DYNAMICS OF THE MOVEMENT OF THE CUTTING ASSEMBLY'S CUTTER BAR

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Abstract
This article presents the dynamics of the cutting assembly's cutter bar movement. The derived mathematical dependencies describing the dynamics' loading of the cutter bar may be used at the stage of designing that type of cutting assemblies and their power transmission systems. The simulation calculations conducted on the derived mathematical dependencies, made it possible for the authors to develop a computational tool for quick identification of demand for power and forces operating in the shear cutting assembly under the influence of changes in the characteristic input values.

Key words: *shear-finger cutting assembly, dynamics of movement, cutter bar, simulation calculations.*

INTRODUCTION

The shear-finger cutting assembly is the basic working assembly occurring in many agrarian machines. It is used in machines like mowing machines, chaff cutters and combine harvesters, the purpose of which is cutting of plant material for the fodder, consumption and energy purposes (Gach, Kuczewski & Waszkiewicz, 1991; Bochat, 2010; Bochat & Zastempowski, 2013)
It results from the generally available literature's analysis, that the subject matter of kinematics and dynamics of cutting units‘ of the machines‘ working assemblies was raised only by Bochat and Zastempowski (2013). Other authors, within the frames of machines' construction, have mainly raised the subject area relating to the rules of designing and the analysis of a construction's resistance (Strzelecki et al., 2016; Tomaszewski et al., 2014), with the rules of use of MES and numerical analysis (Szala, 2014; Knopik et al., 2016) with mathematical modelling and the construction's optimization (Knopik et al., 2016; Knopik & Migawa, 2017; Ligaj & Szala, 2010; Zastempowski and Bochat, 2014, 2015; Keska & Gierz, 2011) as well as the influence of technical devices on the environment (Karwowska et al., 2013, 2014) as well as processing of the harvested plant material (Dulcet et al., 2006).
Fig. 1 presents the shear-finger cutting assembly's construction.
The essence of its construction lies in that it is made of a movable cutter bar making a reciprocating movement and an immovable finger bar. Knives riveted to the cutter bar are of a trapezoid shape. Fingers fastened to the finger bar are used to divide the harvested material into batches.
The principles of the shear-finger cutting assembly's operation constists in that the fingers enter between the harvested plants and divide them into batches. Then, individual knives squash the stalks or plants‘ stems to the side fingers‘ edges and make the plants‘ cut. The shear-finger cutting assemblies are split into: the ones of standard cut with single knives‘ stroke (classical), the ones of standard cut with double knives‘ stroke, of medium cut and of low cut (Bochat, 2010).

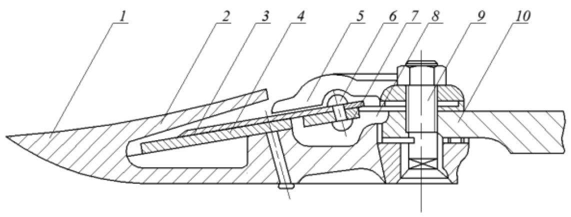

Fig. 1 Shear-finger cutting assembly (Bochat, 2010): 1 – finger, 2 – finger's blade, 3 – knife, 4 – liner, 5 – button, 6 – rivet, 7 – movable cutter bar, 8 – guide, 9 – screw, 10 – immovable finger bar

The purpose of the study is to analyse functioning of the shear-finger cutting assembly in the aspect of dynamic loads of its cutter bar. The knowledge of mathematical dependencies describing the cutter bar's dynamic load is necessary for its correctly designing of power transmission system.

MATERIALS AND METHODS

Conducting of an appropriate analysis of the shear cutting unit's operation in the aspects of dynamic loads of the cutter bar is embarassing, because of the system's complexity and imperfection of dynamic dependences describing it.
However, the experimental surveys of the cutting assembly are insufficient due to the fact, that they concern summary loads having an effect on individual construction elements and do not isolate unequivocally the reasons of their arising.
In literature it is most often assumed, that the force P_1 counteracting the cutter bar's movement (constituting the resistance of its movement) is equal to the sum of forces having an effect on it (fig.2) and is described with a formula:

$$P_1 = P_s + P_B + T_c, \tag{1}$$

where P_S is the average value of cutting resisting forces, P_B is the inertial force of the cutter bar and T_C is the friction force of the cutter bar against guide elements.

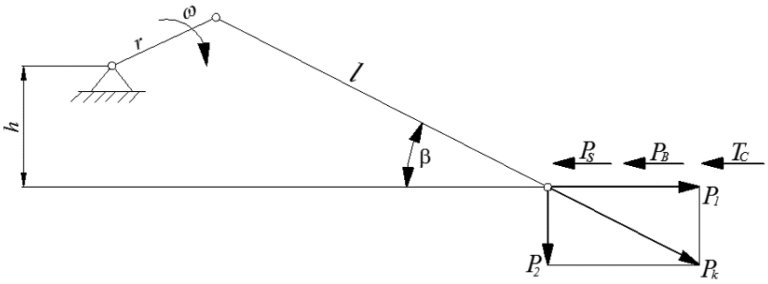

Fig. 2 Forces having an effect on the cutter bar powered with the asymetrical crank mechanism (own study): P_k – force having an effect along the connecting rod, P_1 and P_2 – vertical and horizontal constituent of force P_k, P_S – mean value of cutting resisting force, P_B – inertial force of the cutter bar, T_C – friction force of the cutter bar against guide elements, r – length of crank, l – length of connecting rod (pitman), h – distance of the rotating disc's shaft with the crank from the bar movement's plane, β - the angle of the connecting rod's inclination.

Cutting resistances are theoretically hard to determine, as they depend on many factors just like: species and variation of cut material, stiffness and humidity of individual blades, weediness, intensity of the cutting assembly's feeding, cutting speed, technical condition of the cutting assembly and other ones.
In practice the mean value of the cutting resisting forces is calculated from the dependence:

$$P_s = \frac{L_j \, F_0 \, i}{x_c}, \tag{2}$$

where L_j is the value of work necessary to cut the plants from the area of 1cm^2, F_0 is the fiels of the cutting assembly's loading, i is the number of knives of the cutter bar, and x_C is the knife's way from the beginning till the end of cutting.
The number of plants falling on 1cm^2 of cultivation, shall for the cereals assumed to be equal to z = 0,2 – 0,8 $\text{szt.}\cdot \text{cm}^{-2}$, while for forage grasses z = 1,2 – 2,0 $\text{szt.}\cdot \text{cm}^{-2}$.
It has been experimentally established, that if the value of work needed for cutting plants from 1 cm^2 amounts more or less for cereals, then L_j = 0,01 - 0,02 $\text{J} \cdot \text{cm}^{-2}$, and for forage grasses, L_j = 0,02 – 0,03 $\text{J} \cdot \text{cm}^{-2}$ (Bochat, 2010; Gach, Kuczewski & Waszkiewicz, 1991).

At the time of the analysis of operation of different types of cutting assemblies there stands out the so-called feeding field and load field. The field from which plants are cut during one stroke of knives is called the feeding field *F*, while the field from which plants are cut at the time of one knife's stroke with the use of one finger is called the load field F_0 (according to the formula (2)) as described by Bochat (2010).
For a normal cut assembly with a single stroke of knives we have: $F = F_0$.
Figure 3 presents the load field for a normal cutting assembly with a single stroke of knives.

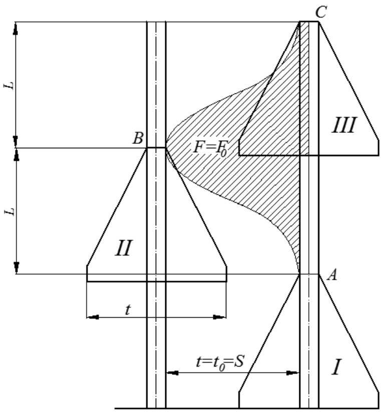

Fig. 3 The load field for a normal cutting assembly with a single stroke of knives (Bochat, 2010)

In the normal cutting assembly with a single stroke of knives, the top of a given knife's cutting edge at the time of a single revolution of a rotating disc with a crank circles the ABC (movement trajectory). The filed limited with that curve and the line *AC* is the feeding field *F* equal to, as earlier mentioned, the load field F_0 . So, taking into consideration the fact, that any point of a knife's edge makes a complex movement (according to the mechanisms' theory) it covers the relative motion and „transportation". In this case, the relative motion is described by the formula of the cutter bar's dislocation (3):

$$x = r(1 - cos\,\omega t). \tag{3}$$

Dependency (3) is received on the basis of the cutter bar's kinematics of movement analysis.
The formula (3) constitutes an equation of a harmonic movement describing translocation of projection of crankpin on the line of the cutter bar's movement. The „transportation" is described with the equation:

$$y = L\frac{\omega t}{\pi} = L\frac{\varphi}{\pi}. \tag{4}$$

The load field F_0 may be expressed with the formula:

$$F_0 = \int_0^{2\pi} x\, dy. \tag{5}$$

Taking into consideration, that:

$$dy = L\frac{d\varphi}{\pi}, \tag{6}$$

we shall receive:

$$F_0 = \int_0^{2\pi} \frac{L\,r}{\pi}(1 - cos\varphi)d\varphi, \tag{7}$$

Having solved the equation (7) finally we shall receive:

$$F_0 = S\,L. \tag{8}$$

So, the load field F_0 in the cutting assemblly of a normal cutting with a single stroke of knives equals to the product of the cutter bar's stroke S and the feeding way L.
According to calculations of the study's authors, for the low cut cutting assembly, the load field amounts to:

$$F_0 = 0{,}32\, S\, L. \tag{9}$$

However, for the normal cut cutting assembly with double stroke of knife, the load field amounts to:

$$F_0 = 0{,}18\, S\, L. \tag{10}$$

Concluding it may be said, that the load fields of cutting assemblies on the stage of their theoretical calculations may be determined based on the derived dependencies: (8), (9) and (10) or for the analysed, special construction of the cutting assembly they should be determined independently.
The inertial force P_B of the cutter bar's mass is the product of the cutter bar's mass m together with the part of the connecting rod's mass making the to-and-fro motions and its acceleration a.
Taking into consideration the fact, that:

$$m = (m_1 + m_2), \tag{11}$$

and

$$a = \omega^2\, r\, cos\omega t = \omega^2\, r\, \left(1 - \frac{x}{r}\right), \tag{12}$$

we have received:

$$P_B = (m_1 + m_2)\, \omega^2\, r\, \left(1 - \frac{x}{r}\right). \tag{13}$$

where m_1 is the cutter bar's mass , m_2 is the part of the connecting rod's mass making to-and-fro motions, r is the length of the crank, ω is the crank's angular speed, $x_{nż}$ is the cutter bar's displacement.
Mass m_1 of the cutter bar is calculated assuming that the mass falling per 1m of its lendth amounts to: 2,20-2,35 $\text{kg} \cdot \text{m}^{-1}$ (Zastempowski & Bochat, 2014).
However, m_2 is calculated from dependences:

$$m_2 = m_k \frac{l_0}{l}, \tag{14}$$

where m_k is the mass of a connecting rod, l_0 is the distance of the centre of the connecting rod's mass from the crankpin, l is the total length of the connecting rod.
From the analysis of the selected crank drives' design solutions it results, that the connecting rod's mass most often amounts to: 2,30–3,25 kg, and in extreme cases it assumes the value equal to 5 kg (Bochat, 2010).
The friction force T_C of the cutter bar against the cutter assembly's guide elements is the sum of the friction force T_1, coming from the weight of the cutter bar and the friction force T_2, coming from the connecting rod's operation.
So:

$$T_c = T_1 + T_2. \tag{15}$$

The friction force T_1 is determined from the dependence:

$$T_1 = \mu\, m_1 g, \tag{16}$$

where μ is the sliding friction co-efficient and assumes the values within the range 0,25-0,30 [2, 3], m_1 is the mass of the cutter bar, and g is the gravitational acceleration.
However, the friction force T_2 is determined on the basis of the depenedence:

$$T_2 = \mu\, P_2, \tag{17}$$

whereas

$$P_2 = P_1\, tg\beta, \tag{18}$$

where P_2 is the normal constituent of the connecting rod's having an effect on the cutting assembly (fig. 2), and β is the temporaty inclination angle of a pitman (connecting rod) to the cutting plane. Taking into consideration that:

$$P_2 = (P_s + P_B + \mu\, m_1 g + \mu\, P_2)\, tg\beta. \tag{19}$$

Following conversion we have received:

$$P_2 = \frac{(P_s + P_B + \mu\, m_1 g)tg\beta}{1 - \mu\, tg\beta}. \qquad (20)$$

From the analysis of the formula (20) it results, that the temporaty inclination angle of a pitman (connecting rod) β to the level, has an important impact on the T_2 force value, which changes its value within a closely determined scope.
The scope of changes determines the value of eccentricity ε of the crank mechanism:

$$\varepsilon = \frac{h}{l}, \qquad (21)$$

where h is the distance of the crank shaft from the cutter bar movement's plane, and l is the length of the pitman.

RESULTS AND DISCUSSION

Within the frames of the task's realization, on the basis of the derived mathematical dependences, an own software has been developed for quick simulation calculations, within the frames of which there have been identified demands for power and forces operating within the frames of shear-finger cutting assemblies under the influence of characteristic input values such as: μ- friction co-efficient, m_1 – mass of the cutter bar, m_2 – mass of the connecting rod, h_k – distance of the symmertry axis of the crank shaft's distance from the cutter bar movement's plane, l – length of the connecting rod (pitman) and n – the shaft's rotational speed. At the stage of the simulation surveys' realization, there have been calculated in order: $v_{snż}$ – mean speed of knives, P_S – mean value of cutting resistances, P_B -maximum inertial force, T_1 – friction force from the cutter bar's weight, T_2 – frinction force on the connecting rod's operation, P_{max} – maximum value of the total force, N – demand for power.
In table 1 there are presented the exemplary results of simulation calculations of the movement dynamics of the cutter bar's movement of the shear finger cutting assembly.

Tab. 1 Results of the exemplary simulation calculations of the shearfinger cutting assembly (own study)

	Fixed input data	
	h_k = 75 mm, *l = 230 mm*	*n = 1020 rev min^{-1}*
	Variable input data	
	μ=0,3 m_1 = 4 kg m_2 = 1,4 kg	*μ=0,19 m_1 = 2 kg m_2 =1,4 kg*
	Results of calculations	
$v_{snż}$ [m s^{-1}]	2,59	2,59
P_S [N]	952,5	952,5
P_B [N]	2347,34	1477,95
T_1 [N]	11,77	3,72
T_2 [N]	2186,62	1498,43
P_{max} [N]	4112,01	3059,80
N [kW]	15,81	11,76

The tool developed for simulation calculations makes allows for quick identification of demand for power and the values of the operating forces, also in other working assemblies of the shear-finger cutting assemblies following implementation of changes in their construction.

CONCLUSIONS

Within the frames of the study's realization, functioning of the shear-finger cutting assembly in the aspect of dynamic loads of its cutter bar has been analysed. According to the assumed purpose of the study, there have been drawn up mathematical dependencies describing the cutter bar's dynamic loads. For the purposes of quick designing of the power transmission systems of the cutter bar and the shear-finger cutting assembly, there have been conducted the simulation calculations as well as the own software for quick identification of the demand for power and forces operating within the shear cutting units under the influence of the characteristic input data's changes has been developed.

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