A Study on the Impact Force in Case of Tamping Rammers

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Abstract: - The most important technical specification of the tamping rammers is represented by the impact force whose value is in accordance with the compaction degree of the soil. Also, the tamping force is influenced by a series of factors such as: the engine power and speed, the dimensions of the crank and connecting rod assembly, the spring's elasticity constant, the damping coefficient, and so on. The objective of the paper is to conduct a study on the impact force generated by tamping rammers during the soil compaction process.

Keywords: - impact force, tamping rammers, vibrations.

1. INTRODUCTION

Soil tamping is defined as the method of mechanically increasing the density of soil. There are five principle reasons to compact soil: increases load-bearing capacity; reduces settling of soil; reduces shrinkage of soil; reduces water seepage, swelling and contraction; provides better stability.

The tamping rammers deliver a high impact force (high amplitude) making them an excellent choice for cohesive and semi-cohesive soils. Rammers cover three types of compaction: impact, vibration and kneading. The tamping rammers are machineries for compacting the soil by tapping, that are intended for use in narrow spaces, ensuring a compaction depth of 0.6 ÷ 0.8 m. By tilting the axis of rammer cylinder at an angle of 10 to 17º from the vertical, it will perform a forward displacement with 10 ÷ 15 cm at each jump.

The lifting height of the compacting device is of 30 ÷ 100 mm, the number of hits / minute being between 400 ÷ 800. The total weight of the rammer is between 50 and 200 kg.

The compacting effect is obtained by:
- the effect produced on the compacting plate by the crank and connecting rod assembly;
- the shock caused by the fall of the rammer due to the performed lifting;
- the effect of low frequency vibrations transmitted to the earth at each fall of the tamping rammer;

From the constructive point of view, the tamping rammer (figure 1) consists of: compacting plate (1), oil cylinder (2), bellows device (3), gear box (4), anti-vibration rubber elements (5), drive engine (6), frame (7), drive handle (8), fuel tank (9).

The drive engine can be diesel or on gasoline, in two or four stroke type, with a driving power between 1.4 and 4 kW and revolution speeds between 2300 ÷ 4500 rpm. The transfer of the motion from the drive motor (6) is achieved by means of a centrifugal coupling (not shown in the figure) to the gear box (4), which reduces the speed down to 400 ÷ 800 rpm.

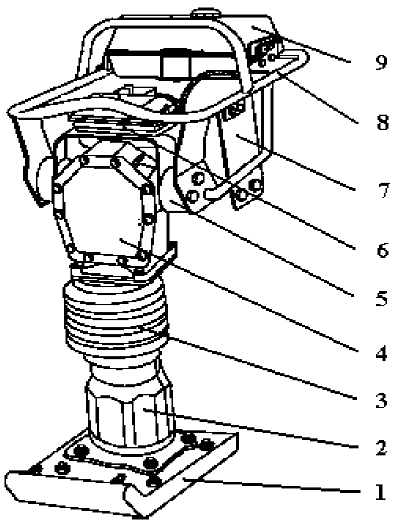

Figure 1. The components of a tamping rammer

The speed reducer (4) drives a crank and connecting rod mechanism connected to a piston, which acts by means of elastic springs on the compacting plate (1) and to which an oscillatory movement is imparted.

The bellows cylinder (2) is filled with oil and provides the damping effect, the lack of oil leading to the malfunction of the machine.

The drive handle (8) is used to control the machine during operation, being insulated from the frame (7) by means of rubber anti-vibration elements (5).

During the operation of the machine, the compacting plate performs leaps (with detachment from the soil) the height of the leap being 30 ÷ 100 mm. The frame is performing up and down displacements, on upright direction, of 30 ÷ 50 mm, but the vibrations of the handle are lower due to the anti-vibration elements which ensure the isolation of the handle against the machine frame.

The compacting plate dimensions are within the following limits: (150 ... 300) x (270 ... 350) mm. The working speed of the tapping rammers is between 8 ÷ 20 m/min and the productivity is between 175 ÷ 350 m^2/h.

The mass of the machine, along with the force generated by the springs in the cylinder, produces the impact force with which the compacting plate acts upon the ground, the value of this force being between 9 ÷ 21 kN.

2. THE MATHEMATICAL MODEL

In the process of achieving the mathematical model, the following simplifying hypotheses were taken into account:
- the system consists of three masses;
- the displacement is only in the vertical direction;
- all parts, except for the springs, are rigid;
- the driving engine and the coupling parts are modeled as a rotating disc;
- the moment of inertia I and the driving moment M are constant;
- the elastic constants and the damping coefficient are linear;

In the mechanical model the notations are used:
- x_1 and x_2 = the parameters defining the position of the elements which forms the mechanical system;
- M = the torque (constant);
- ω = angular velocity (constant);
- φ = the angle between the crank and the vertical axis (denoted Ox);
- ψ = the angle made by connecting rod with the vertical axis;
- m_1 = the total mass of the frame, engine and gearbox;
- m_2 = the total mass of the piston, connecting rod and of the upper springs assembly;
- m_3 = the total mass of the compacting plate, cylinder and the lower springs assembly;
- m_4 = soil mass shifted after the tamping;
- r = length of crank;
- b = the length of the connecting rod;
- l_1 = the length of the springs in the mounted state;

The mechanical system shown in figure 2 is a system with two degrees of freedom, whose elements perform vertical movements, the geometric parameters defining the movement of the system being x_1, respectively x_2.

The crank rotates at a constant angular velocity in a trigonometric sense, acting on it with a constant torque M. Torque M is replaced by a torque generated by a T_0 force applied to the crank end (figure 2).

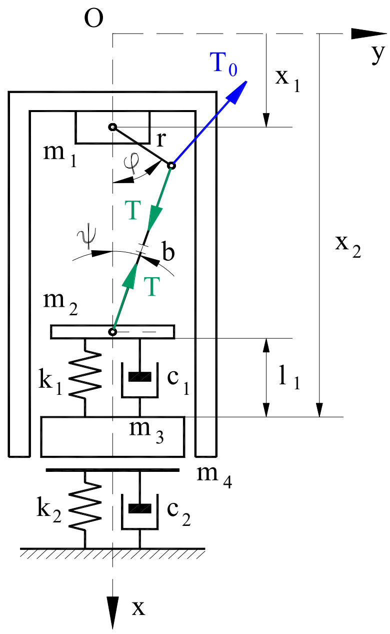

Figure 2. Theoretical model of tamping rammer

Thus, the following expressions are determined:

$$\phi = \omega \cdot t \ ;$$

$$r \cdot \sin\phi = b \cdot \sin\psi \ ;$$

$$\psi = \arcsin\left(\frac{r}{b} \cdot \sin\phi\right); \qquad (1)$$

$$\dot{\psi} = \frac{r \cdot \omega \cdot \cos\phi}{b \cdot \cos\psi} ;$$

$$\ddot{\psi} = \frac{b \cdot \dot{\psi}^2 \cdot \sin\psi - r \cdot \omega^2 \cdot \sin\phi}{b \cdot \cos\psi} ;$$

In this study, the soil is modeled by means of a helical spring and a parallel-mounted shock absorber (Kelvin-Voigt model), the values for the elastic and damping characteristics being expressed according to the type of the soil, but also on the basis of its tamping degree. The elastic constant k and the damping constant c corresponding to the mathematical model of the soil can be determined based on [12], [17]:

$$k=\frac{4\cdot G\cdot r}{1-\nu}\ ;\ c=\frac{3,4}{1-\nu}\cdot r^2\cdot\sqrt{\frac{G\cdot\gamma_t}{g}}\ ; \qquad (2)$$

where: G – shear modulus; r – the equivalent radius of the rammer plate; ν – Poisson's coefficient; γ_t – the specific gravity of the soil.

Also, between the elastic constant k and the damping constant c, there is the relation [4],[8],[11]:

$$c=2\cdot D\cdot\sqrt{m\cdot k} \qquad (3)$$

where D is a damping coefficient (whose value, in this paper was considered D = 0,4).

Between G – the shear modulus and E –Young's elasticity modulus can be established the relation:

$$G=\frac{E}{2\cdot(1+\nu)}\ ; \qquad (4)$$

Applying Newton's law leads to the following differential equations that define the motion of m_1, m_2 and m_3 masses:

$$m_1\cdot\ddot{x}_1=m_1\cdot g-T_0\cdot\sin\phi+\mathrm{T}\cdot\cos\psi\ ; \qquad (5)$$

$$\left\{\begin{array}{c} m_2\cdot\frac{d^2}{dt^2}\cdot\left(x_1+r\cdot\cos\phi+b\cdot\cos\psi\right)= \\ =m_2\cdot g-T\cdot\cos\psi+ \\ +k_1\cdot\left(x_2-x_1-l_1-r\cdot\cos\phi-b\cdot\cos\psi\right)+ \\ +c_1\cdot\left(\dot{x}_2-\dot{x}_1+r\cdot\omega\cdot\sin\phi+b\cdot\dot{\psi}\cdot\sin\psi\right) \end{array}\right\};\quad (6)$$

$$\left\{\begin{array}{l} m_3\cdot\ddot{x}_2=m_3\cdot g-k\cdot\left(x_2-x_1-l_1-r\cdot\cos\cdot\phi-b\cdot\cos\psi\right)- \\ \quad c_1\cdot\left(x_2-x_1+r\cdot\omega\cdot\sin\phi+b\cdot\dot{\psi}\cdot\sin\psi\right)- \\ \quad\left\{\begin{array}{ll} 0, & x_2\le r+b+l_1 \\ \left[k_2\cdot\left(x_2-r-b-l_1+c_2\cdot\dot{x}_2\right)\right], & x_2>r+b+l_1 \end{array}\right\} \end{array}\right\};\ (7)$$

In the equation (7) was considered that the mass m_3 is not in permanent contact with the soil.

Also, the mass m_4 representing the mass of the soil removed at the impact was omitted. In the following, is computed the derivative:

$$\left\{\begin{array}{c} \frac{d^2}{dt^2}\left(x_1+r\cdot\cos\phi+b\cdot\cos\psi\right)= \\ =\frac{d^2}{dt^2}\left(\dot{x}_1-\omega\cdot r\cdot\sin\phi-b\cdot\dot{\psi}\cdot\sin\psi\right)= \\ \ddot{x}_1-\omega^2\cdot r\cdot\cos\phi-b\cdot\ddot{\psi}\cdot\sin\psi-b\cdot\dot{\psi}^2\cdot\cos\psi \end{array}\right\};\ (8)$$

so, the equation (6) becomes:

$$\left\{\begin{array}{c} m_2\cdot\ddot{x}_1-m_2\cdot\omega^2\cdot r\cdot\cos\phi- \\ -m_2\cdot b\cdot\ddot{\psi}\cdot\sin\psi-m_2\cdot b\cdot\dot{\psi}^2\cdot\cos\psi= \\ =m_2\cdot g-T\cdot\cos\psi+ \\ +k_1\cdot\left(x_2-x_1-l_1-r\cdot\cos\phi-b\cdot\cos\psi\right)+ \\ +c_1\cdot\left(\dot{x}_2-\dot{x}_1+r\cdot\omega\cdot\sin\phi+b\cdot\dot{\psi}\cdot\sin\psi\right) \end{array}\right\};\quad (9)$$

The unknown represented by T from equation (5) is replaced in the equation (9), this leading to:

$$\left\{\begin{array}{c} \left(m_1+m_2\right)\cdot\ddot{x}_1=\left(m_1+m_2\right)\cdot g-T_0\cdot\sin\phi+ \\ +k_1\cdot\left(x_2-x_1-l_1-r\cdot\cos\phi-b\cdot\cos\psi\right)+ \\ +c_1\cdot\left(\dot{x}_2-\dot{x}_1+r\cdot\omega\cdot\sin\phi+b\cdot\dot{\psi}\cdot\sin\psi\right)+ \\ +m_2\cdot\left(\omega^2\cdot r\cdot\cos\phi+b\cdot\ddot{\psi}\cdot\sin\psi+b\cdot\dot{\psi}^2\cdot\cos\psi\right) \end{array}\right\};\ (10)$$

The system of differential equations which describe the motion of the rammer plate and of the rammer frame respectively, is presented below:

$$\left\{\begin{array}{c} \left(m_1+m_2\right)\cdot\ddot{x}_1=\left(m_1+m_2\right)\cdot g-T_0\cdot\sin\phi+ \\ +k_1\cdot\left(x_2-x_1-l_1-r\cdot\cos\phi-b\cdot\cos\psi\right)+ \\ +c_1\cdot\left(\dot{x}_2-\dot{x}_1+r\cdot\omega\cdot\sin\phi+b\cdot\dot{\psi}\cdot\sin\psi\right)+ \\ +m_2\cdot\left(\omega^2\cdot r\cdot\cos\phi+b\cdot\ddot{\psi}\cdot\sin\psi+b\cdot\dot{\psi}^2\cdot\cos\psi\right); \\ m_3\ddot{x}_2=m_3\cdot g-k_1\cdot\left(x_2-x_1-l_1-r\cdot\cos\cdot\phi-b\cdot\cos\psi\right)- \\ -c_1\cdot\left(\dot{x}_2-\dot{x}_1+r\cdot\omega\cdot\sin\phi+b\cdot\dot{\psi}\cdot\sin\psi\right)- \\ -\left\{\begin{array}{ll} 0, & x_2\le r+b+l_1 \\ k_2\cdot\left(x_2-r-b-l_1\right)+c_2\cdot\dot{x}_2, & x_2>r+b+l_1 \end{array}\right\}. \end{array}\right\};\ (11)$$

The solutions of the system of differential equations (2), were determined by using the fourth order Runge - Kutta method implemented in a C language application.

The input data are:

- m_1 – the mass of the frame, driving engine and speed reducer altogether;
- m_2 – the mass of the piston, connecting rod and of the upper springs assembly altogether [kg];
- m_3 – the mass of the rammer plate, cylinder and lower springs assembly altogether [kg];
- k_1 – the elasticity constant of the helical springs which are elements of the tapping rammers [N/m];

- c_1 – the damping coefficient of the viscous damper [N*s/m];
- r – the length of the crank [m];
- b – the length of the connecting rod [m];
- k_2 – the elasticity constant of the soil (Kelvin-Voigt model) [N/m];
- c_2 – the damping coefficient considered at the soil modeling (Kelvin – Voigt model) [N*s/m];
- M – the moment applied to the crank [N*m];
- n – the rotation speed of the crank [rot/min];

2. THE NUMERICAL MODELING

The study aimed to determine the impact force for a series of tamping rammer configurations, comparing the values obtained with those specified in the manufacturer's catalogs.

The authors have previously conducted a series of studies that have led to some of the ranges of input data, which ensure a better operation of the machine. It ascertained that the choice of the rammer's constructive characteristics is an important step, although difficult, which involves to find a balance between the required mechanical characteristics of the rammer (for example, the impact force vs. displacement of the frame).

Table 1. MIKASA tamping rammers

Model	Operating Weight [kg]	Impact Force [kN]	Foot Width [mm]	Engine
MTX 60	64	13,6	265	Honda GX100
MTX 70	75	14,9	285	Honda GX100
MTX 80	82	15,6	285	Subaru EH 12
MTX 85 DY	93	15,7	285	Yanmar L48N
MT 55 H	62	9,8	265	Subaru EH09-2F
MT 65 H	66	12,7	285	Honda GX100
MT 66 HL	72	12,7	285	Honda GX100
MT 72 FW	74	13,7	285	Subaru EH 12
MT 77 HRL	77	13,7	285	Honda GRX120
MT 84F	84	15,7	285	Subaru EH 12

Through previous studies [20], it has been found that there is a range of rotational speeds in which a large tread of the compacting plate is achieved, at the same time with the small displacement of the frame.

Also, the values of the elastic constant and the damping coefficient were determined such as to ensure proper operation of the tamping rammer.

Figure 3 illustrates a MTX 60 tamping rammer manufactured by MIKASA, a machine that is part of a family of rammers made by this manufacturer.

Figure 3. Mikasa tamping rammer MTX 60

In the Table 1 are mentioned the technical specifications for ten different variants of tamping rammers made by MIKASA manufacturer.

Analyzing the data presented in Table 1 it is observed that there is a large variety of constructive variants, the impact force being influenced by several factors, presented bellow:

Table 2. The characteristics of the selected rammer

Considered parameter	Symbol	Value
the mass of the frame, driving engine and speed reducer altogether [kg]	m_1	45
the mass of the piston, connecting rod and of the upper springs assembly altogether [kg]	m_2	5
the mass of the rammer plate, cylinder and lower springs assembly altogether [kg]	m_3	25
the length of the crank [m]	r	0,0275
the length of connecting rod [m]	b	0,350
The length the springs in mounted state [m]	l_1	0,150
The elastic constant of the helical springs from the tamping rammer [N/m]	k_1	150000
The damping coefficient of the viscous damper [N*s/m]	c_1	500
The elastic constant of the soil (Kelvin-Voigt model) [N/m]	k_2	750000
The damping coefficient considered at soil modeling (Kelvin-Voigt model) [N*s/m]	c_2	3000
The moment applied to the crank [N*m]	M	15
Crank's rotation speed [rot/min]	n	700

Using the C software application, based on the mathematical model presented above, can be plotted the impact force variation chart. In this respect, is selected a variant of tamper rammer, whose characteristics are presented in Table 2.

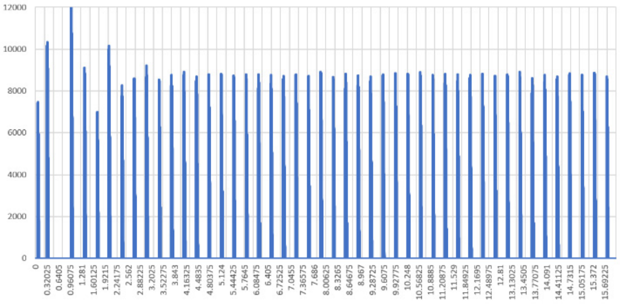

Figure 4. The variation law for the impact force

Figure 4 illustrates the variation law for the impact force in case of the tamping rammer whose characteristics were presented in Table 2.

In the Figure 4 the time [s] is represented on the horizontal axis, while the value of the impact force [N] is represented on the vertical axis.

By analyzing the variation law presented in Figure 4, it can be seen that the variation of the impact force corresponds to the operating mode of the tamping rammer, i.e. there are moments when the impact force is 0 this happening when the compacting plate is not in contact with the ground.

It is also observed that after 3 - 4 seconds, the variation of the impact force stabilizes, which corresponds to the permanent operating regime.

Analyzing Table 1, it is obvious that there are several variants of tamping rammers with different masses which can develop the same impact force.

It is found that are several geometric and constructive parameters which influence the impact force.

Table 3. The characteristics of four rammers

Symbol	Variant 1	Variant 2	Variant 3	Variant 4
m_1 [kg]	45	50	56	60
m_2 [kg]	5	6	8	8
m_3 [kg]	16	18	20	24
r [m]	0,0275	0,032	0,033	0,035
b [m]	0,350	0,350	0,330	0,350
l_1 [m]	0,200	0,180	0,170	0,175
k_1 [N/m]	175000	165000	180000	160000
c_1 [N*s/m]	600	500	490	480
k_2 [N/m]	800000	800000	800000	800000
c_2 [N*s/m]	3500	3500	3500	3500
M [N*m]	15	16	18	20
n [rot/min]	700	600	700	650
Impact force	9,835	13,389	14,326	15,481

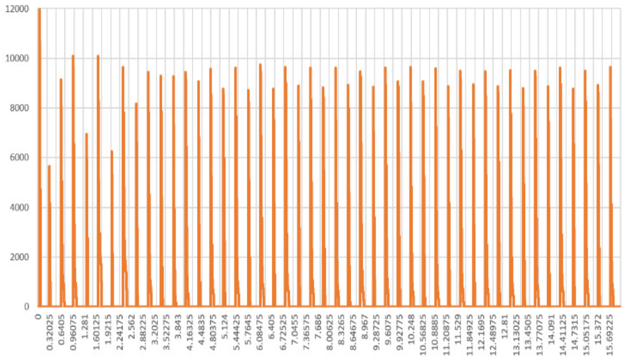

Figure 5. The variation law for the impact force (variant 1)

Table 3 shows the maximum impact force values in case of four different variants of rammers.

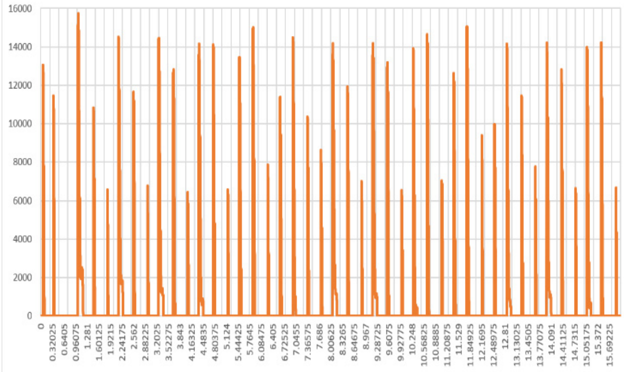

Figure 6. The variation law for the impact force (variant 4)

Analyzing the data in Table 3, is observed that there is a similarity between the data from Table 1 and Table 3. So, for tamping rammers with same total mass, a force similar to the one presented in Table 1 was obtained.

4. CONCLUSIONS

The impact force is one of the most important technical specifications of the tamping rammers, its value being a function of the degree of compaction of the soil and the productivity of the compaction process [21], [22]. As shown in the paper, the value of the impact force depends on a number of factors, such as: the masses of the tamping rammers' components, the power and speed of the driving motor, the geometric characteristics of the crank and connecting rod mechanism, the elastic constants of the rammer's spring assembly, and the damping coefficient of the hydraulic damper.

The previously mentioned parameters also depend on other characteristics of the tamping rammer, such as the displacement of the frame or the displacement of the compacting plate [19].

Based on these considerations, using a mathematical model and a program written in C language, was carried out a study on the variation of the impact force, based on the chosen input data.

The obtained data are similar to those presented in the technical specifications by the manufacturing companies, thus the previously presented mechanical and mathematical model was validated.

Also, by using the C program, any variant of a compactor may be analyzed. The operation of the analyzed rammers is known before its practical execution, so that its constructive characteristics can be established from the beginning.

By the help of the developed C program can be also conducted a study on the influence of some functional parameters on the impact force.

As a further development, the authors intend to conduct studies on the influence of geometric and mechanical parameters of the impact force value.

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