect the manner to provide the compacting machine capability during the technological process for the vibration pulsation in the steady regime $\omega = 314$ rad/s;

● variation curves for the parameters reflecting vibration insulation, such as: $A_2 = A_2(\omega)$, $A_3 = A_3(\omega)$, $T_{12} = T_{12}(\omega)$, and $I_{12} = I_{12}(\omega)$. For the vibration pulsation $\omega = 314$ rad/s the parameters values correspond to a global insulation degree of about 85 %.

Taking into consideration the model has been experimentally tested, the calculus relations can be applied in the conception as well as in the technological and logistic management stage for different works in this category.

REFERENCES

1. BRATU, P., *Elastic system vibrations*, Edit. Tehnică, Bucharest, 2000.
2. BRATU, P., *Vibration insulation and damping in case of construction machinery*, INCERC, Bucharest, 1982.
3. BRATU, P., *Non-linear vibrations*, Edit. Impuls, Bucharest, 2001.
4. COWLEY, R.H., MECH, A.M., *The tandem vibrating roller: its design and development in the field of dynamic compaction,* Proc.Instr.Mechn.Engs., **181**, part 2A, *3*, 1967.
5. FLASS, R., *Compaction Technology in earthwork, Highway and Transportation engineering*, Bomag GmbH & Co, Germany, 2001.
6. LEWIS, A.W., *Road rollers. Full-scale studies of the performance of plant in the compaction of soils and granular base materials,* Proc.Instr.Mechn.Engs., **181**, part 2A, *3*, 1967.
7. MIHĂILESCU, ŞT., BRATU, P., GORAN, V., *Construction machinery*, **II**, Edit. Tehnică, 1985.
8. SELING, E.T., YOO, T.S., *Fundamentals of vibratory roller behaviour,* International Conference of Soil Mechanics and Foundation Engineering, Paris, 1977.