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Practical Computation Models for the Structural Strength of the New Blade Unit in Crowler Bulldozers

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Abstract

For the manufacture of new-generation blade bulldozers with an increased degree of productivity, we propose new computation models which refer to the following cases: shock against an obstacle, elastic/plastic deformation, or the dynamic working regime of the dozer under acceleration upgrade velocity, for earth digging on a given distance. The proposed computation models are accompanied by the rotation laws specific for the new, high performance blade type.

Key words: *next-generation bulldozer dynamics, Sigmadozer blade, calculation model for Sigmadozer blade*

Introduction

New-generation Komatsu bulldozers include advanced technology components, such as the Sigmadozer blade type (manufactured by Komatsu) and K-bogie undercarriage system which ensure a higher level of productivity, energy consumption, of traction force and dozer stability [3].

Only two models from the Komatsu bulldozer offer (D155AX-6 and D-155AX-7) are provided with a wide range of blade and K-bogie systems. On the other hand, K-bogies are also used for larger bulldozer models: D275AX-5 and D375-6 [5, 6].

According to the traction diagram, in the case of D 155AX-6 (fig. 2) the required traction force is maintained at all times and, as a whole, the fuel consumption is reduced by 10% while the engine operability is 15% higher if the Sigmadozer, Komatsu brand blade is used [3]. On the traction diagram in Figure 2 the following notations were used: 1 - continuous line, for the automatic coupling mode, and 2 – dotted line, for the manual coupling mode of the power shaft [3].

The structure of the Sigmadozer-type blade and the advantages of its use, in comparison with a classical type are shown in Figure 3. The central segment of the blade is very strong and its ends are V-shaped, in order to cut the earth previously dug by the center part, when the shoveled material advances sideways.

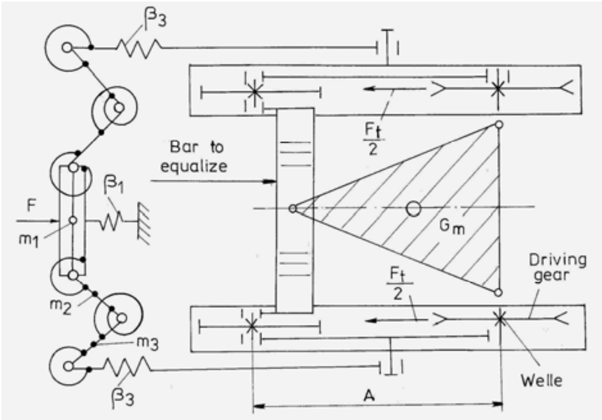

Fig. 1. Configuration of the bulldozer with Sigmadozer blade.

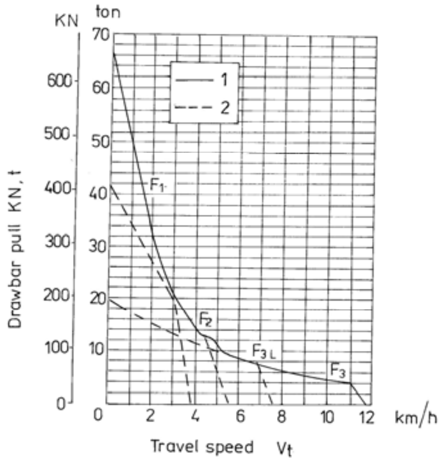

Fig. 2. Traction diagram of D155 AX-6: 1 - automatic coupling; 2 - manual coupling mode [4].

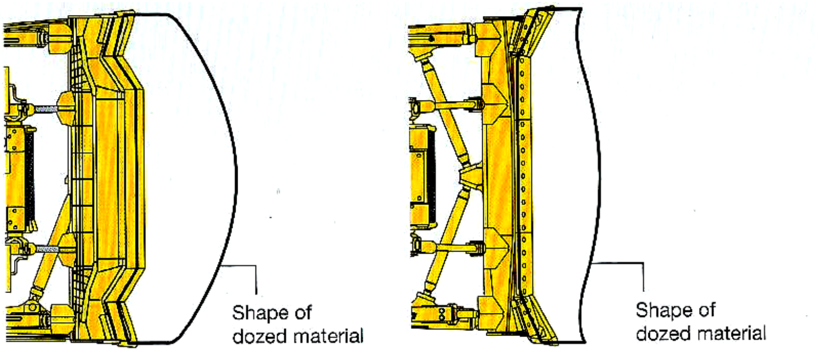

Fig. 3. The structure of the Sigmadozer blade in comparison with a classic type [4].

Komatsu Sigmadozer Bulldozer Blade Modelling

Take the elastic model for a blade of the shape shown in Figure 4, made of a central mass m_1 elastically mounted on the curved area having an elastic constant β_1; the digging force F is applied on the mass, together with the inertia of mass force $F_{1j} = m_1 \cdot \ddot{y}_1$ and the inertia moment $\dot{y}_{1i} \cdot \ddot{\varphi}_1$. The mass length is marked by a_1. A concentrated force F is applied on the mass centre; it stands for the earth load over the blade. The convergent action of these forces determines a swing of mass m_1 with an angle φ_1.

The V-shaped structure at both ends of the central blade is made of two masses m_2 and m_3 bound together elastically and connected to mass m_1 through springs with a cylindrical bending strength K_1, K_2 and K_3. Mass m_3 is placed elastically on the blade end on the spring with strength constant β_3. Inertia forces $F_{21} = m_2 \cdot \ddot{y}_2$ and $F_{3i} = m_3 \cdot \ddot{y}_3$ as well as inertia moments $J_{2i} \cdot \ddot{\varphi}_2$ şi $J_{3i} \cdot \ddot{y}_3$ apply over masses m_2 şi m_3 respectively.

Mass m_2 takes an angle α_1 in comparison to mass m_1, while mass m_3 takes an angle α_2 in comparison to m_2.

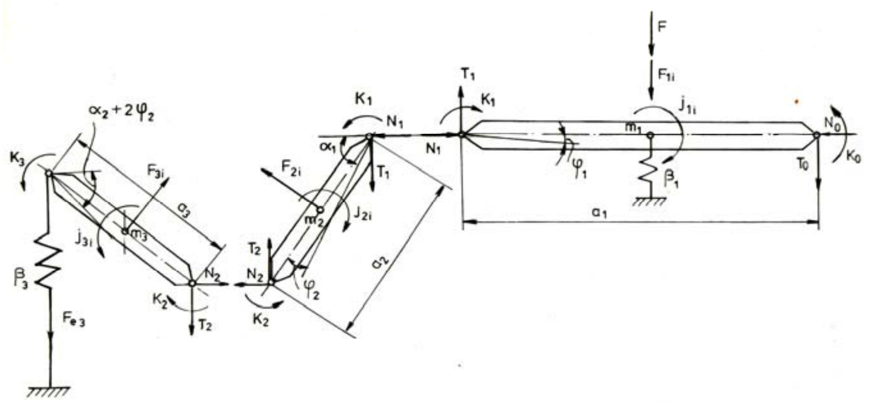

Fig. 4. The elastic model for a blade

If masses m_1, m_2 and m_3 are no longer connected, normal forces N_1, N_2 and N_3 and tangent forces T_1, T_2 and T_3 apply in the joints, along with moments K_1, K_2, K_3. Because the blade is symmetrical, forces N_0, T_0 and moment M_0 apply at the right end of mass m_1.

An elastic force appears along the strength spring β_1 when the central segment of the blade $F_{e1}=\beta_1\cdot y_1$ is deformed. The elastic force acting at the left end of the blade is $F_{e3}=\beta_3\cdot(y_2+y_3)$.

In order to comply with the symmetry condition of forces and moments applied at the ends of mass m_1 we take: $N_0 = N_1$, $T_0 = T_1$, $K_0 = K_1$. We note $\varphi_2+\varphi_3=2\cdot\varphi_2$ and $\alpha_1=\alpha_2$.

The maximum digging force is generated by the blade thrust force, which is the traction force of the basic unit in the engine traction diagram (fig. 2).

Level (horizontal) projection equations for the three mass values are ignored. Here is the resulting simplified system of differential equations:

$$F=m_1\cdot\frac{a_1}{2}\cdot\ddot{\varphi}_1+\beta_1\cdot\frac{a_1}{2}\cdot\varphi_1\ ; \tag{1}$$

$$T_1\cdot a_1-\frac{1}{6}m_1\cdot a_1^2\cdot\ddot{\varphi}_1+\beta_1\cdot\frac{a_1^2}{4}\cdot\varphi_1+F\cdot\frac{a_1}{2}=0\ ; \tag{2}$$

$$T_1-T_2+m_2\cdot\frac{a_2}{2}\cdot\ddot{\varphi}_2\cdot\cos\alpha_1=0\ ; \tag{3}$$

$$T_2\cdot a_2\cdot\cos\alpha_1+\frac{1}{3}\cdot m_2\cdot a_2^2\cdot\ddot{\varphi}_2-\beta_1\cdot\frac{a_1^2}{4}\cdot\varphi_1-\beta_3\cdot\frac{a_3^2}{2}\cdot\varphi_2=0 \tag{4}$$

$$2\cdot\beta_3\cdot a_3\cdot\varphi_2+m_3\cdot a_3\cdot\ddot{\varphi}_2\cdot\cos\alpha_2=0\ ; \tag{5}$$

$$-\frac{1}{3}\cdot m_3\cdot a_3^2\cdot\ddot{\varphi}_2+2\cdot\beta_3\cdot a_3^2\cdot\varphi_2=0 \tag{6}$$

The result is a differential equation with the following form:

$$\begin{aligned}&-\frac{1}{12}\cdot m_1\cdot a_1\cdot a_2\cdot\cos\alpha_1\cdot\ddot{\varphi}_1-\frac{\beta_1\cdot a_1}{2}\left(a_2\cdot\cos\alpha_1+\frac{a_1}{2}\right)\cdot\varphi_1+\\&+\beta_3\cdot a_3^2\cdot\left[\frac{m_2}{m_3}\cdot\left(3\cdot\cos^2\alpha_1+2\right)-\frac{1}{2}\right]\cdot\varphi_2=0\end{aligned} \tag{7}$$

Take a general solution with the form forma $\varphi_1 = e^{u_1 \cdot t}$, $\varphi_2 = e^{u_2 \cdot t}$. The characteristic equation is $a \cdot x^2 + c = 0$, that is:

$$\left(-\frac{1}{12} \cdot m_1 \cdot a_1 \cdot a_2 \cdot \cos\alpha_1\right) \cdot U_1^2 - \frac{\beta_1 \cdot a_1}{2} \cdot \left(a_2 \cdot \cos\alpha_1 + \frac{a_2}{2}\right) = 0 .$$

Where the solutions are: $x_{1,2} = \pm\sqrt{-(c/a)}$, that is:

$$U_{1,2} = \pm\sqrt{-6 \cdot \beta_1 \cdot \left(a_2 \cdot \cos\alpha_1 + \frac{a_e}{2}\right) \Big/ \left(m_1 \cdot a_2 \cdot \cos\alpha_1\right)} \tag{8}$$

While the specific solution is:

$$U_3 = -\frac{12 \cdot \beta_3 \cdot a_3^2 \cdot \left[\frac{m_2}{m_3}\left(3 \cdot \cos\alpha_1^2 + 2\right) - \frac{1}{2}\right]}{m_1 \cdot a_1 \cdot a_2 \cdot \cos\alpha_1} \tag{9}$$

The blade swing law will take the form:

$$\begin{aligned}\varphi_1 = {} & A \cdot \cos\sqrt{-\frac{6 \cdot \beta_1\left(a_2 \cdot \cos\alpha_1 + \frac{a_2}{2}\right)}{m_2 \cdot a_2 \cos\alpha_1}} \cdot t - B \cdot \sin\sqrt{-\frac{6 \cdot \beta_1\left(a_2 \cdot \cos\alpha_1 + \frac{a_2}{2}\right)}{m_2 \cdot a_2 \cdot \cos\alpha_1}} + \\ & + C\cos\left[-\frac{12 \cdot \beta_3 \cdot a_3^2 \cdot \left[\frac{m_2}{m_3}\left(3 \cdot \cos\alpha_1^2 + 2\right)\right] - \frac{1}{2}}{m_1 \cdot a_1 \cdot a_2 \cdot \cos\alpha_1}\right] \cdot t^2 + D\end{aligned} ; \tag{10}$$

$$t = 0; \varphi_1 = \frac{2 \cdot F}{\beta_1 \cdot a_1} - \frac{m_1}{\beta_1} \cdot \ddot{\varphi}_1; \ \varphi_1 = \frac{2 \cdot F}{a_1 \cdot \beta_1} + \frac{12 \cdot m_1}{m_2} \cdot \frac{\beta_3^2}{\beta_1^2} \cdot \frac{a_3^2}{a_1^2} \cdot \varphi_2; \varphi_2 = \frac{2 \cdot K_3}{\beta_3 \cdot a_3^2}; \ddot{\varphi}_2 = \frac{6 \cdot \beta_3}{m_3} \cdot \varphi_2 \tag{11}$$

Frontal Impact Stress of the Vehicle with an Obstacle

Let us present the initial requirements for the bulldozer blade at impact with an obstacle having the mass *m*.

Problem enunciation [1]: Calculate the common movements of a mass *m* (for instance a heavy boulder) with the bulldozer blade represented by an elastic mass system M_1. In this case M_1 stands for the total undercarriage plus equipment mass.

Following the impact, the two masses remain in contact and move with a V_1 speed.

$$V_1 = m \cdot V_0 \Big/ \left(m + M_1\right) \tag{12}$$

where V_0 is the speed of the obstacle mass; V_1 is the vehicle transport speed.

After impact, the system has a single degree of non-mass freedom being hit by a common mass $m + M_1$. When set into motion, the elastic system has a V_1 speed.

The rhythmic frequency is:

$$p = \sqrt{k \Big/ \left(m + M_1\right)} \tag{13}$$

The maximum displacement becomes:

$$\delta_{\max} = \delta_{st} \cdot \sqrt{\frac{m \cdot V_0^2}{g \cdot \delta_{st} \cdot (m + M_1)}}, \tag{14}$$

where: $\delta_{st} = G / k$; G – is the obstacle weight, kN; k – is the elastic constant of the system, which can be assessed according to the deformation of the central blade segment undergoing a cylindrical bending due to a uniform load. The maximum deformation at the centre of the blade is W_{max}.

The shock time is:

$$T = \sqrt{\delta_{st} \cdot (m + M_1) / (g \cdot m)} . \tag{15}$$

Here it is considered that the whole bulldozer mass is reduced for the elastic blade plate with the respective deformation W_{max}.

If it is accepted that the deformation of the two masses takes a very short time, being in fact instantaneous, thus justifying the cylindrical deformation being visible, at first, only on the plate mounted on the two long sides of the blade and free at the high ends, the result is the post-shock speed function of the speed before shock.

For the obstacle:

$$V'' = V' \cdot \frac{m - \eta \cdot M_1}{m + M_1} + V' \cdot M_1 \cdot \frac{1+\eta}{m + M_1} \tag{16}$$

For the bulldozer:

$$W_1'' = V' \cdot \frac{m \cdot (1+\eta)}{m + M_1} + V_1' \cdot \frac{M_1 - \eta \cdot m}{m + M_1} , \tag{17}$$

where η is a coefficient of restitution, $\eta = 0.2 \div 0.5$,

For the case under study: the total mass of the vehicle M_1 is initially moving and the mass which produces the shock has a $V' = V_0$ speed. Following the shock, mass m has the following speed:

$$V'' = V_0 \cdot (m - \eta \cdot M_1) / (m + M_1), \tag{18}$$

while the speed of the mass M_1 is:

$$V' = V_0 \cdot m \cdot (1+\eta) / (m + M_1). \tag{19}$$

If $V'' > V'$, a correct situation from a physical point of view because the boulder mass is smaller than the bulldozer mass, the boulder mass m moves with a constant speed according to $\delta = V' \cdot t$, while the bulldozer mass M_1 starts oscillating:

$$\delta_1' = (V'' / p_1) \cdot \sin p_1 \cdot t; \quad p_1 = \sqrt{k / M_1} . \tag{20}$$

In practice, this is the basis for an assessment of the value of maximum displacement during shock (bulldozer and boulder in the first phase).

For the first phase shock where $t = 0$, the blade displacement is:

$$y_1 = 0,5 \cdot \varphi_1 \cdot a_1 = \delta_{\max} = \delta_{st} \cdot \sqrt{m \cdot V_0^2 / [g \cdot \delta_{st} (m + M_1)]} . \tag{21}$$

For the second phase, the blade mass starts oscillating according to:

$$\delta_1' = \left(V_1''/p_1\right)\cdot \sin\, p_1\cdot t,$$

which means that the initial conditions continue to change:

$$y_1 = 0{,}5\cdot\varphi_1\cdot a_1;\quad \delta_1' = \left(V_1''/p_1\right)\cdot \sin\, p_1\cdot t;\; p_1 = \sqrt{k/M_1} \tag{22}$$

In the second phase this oscillation is completely absorbed by the elastic (Sigmadozer type) blade system as in Figure 5,b.

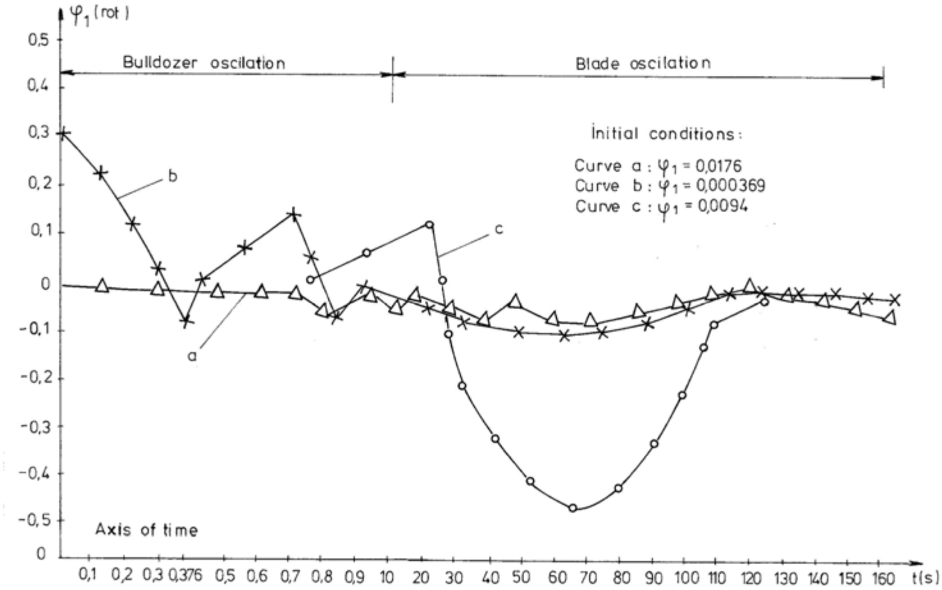

Fig. 5. a - Pulsation at the blade end generated by the oscillation movement of the bulldozer after impact (in sin and cos); b- idem with the blade rotation (in cos); c- the blade rotation to the action of a traction force in absence of a contact with the obstacle.

In this case k is the elastic constant generated by the deformation of the front plate of the blade, for the case of a cylindrical U-type blade.

We now have the initial conditions resulting from the structure of the elastic system (see 11), that is:

$$\varphi_1 = \frac{2\cdot F}{a_1\cdot\beta_1} + \frac{24\cdot m_1}{m_2}\cdot\frac{K_3\cdot\beta_3}{\beta_1^2\cdot a_1^2};\qquad \ddot{\varphi}_1 = \frac{2\cdot F}{a_1^2\cdot m_1} - \frac{\beta_1}{m_1}\cdot\varphi_1; \tag{23}$$

$$\varphi_2 = \frac{2\cdot K_3}{\beta_3\cdot a_3^2};\quad \ddot{\varphi}_2 = \frac{12\cdot K_3}{m_3\cdot a_3^2}\;. \tag{24}$$

From (23) we can assess the initial conditions for $t = 0$. The potential φ_1 swing depends on the vehicle traction force F_t and on the central blade hardness, at the point where the spring with a hardness coefficient β_1 is placed at the middle of m_1 mass, while the hardness coefficient β_3 operates at the end of mass m_3.

The β_1 hardness coefficient is calculated according to the maximum blade deformation W_{max}, while

$$\beta_3 = 2\cdot K_3\Big/\left(\varphi_2\cdot a_3^2\right). \tag{25}$$

Dynamic Stress Values for Different Vehicle Parts, Based on the Variation of the Digging Drag and on the Unevenness of the Ground

Paper [2] analyzes the dynamic stress underwent by the vehicle parts, based on the blade digging variation as well as on potential ground unevenness. As for the working equipment, the dynamic stress applies when an obstacle appears in front of the blade. The model takes into account the traction power of the undercarriage, as shown in the bulldozer traction diagram (D155AX-6) for an adaptability coefficient of the Diesel engine *Ka* = 1.2; the nominal speed and the speed for a maximum couple while the bulldozer moves in phase I (see fig. 2) are the following: V_0 = 1 km/h; and V = 5.5 km/h. The apparent mass of the vehicle, taking the form $m^x = \delta \cdot m$, is computed with the help of the influence coefficient, during the rotation movement $\delta = 1{,}4$ (this coefficient takes into account the moving engine parts, the transmission and undercarriage components); the overall resistance to movement $F_f = f \cdot G_a$ (where the overall coefficient of resistance to movement is f = 0.08), the resistance to platform climbing is $F_p = m \cdot g \cdot \sin\theta$ (maximum platform angle $\theta = 20^0$) and the digging distance is S = 40 m.

Examples of computation for D155AX-6 bulldozer with an engine power of P_m = 268 kW for an engine rotation cycle of 1900 rot/min, the overall operation mass of the vehicle M_1 = 39500 kg. The calculated values of space, speed and acceleration (in this case) are:

$$X = 0{,}01\ m;\ \dot{X} = 160{,}362\ m/s;\ \ddot{X} = -24{,}72\ m/s^2\ .$$

Modeling Results for the Bulldozer Blade Equipment

The elastic behaviour of the bulldozer blade structure was assessed for the following specific working conditions:

a) Blade impact with an obstacle. The deformation of the elastic system for a traction force $F_t / 2 = 250\ kN$, for half the structure (for one undercarriage), is taken into account (as the model is symmetrical);

b) The singular bulldozer oscillation after impact is continued with the oscillation of the blade after separating the blade from the respective obstacle:

c) Following its own oscillation, the bulldozer continues to operate with a traction force F_t = 500 kN. Several 4th order oscillations are also transmitted from the rolling system to the Sigmadozer blade frame;

d) The rotation law φ_1 of the straight blade must be determined. Its rigidity is computed according to the W_{max} distortion of the central blade plate, imagined as a plate which is inserted on one side and just propped on the other three sides. In this case the initial condition is $\varphi_1(0) = 0$.

e) The rotation law φ_1 of the blade must be determined in the case of a bulldozer operating in a dynamic regime, at maximum platform angle, using the calculation model described in the previous paragraph.

From here on the resulting values for the movement, speed and acceleration of the vehicle (x, $\dot{x}$ and $\ddot{x}$) are considered as the initial conditions x(0), $\dot{x}(0)$ and $\ddot{x}(0)$ for the rotation law applied to the elastic structure of Sigmadozer blades.

Case A. Special type Sigmadozer blade (fig. 4): In the blade rotation law φ_1 shown at (10) we introduce the formula of bulldozer oscillation after the shock:

$$\delta_1^{'} = \left(V_1^{"}/p_1\right)\cdot \sin p_1\cdot t;\quad p_1 = \sqrt{k/M_1}\ .$$

The chosen restitution coefficient has the value $\eta = 0,2 \div 0,5$. We have the values:

$$\begin{aligned} \varphi_1 &= \frac{2\cdot V_1^{"}}{a_1\cdot p_1}\cdot \sin p_1\cdot t;\ \dot{\varphi}_1 = \frac{2\cdot V_1^{"}}{a_1}\cdot \cos p_1\cdot t;\\ \ddot{\varphi}_1 &= -\frac{2\cdot V_1^{"}}{a_1}\cdot p_1\cdot \sin p_1\cdot t;\ \dddot{\varphi}_1 = -\frac{2\cdot V_1^{"}}{a_1}\cdot p_1^2\cdot \cos p_1\cdot t. \end{aligned} \tag{26}$$

We include (26) in equation (7) derived once. We use $\dot{v} = k\cdot e^{k\cdot t}$ for $\dot{\varphi}_2$.

If t =0, the pulsation at the end of the V-shaped blade, determined by the oscillating movement of the bulldozer is:

$$\bar{K} = -\frac{V_1^{"}\cdot\left[a_2\cdot\left(m_1\cdot p_1^2/3+\beta_1\right)\cdot\cos\alpha_1 - a_1\cdot\beta_1\right]}{\beta_3\cdot a_3^2\cdot\left[2\cdot\left(m_2/m_3\right)\cdot\left(3\cdot\cos^2\alpha_1+2\right)-1\right]} \tag{27}$$

This pulsation which occurs at the blade end is generated by the oscillating movement of the bulldozer after its impact with the obstacle. It will influence the blade rotation law φ_1 because it depends on speed $V_1^{"}$, on the square value of the bulldozer pulsation p_1^2 determined by the impact of the bulldozer with the obstacle, by the masses of blade components m_1, m_2, m_3, by distances a_1, a_2, a_3 as well as by the blade stiffness constants β_1, β_3.

The blade rotation law φ_1 will take the form:

$$\begin{aligned} \varphi_1(t) &= A\cdot\cos\sqrt{-\frac{3\cdot\beta_1\left(2\cdot a_2\cos\alpha_1+a_2\right)}{m_2\cdot a_2\cdot\cos\alpha_1}}\cdot t - B\sin\sqrt{-\frac{3\cdot\beta_1\left(2\cdot a_2\cdot\cos\alpha_1+a_2\right)}{m_2\cdot a_2\cdot\cos\alpha_1}}t+\\ &+C\cdot\cos\left\{-\frac{V_1^{"}\left[a_2\cdot\left(m_1\cdot p_1^2/3+2\cdot\beta_1\right)\cdot\cos\alpha_1-a_1\cdot\beta_1\right]}{\beta_3 a_3^2\left[2\cdot\left(m_2/m_3\right)\cdot\left(3\cos^2\alpha_1+2\right)-1\right]}\right\}\cdot t^2+D \end{aligned} \tag{28}$$

The blade oscillation shall continue after t = 0,376 s, that is, when the bulldozer oscillation caused by the plastic impact with the obstacle ceases.

Example: The component masses of the Sigmadozer half blade taken into consideration are : m_1 = 2558.55 kg, $m_2 = m_3$ = 615.36 kg; distances : a_1 = 2m; $a_2 = a_3$ =0,8 m; $V_1^{"} 0,0553 m/s$; p_1 = 2,9162 1/s; blade stiffness constants $\beta_1 = 335,997$ kN/m^2 and $\beta_3 = 58196,748$ kN/m^2.

Initial conditions: $\varphi_1 = 0,0176; \ddot{\varphi}_1(0) = 195,423; \varphi_2(0) = 0,0007756$. The graphic of blade rotation law φ_1 is shown in Fig. 5,a for the solution that uses sin and cos integration constants.

If the blade rotation law φ_1 includes only the components of cos integration constants, then the integration constants *A*, *C*, *D* have identical values, while *B* = 0 (see the graphic in fig. 5,b).

Case B: If an undercarriage traction force F_t = 500 kN (from fig.2 of the traction diagram) is applied on the oscillating bulldozer after the shock, it will be transmitted through the stiffness spring β_4 to the blade model. The initial conditions were computed according to the internal parameters of the elastic system of the blade (see the curve in fig. 5,c). The force transmitted from the undercarriage:

$$F_t = N_4 = \left(4 \cdot 43750\right) / \left(0{,}7\right) = 250\,kN .$$

Here the assessment is made taking into account the stiffness resulting from W_{max} for the blade submitted to a cylindrical deformation (bending) under the action of a uniformly distributed force. It is perfectly identifiable with the traction force for one undercarriage (see the traction diagram in fig. 2). The initial conditions are:

$$\varphi_1\left(0\right) = 0;\ \ddot{\varphi}_1\left(0\right) = 194{,}2;\ \varphi_3\left(0\right) = \frac{6 \cdot F_t \cdot a_2 \beta_3}{m_3^2 \cdot a_3^2} = 69832402 \frac{1}{s^4};\ \varphi_2 = 0{,}0007756 \qquad (29)$$

The blade rotation law φ_1 submitted to the action of a traction force F_t = 250 kN in absence of a contact with the obstacle is shown in Figure 5,c.

Case C: When the blade comes into contact with the obstacle the initial conditions calculated according to the blade system equations are the following: $\varphi_1\left(0\right) = 59{,}0584; \ddot{\varphi}\left(0\right) = -7558{,}47; \varphi_2\left(0\right) = 0{,}003$.

The blade rotation law φ_1 (fig. 6,a) shows a significant deformation determined by the load applied on the central Sigmadozer blade structure, if we apply an undercarriage traction force F_t = 500 kN for a speed of V = 1 km/h. The 100 s calculation time corresponds to the required interval admissible for the K bogie components of the undercarriage rolling surface.

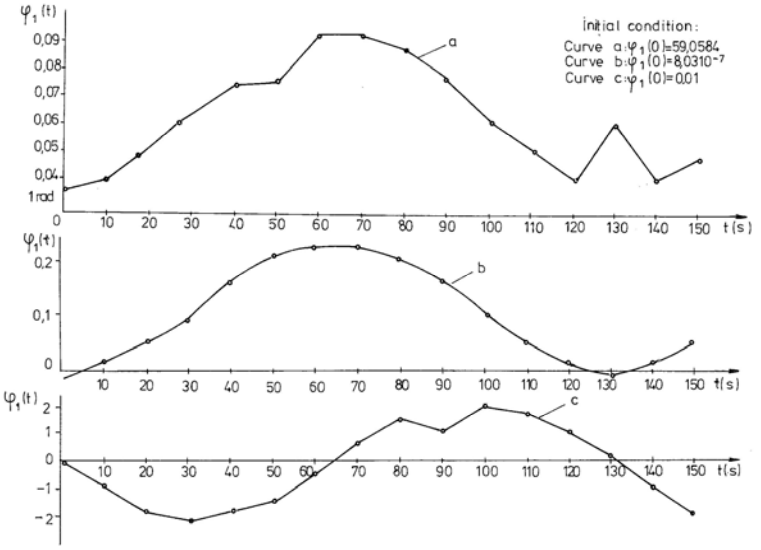

Fig.6. a - Deformation determinated by the load applied on a central Sigmadozer blade structure; b- totally rigid straight blade; c- the blade rotation low under dynamic operating conditions.

Case D: The use of a totally rigid straight blade. If the central segment of the blade is a plate whose long side is built-in and the other sides are only propped, and it holds a uniformly distributed load, then the maximum curvature at the centre of the plate is:

$$W_{max} = -\alpha \cdot p \cdot b^4 / \left(E \cdot h^3\right). \qquad (30)$$

For p = 0.135 N/mm^2, h = 20 mm, b = 1850 mm and a = 2000 mm, (W_{max} = 8.0276 · 10^{-10} mm) the result is: Stiffness constant $\beta_1 = F / W_{max} = 623000 \cdot 10^6\ N/m$. If we use $\varphi_1 = f\left(\beta_1, \beta_3\right)$, the result is: $\varphi_1\left(0\right) = 8{,}03 \cdot 10^{-7} \approx 0;\ \varphi_2\left(0\right) = C = 0{,}0007756$. The rotation law will take the form shown in Figure 6,b. From the condition implied in the blade rotation:

$$\begin{aligned} \varphi_1 &= 16 \cdot K_3 / \left(a_1 \cdot a_3 \cdot \beta_1\right) + 24 \cdot m_1 \cdot K_3 \cdot \beta_3 / \left(m_2 \cdot \beta_1^2 \cdot a_1^2\right) = 0; \\ & 8 \cdot K_3 \cdot \left[2/a_3 + 3 \cdot m_1 \cdot \beta_2 / \left(m_2 \cdot \beta_1 \cdot a_1\right)\right] / \left(\beta_1 \cdot a_1\right) = 0 \end{aligned} \qquad (31)$$

We have: $16 \cdot m_2 \cdot a_1 \cdot \beta_1 = -24 \cdot m_1 \cdot a_3 \cdot \beta_3$, that is: $\beta_1 = -\ 3 \cdot m_1 \cdot a_3 \cdot \beta_3 / \left(2 \cdot m_2 \cdot a_1 \right)$.
By replacing the given values, the potential value for β_1 is:

$$|\beta_1| = 145182{,}61 > \beta_3 = 58196{,}748\ [N/m] \qquad (32)$$

Case E: For the blade rotation law of the form (10) in the mentioned dynamic working conditions of the vehicle, the initial requirements are: $\varphi_1(0) = 0{,}01; \dot{\varphi}_1(0) = 160{,}386; \ddot{\varphi}_1(0) = -24{,}72$. They are used to calculate the values of the blade integration constants, if we also admit the condition $\varphi_2(0) = 0{,}0007756$ for the V-shaped blade end part. The blade rotation law under dynamic bulldozer operating conditions – an acceleration regime for climbing a platform in the first velocity stage (according to traction diagram in fig. 2) is shown in Figure 6,c.

General Conclusions

In the event of an impact with an obstacle, the central segment of the blade works together with the entire bulldozer mass, then free oscillations occur due to the shock; finally, the earth load is spilled and cut by the V-shaped structure at the ends of the blade. This can be noted from the shape of the graphics showing the blade rotation φ_1 in Figures 5,c and 6,a,b and c respectively.

The rotation law φ_1 for an m_1 mass is a significant deformation of the central blade structure, if in static conditions a traction force of F_t = 500 kN is applied on the undercarriage; however, this is better shown in a dynamic vehicle operation. The accepted time needed to calculate the rotation $\varphi_1(t)$ was 100 s, because it corresponds to the stress interval computed for the elastic model of the K bogie undercarriage.

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Modele de calcul pentru structura nouă a echipamentului cu lamă, la buldozerele pe şenile

Rezumat

Pentru construcţia echipamentului cu lamă de buldozer de o nouă generaţie, care asigură o productivitate mărită, se propune construcţia unor modele de calcul care se referă la: şoc cu un obstacol, deformare elasto-plastică sau regimul dinamic de lucru al buldozerului la urcare rampă în regim de accelerare, pentru executarea săpării pământului pe o distanţă dată. Pentru modelele de calcul propuse se dau legile de rotire ale noii constructii de lamă cu performanţe superioare de lucru.