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STUDY OF THE INFLUENCE OF PLATE COMPACTOR WEIGHT ON ITS FUNCTIONING

Radu Mircea MORARIU-GLIGOR

Abstract: *The functioning of the plate compactor is influenced by a series of factors: the material which will be compacted, the power of the engine, the engine's revolution, vibrations generator's rotation speed etc. The construction pattern of the plate compactor has also a big influence in its functioning. A study regarding the selection method of the plate compactor weight and its influence on the plate compactor functioning is presented in this paper. Selecting the weight and knowing the way the plate compactor behaves according to the weight of its components are very important in the design stage of the plate compactor.*
Key words: *plate compactors, behavior*

1. INTRODUCTION

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

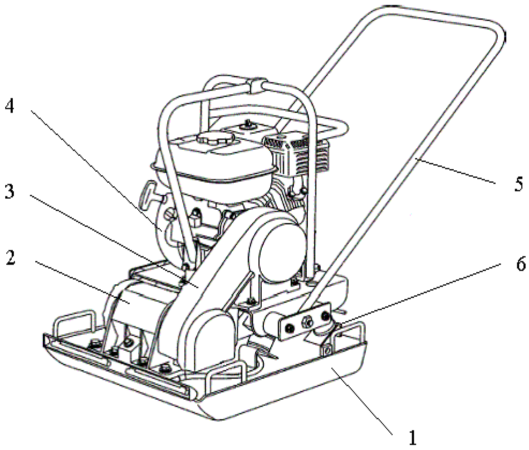

Fig.1. One – way plate compactor

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.

In order to achieve an efficient compactness by vibration, the following factors must be taken into consideration: the properties of the material to be compacted (soil, concrete…), the width of the compaction band, the thickness of the material layer, the final compacting degree of the material.

When designing a vibrating plate the following parameters must be studied [5]: the weight of the vibrating elements, the size of the disturbing force, the vibrations frequency, the foot surface dimensions, the driving motor power.

An extremely important parameter is represented by the total weight of the compactor, but also the foot weight together with the vibration generator, respectively the frame weight together with the motor and the operating handler.

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. It has been proven experimentally that, for the same values of the vibrations amplitude, the compaction effect depends on the vibrator weight [2].

If the vibrator weight is below a certain limit, the compaction is not achieved. Increasing the machinery weight above a certain limit will lead to the decrease of the compaction level or even to the lack of compaction effect. The assurance of an optimal compaction effect is based on developing a *p* static pressure on the material. The compactor weight is calculated according to the nature of the material to be compacted and to the support surface (foot surface) size, as follows [2], [5]:

$$G_t \geq p \cdot S \ \text{[daN]} \tag{1}$$

where *S* represents the support surface of the vibrating plate, in [m^2], and the *p* static pressure is expressed in [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:
- lightweight plates (G = 40 ÷ 250 kg), c_1 = 0,4 ÷ 0,5;
- middleweight plates (G = 251 ÷ 1000 kg), c_1 = 0,4 ÷ 0,6;
- heavy plates (G = 1001 ÷ 2500 kg)
- c_1 = 0,5 ÷ 0,8;
- really heavy plates (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 foot weight together with the vibration generator weight and it is calculated using one of the following equations:

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

2. MATHEMATICAL MODEL

In some other paper [5], this type of equipment is modeled very much simplified by means of two masses system, with 2 degrees of freedom whose analytical answer is to be found into the general theory of vibrations of linear systems.

These systems, easy to be solved, do not wholly satisfy the necessities of determination the behavior of this type of equipment. In fact, the movements of components are complex (not only one directional but planar), the respective system of differential equations resulting nonlinear.

In the papers [11], [12] it is presented a mathematical model formed of 4 non-linear differential equations that makes possible the more accurate study of the behavior of this type of equipment.

The results obtained with the help of this mathematical model indicate a more real behavior of the equipment in comparison with the other models met in the technical literature.

The forces which act on the plate compactor are represented in figure 3. Starting from this mathematical model it was developed the study proposed in this work.

On the basis of the mechanical model a mathematical model it was developed composed of four non-linear differential equations corresponding to the movements performed by each component part of the mechanical system, as follows:
- an equation corresponding to the plate's movement on horizontal direction;
- three equations corresponding to the frame's horizontal, vertical and rotating movements;

Into the mechanical system the followings were considered as generalized coordinates:
- x_{O2_1} – displacement on the horizontal direction of the origin of the system referring to the vibrating plate;
- x_{O3_1}, respectively y_{O3_1} – displacements on the horizontal direction, and the vertical one respectively, of the O_3 origin of the mobile system referring to the frame;
- φ – turning angle of the frame (counterclockwise) due to the horizontal line;

Significance of the quantities in the mathematical system is as follows: m_0 – eccentric mass; m_2 – plate mass; m_3 – mass of the frame and motor; k_1 – stiffness; k_2 – axial factor of rigidity; x_{A_B}, respectively y_{A_B} – distances on the axis O_1x_1, respectively O_1y_1 between points A and B. In order to solve the mathematical model previously presented it was written a program in C language [10]. Resolution on the system of differential equations is found with the help of Runge-Kutta fourth order method [9]. The equations corresponding to the mathematical model are [1][8]:

$$\frac{d^2 x_{O2\_1}}{dt^2} = \frac{1}{m_2 + m_0}\left[m_0\, r\, \omega^2 \cos\omega\, t + T\, \cos\psi - \mu\, N\, sign\left(\frac{d\, x_{O2\_1}}{d\, t}\right) + \left(x_{L2\_1} - x_{L1\_1}\right) k_1 + \left(x_{L4\_1} - x_{L3\_1}\right) k_1 \right] \tag{5}$$

$$m_3 \left[\frac{d^2 x_{O3\_1}}{dt^2} + \left(-x_{C3\_3}\, \sin\varphi - y_{C3\_3}\, \cos\varphi\right)\frac{d^2\varphi}{dt^2} + \left(-x_{C3\_3}\, \cos\varphi + y_{C3\_3}\, \sin\varphi\right)\left(\frac{d\varphi}{dt}\right)^2 \right] = \dots \tag{6}$$

$$\dots = -\left(x_{L2\_1} - x_{L1\_1}\right)k_1 - \left(x_{L4\_1} - x_{L3\_1}\right)k_1 - T\, \cos\psi$$

$$m_3 \left[\begin{array}{l} \frac{d^2 y_{O3\_1}}{dt^2} + \left(x_{C3\_3}\, \cos\varphi - y_{C3\_3}\, \sin\varphi \right)\frac{d^2\varphi}{dt^2} + .. \\ .. + \left(-x_{C3\_3}\, \sin\varphi - y_{C3\_3}\, \cos\varphi\right)\left(\frac{d\varphi}{dt}\right)^2 \end{array} \right] = -\left(y_{L2\_1} - y_{L1\_1}\right)k_1 - \left(y_{L4\_1} - y_{L3\_1}\right)k_1 - T\, \sin\psi - m_3\, g + .. \tag{7}$$

$$+ \begin{cases} 0 & daca \quad y_{L6\_1} - y_{L5\_1} > d_{\max} \\ k_2 \cdot \left[d_{\max} - \left|y_{L6\_1} - y_{L5\_1}\right|\right] & daca \quad y_{L6\_1} - y_{L5\_1} \le d_{\max} \end{cases},$$

$$J_{C3}\, \frac{d^2\varphi}{dt^2} = \left[-\left(x_{L2\_1} - x_{C3\_1}\right)\left(y_{L2\_1} - y_{L1\_1}\right) + \left(y_{L2\_1} - y_{C3\_1}\right)\left(x_{L2\_1} - x_{L1\_1}\right)\right] k_1 + \dots$$

$$\dots + \left[-\left(x_{L4\_1} - x_{C3\_1}\right)\left(y_{L4\_1} - y_{L3\_1}\right) + \left(y_{L4\_1} - y_{C3\_1}\right)\left(x_{L4\_1} - x_{L3\_1}\right)\right] k_1 + \left(x_{L6\_1} - x_{C3\_1}\right)\left\{ \begin{array}{c} \left[d_{\max} - |\, y_{L6\_1} - y_{L5\_1}\,|\,\right] k_2 \\ sau \\ 0 \end{array} \right\} + \dots \tag{8}$$

$$\dots + \left[\begin{array}{l} -\left(x_{03\_1} + r_{sm} \cdot \sin\psi \cdot \cos\varphi - \left(h_4 - r_{sm} \cdot \cos\psi\right) \cdot \sin\varphi - x_{C3\_1}\right) \sin\psi + \dots \\ \dots + \left(y_{03\_1} + r_{sm} \cdot \sin\psi \cdot \sin\varphi + \left(h_4 - r_{sm} \cdot \cos\psi\right) \cdot \cos\varphi - y_{C3\_1}\right) \cos\psi \end{array} \right] T$$

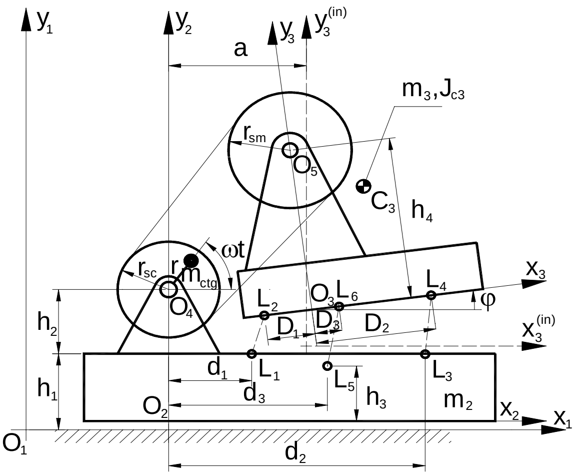

Fig. 2. Mechanical model

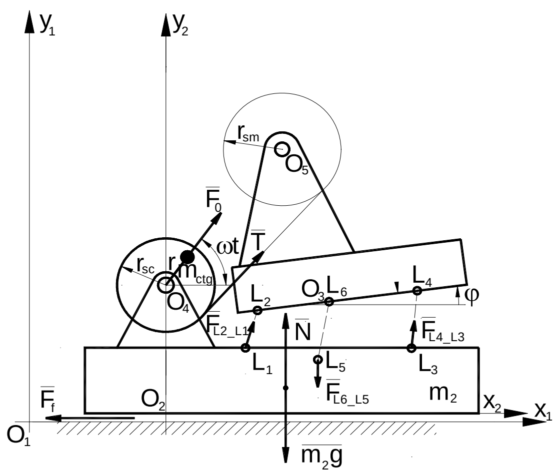

Fig 3. Forces on the components of the plate compactor

3. NUMERICAL STUDY

Due to the fact that the compactor weight is an important parameter that must be taken into consideration in the design stage, it is necessary to conduct a study regarding its influence on the machinery operation.

The study was conducted for the case of a compacting plate having the following characteristics: the mass of the eccentric: m_0 = 3,465 [kg], the eccentricity r = 0,0134 [m], the elasticity constant k_1 = 58110*2 [N/m] and k_2 = 608250 [N/m], the friction coefficient between the base plate and ground is: μ = 0,35.

Three plate compactor types were taken into consideration for this study, with the same total weight, but distributed differently between its two main components, as follows:

- case 1: the mass of the bed plate: m_1 = 70 [kg], the total mass of the frame and driving engine: m_2 = 55 [kg];
- case 2: the mass of the bed plate: m_1 = 62,5 [kg], the total mass of the frame and driving engine: m_2 = 62,5 [kg];

- case 3: the mass of the bed plate: $m_1 = 55$ [kg], the total mass of the frame and driving engine: $m_2 = 70$ [kg];

In the figure 4 are presented the displacements of the compacting plates.

It is noteworthy that the change of the ratio between the two weights leads to a change of the plate compactor travel speed.

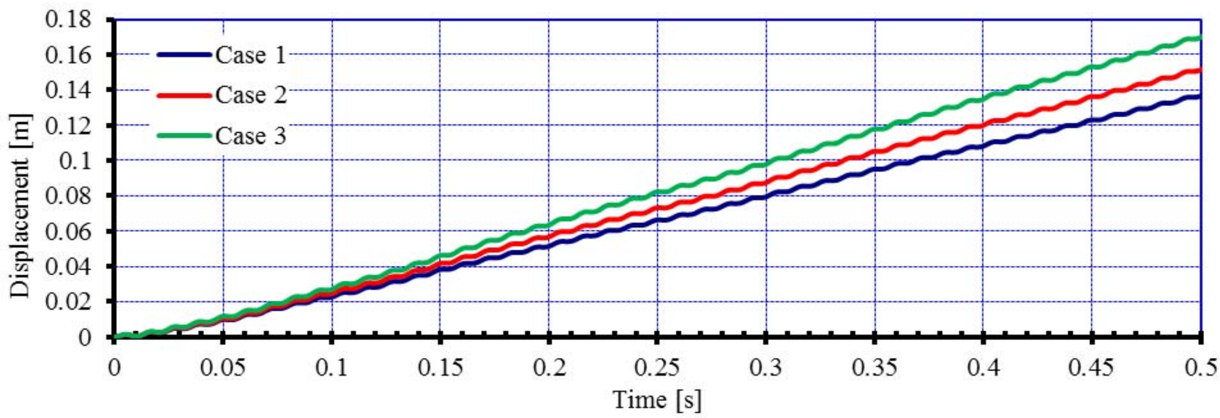

Fig. 4. The time variation diagram for the space covered by the compacting plate

The figure 5 present the variation law of vertical displacement for the mobile reference system attached to the frame.

Analyzing this figure, it can be observed that the change of the weight of the vibrating plate components influences the vertical displacement of the frame and handler.

It can be seen that the angular displacement of the frame is influenced by the values of the two weights (figure 6).

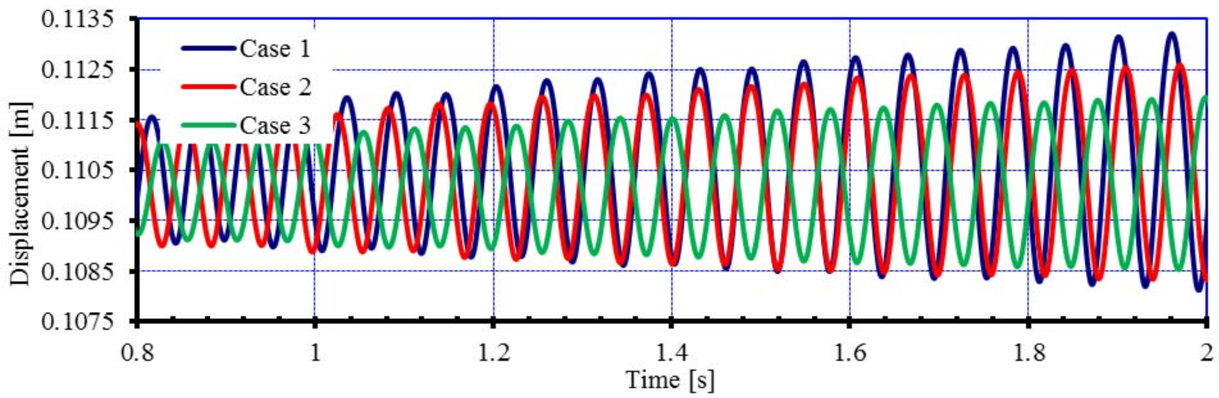

Fig. 5. The time variation diagram for the vertical displacement of the frame

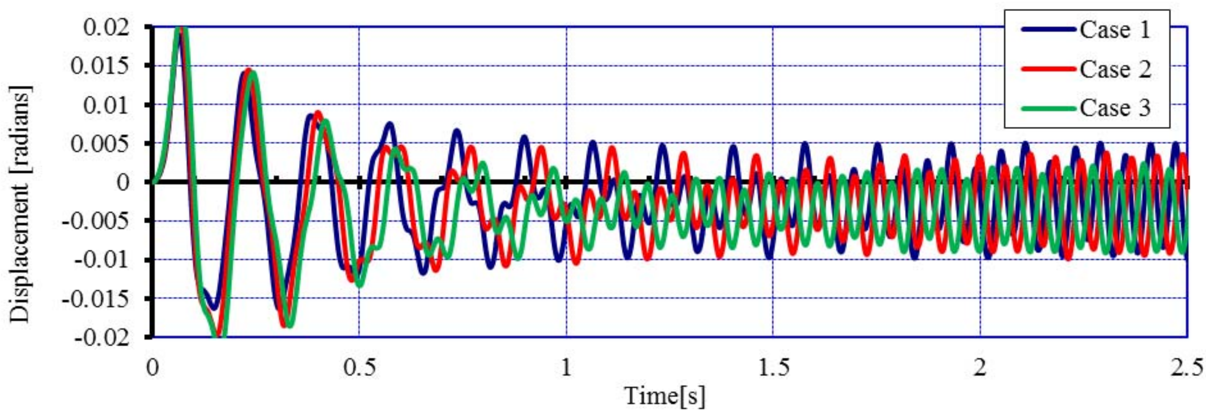

Fig. 6. The time variation diagram for the rotation angle of the frame

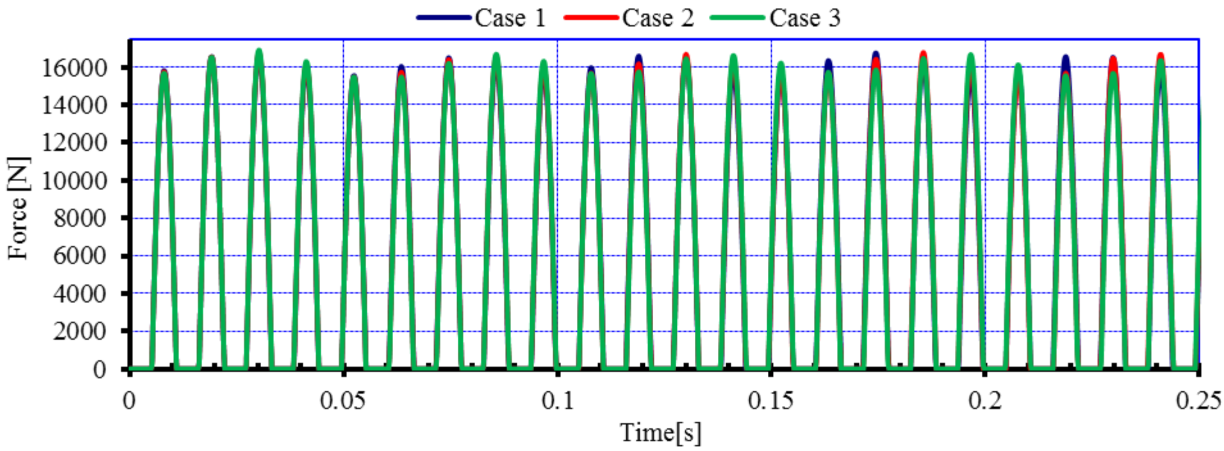

Fig. 7. The time variation diagram for the normal force applied to the ground

Another important parameter that must be considered in the study of the behavior of a compacting plate is the variation of the pressure applied to the ground by the compactor bed plate.

The figure 7 illustrates the variation in time of the pressure applied in the three analyzed situations.

4. CONCLUSIONS

The mathematical model performed for the study of the one way plate compactor allows the analysis of the factors that can also influence the equipment performance.

Analyzing the previous figures, the following conclusions can be drawn:

- the travel speed increases when the foot weight decreases;
- the vertical displacement of the frame and the angular displacement decrease as the frame and motor weight increases;
- the variation of the down force is not influenced significantly by the change of the ratio between the two weights.

Using the program developed in Borland C programming environment and in 3D modeling environment Solid Edge, more types of plate compactors can be configured and analyzed, in the end obtaining an optimized plate compactor.

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STUDIUL INFLUENŢEI MASEI PLĂCII COMPACTOARE ASUPRA FUNCŢIONĂRII ACESTEIA

Rezumat: Funcţionarea unei plăci compactoare este influenţată de o serie de factori: materialul de compactat, puterea motorului, turaţia generatorului de vibraţii etc. Un alt parametru important care influenţează funcţionarea plăcii compactoare îl reprezintă masa totală a acesteia şi modul în care aceasta este distribuită între cele două componente: motorul împreună cu cadrul şi cu mânerul de acţionare, respectiv talpa plăcii împreună cu generatorul de vibraţii. Alegerea valorilor celor două mase este foarte importantă, acestea influenţând viteza de deplasare dar şi nivelul vibraţiilor transmise cadrului şi operatorului uman.

Radu Mircea MORARIU-GLIGOR, Lecturer, Technical University of Cluj-Napoca, Department of Mechanical Engineering Systems, Faculty of Machine Building, rmogli70@yahoo.com, Ariesului 102/107, Cluj-Napoca, 0743 - 120463.