THE DYNAMICS OF THE DEEP SOIL LOOSENING MACHINE – TRACTOR UNIT

Alexandru-Dorin DAVID¹, Gheorghe VOICU², Mihaela DUȚU³

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 $c_1, c_2$ - the damping coefficients of the front and rear tires.

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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:

$$m_s = \lambda_{din}(m_t + m_l + m_m) \quad m_f = m_t + m_l + m_m - m_s$$  \hspace{1cm} (1)

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

$$\lambda_{mdin} = \frac{m_l[(L - b)\cos\alpha + h_g\sin\alpha] + m_l[h_g\sin\alpha - (l_l - L)\cos\alpha]}{L(m_t + m_l + m_m)} + \frac{m_m[h_m\sin\alpha + (l_m + L)\cos\alpha]}{L(m_t + m_l + m_m)}$$  \hspace{1cm} (2)

The following notations were made:

- $\alpha$ – 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, $c_{dfs} = 0.07 - 0.025$ [5] the tires deformations can be calculated:

$$\delta_1 = c_{df} \cdot r_{01} \quad \delta_2 = c_{dfs} \cdot r_{02}$$  \hspace{1cm} (3)

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

$$k_1 = \frac{m_f \cdot 9.81}{2 \cdot \delta_1} \quad k_2 = \frac{m_s \cdot 9.81}{2 \cdot \delta_2}$$  \hspace{1cm} (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 ($\phi$ - is the phase) and can be calculated with the relations [1]:

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Fig.1. The mechanical model with two degrees of freedom for the vibration study
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 \( \theta \) 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:

\[
z_1 = z + a \cdot \theta \\
z_2 = z - b \cdot \theta
\]

(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:

\[
E = \frac{1}{2} m_1 \dot{z}_1^2 + \frac{1}{2} J \dot{\theta}^2 + \frac{1}{2} m_2 \dot{z}_2^2
\]

(7)

where:

\( m_1 = m_f/2 \) and \( m_2 = m_r/2 \) 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 \( m_f \) and the mass of the front wheel axis \( m_{af} \) respectively \( m_r \) and \( m_{as} \) for the rear axle, the following relations are used:

\[
m_{pf} = 2 \cdot m_{rf} + m_{af} \\
m_{ps} = 2 \cdot m_{rs} + m_{as}
\]

(8)

Mass of the tractor chassis is:

\[
m_{st} = m_t + m_i + m_m - m_{pf} - m_{ps}
\]

(9)

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

\[
J = \frac{1}{2} \left\{ m_{pf} \cdot a^2 + m_{ps} \cdot b^2 + m_{m} \left( a - \frac{L}{2} \right)^2 + m_{m} \cdot \frac{L^2}{12} \right\}
\]

(10)

The potential energy of the system is:

\[
U = -\left( \frac{1}{2} k_1 z_1^2 + \frac{1}{2} k_2 z_2^2 \right)
\]

(11)

The energy dissipated in shock is given by:

\[
D = -\left( \frac{1}{2} c_1 z_1^2 + \frac{1}{2} c_2 z_2^2 \right)
\]

(12)

The inertia matrix \( (M) \), the stiffness matrix \( (K) \) and the damping matrix \( (C) \) are given by the equations:
The forces that act during the movement are:

\[ F = \begin{bmatrix} k_1 \cdot \dot{h}_1 + c_1 \cdot \ddot{h}_1 \\ k_2 \cdot \dot{h}_2 + c_2 \cdot \ddot{h}_2 \end{bmatrix} \]

The motion parameters matrix is:

\[ Z = \begin{bmatrix} z \\ 0 \end{bmatrix} \]

The differential equation matrix of equilibrium is:

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

The differential scalar equations of equilibrium are:

\[ (m_1 + \frac{J}{L^2}) \ddot{z}_1 + c_1 \dot{z}_1 + k_1 z_1 - \frac{J}{L^2} \ddot{z}_2 = F_{01} \cos(\omega t - \varphi_1) \]

\[ \frac{J}{L^2} \ddot{z}_1 + (m_2 + \frac{J}{L^2}) \ddot{z}_2 + c_2 \dot{z}_2 + k_2 z_2 = F_{02} \cos(\omega t - \varphi_2) \]

where:

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

\[ \varphi_1 = \arctan \frac{k_1}{c_1 \omega} ; \quad \varphi_2 = \arctan \frac{k_2}{c_2 \omega} \]

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:

\[ (k_1 - M_{11} \omega^2) A - M_{12} \omega^2 B = 0 \quad -M_{21} \omega^2 A + (k_2 - M_{22} \omega^2) B = 0 \]

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:

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

Expanding the determinant we obtain the equation of the natural frequency of vibration:
The dynamics of the deep soil loosening machine – tractor unit

\[(M_{11}M_{22} - M_{12}M_{21})\omega^4 - (k_1M_{22} + k_2M_{11})\omega^2 + k_1k_2 = 0 \quad (20)\]

The following notations are made:
\[a_1 = M_{11}M_{22} - M_{12}M_{21} \quad b_1 = k_1M_{22} + k_2M_{11} \quad (21)\]

Substituting in relation (20) we obtain:
\[\omega^4 - \frac{b_1}{a_1} \omega^2 + \frac{k_1k_2}{a_1} = 0 \quad (22)\]

The natural frequencies of vibration are:
\[\omega_{1,2}^2 = \frac{b_1}{2a_1} \pm \sqrt{\frac{b_1^2}{4a_1^2} - \frac{k_1k_2}{a_1}} \quad (23)\]

The free linear vibration damping equations are:
\[M_{11}\ddot{z}_1 + c_1\ddot{z}_1 + k_1z_1 + M_{12}\ddot{z}_2 = 0 \quad M_{12}\ddot{z}_1 + c_2\ddot{z}_2 + k_2z_2 + M_{22}\ddot{z}_2 = 0 \quad (24)\]

We try a solutions of exponential form:
\[z_1 = A_1 e^{\lambda t} \quad z_2 = A_2 e^{\lambda t} \quad (25)\]

The solutions for the amplitudes will exist if the determinant formed by the coefficients of these unknowns (Lagrange determinant) is zero:
\[\Delta = \begin{vmatrix} M_{11}\lambda^2 + c_1\lambda + k_1 & M_{12}\lambda^2 \\ M_{21}\lambda + c_2 & M_{22}\lambda^2 + c_2\lambda + k_2 \end{vmatrix} = 0 \quad (26)\]

The motion equations of the system are:

- if the roots are real: \( \lambda = \alpha \) when \( z = C_1 e^{\omega t} \);
- if the roots are imaginary:
  \[\lambda_{1,2} = \pm \beta i \quad z = C_1 \cos \beta t + C_2 \sin \beta t \quad (27)\]
  \[\lambda_{1,2} = -\alpha \pm \beta i \quad z = e^{-\alpha t} (C_1 \cos \beta t + C_2 \sin \beta t) \quad (28)\]

The differential equations for forced vibration with damping are:
\[M_{11}\ddot{z}_1 + c_1\ddot{z}_1 + c_1\ddot{z}_1 + k_1z_1 + M_{12}\ddot{z}_2 = F_1 \quad M_{21}\ddot{z}_1 + M_{22}\ddot{z}_2 + c_2\ddot{z}_2 + k_2z_2 = F_2 \quad (29)\]

We consider the perturbation due to the road (7 relationships) and study the steady harmonic motion. In this particular case the solutions are trigonometric:
\[z_1 = q_1 = A_1 \sin \omega t + B_1 \cos \omega t \quad (30)\]
\[z_2 = q_2 = D_2 \sin \omega t + E_2 \cos \omega t \]

The following notations are made:
\[D_2 = A_2 \cos \phi - B_2 \sin \phi \quad E_2 = -A_2 \sin \phi + B_2 \cos \phi \quad (31)\]

The solutions are:
\[z_1 = A_1 \sin \omega t + B_1 \cos \omega t \quad z_2 = A_2 \sin \left(\omega t + \phi\right) + B_2 \cos \left(\omega t + \phi\right) \quad (32)\]
where:

\[ A_2 = D_2 \cos \varphi + E_2 \sin \varphi; \quad B_2 = \frac{E_2 - A_2 \sin \varphi}{\cos \varphi} \]

The vertical forces on the wheels are:

\[ Z_1 = Z_{01} + Z_{d1} \quad Z_2 = Z_{02} + Z_{d2} \quad (33) \]

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 the rear drive wheel;
- \( z_{d1} \) - the dynamic vertical force on the front motor wheel.

\[ Z_{d1} = k_1 z_1 + c_1 z_1 + \frac{m}{2} \ddot{z}_1 \quad Z_{d2} = k_2 z_2 + c_2 \dddot{z}_2 + \frac{m}{2} \dddot{z}_2 \quad (34) \]

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

\[ z = \frac{(L - l) z_1 - l z_2}{L} \quad \theta = \frac{z_1 - z_2}{L} \quad (35) \]

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;
- \( l_l = 2 \) 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;
\[ h_I = 0.9 \text{ m}, \text{ distance from the support surface to the center of gravity of the ballast}; \]
\[ \text{Front wheels } 14.9/28, \text{ rear driving wheels } 18.4/38. \]

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

- The tractor’s natural frequency of vibration \( \omega_1 \) and \( \omega_2 \);
- pulsation \( \omega (\text{s}^{-1}) \) and the frequency of oscillation due to the terrain (Hz);
- the magnitude of the rotation angle of the tractor around the center of mass \( \theta \), (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 \( Z_{1\text{max}}, Z_{1\text{min}} \) (daN) and on the rear wheel \( Z_{2\text{max}}, Z_{2\text{min}} \) (daN) of the tractor.

Thus, the tractor – deep soil loosening machine system moving on asphalt road has a natural frequency of \( \omega_1 = 12.852 \text{ s}^{-1} \) corresponding to the speed of \( v_1 = 13.844 \text{ km/h} \) and \( \omega_2 = 16.016 \text{ s}^{-1} \) for \( v_2 = 17.252 \text{ km/h} \).
The parameters of oscillations of the aggregate for asphalt road

<table>
<thead>
<tr>
<th>Speed (km/h)</th>
<th>Own pulsations (s⁻¹)</th>
<th>Center of mass oscillations (m)</th>
<th>Wheels oscillations (m)</th>
<th>Vertical force on the wheels (daN)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Z(m)</td>
<td>θ(rad)</td>
<td>z₁(m)</td>
<td>z₂(m)</td>
</tr>
<tr>
<td>1.5</td>
<td>1.393</td>
<td>0.0168</td>
<td>0.0000256</td>
<td>0.01682</td>
</tr>
<tr>
<td>5</td>
<td>4.642</td>
<td>0.0182</td>
<td>0.0000287</td>
<td>0.0183</td>
</tr>
<tr>
<td>10</td>
<td>9.248</td>
<td>0.0251</td>
<td>0.00155</td>
<td>0.0262</td>
</tr>
<tr>
<td>13.844</td>
<td>12.852</td>
<td>0.0547</td>
<td>0.029</td>
<td>0.103</td>
</tr>
<tr>
<td>17.252</td>
<td>16.016</td>
<td>0.12</td>
<td>0.01</td>
<td>0.115</td>
</tr>
<tr>
<td>20</td>
<td>18.567</td>
<td>0.0434</td>
<td>0.00212</td>
<td>0.045</td>
</tr>
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<td>25</td>
<td>23.209</td>
<td>0.0148</td>
<td>0.0004036</td>
<td>0.0145</td>
</tr>
<tr>
<td>30</td>
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<td>0.0081</td>
<td>0.000166</td>
<td>0.008</td>
</tr>
<tr>
<td>35</td>
<td>32.5</td>
<td>0.0053</td>
<td>0.000103</td>
<td>0.0052</td>
</tr>
</tbody>
</table>

The oscillations 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 \( \omega_2 = 16.016 \) s⁻¹, the oscillation amplitude is about six times higher than the minimum speed and 22 times higher than the maximum speed.

\[
z_2 = -1 \cdot 10^{-6} \cdot \nu^3 - 0.0001 \cdot \nu^2 + 0.0019 \cdot \nu + 0.0132 \quad (36)
\]

The vertical force on the front wheel of the tractor varies between \( Z_{1\text{max}} = 1800 \) daN and \( Z_{1\text{min}} = 720 \) daN (Figure 5a) and on the rear wheel varies between \( Z_{2\text{min}} = 1325 \) daN and \( Z_{2\text{max}} = 4550 \) daN (Figure 5b). On its own pulsation \( \omega_1 = 12.852 \) 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 \( v_1 \) 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:
The dynamics of the deep soil loosening machine – tractor unit

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

On its own pulsation \( \omega_2 = 16.016 \text{ s}^{-1} \) \( (v_2 = 17.252 \text{ km/h}) \) the vertical force on the rear drive wheel varies mostly between 4550 and 1325 daN and the own pulsation \( \omega_1 = 12.852 \text{ s}^{-1} \) \( (v_1 = 13.884 \text{ km/h}) \) 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:

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

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

<table>
<thead>
<tr>
<th>The vertical accelerations amplitudes for the axles of the tractor moving on asphalt road</th>
</tr>
</thead>
<tbody>
<tr>
<td>Accelerations [m/s²]</td>
</tr>
<tr>
<td>----------------------</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Front axle</td>
</tr>
<tr>
<td>Rear axle</td>
</tr>
</tbody>
</table>

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 different movement speed.

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

The acceleration amplitude of the front and rear axes out of the range speeds of \( v = 15 – 25 \text{ km/h} \) can be calculated with the following equations:

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

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/s² from the calculated value of 6.1 m/s² and for the rear axle 6.84 m/s² from the calculated value of 6.44 m/s².

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.
The vertical accelerations amplitude of the tractor wheels

<table>
<thead>
<tr>
<th>Sample</th>
<th>Land</th>
<th>Speed [km/h]</th>
<th>Acceleration front axle (Y) [m/s²]</th>
<th>Acceleration rear axle (Y) [m/s²]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Maximum</td>
<td>Average</td>
</tr>
<tr>
<td>1</td>
<td>Asphalt road</td>
<td>30</td>
<td>4.5577</td>
<td>0.098293</td>
</tr>
<tr>
<td>2</td>
<td>Asphalt road</td>
<td>13</td>
<td>18.9295</td>
<td>0.096727</td>
</tr>
<tr>
<td>3</td>
<td>Asphalt road</td>
<td>20</td>
<td>13.1864</td>
<td>0.084392</td>
</tr>
<tr>
<td>4</td>
<td>Asphalt road</td>
<td>17</td>
<td>28.3295</td>
<td>0.080716</td>
</tr>
</tbody>
</table>

Figure 8a shows that the magnitude of the vertical acceleration of the front axle is 4.55 m/s² which differs from the calculated value of 6.1 m/s² and for the rear axle 6.84 m/s² which also differs from the calculated value of 6.44 m/s².

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.
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 $\omega_2 = 16.016$ 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 $\omega_1 = 12.852$ 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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