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FIRST ELASTIC STEP VIBRATION ISOLATION OF SINGLE DRUM VIBRATORY ROLLER COMPACTOR

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Abstract: *The discrete variation of equivalent stiffness coefficient in the first elastic step is performed in order to determine the optimum solution of the dynamic regime in the vibratory compaction process. Research results on the influence of equivalent stiffness coefficient for the first elastic step over the dynamic response of single drum vibratory roller equipment are evidenced in this article. This influence is evidenced by the variation curves of vibration amplitude function of excitation pulsation. The favorable dynamic regime is defined by the discrete variation interval of excitation pulsation so that the vibration amplitude of the compaction vibratory roller to be high enough and adequate for the compaction technological process, as long as the other vibration amplitudes are small enough, fit to be neglected.*
The variation curves of vibration amplitude have been determined for the functional and constructive model of a self-propelled vibratory compactor with one vibratory roller (CVA 4-5) and anti-vibration elements designed and manufactured in homologated technology. The elastomer characteristics have been determined according to ongoing standards.
Keywords: *vibratory roller, vibration amplitude, isolation step, natural frequency, technological vibrations*

1. INTRODUCTION

The paper presents how to adjust the forced vibrations regime for 5 tones vibrating compactor, made and used in Romania, for road construction works. This compactor has one vibratory roller for the dynamic compaction and the other subassemblies that must be isolated from roller's vibrations. In fact, for the compaction equipment, there has to be solved simultaneously two problems that are: to produce and maintain the vibrations required for compaction technological process; to isolate the subassemblies from transmitted roller's vibrations.

The self-propelled vibrating compactors are characterized, on one hand by the work vibrations generated by the front roller that produce dynamic compaction of materials and, on the other hand by the chassis transmitted vibrations. These last ones have to be small enough, bellow the limits admitted or recommended by the mechanical equipment's producers. The self-propelled vibrating compactors (single drum vibratory roller) have, among other important components, the rubber anti-vibration elements that ensure elastic connection between subassemblies. They enable to generate and maintain vibration at the vibratory roller while isolating and damping unwanted vibrations at the other equipment's subassemblies, as: the cabin; engine – hydraulic pump drive group; auxiliary installation etc.

In Romania, at ICECON Bucharest, there were designed and manufactured this type of compaction equipment's, with elastic systems made of special rubber anti-vibration elements. The rubber material is designed and obtained by a homologated technology [1-7]. An important aspect to be mentioned is that the equipment's design includes two elastic steps, while most of the classic equipment's models (made by respected companies) include just one elastic step for isolation of vibrations.

This innovative way of conceiving the single drum vibratory compactor's structure lead to significant improvement of, both, vibration isolation parameters and working parameters.

In the design and, further, the exploitation phases there is the need to know the phenomena and rules that change the resonance zones and, also, the isolation effect as function of stiffness characteristics. So, there is the need for determination of stiffness discrete variation influence on the vibrations' amplitude and natural frequencies

For this purpose, there were conceived and applied three different recipes for manufacturing the rubber and, consequently, there were designed, manufactured and experimented several constructive ties of anti-vibration elements [8-11]. In the design phase, there is the possibility of choosing technical solutions for support, rubber type and geometric characteristics of anti-vibration element. In the exploitation phase there can be estimated the effect on functioning and isolation determined by changes of system's stiffness characteristic. More of it, because of rubber aging phenomenon, temperature variation and fatigue at cyclic loading, the stiffness characteristic of the system could vary in time [12-16].

This paper evidence aspects on how the operating dynamic regime and the vibrations amplitudes are influenced by the changes of stiffness coefficients values for the anti-vibration rubber elements – when the first step of isolation is studied. The equipment is the Romanian made self-propelled vibratory compactor with one vibratory roller, CVA 4-5.

2. METHOD FOR EVALUATION OF FORCED VIBRATIONS AMPLITUDE

The constructive model for vibratory roller compactor considered in research is shown in figure 1.

The simplified model of single drum vibratory roller [17-19] is evidenced below (figure 2), considering the following aspects: high degree of geometric and mass symmetry of the equipment with respect to the median longitudinal plane; symmetric construction of front chassis with respect to the axis of vibratory roller; equivalence of physical characteristics of the rubber elements.

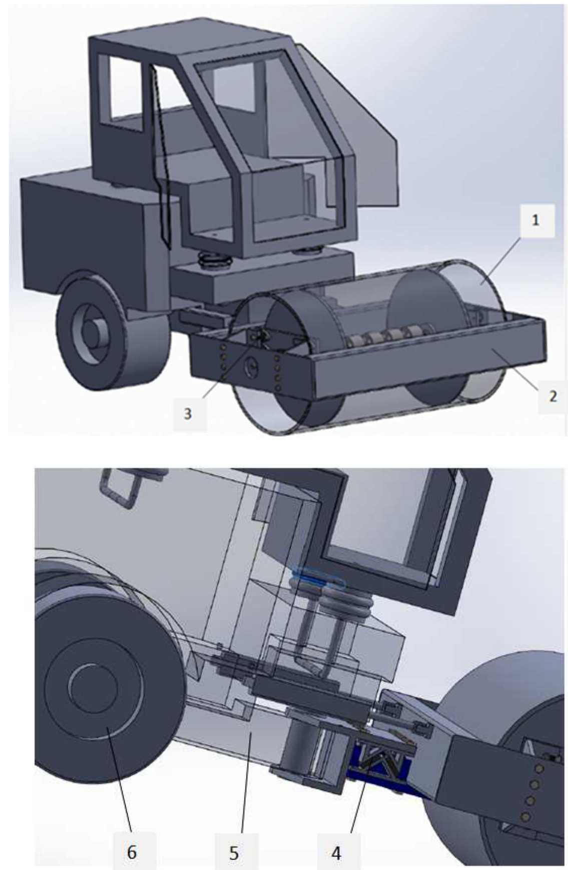


Figure 1. Constructive scheme for the vibrating compactor; 1 – vibratory roller; 2 – front chassis; 3 – first elastic step for vibration isolation and damping; 4 – second elastic step for vibration isolation and damping; 5 – rear chassis; 6 – propelling wheels.

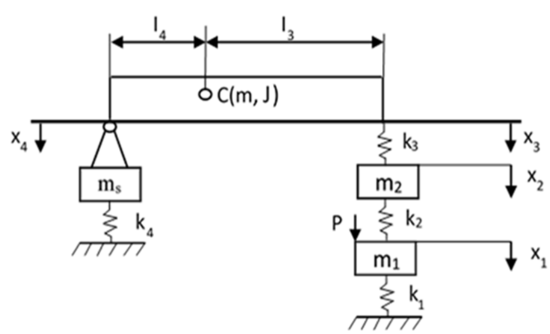


Figure 2. Dynamic model for the vibrating compactor

The two elastic steps for vibration isolation and damping were modeled only by elastic elements (HOOKE) due to the fact that natural frequency values are slightly affected (not more than 5%) when compared to visco-elastic modeling of elements (VOIGT-KELVIN).

For the considered dynamic model, kinetic energy is given by:

$$2T = m_1\dot{x}_1^2 + m_2\dot{x}_2^2 + m_3\dot{x}_3^2 + 2m_{34}\dot{x}_3\dot{x}_4 \quad (1)$$

where:
m_1 are stands for the mass of vibratory roller;
m_2 – mass of front chassis;
$m_3 = \frac{ml_4^2+J}{l^2}$ – reduced mass of rear chassis when referred to the support point 3;
$m_4 = \frac{ml_3^2+J}{l^2} + m_6$ - reduced mass of rear chassis, added by the mass of propelling wheels, m_6;
$m_{34} = \frac{ml_3l_4-J}{l^2}$ - reduced mass of rear chassis.

The potential energy for deformation of the mechanic system, when referred to the static equilibrium position is given by:

$$2v = k_1x_1^2 + k_2(x_1 - x_2)^2 + k_3(x_2 - x_3)^2 + k_4x_4^2 \quad (2)$$

If forced vibrations, the elementary mechanical work of perturbation (exterior) forces is given by:

$$dL = < \boldsymbol{f}, \boldsymbol{dx} > dL = < f, dx > \quad (3)$$

where: f stands for the vector of perturbation exterior forces, $f^T = [P_1 \quad 0 \quad \ldots \quad 0]$

$P_1 = P_0 sin\omega t$ – the force assigned to generalized coordinate, x_1.

As result of the mentioned above, second order Lagrange differential equations are:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{x}_j}\right) - \frac{\partial T}{\partial x_j} = -\frac{\partial v}{\partial x_j} + \frac{\partial L}{\partial x_j} \quad \frac{d}{dt}\left(\frac{\partial T}{\partial \dot{x}_j}\right) - \frac{\partial T}{\partial x_j} = -\frac{\partial Y}{\partial x_j} + \frac{\partial L}{\partial x_j}$$

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{x}_j}\right) - \frac{\partial T}{\partial x_j} = -\frac{\partial V}{\partial x_j} + \frac{\partial L}{\partial x_j} \quad (4)$$

From relations (1), (2), (4) it results the motion equation written in matrix form, that is:

$$\boldsymbol{M\ddot{x}} + \boldsymbol{Kx} = f \quad (5)$$

or, similarly, the analytically form:

$$m_1\ddot{x}_1 + (k_1 + k_2)x_1 - k_2x_2 = P_0 sin\omega t$$
$$m_2\ddot{x}_2 - k_2x_1 + (k_2 + k_3)x_2 - k_3x_3 = 0 \quad (6)$$
$$m_3\ddot{x}_3 + m_{34}\ddot{x}_4 - k_3x_2 + k_3x_3 = 0$$
$$m_4\ddot{x}_4 + m_{34}\ddot{x}_3 + k_3x_4 = 0$$

The unidirectional perturbation force generated by an inertial vibratory exciter is determined by:

$$P_1 = P_0 sin\omega t \quad (7)$$

of amplitude, $P_0 = m_0 r\omega^2$
where:
$m_0 r$ is the static moment for system's imbalance (equilibrium loss);
ω – pulsation (angular speed) of the excitation system.

If the selected solution is of type: $x_j = A_j sin\omega t$ (j = 1,…,4), it results the vibrations amplitudes, A_1, A_2, A_3 and, respectively, A_4 as solutions of an algebraic system.

3. RESULTS

The values of natural pulsations, p_i (i = 1, …, 4) are determined by the condition of D = 0; where D is the characteristic determinant of equation system (6).

In order to solve the equation, there are considered the notations that follow:

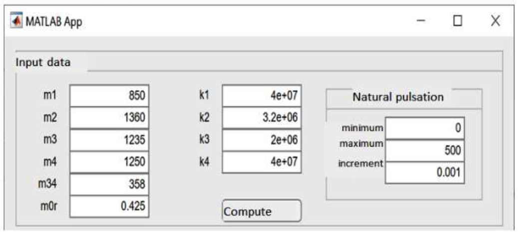
$$P = k_1 + k_2 - m_1\omega^2;$$
$$Q = k_2 + k_3 - m_2\omega^2;$$
$$R = k_3 - m_3\omega^2; \quad (8)$$
$$S = k_4 - m_4\omega^2$$

The analytical expression of D turns into:

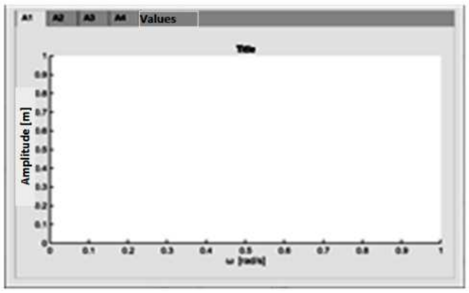
$$D = P(QRS - Qm_{34}^2\omega^4 - k_3^2S) + k_2^2(m_{34}^2\omega^4 - RS) \quad (9)$$

Considering the above mentioned relations, it was created the PULS-AMPL program that enables calculation of the values for natural pulsation. The application is developed in App Designer, from MATLAB [20, 21].

Graphical user interface (GUI) "works" in two zones, one for data input (figure 3, a.) and the other for showing natural pulsation values and plotting the vibration amplitude graphs (figure 3, b.).



a.



b.

Figure 3. Graphical interface for the PULS-AMPL program

From relations (6), (8) and (9) there were determined the mathematical expression for the calculi of forced vibrations amplitudes, that are mentioned next:

$$A_1 = \left|\frac{D_1}{D}\right| m_0 r\omega^2\ ;\ A_2 = \left|\frac{D_2}{D}\right| m_0 r\omega^2$$
$$A_3 = \left|\frac{D_3}{D}\right| m_0 r\omega^2;\ A_4 = \left|\frac{D_4}{D}\right| m_0 r\omega^2 \tag{10}$$

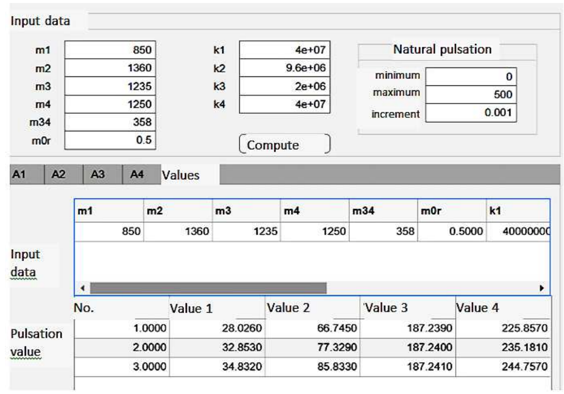
where:

$$D_1 = QRS - Qm_{34}^2\omega^4 - k_3^2 S$$
$$D_2 = -k_2 RS + k_2 m_{34}^2\omega^4 \tag{11}$$
$$D_3 = k_2 k_3 S\ ;\ D_4 = -k_2 k_3 m_{34}\omega^2$$

The pulsation (angular speed) values for excitation system are within the interval 1 ÷ 500 [s^{-1}] (considering the real values for pulsation of vibratory roller), with increment of 0,001 (for precise determination of natural pulsations values).

Input data, meaning the values for specific parameters of the vibration compactor, were experimentally determined and resulted as:
$k_1 = k_4 = 4\cdot10^7$ N/m; $k_3 = 2\cdot10^6$ N/m; $k_2 = n\cdot3{,}2\cdot10^6$ N/m; where n=1,2,3.
$m_1 = 850$ kg; $m_2 = 1360$ kg; $m_3 = 1235$ kg;
$m_4 = 1250$ kg; $m_{34} = 358$ kg; $m_0 r = 0{,}5$ kg·m

Using PULS-AMPL program, there were calculated the values for natural pulsations, p_i, i = 1, …, 4. An example is shown in figure 4.



where each raw of pulsation value corresponds to parameter k_2 values: $k_2 = 1\cdot3{,}2\cdot10^6$ N/m; $k_2 = 2\cdot3{,}2\cdot10^6$ N/m , respectively, $k_2 = 3\cdot3{,}2\cdot10^6$ N/m

Figure 4. Values of natural pulsations – by PULS-AMPL program

The graphical representation of $A_i = f(\omega)$ curves is shown in figure 5, for PULS-AMPL program and in figure 6, for a similar program (AMPL) created in MATLAB.

One particular thing to be mentioned is that the PULS-AMPL program enables plotting graphs for each value of k_2 (stiffness coefficient).

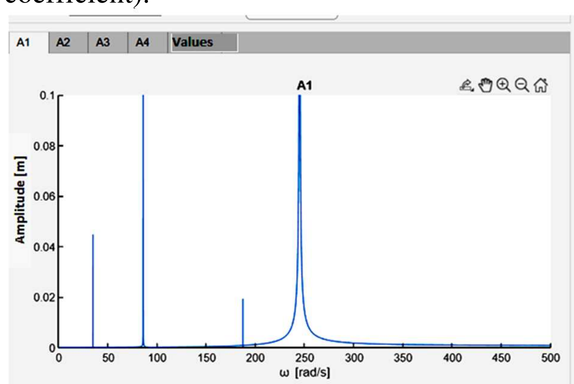


Figure 5. Graph for amplitude curve, A_1 –PULS-AMP program

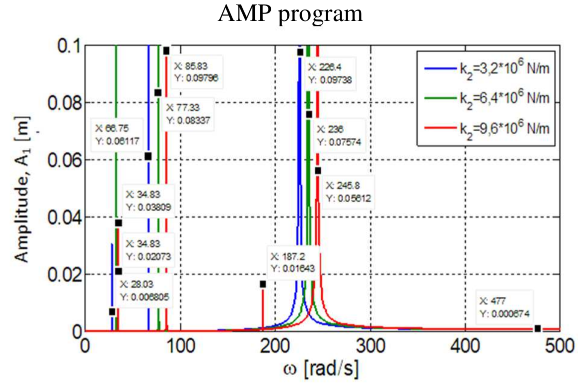


Figure 6. Graph for amplitude curve, A_1 – MatLab program

4. DISCUSSION

Considering the importance of dynamic answer of the vibration roller in the compaction process, this paper presents research results on the influence of discrete variation of first elastic step equivalent stiffness coefficient. This sudied influence is on the mechanical system's dynamic behavior evidenced by amplitude variation curves function of the values for excitation pulsation, The variation of stiffenss coefficient is done by using different anti-vibration rubber elements. The chararcteristics of this rubber are obtained and, thus varied, by homologated manufacturing technology.

Discrete variation of the values for equivalent stiffness coefficient in first elastic step is studied in order to determine the optimum solution for the dynamic regime in the compaction process with single drum vibratory roller.

Natural frequencies values are obtained by using PULS-AMPL program, an application developed in App Designer of MATLAB. The main advantage of this application is that it can run on any computer, as executable (.exe), even there is not MatLab software installed.

The calculated values for natural pulsations, p_i depend on the values of stiffness coefficient, k_2, whose higher values generate higher values for each of the four natural pulsations.

The plotted graphs evidence, for each of k_2 studied values, the four values for natural pulsations, p_i ($i = 1, \ldots, 4$) and, associated to each of them the vibrations amplitudes (that asymptotically tends to infinity ($\rightarrow \infty$).

5. CONCLUSION

The obtained results points out that the technological amplitude, $A_{1,\text{ for}}$ pulsation values within the interval (250 – 350) rad/s, proves almost constant values and, consequently, a stable working regime and adequate technological parameters values for the compaction process.

The values of A_2, A_3, A_4 amplitudes,-for $_{\text{pulsation}}$ values within the interval (250 – 350) rad/s are very low, almost fit to be neglected. Thus, stands as "proof" that the vibrations transmitted equipment's cabin are small, within the admitted norm limits.

Further development of this research will consider other material (rubber / elastomer) types and other constructive forms for anti-vibration elements, so that cabin isolation norms to be fulfilled while the optimum compaction process is on.

6. REFERENCES

[1] Bratu, P., *Multibody System with Elastic Connections for Dynamic Modeling of Compactor Vibratory Rollers*, Symmetry, 12(10), 167, 2020.

[2] Bratu, P., Dobrescu, C.F., *Dynamic Reponses of Zener-Modelled Linearly Viscoelastic Systems under Harmonic Excitation*, Symmetry 11, 1050, 2019.

[3] Dobrescu, C.F., *The Zener rheological viscoelastic modelling of the dynamic compaction of the ecologically stabilized soils,* Acta Tech. Napoc., 63, 2019)

[4] Tonciu, O., Dobrescu, C.F., *Optimal dynamic regimes of vibration compaction equipment*, Acta Tech. Napoc. 63, 2020.

[5] Shabana, A.A., *Dynamic of Multibody Systems*; Cambridge University Press: Cambridge, UK, 2014.

[6] Bratu, P., *Dynamique des compacters autopropulses avec deux rouleaux vibrateurs*, Rev. Roum. Sci. Tech. Mec. Appl., 33, 349-366, 1998.

[7] Bratu, P., *Establishing the equations of motions for the rigid with elastic connections*. Studies and research of applied mechanics, SCMA, Romanian Academy, Volume 50, no.6, 1990.

[8] Dobrescu, C.F., *Analysis of the dynamic regime of forced vibrations in the dynamic compacting process with vibrating roller*, Acta Tech. Napoc. 62, 1, 2019.

[9] Dobrescu, C.F., *Highlighting the Change of the Dynamic Reponses to Discrete Variation of Soil Stiffness in the Process of Dynamic Compaction with Roller Compactors Based on Linear Rheological Modelling*. Appl. Mech. Mater, 801, 242-248, 2015.

[10] Mooney, M.A., Rinehart, R.V., *Field Monitoring of Roller Vibration During Compaction of Subgrade Soil*. J. Geotech. Geoenviron. Eng. ASCE, 133, 257-265, 2007.

[11] Mooney, M.A., Rinehart, R.V., *In-Situ Soil Response to Vibratory Loading and Its Relationship to Roller-Measured Soil*

Stiffness. J. Geotech. Geoenviron. Eng. ASCE, 135, 1022-1031, 2009.
[12] Bratu, P., *Vibrațiile sistemelor elastice (Vibrations of elastic systems)*, Technical Publishing House, Bucharest, 2000.
[13] Bratu, P., Dobrescu, C., *Evaluation of the Dissipated Energy in Vicinity of the Resonance Depending on the Nature of Dynamic Excitation,* Romanian J of Acoustics and Vibrations, 16. 1, 2019.
[14] Bratu, P., *Evaluation of the Dissipated Energy in Viscoelastic or Hysteretic Seismic Isolators,* Romanian Journal of Acoustics and Vibration (1), 2019.
[15] Bratu, P., *Dynamic Stress Dissipated Energy Rating of Materials with Maxwell Rheological Behavior*, Applied Mechanics and Materials, Vol.801, pp 115-121, 2015.
[16] Bratu, P., *Hysteretic Loops in Correlation with the Maximum Dissipated Energy, for Linear Dynamic Systems*, Symmetry, 2019.
[17] Bratu, P., *Elastic structures analysis*, Ed. Impuls, Bucharest, Romania, 2011.
[18] Dobrescu, C. F., Braguța, E., *Optimization of Vibro-Compaction Technological Process Considering Rheological Properties*, Proceedings of the 14th AVMS Conference, Timişoara, Romania, May 25-26, Springer proceedings in Physics ISBN 978-3-319-69822-9.
[19] Wersäll, C., Larsson, S., Ryden, N., Nordfeld, I., *Frequency Variable Surface Compaction of Sand Using Rotating Mass Oscillators*, Geotechnical Testing Journal, Vol. 38, No.2, pp 1-10, 2015.
[20] Matei, L., Dumitru, I., Racilă, L., Vînatoru, M., *Adaptive Traffic Signal Control on a National Road Intersection,* AMM, 822:455–60, 2016.
[21] Matei, L., Dumitru, I., Racila, L., *Smart Signalization and Public Transport Priority, a First Step to Smart Mobility in a Smart City*. AMM 880:383–8, 2018.

PRIMA TREAPTĂ DE IZOLARE A VIBRAȚIILOR CU ELEMENTE ELASTICE PENTRU COMPACTOR CU UN SINGUR CILINDRU VIBRATOR

Abstract: Variația discretă a coeficientului de rigiditate echivalent în prima treaptă de izolarecu elemente elastice este efectuată pentru a determina soluția optimă a regimului dinamic în procesul de compactare vibratorie. Rezultatele cercetării privind influența coeficientului de rigiditate echivalent pentru prima etapă elastică asupra răspunsului dinamic al echipamentelor cu role vibratoare cu un singur tambur sunt evidențiate în acest articol. Această influență este evidențiată de curbele de variație ale amplitudinii vibrației în funcție de pulsația de excitație. Regimul dinamic favorabil este definit de intervalul discret de variație a pulsației de excitație, astfel încât amplitudinea vibrației cilindrului vibrator de compactare să fie suficient de mare și adecvată procesului tehnologic de compactare, atâta timp cât celelalte amplitudini de vibrație sunt suficient de mici, potrivite pentru a fi neglijat.
Au fost determinate curbele de variație a amplitudinii vibrațiilor pentru modelul funcțional și constructiv al unui compactor vibrator autopropulsat cu o rolă vibratoare (CVA 4-5) și elemente antivibrații proiectate și fabricate în tehnologie omologata. Caracteristica elastomerului a fost determinată conform standardelor în vigoare.

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