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OPTIMAL DESIGN OF AN ONE-WAY PLATE COMPACTOR

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Abstract: *This paper presents a software tool and a method for optimizing the design of a one-way plate compactor. Starting from the mechanical model of a plate compactor, a mathematical model was made. Based on the mathematical model, a software was developed to study the influence of constructive parameters on the machine's behaviour. Studying the influence of constructive parameters, an optimal variant can be achieved.*

Key words: *one-way plate compactor, optimal design.*

1. INTRODUCTION

The plate compactors are devices used to compact the soil, broken stones (non-cohesive materials), concrete or asphalt coatings.

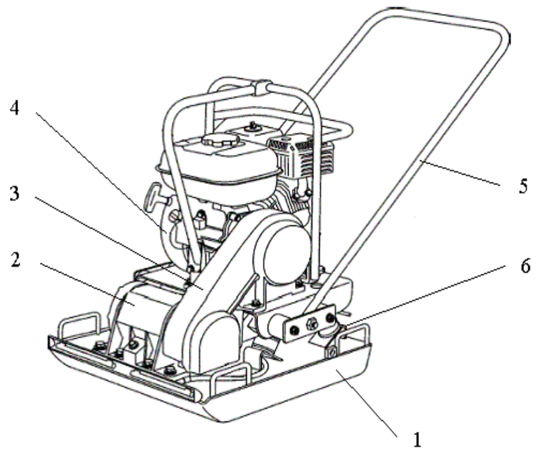

Fig.1. One – way plate compactor

From constructive point of view the plate compactor (figure 1) consists of: base plate (1), vibrations generator (2) (fitted on the base plate) driven by a combustion engine (4) by means of V-belt gearing (3), the equipment being oriented by a handle (5).

The plate compactors use a mechanism (driven by motor oil or a diesel one) which creates a descending force added to the equipment's static weight. One or two eccentrically weights turning around usually form the vibrating generator. The resulted vibrations generate the equipment's advancing movement.

In order to get compaction it is necessary to reach a certain mode of operation mainly characterized by the frequency and amplitude of vibrations due by the vibration generator.

When designing a plate compactor the following parameters must be studied: the weight of the vibrating elements, the size of the disturbing force, the vibrations frequency, the foot surface dimensions, the driving motor power and also a series of dimensional parameters.

The rational choice of these basic parameters of the machinery aims to obtain an efficient compactness, but also to decrease the level of vibrations transmitted to the motor and the human operator.

Ensuring an optimal compaction effect is based on the development of a certain static pressure on the material.

The weight of the plate is calculated according to the type of compacted material as well as the surface of the sole plate as follows:

$$G_t \geq p \cdot S \ [daN] \qquad (1)$$

where S is the surface of the sole of the plate compactor [m^2], and p is the required static pressure [daN / m^2].

The weight of the upper part of the compactor is the weight of the frame and of the motor mounting plate, together with the motor weight:

$$G_s = G_t \cdot (1 - c_1) \text{ [daN]} \tag{2}$$

where c_1 represents the ratio between the lower part weight during vibration and the total weight of the plate.

The value of c_1 ratio is chosen according to the plate weight, as follows:

- G = 40 ÷ 250 kg, c_1 = 0,4 ÷ 0,5;
- G = 251 ÷ 1000 kg, c_1 = 0,4 ÷ 0,6;
- G = 1001 ÷ 2500 kg, c_1 = 0,5 ÷ 0,8;
- G = 2501 ÷ 8000 kg, c_1 = 0,6 ÷ 0,8.

Also, the value of c_1 ratio must be chosen so that it meets the following condition:

$$G_s > 2 \cdot G_M \text{ [daN]} \tag{3}$$

where G_M represents the driving motor weight.

The weight of the lower part represents the sole weight together with the vibration generator weight and it is calculated using one of the following equations:

$$G_i = G_t - G_s, G_i = c_1 \cdot G_t \text{ [daN]} \tag{4}$$

2. MATHEMATICAL MODEL AND SOFT ANALYSIS

To study the behavior of a unidirectional plate compactor, a mechanical model was created (figure 2). The forces acting on the plate are represented in figure 3.

Based on the mechanical model, a mathematical model composed of four non-linear differential equations corresponding to the movements of the plate components was developed as follows:

- an equation corresponding to the horizontal movement of the plate;
- three equations describing the horizontal and vertical motion of the frame, as well as the variation of the rotation angle of the frame.

Taking into account the mechanical model, in the elaboration of the mathematical model the following were considered as generalized coordinates:

- x_{O2_1} the horizontal displacement of the origin of the coordinate system connected to the sole of the plate compactor;
- x_{O3_1}, respectively y_{O3_1} the horizontal and the vertical displacement of the O_3 point (the origin of the mobile coordinate system connected to the frame);
- φ - the rotation angle of the frame measured in the vertical plane;

When writing the mathematical model, the following notations were also used: m_0 – eccentric mass; m_2 – plate mass; m_3 – mass of the frame and motor; k_1 – stiffnesses; k_2 – axial factor of rigidity;

To solve the mathematical model and to study the influence of various parameters, a program was developed in the Visual Basic language. The solution of the differential equation system was achieved by using the Runge-Kutta method of 4th order.

The equations corresponding to the mathematical model are:

$$\frac{d^2 x_{O2\_1}}{dt^2} = \frac{1}{m_2 + m_0} \begin{bmatrix} m_0\, r\, \omega^2 \cos\omega t + T \cos\psi - \\ ..-\mu\, N\, sign\left(\frac{d\, x_{O2\_1}}{d\, t}\right) + .. \\ ..+\left(x_{L2\_1} - x_{L1\_1}\right) k_1 + .. \\ ..+\left(x_{L4\_1} - x_{L3\_1}\right) k_1 + .. \end{bmatrix} \tag{1}$$

$$m_3 \begin{bmatrix} \frac{d^2 x_{O3\_1}}{dt^2} + \left(-x_{C3\_3} \sin\varphi - y_{C3\_3} \cos\varphi\right) \frac{d^2\varphi}{dt^2} + \\ + \left(-x_{C3\_3} \cos\varphi + y_{C3\_3} \sin\varphi\right) \left(\frac{d\varphi}{dt}\right)^2 \end{bmatrix} = \tag{2}$$
$$= -\left(x_{L2\_1} - x_{L1\_1}\right) k_1 - \left(x_{L4\_1} - x_{L3\_1}\right) k_1 - T \cos\psi$$

$$m_3 \begin{bmatrix} \frac{d^2 y_{O3\_1}}{d t^2} + \left(x_{C3\_3} \cos\varphi - y_{C3\_3} \sin\varphi\right) \frac{d^2 \varphi}{d t^2} + .. \\ ..+\left(-x_{C3\_3} \sin\varphi - y_{C3\_3} \cos\varphi\right) \left(\frac{d\varphi}{d t}\right)^2 \end{bmatrix} = \tag{3}$$
$$= -\left(y_{L2\_1} - y_{L1\_1}\right) k_1 - \left(y_{L4\_1} - y_{L3\_1}\right) k_1 - ..$$
$$..- T \sin\psi - m_3\, g + ..$$
$$+ \begin{cases} 0 & daca \quad y_{L6\_1} - y_{L5\_1} > d_{\max} \\ k_2 \cdot \left[d_{\max} - \left|y_{L6\_1} - y_{L5\_1}\right|\right] & daca \quad y_{L6\_1} - y_{L5\_1} \le d_{\max} \end{cases},$$

$$J_{C3} \frac{d^2\varphi}{d t^2} = \begin{bmatrix} -\left(x_{L2\_1} - x_{C3\_1}\right)\left(y_{L2\_1} - y_{L1\_1}\right) + .. \\ ..+\left(y_{L2\_1} - y_{C3\_1}\right)\left(x_{L2\_1} - x_{L1\_1}\right) \end{bmatrix} k_1 +$$
$$+ \begin{bmatrix} -\left(x_{L4\_1} - x_{C3\_1}\right)\left(y_{L4\_1} - y_{L3\_1}\right) + .. \\ ..+\left(y_{L4\_1} - y_{C3\_1}\right)\left(x_{L4\_1} - x_{L3\_1}\right) \end{bmatrix} k_1 + \tag{4}$$
$$+ \left(x_{L6\_1} - x_{C3\_1}\right) \begin{Bmatrix} \left[d_{\max} - \left| y_{L6\_1} - y_{L5\_1} \right|\right] k_2 \\ sau \\ 0 \end{Bmatrix} +$$
$$+ \begin{bmatrix} -\begin{pmatrix} x_{03\_1} + r_{sm} \cdot \sin\psi \cdot \cos\varphi - .. \\ ..-\left(h_4 - r_{sm} \cdot \cos\psi\right) \cdot \sin\varphi - x_{C3\_1} \end{pmatrix} \sin\psi + \\ +\begin{pmatrix} y_{03\_1} + r_{sm} \cdot \sin\psi \cdot \sin\varphi + .. \\ ..+\left(h_4 - r_{sm} \cdot \cos\psi\right) \cdot \cos\varphi - y_{C3\_1} \end{pmatrix} \cos\psi \end{bmatrix} T$$

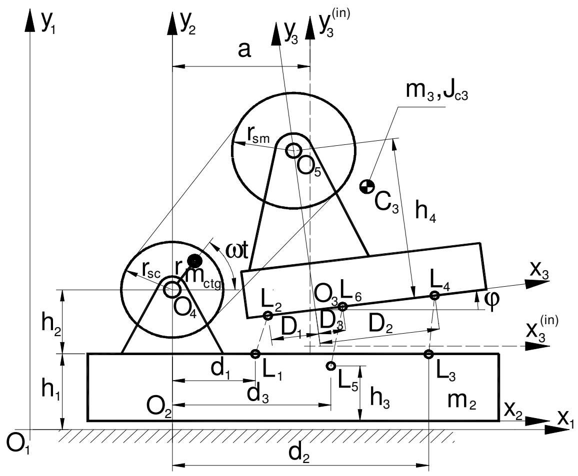

Fig. 2. Mechanical model

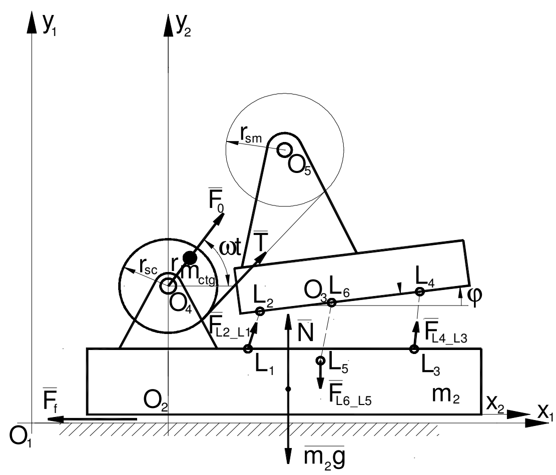

Fig 3. Forces on the components of the plate

The program allows plotting the variations of horizontal displacement of the sole, the horizontal and vertical displacements of the frame and also the variation graph of the normal pressing force of the ground.

The program also allows drawing the speed variation diagrams for the soles and frame movements.

The program consists of several modules and interacts directly with the three dimensional modeling software Solid Edge.

The input data from the analysis program are provided by the three-dimensional model of the compacting plate, therefore any modification to the three-dimensional model is found in the program that analyses the behavior of the machine.

The interaction is in both ways; for this purpose, program modules have been created that allow to change the virtual machine components without interfering with the 3D model through the Solid Edge software.

The program, together with the three-dimensional model of the unidirectional compacting plate, forms an extremely powerful tool that can be used for the analysis and design of unidirectional compacting plates.

Figure 4 illustrates the main window of the program that analyses the behavior of the unidirectional compaction plates.

The window contains five buttons to define the three-dimensional model of the compacting plate. When you operate one of these buttons, a new module opens with which you can change the coresponding component.

Fig. 4. Behavior analysis program for one-way plate compactor

Based on the input data obtained from the three-dimensional model of the plate, it is possible to plot the variations of the horizontal and vertical displacements of the angular displacements as well as the pressing force of the ground.

The display options (Figure 5) refer to the simulation duration, the simulation step and the number of subintervals in which the values on the two axes (horizontal and vertical) are divided.

With the radio buttons you can select which chart to be displayed: horizontal movement of the sole, horizontal movement of the origin of the frame coordinate system, vertical movement of the origin of the coordinate system related to the frame, angular displacement of the frame as well and the normal pressing force.

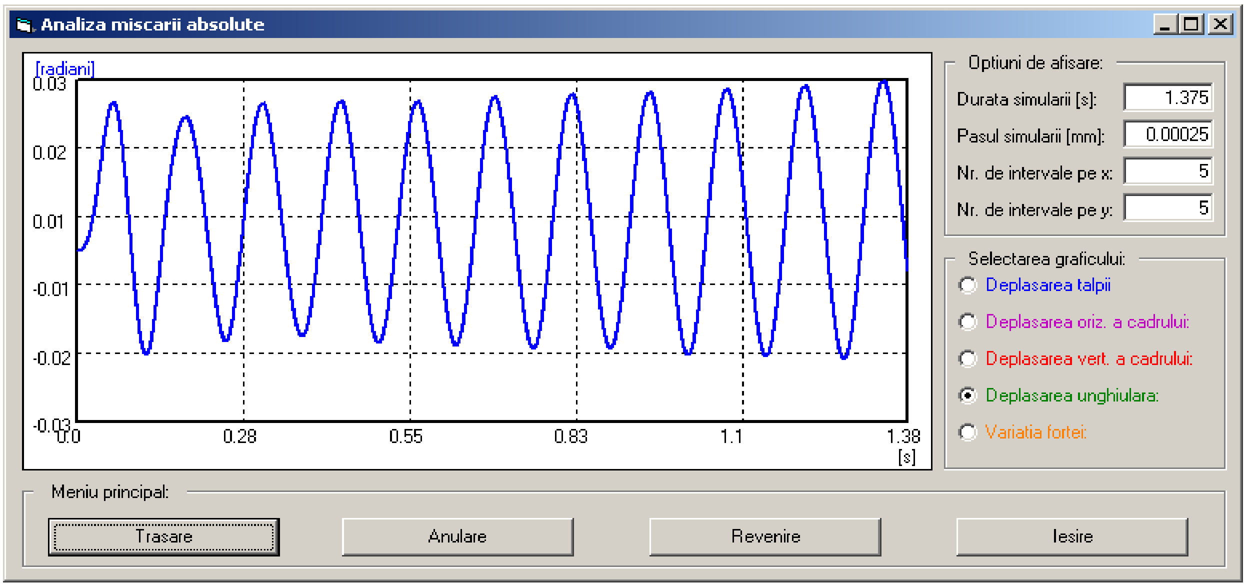

Fig. 5. Module for plotting variation graphs

After setting the display options and selecting the graph, the Trace button will be pressed in order to draw it.

If you want to draw a new graph, use the ***Cancel*** button. The effect of this action is to delete the drawn graph and to cancel the values in the text boxes.

If you want to modify the components of the three-dimensional model, the ***Return*** button has to be pressed. When this button is pressed, it returns to the main module where the components in order to make changes. To exit the application, press the ***Exit*** (*Ieşire*) button.

3. NUMERICAL STUDY

The study was carried out for a compacting plate with the following characteristics: sole mass: m1 = 70 [kg], mass of the frame together with the driving motor: m2 = 55 [kg], vibration generator speed: 5400 [rpm] : eccentric mass m0 = 3,465 [kg], eccentricity r = 0,0134 [m], elastic constant k1 = 58110 * 2 [N / m], elastic constant k2 = 608250 [N / m] friction coefficient between sole and ground $\mu = 0.35$.

It is desirable to design an optimal compaction plate that meets the following requirements: the frame vibrations to be minimum, respectively the developed force to be maximum.

Six versions for positioning of the anti-vibration elements mounted horizontally were considered: Version I (d1 = 0.135 [m], d2 = 0.500 [m]), version II (d1 = 0.140 [m]], version III (d1 = 0.145 [m], d2 = 0.400 [m]), version IV (d1 = 0.150 [m], d2 = 0.350 [m] = 0.300 [m]), version VI (d1 = 0.160 [m], d2 = 0.250 [m]).

Figure 6 illustrates the variation of the movement of the sole as a function of the mounting version. It can be noted that the displacement speeds have close values, with the observation that in case of version VI a different displacement of the machine is recorded compared to the other version.

The positioning in various version of the anti-vibration elements leads to large variations of the vertical displacements of the frame (fig. 7), and it is observed that in case of the versions IV, V and VI the displacement amplitudes are large, even very large in the case of version VI.

Therefore, to avoid large frame vibrations, one of the mounting versions I, II or III will be used.

From the angular displacement analysis (Figure 8) it is found that for the versions IV, V and VI there are recorded large angular displacements (up to 4 degrees in the case of version VI).

The fitting of the anti-vibration elements in different versions does not bring any significant changes in the variation of the soil compaction force (Figure 9), depending mainly on the disturbing force, the vibration generator speed and the table mass.

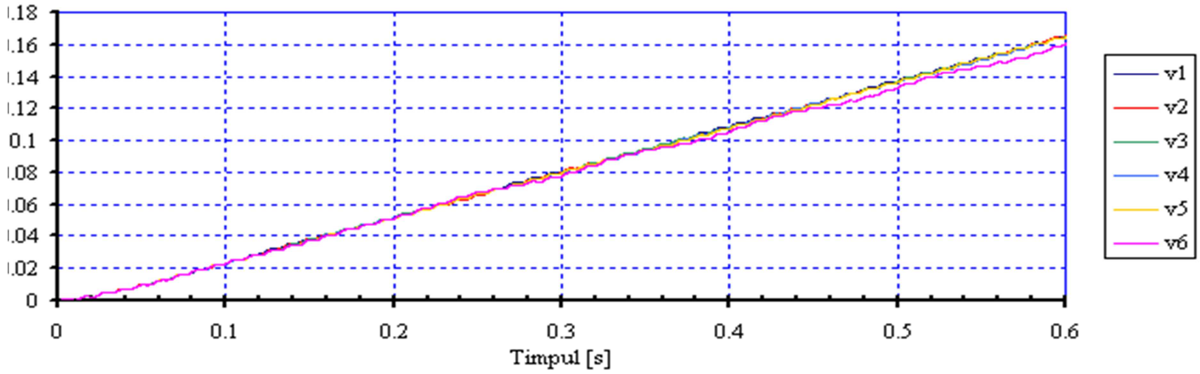

Fig. 6. Diagram of variation of plate movement depending on the mounting version of the antivibration elements

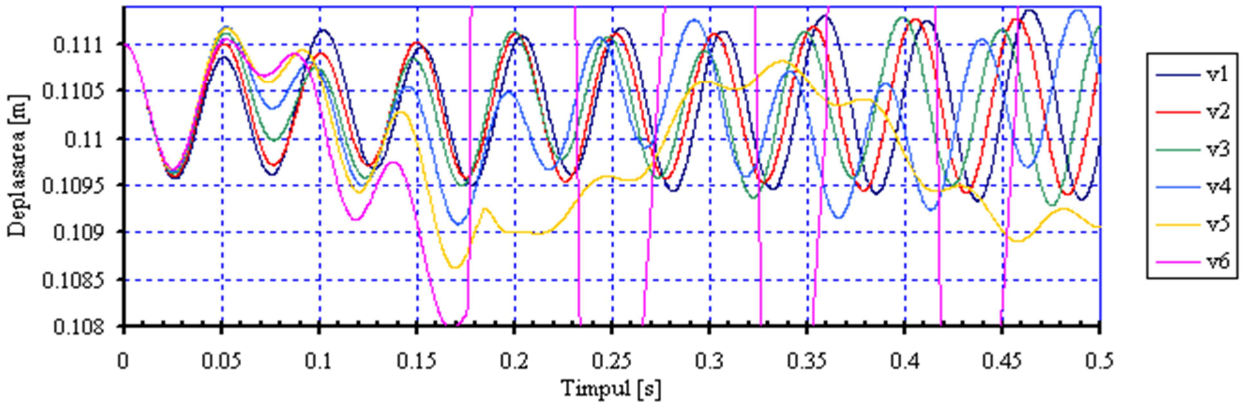

Fig. 7. Variation chart of the vertical movement of the frame, depending on the mounting variant

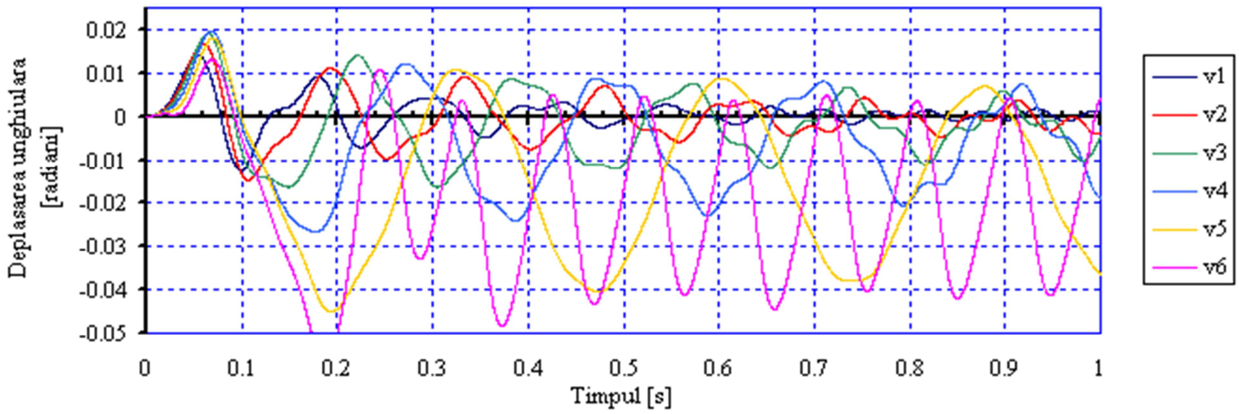

Fig. 8. Variation of angular displacements depending on mounting variant

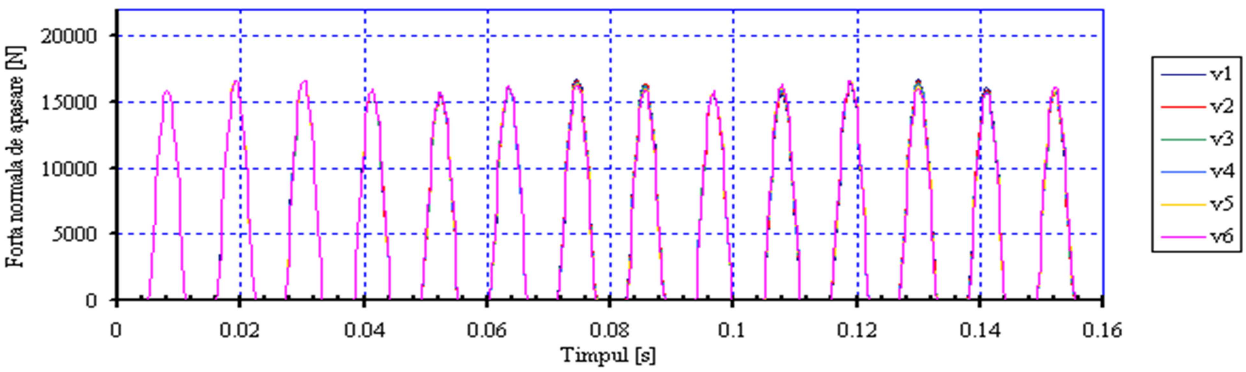

Fig. 9. Variation of the presure force depending on the mounting variant

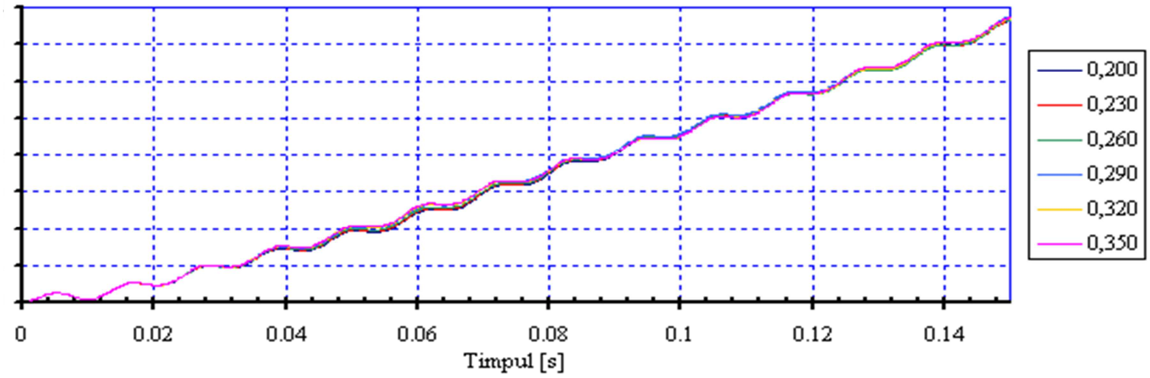

Fig. 10. Variation of the movement of the sole according to the position of the vertical anti-vibration element

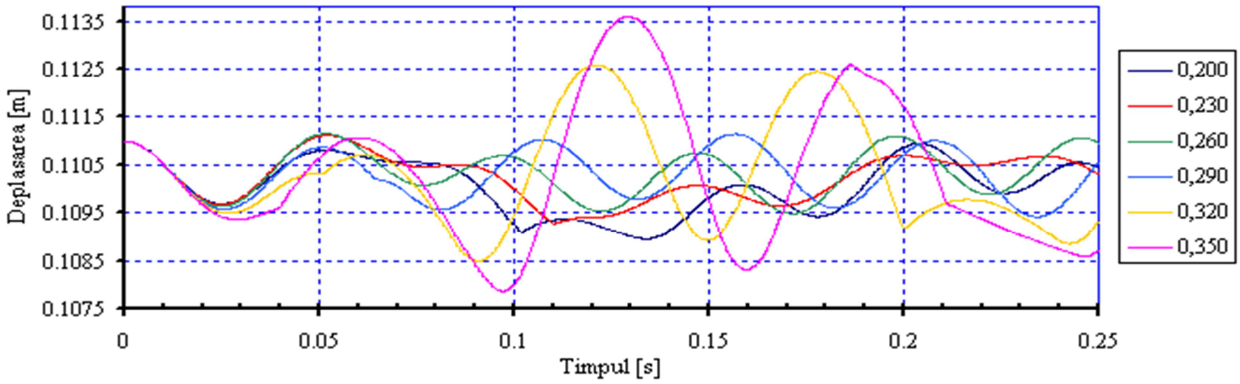

Fig. 11. Variation of the vertical movement of the frame according to the position of the vertical

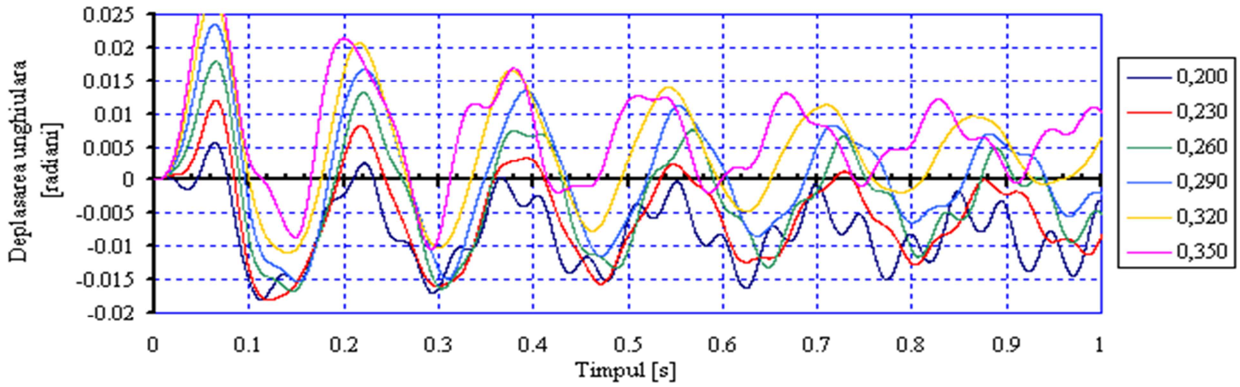

Fig. 12. Variation of the angular displacement of the frame depending on the position of the vertical

In order to study the influence of the position of the vertical anti-vibration element, the following values of the distance from the origin of the mobile coordinate system, connected to the frame (figure 2) are considered: version I - d3 = 0.200 [m], version II - d3 =], version III - d3 = 0,260 [m], version IV - d3 = 0,290 [m], version V - d3 = 0,320 [m], version VI - d3 = 0,350 [m].

Figure 10 illustrates the variation of the horizontal movement of the plate, as a function of the mounting version. The positioning of the vertical antivibration element does not

significantly affect how the plate moves in the plane and the displacement speed.

From the analysis of the vertical displacements and the angular displacements carried out by the frame (Figures 11 and 12 respectively) it is found that for positioning the vertical antivibration element between the limits of 0.260 [m] and 0.290 [m] the frame exhibits small displacements both in the vertical direction aa well as angular ones.

Therefore, for a compacting plate with the characteristics of the one analyzed, it is recommended to position the vertical antivibration element at a distance between 0.260 [m] and 0.290 [m] relative to the origin of the coordinate system (from the vibration generator rotation center).

4. CONCLUSION

The mathematical model developed for the study of the behavior of unidirectional compacting plates allows the study of the influence of several factors on the behavior of the machine.

From the large number of input parameters, one part can be experimentally determined, and one part can be determined based on studies conducted using the program created by the author.

The study presented in this paper allows the design of an optimal version of the compacting plate by determining the position of the antivibration elements.

It can be observed that by increasing the vibration generator speed, the speed of movement, the pressing force of the ground and the vibrations transmitted to the frame and, implicitly, to the human operator increase.

For certain selected values of the vibration generator speeds, using the above preseted program, the positions of the damping elements are determined in order to obtain the lowest vibrations for the frame (and motor, respectively).

Similarly, with the help of the program created by the author, a series of studies can be carried out on optimal versions of compacting plates, imposing some of the input parameters, the others being determined based on studies made using the program.

This results in high technological performance with low vibration level transmitted to the frame and the human operator.

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OPTIMIZAREA CONSTRUCTIVĂ A UNEI VARIANTE DE PLACĂ COMPACTOARE UNIDIRECȚIONALĂ

Rezumat: *În această lucrare se prezintă un instrument software şi o metodă de optimizare a proiectării constructive a unei plăci compactoare unidirecţionale. Pornind de la modelul mecanic al unei plăci compactoare, s-a realizat un model matematic al acesteia. Pe baza modelului matematic s-a creat un soft cu ajutorul căruia se studiază influenţa parametrilor constructivi asupra funcţionării utilajului. Studiind influenţa parametrilor constructivi se poate realiza o variantă constructivă optimală.*

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