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DYNAMICAL AND MATHEMATICAL MODELLING OF THE WORKING PROCESS OF THE PRESSING MECHANISM OF THE HAY AND STRAW BALER MACHINE

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Abstract: *In this paper the working resistance of the pressing mechanism is presented by means of specific relations, allowing their introduction as disturbing factors within the mathematical models of the system tractor functioning – namely the hay and straw baler. After the completion of the dynamical model of this system, the next stage was to write a mathematical model by means of which the dynamic solicitation level in the elements of the pressing mechanism and in the components that ensure motion transmission could be assessed. Solving the mathematical model in multiple versions has been achieved with the Runge-Kutta method of integration. Experiments have validated both the proposed mathematical model and the method to solve it.*

Key words: *baler for hay and straw, dynamical and mathematical modelling.*

1. Introduction

Accurate knowledge of the nature and extent of stresses to which the straw and hay baler's pressing mechanism is subjected is of particular importance for choosing the manner of design and for the correct selection of operating parameters for the baler. Experimental investigation is necessary to confirm that the requirements imposed by the design theme as well as to bring credibility to the results provided that the most rigorous methods and precise devices are used. In comparison to theoretical research, it still has a big disadvantage regarding preparation time and the difficulty of the reproduction test for the full range of possible working conditions.

Hay and straw pressing machines perform this operation using a crank type mechanism driven by the PTO shaft of the tractor by means of its own agricultural machine transmission components. At a constant angular velocity of the crank, the piston moves into a particular channel, thus during the active stroke the material is pressed and shaped into bales and at the passive stroke, the piston retracts to allow refilling with material. It follows that the

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movements and stresses within the mechanism and the motion transmission components are cyclical, reaching maximum values during active strokes.

The research of the evolution of these values in time can be made in writing, by means of appropriate dynamic models for tractor-agricultural machines, namely using systems of differential equations - mathematical models - solvable by the use of appropriate programming methods.

2. Material and Methods

Because the behaviour of the pressing mechanism within a mobile or stationary agricultural system can be considered the same, this research has opted, for the sake of simplicity, for a tractor-baler system working in a steady-state condition [1]. A sufficiently detailed dynamic model for this purpose is shown in Figure l.

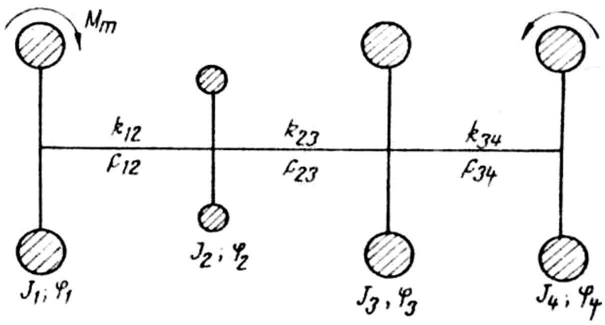

Fig. 1. *The dynamic model of the transmission mechanism between tractor and baler*

Concentrated masses, having moments of inertia $J_1...J_4$ are equivalent to the mobile masses of the following: motor tractor (J_1) gear in the transmission PTO (J_2), flywheel uniform movement of the press (J_3) and mechanism of pressing (J_4). The real connections between these masses are equated, as well as the elastic constants k_{ij} and amortization c_{ij}, through conventional shafts. The angular movements of the concentrated masses and the generalized coordinates of the movement were noted φ, which was individualized with the mass index.

The sizes of the dynamic model, except for the amortization constants, which were established by analogy with the data in the literature, were determined analytically or in a graphic - analytical or in an experimental manner [3], being synthetically rendered in Table 1 below, with low values of the tractor engine crankshaft (the data correspond to the system consisting of the U-650 tractor and the PPF straw and hay baler).

Specific sizes of the dynamic model of Figure 1 Table 1

Size	**The low value,** *kg·m²*	**Size**	**The low value,** *N·m*	**Size**	**The low value,** *N·m·s*
J_1	2.06	k_{12}	1065	c_{12}	1
J_2	0.005	k_{23}	800	c_{23}	1
J_3	0.0885	k_{34}	1010	c_{34}	1
J_4	0.0092				

The disturbing factors of the model are the torque developed by the engine of the tractor (M_m) and the resistant torque (M_p) applied to the rod-crank mechanism of the press.

The mathematical expression of the couple M_m can be achieved through two straight sections of the torque curve of the external characteristics, namely the strand of regulator and the portion between the nominal and the maximum torque moment. With the characteristics of the D-103 engine, fitted on the U-650 tractor there follows [3]:

$$M_m = \begin{cases} 2794 - 13.7 \cdot \dot{\varphi}_1 & for\ 185.5 \le \dot{\varphi}_1 \le 204 \\ 445 - 1.03 \cdot \dot{\varphi}_1 & for\ 145.5 \le \dot{\varphi}_1 \le 185.5 \end{cases} \quad (1)$$

To establish a relationship in order to express the disturbing factor of the timing crank mechanism resistant to pressure, it must be concluded that this is mainly due to inertial forces of reciprocating masses and to the pressing force [4]. Calculation of the resistance moment Mp can be done with the relationship [1], [2]:

$$M_p = M_j + M_{pp}, \quad (2)$$

where M_j is the moment of inertia forces; M_{pp}-moment of resistance during actual pressing.

The m_j masses in the reciprocating rectilinear motion of the mechanism are the m_p-mass of the piston (considered as focusing on the bolt axis) and a part of the rod mass. It is recommended [5] that for practical calculations, the mass m_b of the rod to replace the two concentrated masses: m_{b1}, of the bolt axis, and m_{b2} of the crankpin axis.

Thus:

$$m_b = m_{b1} + m_{b2} \quad (3)$$

and

$$m_j = m_p + m_{b1}. \quad (4)$$

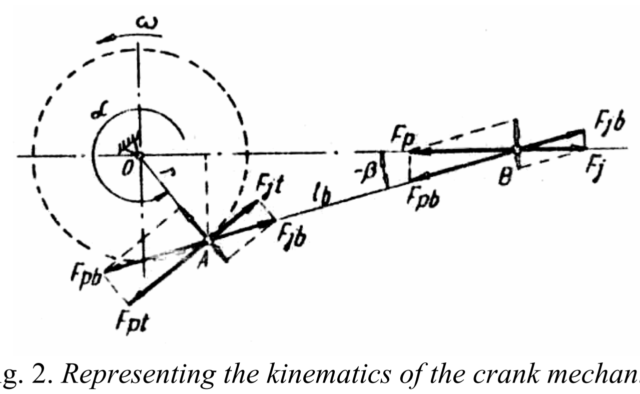

Fig. 2. *Representing the kinematics of the crank mechanism*

Corresponding to the representation in Figure 2, the inertia force related to the mass m_j can be established by means of the following relationship:

$$F_j = m_j \cdot r \cdot \omega^2 (\cos\alpha + \lambda \cdot \cos 2\alpha) \quad (5)$$

where:

r is the radius of the crank; ω-the angular velocity of the crank; λ-the ratio between the radius of the crank and the connecting rod length l_b [3]; α-current angle position.

This force manifests itself through the M_j moment acting on the crank mechanism and is calculated by means of the formula:

$$M_j = F_j \cdot t \cdot r = m_j \cdot r^2 \cdot \omega^2 (\cos\alpha + \lambda \cdot \cos 2\alpha)\frac{\sin(\alpha+\beta)}{\cos\beta} \quad (6)$$

$$M_j = m_j \cdot r^2 \cdot \omega^2 (\cos\alpha + \lambda \cdot \cos 2\alpha)(\sin\alpha + \frac{\lambda}{2}\sin 2\alpha) \quad (7)$$

Considering the measurement of the angle of rotation of the crank as shown in Figure 2, the pressing of the material in the PPF press, takes place only within the range $3\pi/2 + 2\pi n \le \alpha \le 2\pi(n+1)$ of the α angle values, where n = 0, 1, 2, 3, ...

Based on this observation and knowing that during pressing, the force F_p exerted by the piston on the material increases from 0 to the maximum F_{pmax} (reached at the end of the piston), the following approximation can be made:

$$F_p = F_{p\max} \cdot \cos\alpha . \quad (8)$$

The maximum pressing force can be calculated using the expression:

$$F_p = F_{p\,max} \cdot S_p \quad (9)$$

where p_{max} is the maximum pressure applied to the piston; S_p - the surface of the piston head.

On pressing a material with a humidity of 20-25% and a bulk density of the bales of γ = 1600-2000 N/m^3, we indicate [3] for p_{max}:

$$p_{max} = \gamma^{2.32}\ [N/m^2]. \quad (10)$$

The indication of the *α* angle variation interval that occurs during the pressing of the material, can be made by designating it by the symbols of its two trigonometric functions: *tg α* and *cos α*.

Based on Figure 2, given the relationship (7) and the above considerations, we can write:

$$M_{pp} = \begin{cases} -r \cdot F_{p\max}(\sin\alpha + \frac{\lambda}{2} \cdot \sin 2\alpha)\cos\alpha, \ \textit{for } tg\alpha < 0 \textit{ si } \cos\alpha > 0 , \\ 0, \text{ for the rest of the } \alpha \text{ angle ranges}. \end{cases} \quad (11)$$

Given the building characteristics of the PPF-baler and considering an average bale density, the M_p low resistance moment components,the relations (7) and (11) become:

$$M_j = 0.0004 \cdot \dot{\varphi}_4^2 (\cos\frac{\varphi_4}{29.65} + 0.334 \cdot \cos\frac{\varphi_4}{14.82})(\sin\frac{\varphi_4}{29.65} + 0.167 \cdot \sin\frac{\varphi_4}{14.82}), \quad (12)$$

$$M_{pp} = \begin{cases} -13(\sin\frac{\varphi_4}{29.65} + 0.167 \cdot \sin\frac{\varphi_4}{14.82})\cos\frac{\varphi_4}{14.82}, \ \textit{for } tg\frac{\varphi_4}{29.65} < 0 \textit{ si } \cos\frac{\varphi_4}{29.65} > 0; \\ 0, \text{ for the rest of the } \alpha \text{ angle ranges}. \end{cases} \quad (13)$$

The equations, written on the basis of the representation in Figure 1, which illustrate the influence of the pressing mechanism, are the following:

$$\begin{cases} J_1\ddot{\varphi}_1 + k_{12}(\varphi_1 - \varphi_2) + p_{12}(\dot{\varphi}_1 - \dot{\varphi}_2) = M_m, \\ J_2\ddot{\varphi}_2 - k_{12}(\varphi_1 - \varphi_2) - p_{12}(\dot{\varphi}_1 - \dot{\varphi}_2) + k_{23}(\varphi_2 - \varphi_3) + p_{23}(\dot{\varphi}_2 - \dot{\varphi}_3) = 0, \\ J_3\ddot{\varphi}_3 - k_{23}(\varphi_2 - \varphi_3) - p_{23}(\dot{\varphi}_2 - \dot{\varphi}_3) + k_{34}(\varphi_3 - \varphi_4) + p_{34}(\dot{\varphi}_3 - \dot{\varphi}_4) = 0, \\ J_4\ddot{\varphi}_4 - k_{34}(\varphi_3 - \varphi_4) - p_{34}(\dot{\varphi}_3 - \dot{\varphi}_4) = -M_p. \end{cases} \tag{14}$$

In the mathematical model (14) the perturbing factors M_m and M_p are expressed by (1) or (11) and (13).

3. Results and Discussion

The devised mathematical model allows, on the one hand, the research of the influence of operating conditions of the mechanism of compression and its dimensional characteristics on the stresses applied to the tractor gear and to the press (through elastic torsion moments, $k_{i,i+1}(\varphi_i - \varphi_{i+1})$ and, on the other hand, the determination of the stresses on the elements of the mechanism-by calculating the values of the M_p moment in time.

The solving of the system (14), in multiple variations, was achieved using the Runge-Kutta method of integration [1], [3].

In order to exemplify and to highlight the accuracy of the modelling and the admissibility of the simplifying assumptions, a comparison of the torque shaft PTO-$M_{23} = k_{23}(\varphi_2 - \varphi_3)$ while performing a pressing session, determined by calculation and experimentally by means of strain methods, is presented in Figure 3.

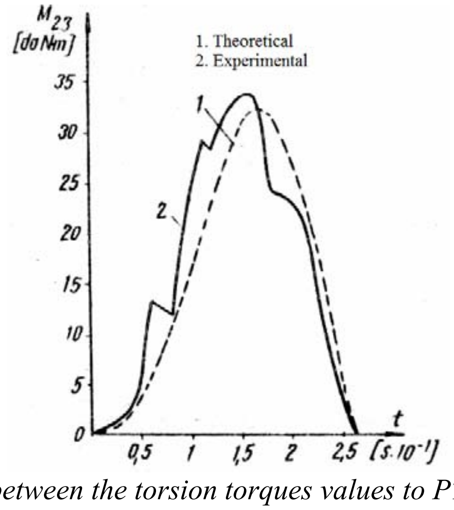

Fig. 3. *Comparison between the torsion torques values to PTO shaft determined by calculation and experimental methods*

There is an obvious qualitative and quantitative similarity between the calculated curve (1) and experimental curve (2). Slight differences in modeling come from neglecting some aspects such as the influence of clearance in splined shafts, in gears, etc.

4. Conclusions

1. The theoretical determination of the behaviour of baler pressing mechanisms as well as of the stress that this mechanism exerts on the transmission of the tractor and of the agricultural machine, can be

made on the basis of mathematical models consisting of systems of differential equations, which require an accurate expression of the disturbances caused by the working resistances of the mechanism.

2. The modeling of this resistance, by taking into account the effect of inertial forces and of the pressing force, with added simplifications, proves to be sufficient in order to obtain results confirmed by experience, and, at the same time, displaying a broad applicability in the sense that it can be used on other types of existing presses or presses that are being designed.

3. The different variants of solving the devised mathematical model, leads to the conclusion that the poignang role in determining the level of stress exerted upon the elements of the pressing mechanism and of the movement transmission components, pertains to pressing force of the material. The specificity of the operation of the pressing mechanism determines this force to manifest itself periodically, at precise time intervals; this is also valid for the stresses within the components that ensure the achievement of useful work with intermittent peaks, whose value depends on the density of pressing, being separated by intervals of relatively reduced values.

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