THE DYNAMICS OF THE DEEP SOIL LOOSENING MACHINE – TRACTOR UNIT

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The tractor's vibrations with an agricultural equipment were analyzed and showed a particular interest to researchers [6, 7, 8]

The unit consists of a New Holland T6070 tractor with and a deep soil loosening machine EAA – 220, which is subjected to shocks and oscillations that depend on both the characteristics of the road as well as the elasticity of the tires. The paper will study the vibrations that occur in the vertical plane of symmetry of the unit due to vertical movement (bouncing or vertical oscillations) and rotation around the transverse axis passing through the center of gravity (pitch oscillations). When driving on paved road, the experimental values for the measured acceleration are given at the two bridges on the tractor and are compared with calculated values.

Keywords: vibrations, shock, tractor, soil loosening, EAA-220 equipment

1. Theoretical consideration

To study the oscillations and pitching of the deep soil loosening machine - tractor unit [1, 2, 3, 4], a mechanical model with two degrees of freedom is shown in Fig. 1. It is a bar of mass m having the center of gravity C at a distance a from the center of the front axle and $b = L - a$ from the center of the rear axle, measured horizontally. The wheelbase of the tractor is L. The tractor wheel is not rigid and can be equated with a spring with linear characteristic k and a damper with damping coefficient c. The unit mass m consists of tractor mass m_t, the ballast m_l, and the mass of the deep soil loosening machine EAA – 220 and distribution on both axles will be made according to the dynamic weight distribution coefficient on the rear drive axle of the tractor.

The following parameters of the tractor wheels are considered: k_1, k_2 - the linear characteristics of the front and the rear tires and $c1$, $c2$ - the damping coefficients of the front and rear tires.

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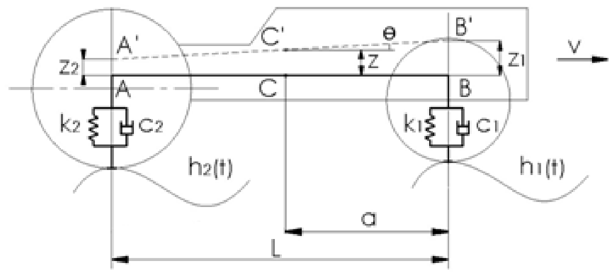


Fig.1.The mechanical model with two degrees of freedom for the vibration study

The units weight distributed on the rear drive axle m_s and the front drive axle m_f of the tractor can be calculated with the equation:

$$m_s = \lambda_{din}(m_t + m_l + m_m) \qquad m_f = m_t + m_l + m_m - m_s \tag{1}$$

The dynamic coefficient of distribution of the total weight on the rear drive axle will be [5]:

$$\lambda_{mdin} = \frac{m_t\left[(L-b)cos\alpha + h_g sin\alpha\right] + m_l[h_l sin\alpha - (l_l - L)cos\alpha]}{L(m_t + m_l + m_m)} + \frac{m_m[h_m sin\alpha + (l_m + L)cos\alpha]}{L(m_t + m_l + m_m)} \tag{2}$$

The following notations were made:

- α – field angle;
- l_m – distance from the center of the rear drive wheels to the machine's center of gravity;
- h_g –distance from the support surface to the tractor's center of gravity;
- h_m – distance from the support surface to the machine's center of gravity;
- h_l – distance from the support surface to the center of gravity of the ballast;

The free radius of the front driving wheels r_{01} and rear driving wheels r_{03}, measured in meters, can be calculated if the dimensions H_1 and H_2 and rim diameter D_1, *and* D_2 are known.

Knowing the tire deformation coefficient, $c_{df,s}$ = 0.07 – 0.025 [5] the tires deformations can be calculated:

$$\delta_1 = c_{df} \,.\, r_{01} \qquad \delta_2 = c_{ds} \,.\, r_{02} \tag{3}$$

The linear characteristics for the tires can be obtained with these equations:

$$k_1 = \frac{\mathrm{m_f} \cdot 9.81}{2 \cdot \delta_1} \qquad k_2 = \frac{\mathrm{m_s} \cdot 9.81}{2 \cdot \delta_2} \tag{4}$$

The shock dampers, characterized by viscous damping, have damping coefficients of about 25% of the critical damping coefficients [1].

Field profile irregularities are considered sinusoidal and their heights are h_1 and h_2 for the front wheels and for the rear wheels (ϕ - is the phase) and can be calculated with the relations [1]:

$$h_2 = h_0 \sin(\omega \cdot t + \phi) \qquad h_1 = h_0 \cdot \sin(\omega \cdot t) \qquad (5)$$

where h_0 is the amplitude of the field irregularities.

The movement is studied in relation to the position of static balance AB.

The following parameters are chosen: the mass center displacement in relation to z and the rotation angle θ of the tractor relative to the mass center. The displacement of the front axle z_1 and of the rear axle z_2 is calculated using the equation:

$$z_1 = z + a \cdot \theta \qquad z_2 = z - b \cdot \theta \qquad (6)$$

The kinetic energy of the model consists of the kinetic energy of the rotation and the kinetic energy of translational motion and is calculated using the equation:

$$E = \frac{1}{2} m_1 \dot{z}_1^2 + \frac{1}{2} J \dot{\theta}^2 + \frac{1}{2} m_2 \dot{z}_2^2 \qquad (7)$$

where:

- $m_1 = m_f/2$ and $m_2 = m_s/2$ represent the mass distributed to the front wheels, respectively the rear wheels;
- J – half of the moment of inertia on the transverse axis passing through the center of mass C.

To determine the mass of the front axle m_{pf} and of the rear axle m_{ps} as functions of the front wheel mass m_{rf} and the mass of the front whell axis m_{af} respectively m_{rs} and m_{as} for the rear axle, the following relations are used:

$$m_{pf} = 2 \cdot m_{rf} + m_{af} \qquad m_{ps} = 2 \cdot m_{rs} + m_{as} \qquad (8)$$

Mass of the tractor chassis is:

$$m_{st} = m_t + m_l + m_m - m_{pf} - m_{ps} \qquad (9)$$

Half of the moment of inertia on the transverse axis passing through the center of mass C is:

$$J = \frac{1}{2} \cdot \left(m_{pf} \cdot a^2 + m_{ps} \cdot b^2 + m_{st} \cdot \left(a - \frac{L}{2} \right)^2 + m_{st} \cdot \frac{L^2}{12} \right) \qquad (10)$$

The potential energy of the system is:

$$U = -\left(\frac{1}{2} k_1 z_1^2 + \frac{1}{2} k_2 z_2^2 \right) \qquad (11)$$

The energy dissipated in shock is given by:

$$D = -\left(\frac{1}{2} c_1 \dot{z}_1^2 + \frac{1}{2} c_2 \dot{z}_2^2 \right) \qquad (12)$$

The inertia matrix (M), the stiffness matrix (K) and the damping matrix (C) are given by the equations:

$$M = \begin{vmatrix} m_1 + \frac{J}{L^2} & -\frac{\mathrm{J}}{\mathrm{L}^2} \\ -\frac{\mathrm{J}}{\mathrm{L}^2} & m_2 + \frac{J}{L^2} \end{vmatrix} \qquad K = \begin{vmatrix} k_1 & 0 \\ 0 & k_2 \end{vmatrix} \qquad C = \begin{vmatrix} c_1 & 0 \\ 0 & c_2 \end{vmatrix} \tag{13}$$

The forces that act during the movement are:

$$F = \begin{vmatrix} k_1 \cdot h_1 + c_1 \cdot \dot{h}_1 \\ k_2 \cdot h_2 + c_2 \cdot \dot{h}_2 \end{vmatrix} \tag{14}$$

The motion parameters matrix is:

$$Z = \begin{vmatrix} z \\ \theta \end{vmatrix} \tag{15}$$

The differential equation matrix of equilibrium is:

$$[M]\{\ddot{Z}\} + [C]\{\dot{Z}\} + [K]\{Z\} = \{F\} \tag{16}$$

The differential scalar equations of equilibrium are:

$$\begin{aligned} &(m_1 + \frac{J}{L^2})\ddot{z}_1 + c_1 \dot{z}_1 + k_1 z_1 - \frac{\mathrm{J}}{\mathrm{L}^2}\ddot{z}_2 = F_{01}\cos(\omega \cdot t - \varphi_1) \\ &-\frac{\mathrm{J}}{\mathrm{L}^2}\ddot{z}_1 + (m_2 + \frac{J}{L^2})\ddot{z}_2 + c_2 \dot{z}_2 + k_2 z_2 = F_{02}\cos(\omega \cdot t - \varphi_2) \end{aligned} \tag{17}$$

where :

$$F_{01} = \sqrt{(k_1 - h_0)^2 + (c_1 h_0 \omega)^2}\ ; \ F_{02} = \sqrt{(k_2 h_0)^2 + (c_2 h_0 \omega)^2}$$

$$\varphi_1 = arctg\frac{k_1}{c_1\omega}\ ; \ \varphi_2 = arctg\frac{k_2}{c_2\omega}$$

The undamped free vibration equations are obtained from the scalar differential equations. The parameters c_1, c_2, h_1, and h_2 are zero.

Substituting the differential equations and simplifying with $\sin(\omega t + \varphi)$ we obtain:

$$(k_1 - \mathrm{M}_{11}\omega^2)A - \mathrm{M}_{12}\omega^2 B = 0 \qquad -\mathrm{M}_{21}\omega^2 A + (k_2 - \mathrm{M}_{22}\omega^2)B = 0 \tag{18}$$

For the system to recognize different solutions from the trivial solution $A = B = 0$, we impose the condition that the determinant of the system is cancelled :

$$\begin{vmatrix} k_1 - M_{11}\omega^2 & -M_{12}\omega^2 \\ -M_{21}\omega^2 & k_2 - M_{22}\omega^2 \end{vmatrix} = 0 \tag{19}$$

Expanding the determinant we obtain the equation of the natural frequency of vibration:

$$(M_{11}M_{22} - M_{12}M_{21})\omega^4 - (k_1 M_{22} + k_2 M_{11})\omega^2 + k_1 k_2 = 0 \tag{20}$$

The following notations are made:

$$a_1 = M_{11}M_{22} - M_{12}M_{21} \qquad b_1 = k_1 M_{22} + k_2 M_{11} \tag{21}$$

Substituting in relation (20) we obtain:

$$\omega^4 - \frac{b_1}{a_1}\omega^2 + \frac{k_1 k_2}{a_1} = 0 \tag{22}$$

The natural frequencies of vibration are:

$$\omega_{1,2}^2 = \frac{b_1}{2a_1} \pm \sqrt{\frac{b_1^2}{4a_1^2} - \frac{k_1 k_2}{a_1}} \tag{23}$$

The free linear vibration damping equations are:

$$M_{11}\ddot{z}_1 + c_1\dot{z}_1 + k_1 z_1 + M_{12}\ddot{z}_2 = 0 \qquad M_{12}\ddot{z}_1 + c_2\dot{z}_2 + k_2 z_2 + M_{22}\ddot{z}_2 = 0 \tag{24}$$

We try a solutions of exponential form:

$$z_1 = A_1 e^{\lambda t} \qquad z_2 = A_2 e^{\lambda t} \tag{25}$$

The solutions for the amplitudes will exist if the determinant formed by the coefficients of these unknowns (Lagrange determinant) is zero:

$$\Delta = \begin{vmatrix} M_{11}\lambda^2 + c_1\lambda + k_1 & M_{12}\lambda^2 \\ M_{21}\lambda_2 & M_{22}\lambda^2 + c_2\lambda + k_2 \end{vmatrix} = 0 \tag{26}$$

The motion equations of the system are:

- if the roots are real: $\lambda_1 = \alpha$ when $z = C_1 e^{\alpha t}$;
- if the roots are imaginary:

$$\lambda_{1,2} = \pm\beta i \quad z = C_1 \cos\beta t + C_2 \sin\beta t \tag{27}$$

$$\lambda_{1,2} = -\alpha \pm \beta i \quad z = e^{-\alpha t}(C_1 \cos\beta t + C_2 \sin\beta t) \tag{28}$$

The differential equations for forced vibration with damping are:

$$M_{11}\ddot{z}_1 + c_1\dot{z}_1 + c_1\dot{z}_1 + k_1 z + M_{12}\ddot{z}_2 = F_1 \qquad M_{21}\ddot{z}_1 + M_{22}\ddot{z}_2 + c_2\dot{z}_2 + k_2 z_2 = F_2 \tag{29}$$

We consider the perturbation due to the road (7 relationships) and study the steady harmonic motion. In this particular case the solutions are trigonometric:

$$z_1 = q_1 = A_1 \sin\omega\cdot t + B_1 \cos\omega\cdot t \tag{30}$$

$$z_2 = q_2 = D_2 \sin\omega\cdot t + E_2 \cos\omega\cdot t$$

The following notations are made:

$$D_2 = A_2\cos\phi - B_2\sin\phi \qquad E_2 = -A_2\sin\phi + B_2\cos\phi \tag{31}$$

The solutions are:

$$\begin{aligned} z_1 &= A_1 \sin\omega t + B_1 \cos\omega t \\ z_2 &= A_2 \sin(\omega t + \varphi) + B_2 \cos(\omega t + \varphi) \end{aligned} \tag{32}$$

where:

$$A_2 = D_2 \cos\varphi + E_2 \sin\varphi; \qquad B_2 = \frac{E_2 - A_2 \sin\varphi}{\cos\varphi}$$

The vertical forces on the wheels are:

$$Z_1 = Z_{01} + Z_{d1} \qquad\qquad Z_2 = Z_{02} + Z_{d2} \tag{33}$$

where:

- Z_1 - the vertical force on the front driving wheel;
- Z_2 - the vertical force on the rear drive wheel;
- Z_{01} - the static vertical force on the front motor wheel;
- Z_{02} - the static vertical force on the rear drive wheel;
- Z_{d2} - the dynamic vertical force on on the rear drive wheel;
- Z_{d1} - the dynamic vertical force on the front motor wheel

$$Z_{d1} = k_1 z_1 + c_1 \dot{z}_1 + \frac{m_f}{2}\ddot{z}_1 \qquad\qquad Z_{d2} = k_2 z_2 + c_2 \dot{z}_2 + \frac{m_s}{2}\ddot{z}_2 \tag{34}$$

The movement of the tractor mass center (z) and the pitching motion (θ) are given by the expressions:

$$z = \frac{(L-1)z_1 - z_2}{L} \qquad\qquad \theta = \frac{z_1 - z_2}{L} \tag{35}$$

2. The oscillations of the unit on asphalt road

The unit consisting of the New Holland T6070 tractor and deep soil loosening machine EAA - 220 (diagram is shown in Figure 2) has the following design parameters:

- M_t = 5410 Kg, mass of tractor;
- h_g = 1.05 m, distance from the support surface to the center of gravity of the tractor;
- M_m = 1290 Kg, mass of the machine;
- M_l = 450 Kg, ballast mass;
- L = 2.723 m, tractor wheelbase;
- l_l = 3.775 m, distance from the center of the rear wheel to the ballast center of gravity;
- l_m =2 m, distance from the center of the rear wheel to center of gravity of the machine;
- h_g = 1.05 m, distance from the support surface to the center of gravity of the tractor;
- h_m = 1.5 m, distance from the support surface to the center of gravity of the machine;

- h_l = 0.9 m, distance from the support surface to the center of gravity of the ballast;
- Front wheels 14.9/28, rear driving wheels 18.4/38.

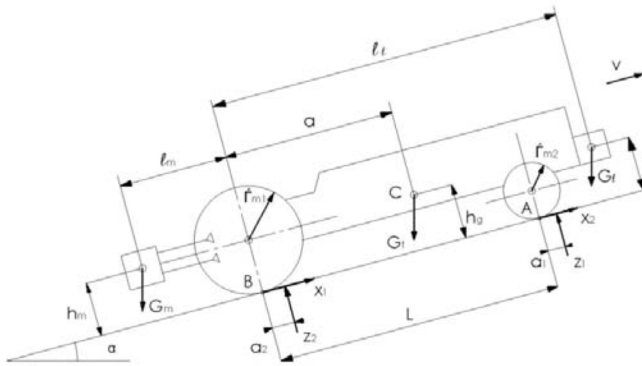


Fig.2. The tractor – deep soil lossening machine unit

Movement of the unit is on asphalt road with height of irregularities h = 16.69 mm and length L = 1.88 m

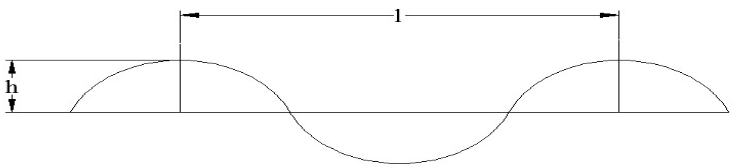


Fig.3. Sizes that characterize the road profile [3]

A MathCAD program that models the dynamics of the unit was developed by the authors of the paper based on the presented algorithm. The imput values are given for the design parameters of the unit, and for the terrain and speed parameters. The calculated parameters are:

- The tractors natural frequency of vibration ω_1 and ω_2;
- pulsation ω (s^{-1}) and the frequency of oscillation due to the terrain (Hz);
- the magnitude of the rotation angle of the tractor around the center of mass θ, (rad);
- the amplitude of the oscillation for the tractor wheels z_1 (m) and z_2 (m);
- magnitude of vertical forces on the front wheel Z_{1max} ,Z_{1min} (daN) and on the rear wheel Z_{2max} ,Z_{2min} (daN) of the tractor.

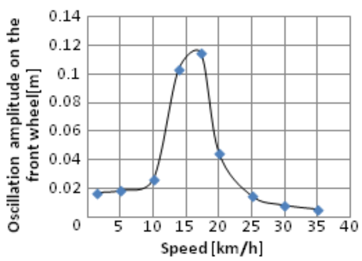
Thus, the tractor – deep soil loosening machine system moving on asphalt road has a natural frequency of ω_1 = 12.852 s^{-1} corresponding to the speed of v_1 = 13.844 km/h and ω_2 = 16.016 s^{-1} for v_2 = 17.252 km/h.

Tabel 1

The parameters of oscillations of the aggregate for asphalt road

Speed (km/h)	Own pulsations (s^{-1})	Center of mass oscilations		Wheels oscilations		Vertical force on the wheels (daN)			
		z (m)	θ(rad)	z_1(m)	z_2(m)	Z_{1max}	Z_{1min}	Z_{2max}	Z_{2min}
1.5	1.393	0.0168	0.0000256	0.01682	0.0168	1801	747	4246	1630
5	4.642	0.0182	0.0000287	0.0183	0.0183	1801	747	4246	1630
10	9.248	0.0251	0.00155	0.0262	0.024	1810	739	4238	1638
13.844	12.852	0.0547	0.029	0.103	0.032	2325	223	3853	2023
17.252	16.016	0.12	0.01	0.115	0.124	1553	994	4550	1325
20	18.567	0.0434	0.00212	0.045	0.046	1828	720	4234	1642
25	23.209	0.0148	0.0004036	0.0145	0.015	1822	725	4227	1649
30	27.85	0.0081	0.000166	0.008	0.0083	1818	729	4230	1645
35	32.5	0.0053	0.000103	0.0052	0.0054	1817	731	4232	1643

The oscilations amplitude of the front driving wheels of the tractor varies depending on the operating speed shown in Figure 4a and the rear wheel in Figure 4b. At a speed of v_2 = 17.252 km/h, corresponding to their pulsation ω_2 = 16.016 s^{-1}, the oscillation amplitude is about six times higher than the minimum speed and 22 times higher than the maximum speed.



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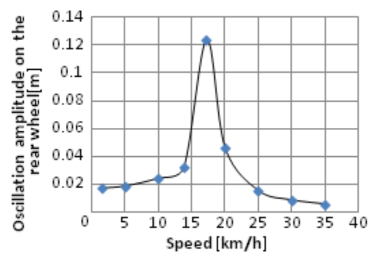


b

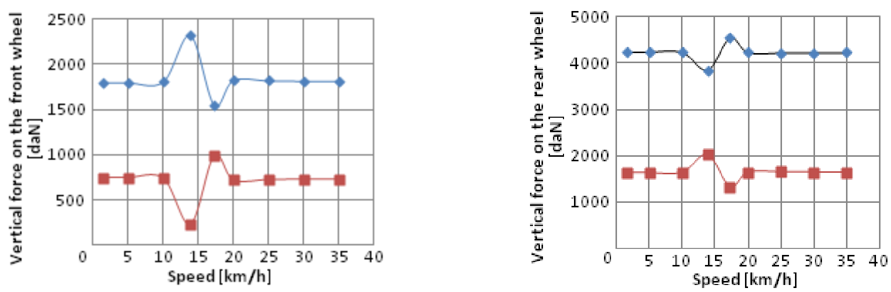
Fig.4. The oscilations amplitude variation on the tractor wheels

For the examined speed range, except the value v_2 = 17.252 km/h, the oscillation amplitude of the tractor's rear driving wheel can be calculated using the equation:

$$z_2 = -1 \cdot 10^{-6} \cdot v^3 - 0.0001 \cdot v^2 + 0.0019 \cdot v + 0.0132 \qquad (36)$$

The vertical force on the front wheel of the tractor varies between Z_{1max} = 1800 daN and Z_{1min} = 720 daN (Figure 5a) and on the rear wheel varies between Z_{2min} = 1325 daN and Z_{2max} = 4550 daN (Figure 5b) . On its own pulsation ω_1 = 12.852 s^{-1} (v_1 = 13.884 km/h) the vertical forces vary mostly between 2325 and 223 daN and for v_2 = 17.252 km/h the smallest variation is between 1553 and 994 daN. The travel speed v_1 = 13.884 km/h should be avoided because the front wheel load is very small and the tractor may lose stability. Maximum and minimum vertical forces on the front driving wheel can be calculated with the following equations:

$$Z_{1max} = -0.001 \cdot v^3 - 0.017 \cdot v^2 - 1.431 \cdot v + 1796$$
$$Z_{1min} = -0.001 \cdot v^3 - 0.046 \cdot v^2 - 1.037 \cdot v + 750.6 \quad (37)$$



a b

Fig.5. The amplitude minimum and maximum variation of the vertical forces on tractor wheels

On its own pulsation $\omega_2 = 16.016$ s^{-1} ($v_2 = 17.252$ km/h) the vertical force on the rear drive wheel varies mostly between 4550 and 1325 daN and the own pulsation $\omega_1 = 12.852$ s^{-1} ($v_1 = 13.884$ km/h) has the smallest variation between 3853 and 2023 daN. Maximum and minimum vertical forces on the rear motor wheel can be calculated with the following equations:

$$Z_{2max} = -0.001 \cdot v^3 - 0.038 \cdot v^2 - 0.471 \cdot v + 4247$$
$$Z_{2min} = -0.001 \cdot v^3 + 0.037 \cdot v^2 + 0.498 \cdot v + 1628 \quad (38)$$

The accelerations amplitude of the rear and front axles, calculated with the mathmatical modeling program are presented in Table 2.

Tabel 2

The vertical accelerations amplitudes for the axles of the tractor moving on asphalt road

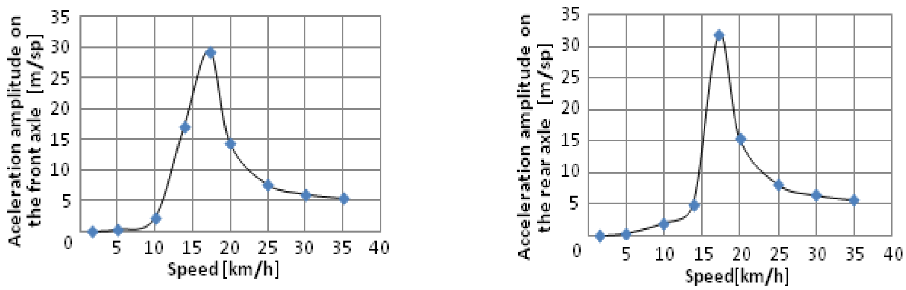
Accelerations [m/s^2]	Movement speed [km/h]								
	1.5	5	10	13.844	17.252	20	25	30	35
Front axle	0.033	0.395	2.264	17.03	29.4	14.3	7.68	6.1	5.42
Rear axle	0.033	0.391	2.11	4.92	3.2	15.4	8.16	6.44	5.71

Figure 6a presents the variation of the acceleration of the tractor front axle and Figure 6b shows the variation of the acceleration on the rear axle of the tractor according to diferent movement speed.

One can notice that the accelerations for the tractor wheels have a maximum value for its own pulsation $\omega_2 = 16.016$ s^{-1} in case the speed is $v_2 =$ 17.252 km/h.

The acceleration amplitude of the front and rear axles out of the range speeds of $v = 15 - 25$ km/h can be calculated with the following equations:

$$a_{PF} = 0.0006 \cdot v^3 + 0.0249 \cdot v^2 + 0.075 \cdot v - 0.296$$
$$a_{PS} = -0.0008 \cdot v^3 + 0.0308 \cdot v^2 + 0.0069 \cdot v + 0.1764 \quad (39)$$



a b

Fig.6. The acceleration amplitude variation for the tractors front and rear axle

To validate the theoretical results for the unit moving on asphalt road, we determined experimentally the variation of acceleration for the front and rear tractor wheels. Two piezoelectric transducers mounted as in Figure 7 and a PCB Piezoelectics amplifier were used. Each amplifier uses two channels, one for the signal in the horizontal direction x and the second channel signal obtained in the vertical direction y.

a b

Fig.7. Set-up of the piezoelectric acceleration transducers

Accelerations amplitudes were measured on the vertical and horizontal directions of the tractor wheel moving on asphalt road. The vertical acceleration data are presented in Table 3.

The amplitude recordings of the vertical acceleration of the tractor bridges (Table 3) are shown in figure 8a for sample 1, figure 8b for sample 2, figure 9a for sample 3 and figure 9b for sample 4.

From figure 8a one can notice that the magnitude of the vertical acceleration of the front axle is 4.55 m/s^2 from the calculated value of 6.1 m/s^2 and for the rear axle 6.84 m/s^2 from the calculated value of 6.44 m/s^2.

In this sample, the units displacement and the amplitudes and accelerations measurement were done on the area where the measurements to determine the terrain irregularities were made.

Table 3

The vertical accelerations amplitude of the tractor wheels

Sample	Land	Speed [km/h]	Acceleration front axle (Y) [m/s^2]			Acceleration rear axle (Y) [m/s^2]		
			Maximum	Average	Minimum	Maximum	Average	Minimum
1	Asphalt road	30	4.5577	0.098293	-10.5486	6.84822	0.11004	-7.73081
2		13	18.9295	0.096727	-54.2739	9.17268	0.111465	-8.77852
3		20	13.1864	0.084392	-28.9796	5.98432	0.093993	-5.24089
4		17	28.3295	0.080716	-196.794	6.03521	0.091387	-5.45512

Figure 8a shows that the magnitude of the vertical acceleration of the front axle is 4.55 m/s^2 which differs from the calculated value of 6.1 m/s^2 and for the rear axle 6.84 m/s^2 which also differs from the calculated value of 6.44 m/s^2.

In this sample the units displacement and the amplitudes and accelerations measurements were done on the area where the measurements to determine the terrain irregularities were made.

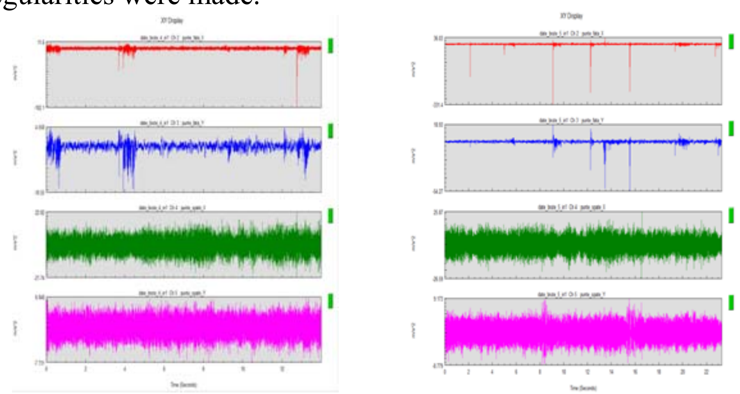


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Fig.8. The accelerations recorded in sample 1 (a) and sample 2 (b)

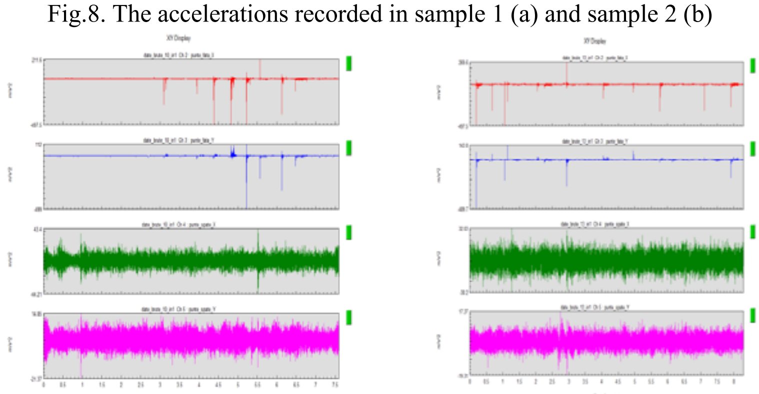

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Fig.9. The accelerations recorded in sample 3 (a) and sample 4 (b)

3. Conclusions

1. Modeling the oscillations of the deep soil loosening machine allows assessing trends in the variation of oscillation movement parameters when driving on various roads within the determined speed limits;
2. At a speed v_2 = 17.252 km/h, corresponding to their own pulsation ω_2 = 16.016 s^{-1}, the tractor wheels oscillation amplitude is about 6 times higher than the minimum speed and 22 times higher than the maximum speed.
3. At a natural frequency of vibration ω_1 = 12.852 s^{-1} (v_1 = 13.884 km/h) the vertical force varies mostly between 2325 daN and 223 daN. This working speed must be avoided because the front wheel load is very small and the tractor may lose stability.
4. In sample 1, which was made on the moving unit where the land irregularities were determined, the calculated and measured tractors wheels vertical accelerations are very close one to the other.

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