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The Substantiation of Parameters of Mechanism for the Drive of Cutting Machine with a Double-Way of the Knife

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Abstract: For kinematic study regarding crank-rocker mechanism the method of vector contours was applied. Analytical equations that describe patterns of motion of knife stroke, speed and acceleration were obtained. They allowed to substantiate the parameters of the crank-rocker mechanism with a single and double strokes of the knife. The correctness of theoretical premises is confirmed by the graphoanalytical method. The developed mechanisms were tested in production conditions. The mechanism for a double-stroke drive allows to reduce the power requirement by 1.6.

Key words: Crank-rocker mechanism, drive, cutting device, double stroke of the knife, inertial forces, rotating speed, crank, rocker, segment, fingers, lever, vector method

INTRODUCTION

In the Republic of Kazakhstan 5.0 million ha of land are occupied by natural grasses with low yields and it is planned to increase acreage areas for sown herbs by more than 3.4 million ha.

The solution to the problem of fodder harvesting largely depends on the availability and condition of mowers and perfection of their design. Therefore, it is necessary to modernize existing and create new mowers taking into account domestic and foreign experience.

According to the principle of cutting plants, cutting machines of mowers are divided into two main groups: apparatuses that cut plant stems with support and without support of stalks by the movement trajectory of cutting elements of apparatuses with reciprocating motion of knives and with rotational motion. Cutting devices with reciprocating motion of knives are divided into sets with fixed fingers, movable fingers and two-knife ones (Osobov and Vasiliev, 1983). In Foreign countries for mowing grasses rotary mowers are mainly used. They allow to work at large forward speeds, limited only by the terrain and the tractor's capabilities. The use of rotary cutting devices improves the efficiency of technological process in mowing high-yielding, coarse and liable grasses (Osobov and Vasiliev, 1983).

The disadvantages of rotary mowers are revealed in high energy intensity (about 12, ..., 15 kW/m of working width), high specific fuel consumption and high cost; risk of injury (significant acceleration of falling Foreign objects) and contamination of grass with soil while mowing grass on sandy and loose soils (Vanecek and Kalab, 2003).

The most common method of harvesting forages from natural and sowing grasses in practice is the technology of mowing grass with the use of mowing segment-finger cutting machines with laying of sloping mass in the mown. For mowing small sections of complex contour farms mainly single-barred mowers CS-F-2,1 are used and in large areas with flat terrain semi-mounted two-bridle mower KD-F-4,0, two-bridle mounted mower KDS-4, 0 and trailed three-axle mowers KP-F-6.0 are used (Khalansky and Gorbachev, 2004).

Mowers with segment-finger cutting apparatuses are produced mainly in CIS: PJSC "Berdyansk Reapers" (Ukraine) produces a single-beam hinged mower KPO-2,1, OJSC "Bobruiskagromash" (Republic of Belarus) produces a segmented-finger mounted mower KS-F -2,1B and mounted two-bridle mower KDS-4,0, JSC "Urgenchkormmash" (Uzbekistan) produces a one-bridle mounted mower KOS-2,1 and two-bridged half-mounted mower KDP-4,0 (Vanecek and Kalab, 2003). The advantages of these mowers are the exact cut, low energy consumption (about 2.0, ..., 2.5 kW/m of working width), low contamination of the forage mass and low cost.

Disadvantages are presented in low productivity (about 0.45 ha/1 h and a meter of working width), high risk of blockage and frequent replacement of knives, high

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sensitivity to mechanical damage, high maintenance costs. Also the disadvantage of the known mowing machines with segmental-toothed cutting devices is revealed in cutting grasses on natural and sowing hayfields with uneven terrain, frequent breakage of knife and connecting rod. The reason is that a crank-conrod mechanism is used for driver of cutting machine in mowers. The driving shaft with pulley and a crankcase box with pulley are fixed to the main frame; the cutting device with inner and outer shoes is fixed to the frame by means of pull rod, yoke while the connecting rod engages with the knife head. Therefore, on uneven fields, when the outer shoe rises more than 0.2 m above the inner shoe, horizontal angle of inclination of connecting rod increases and the stroke of knife changes. With an increase ofinclination angle of connecting rod, the vertical components of forces which act on knife and elements of the finger rod, rise what leads to increase of frictional forces in the cutting device and pinching of the knife in the cutting device. As a consequence, frequent breakages of the knife and the connecting rod occur, it requires additional time to eliminate the drawbacks, it reduces the productivity of the mower. It is possible to increase the productivity of mowers and reapers by increasing the width of gripper unit as well as its translational speed. Increasing the width of the aggregate leads to creation of cumbersome, low-maneuverability and unreliable machines with uneven hayfields.

One of the promising directions for increasing the translational speed is an increase in the rotational speed of the crank shaft. This in turn, leads to a sharp increase in alternating inertial loads and consequently, to a decrease in operational reliability and durability of the machine as a whole.

The purpose of the research is the development and justification of design parameters and operating modes of the driving mechanism of cutting device of mower which is able to work on unevenness of natural and sow hayfields which reduces dynamic loads and high-quality performance of the technological process. In this regards, the following tasks are set in the research:

- Carry out an analysis of conducted studies of various mechanisms for drivers of cutting device, select the most appropriate structural design scheme
- Determine the rational relationship between individual parameters of the drive mechanism of the mower
- Carry out a study of kinematic and dynamic characteristics of the drive mechanisms and determine the main parameters

MATERIALS AND METHODS

The research is carried out using classical methods of theoretical and applied mechanics, theory of mechanisms and machines.

Kinematic study of mechanism is carried out by the vector contour method. According to this method, the scheme of a plane lever mechanism which is located in a rectangular coordinate system xOy, isrepresented as a closed polygon that consists of one or more closed vector contours depending on the complexity of the mechanism. Closure conditions arewritten in vector form or in form of projections on the coordinate axis (Zinoviev, 1972).

Numerical analysis of equation of drive motion of the knife of cutting device is performed using standard Excel programs, Matcad. In processing the results of experimental studies the provisions of mathematical statisticsare used.

RESULTS AND DISCUSSION

To implement the reciprocating motion of knives of cutting machines, various types of drive mechanisms are used. The most widespread oneis the crank mechanism. Due to design features of machines and conditions of their operation, the axis of the crank shaftis located above the line of motion of knife by some amount which is called as displacement (deaxial).

The experience of operating of single-knife cutting machines shows that it is impossible to completely balance the inertial forces of oscillating masses.

Maglioni and Molari investigated the crank-rod mechanism with reciprocating motion of the knife. They determined the impact of kinematic parameters on the movement ofblade mechanism, the quality of grass cutting and swing of the mower.

Using the method of dynamic models, they carried out a dynamic analysis of the mechanism, determined the law of temporal variation of angular velocity of drive mechanism. The changes in drive torque which were obtained experimentally were used to analyze the finite elements with the use of MSC. Visual. Nastran Software (Dumitru and Ungureanu, 2009).

In the mower with movable fingers (Duplex cutting device), the fingers are fixed to the back of toe bar and produce reciprocating motion towards the segments with drive from the crankshaft. At the same time, the beam and the knife of cutting device oscillate in antiphase with the same frequency and thus, provide self-balancing of inertial forces in cutting device (Suranchiev, 2006).

This device of the cutting machine allows to increase the cutting speed by two times at the same speed on the driven pulley which allows to obtain more clean cut of the grass stand. In addition, the reciprocating movement of fingers provides an opportunity to make mowing of grass that lies in the wet grass stand. The disadvantage of this mower is revealed in complexity of design and in imbalance of the inertial forces of pin bar and cutting knife.

Rustamov (1981) detailed the analysis of various mechanisms of knife drives that were used in commercially produced machines, mowing machines, reapers also in proposed new inventions, researcher's certificates and patents. He analyzed all known mechanisms of the drive with an assessment of their compliance with requirements in terms of analysis and synthesis of driving mechanisms.

In grain and forage harvesters the crank shaft and blade of cutting device are located on one rigid frame. All links of this type of flat mechanisms move in the same plane. The displacement is chosen so that the lowest point of trajectory of the crank pin was above the line of knife movement (Reznikov *et al*., 1991).

In mowers the crank shaft with box and crank are located on the main frame, it is connected by means of connecting rod with knife head of the cutting device which moves reciprocally. The finger bar during the work is supported by shoes on the surface of field. To copy the microrelief, the beam is articulated by a pull rod to the frame of machine. Due to elastic deformations of pull rod and gaps in hinges, the finger bar moves back. In order to ensure the efficiency of the mower, the field end of the pallet rod is pushed forward in course of the unit while in the drive of the cutting device ball joints are used (Dolgova, 1987). This allows links to move in different planes; the mechanism becomes spatial. Such drive of the cutting device is used in two-beamed mower KDP-4,0 in mounted mower KS-2,1, etc. The rotational speed of the PTO is 940, ..., 1024 rpm. When the unit is moving with the switched on power selection valley of tractor, the rotation through the cardan and belt drive is fed to the eccentric pulley. From it, through the crank mechanism, the rotation is converted into a reciprocating motion of knife which produces mowing of the plants.

Experiments have established that for high-qualitative cut of plants it is necessary to have a cutting speed for grass of at least 2.15 m/sec and for cereals -1.5 m/sec.

There are known mowers that have a crankcase made in the form of a shaft which is mounted on bearings in a housing at one end a pulley is mounted and at the other end a crank with a crank mechanism is attached. It is mounted on the inner shoe and connected to the pull rod by a pivot shaft which coincides with the axis of rotation of crank input shaft. The connecting rod moves in parallel plane of the movement of knife in cutting machine.

These mowers have low operational reliability of work on the complex micro relief of terrain. The reason is described in the upper part of connecting rod that moves along a circle with radius which is equal to the radius of the crank with finge while the lower part of connecting rod which is connected to the knife, performs oscillatory reciprocating motion in horizontal plane. The larger the radius of the crank, the greater the angle of inclination of connecting rod to the plane of motion of the knife. As the angle of inclination of connecting rod increases, the vertical components of forces which act on knife and elements of the finger rod, increase, what leads to increasein the frictional forces of cutting device.

Also, self-balancing schemes of cutting machines are used in construction of mowers. In Foreign countries, no-finger cutting machines are used with two active knives that are equal in length to the width of the grip and oscillate in opposite directions. A non-paltry cutting device with two movable knives is used for cleaning confused and loosened bread, rice, legumes where cutting machines with fingers usually can not work and also they are used at switching to increased translational speeds of harvesting machines. The cutting device with two movable knives is used on "Don-1500" combines and ZHBR-4, 2 reaping machines (Reznikov *et al*., 1991).

The two-knife cutting device allows to increase the speed of tractor to 12 km/h, what significantly increases the productivity with clean mowing quality and low power demand (about 2.5 kW/m of working width). The design of a double-chevron cutting device allows to balance inertial forces. The drive of cutting device is carried out with the help of crank-rocker mechanism, the links of which are hingedly mounted on rubber blocks, what provides quiet and easy work of the mower (Biocom Technology Ltd., 2017; Brielmaier motor mower GmbH, 2016; Buhler Industries Inc., 2016).

The disadvantages of two-knife cutting device include the difficulty of maintaining a constant gap between segments of upper and lower knives during operation. Increasing this gap will result in clogging the machine. The mechanism of drive of knifes is quite complex. On the areas contaminated with stones the apparatus is not reliable in operation because of insecurity of segments (Osobov and Vasiliev, 1983). At a low cut of grasses on uneven fields segments of knives are buried in the earth, that leads to breakages of segments.

For mowing of sown and natural grasses with the formation of rolls, self-propelled mowers-conditioner E-303 E-304 of firm "Gase IH" (Germany), trailing mower-conditioner KPP-4,2 "Polesset ST 42" Gomselmash are used. In drives of cutting machines for self-propelled machines, crank and rocker mechanisms which are spatial are used because of the need to reduce the cross-sectional dimensions of harvesters. The distinctive feature of crank-rocker mechanism is that between the connecting rod and knife of the cutting machine additional links (a two-arm lever and a leash-crank small) are presented (Adilsheev and Zhortyylov, 2015).

On the front mounted reapers of forage harvesters for driver of knives of grasping devices, the mechanisms of swashplate are used. On reapers of self-propelled harvesters and self-propelled mower-pliers two mechanisms of swinging washer are installed by one on the right and the left knives. The main components of such a mechanism are: crank shaft, spider, washer shaft, fork, connecting rod, suspension and knife head.

The advantage of mechanisms of swinging fork and washer as a drive for the knife of cutting machines in harvesting machines is the compactness which makes it possible to locate the drive practically behind the shield of the field divider.

The disadvantage is that the mechanism has a complex design which requires the preparation of detailesof high class of accuracy and careful assembly. The reliability of the work of mechanism of swash plate is ensured only if the axis of its three parts of crank shaft, washer and fork intersect at one point. Failure to comply with these requirements leads to rapid heating and wear of hinges, breakage of parts, i.e., reducingof reliability of mechanism and consequently to decrease in performance of reapers (Adilloseev, 2010).

For the drive of cutting device of agricultural harvesters N.B. Bokom proposed the planetary mechanism (Bok and Amiryanm, 1967). It consists of a gear crown of drive that is mounted in eccentric shaft which rolls over a fixed toothed wheel crown with internal teeth. The radius of the initial circumference of crooked spike gear is equal to the radius of pin position of crankin relation to its axis. When the eccentric shaft rotates, the crank pin performs a rectilinear reciprocating motion at each revolution which exactly corresponds to diameter of initial circle of the wheel with internal teeth or to double diameter of initial circumference of crankshaft crown.

At the present time, the construction of drive with the use of planetary gear which is perfected by Schumacher is used on reapers of harvesting machines in Germany. Due to the rectilinear knife, beating and vibration which act on reapers are excluded. The number of moves of knife increases to 1120, ..., 1200/min. The society of Limited Liability Company "Astarta-Tehniks" (Russia) provide services for modernization of various types of reapers by replacing standard cuting mechanism to cuting system "Schumacher" with planetary gear mechanism. The reapers "DON-1500", "Niva" (Rosselmash), "Yenisei" (Krasnoyarsk combines), claas, case, Massey-Fergusson, New Holland, Deutzfahr, Challenger, John Deere, Laverda, Biso, Capello, "Lida", "Berdyanskie Harvesters", "Pole-sie" (Gomselmash), KSK-100, E-281, E-292, E-303, KPS-5G, ZhVN-6, ZHVP and others are under the modernisation. According to the data of LLC "Astarta-Technics", productivity increases, losses are reduced during cleaning, time and money spent on maintenance are reduced.

The shortcomings of the planetary mechanism "Schumacher" include a large cost which fluctuates from $700-1,300 depending on the manufacturer. Therefore, the planetary drive is not installed on mower as the cost of drive exceeds the cost of mower itself that equal to 1000$.

Increasing the productivity of the mower, reducing the rotational speed of the crankshaft and thereby, reducing inertial loads is achieved using a double-blade cutter. The studies of operation of the cutting device with double stroke of segments of mowers are devoted to research by Popov (1978). The advantage of cutting machine with a double run of segments in comparison with a single-run machine is that the angular speed of crank is reduced by 1.5, ..., 2.0 times, the lodging of cutting device can be increased by 1.6, inertia forces of knife are reduced in 1.1, ..., 1.3 times (Popov, 1978). But this device has significant drawbacks:

- Insufficient use of the maximum speed of the knife during cutting of stems does not allow to decrease the frequency of rotation with increased radius by 2.0
- The oscillation of toe bar, frame of machine with a double stroke of the knife is much larger than those with a single knife
- With the crank drive of knife in a dangerous section of blade high stresses arise
- Damage of segments of blade occurs faster than in devices with single run of segments
- The power which is required to overcome the frictional forces of the cutting device with a double blade run will be 25% greater than that of a single-run cutting machine

It should be noted that these results were obtained for a cutting device with a double blade stroke that was driven by de-axial crank-conrod mechanism. The double stroke of knife is ensured at a radius of crank that is 2 times greater than in the normal cutting machines with single run segments. In this case, the angle of inclination of connecting rod increases, what leads to increase in vertical components of forces what cause an increase in friction force against the guides.

Some disadvantages can be eliminated using stamped double fingers, segments covered with ceramics. The two-sided cutting device of "Schumacher" type eliminates the need to install clamping devices and allows to produce a quality cut both of sown and natural grass.

At the Kazakh Scientific Research Institute of Mechanization and Electrification of Agriculture a mechanism of driver of the mower has been developed in which the reciprocating motion of knife of cutting device is carried out by a crank-rocker mechanism. The technical novelty of invention is protected by innovative patents of Republic of Kazakhstan No. 26421 and 29916 (Zhortyylov *et al*., 2013, 2015). The mechanism is mounted on the inner shoe and consists of drive shaft with a crank connected via. connecting rod to a rocker, one end of which it is pivotally connected to the stand. The other end of the rocker is connected through a leash with a knife head (Zhortyylov *et al*., 2011). Determining the parameters of the drive mechanism of the cutting device.

In the mower for drive of the knife a crank-rocker mechanism is used. It converts the rotational motion of crank into oscillatory motion of rocker and reciprocating motion of knife. The mechanism consists of crank 1, connecting rod 2, rocker 3, knife head 4.

The scheme of the crank-rocker mechanism of the drive in cutting device with single and double run of the knife is shown in Fig. 1. The mechanism consists of crank 1, connecting rod 2, rocker 3, leash 4, slide-head of knife 5 and post 6. The rocker has a length, DE = R. Coefficient of lengths of rocker:

$$K = \frac{DE}{DC} = \frac{R}{R_1}$$

A lead of length of l_4 is connected to the point E. To determine the kinematic parameters of the mechanism, let's consider the four-link mechanism and compose the vector equation of closed contour ABCD (Fig. 2). When the mechanism is operating, the rocker CD which swings about the D axis, describes an arc C_1, C_2.

The position of the CD rocker which corresponds to extreme position of knife, must be symmetrical about the axis O_Y, So, $OC_1 = OC_2$. This installation of the rocker provides reliable operation of cutting device. If cranks are aligned with connecting rod, the knife will take extreme positions. In the initial position at $\varphi = 0$, the crank AB and connecting rod BC lie along one straight line $l = d \cos \alpha$. The stroke of knife depends on the length of rocker and

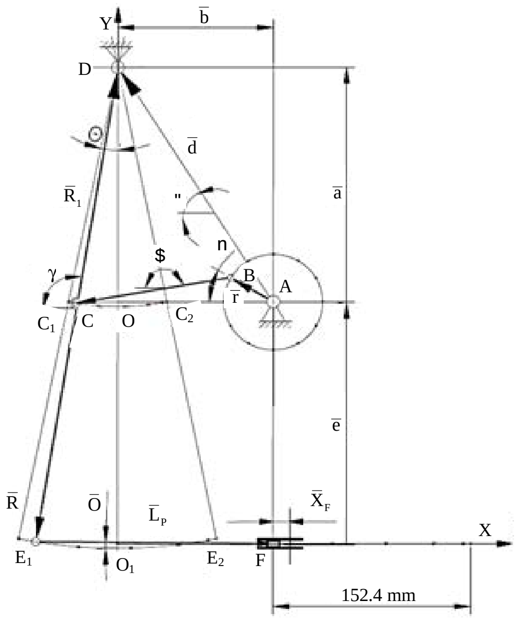

Fig. 1: Scheme of the crank-rocker mechanism of driver of cutting mechanism with double blade run

angle of its tilt (rocking). The single stroke of knife of the cutting device S = t = 76.2 mm. Assuming the length of rocker arm R, determine the rolling angle of rocker arm θ by equation:

$$\theta = \arccos S/2R_1 = \gamma\ 90°$$

The working condition of the crank is expressed by the following inequalities:

$$r \prec l; r+l \prec R_1+d; l-r \succ d-R_1$$

The vector equation can be written in the following form:

$$\vec{r}+\vec{l}+\vec{R}_1 = \vec{d} \tag{1}$$

Where:

r = The crank radius AB
l = The conrod length BC
R_1 = The length of the rocker CD
d = The distance between supports A and D

Projecting Eq. 1 on the coordinate axis OX and OY, we obtain:

$$\left.\begin{aligned} r\cos\varphi + l\cos\beta + R_1\cos\gamma = d\cos\alpha \\ r\sin\varphi + l\sin\beta + R_1\sin\gamma = d\sin\alpha \end{aligned}\right\} \tag{2}$$

Where:

φ = The rotation angle of the crank
β, γ, α = The angles that are formed by the vectors $\vec{l}, \vec{R}, \vec{d}$ with the O_x axis at $\varphi = 0$; $d \cos \alpha = l$; $d \sin \alpha = R_1 \sin \gamma$

The movement of the point of swing rocker CD is determined by the expression:

$$x_1 = R_1 \cos\gamma \tag{3}$$

From the first Equation of the system 2, the value of the expression $R_1 \cos \gamma$ is substituted into equation:

$$x_1 = r\cos\varphi - l\cos\beta + d\cos\alpha \tag{4}$$

From the second Eq. of system 2:

$$\sin\beta = \frac{-r\sin\varphi - R_1\sin\gamma + d\sin\alpha}{l}$$

Then:

$$\cos\beta = \frac{\sqrt{l^2 - (-r\sin\varphi - R_1\sin\gamma + d\sin\alpha)^2}}{l}$$

Substituting the value of the expression $\cos \beta$ into Eq. 4, we obtain:

$$x_1 = -r\cos\varphi - \sqrt{l^2 - (-r\sin\varphi - R_1\sin\gamma + d\sin\alpha)^2} + d\cos\alpha \tag{5}$$

With sufficient accuracy, we can assume that $R_1 \sin \gamma \approx d \sin \alpha$, then we get:

$$x_1 = -r\cos\varphi - \sqrt{l^2 - r^2\sin^2\varphi} + d\cos\alpha \tag{6}$$

Equation determining the analogues of velocity and acceleration of point C are obtained by two-fold differentiation by Eq. 6 over the generalized coordinate. Then we obtain analytical expressions which determine the displacement, speed and acceleration of knife of cutting machine of the mower:

$$x = \frac{R}{R_1}\left(-r\cos\varphi - \sqrt{l^2 - r^2\sin^2\varphi} + d\cos\varepsilon\right) \tag{7}$$

$$V = \frac{R}{R_1}\omega\left(r\sin\varphi + \frac{r^2\sin 2\varphi}{2\sqrt{l^2 - r^2\sin^2\varphi}}\right) \tag{8}$$

$$a = \frac{R}{R_1}\omega^2\left[r\cos\varphi + \frac{r^2}{4}\left(\frac{4\sqrt{(l^2 - r^2\sin^2\varphi)}\cos 2\varphi + r^2\sin^2 2\varphi}{\sqrt{(l^2 - r^2\sin^2\varphi)^3}}\right)\right] \tag{9}$$

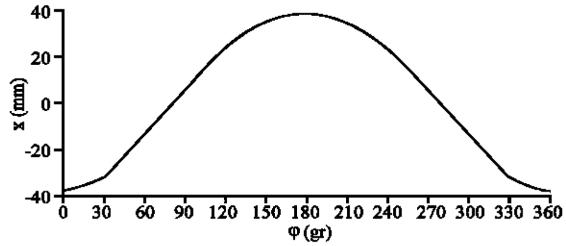

Fig. 2: Chart of movement with single run of the knife in dependence on the rotation angle of crank

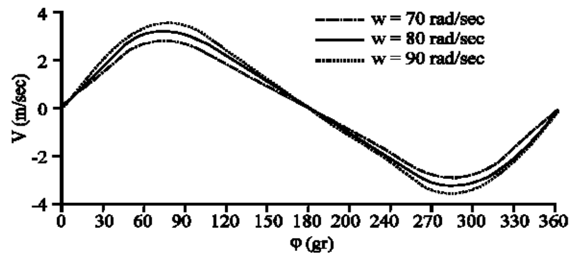

Fig. 3: Graphs of changing of speed of knife with a single run in dependence on rotation angle of crank

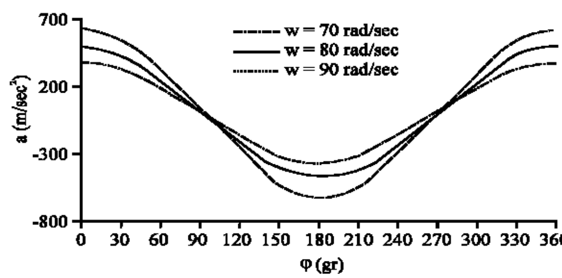

Fig. 4: Graphs of the change in acceleration of knife with a single run in dependenceon rotation angle of crank

With $R = R_1 = D_C$ equation are described for a single blade run and with $R/R_1 = 2$ for a double blade run.

Figure 2 shows the movement of the knife in dependence onrotation angle of crank for a cutting device with a single blade run.

Figure 3 and 4 show graphs of speed and acceleration of theknife with a single blade run at various crank shaft speeds. At a speed of ω = 70 rad/sec, the maximum speed of the knife is 2.8 m/sec and at rotational speeds ω = 80, 90 rad/sec, the maximum speeds reach 3.1 and 3.5 m/sec that is above the minimum allowable value (2.5 m/sec). Therefore, to perform the mowing process, the rotation speed must be at least 70 rad/sec. At rotational speeds of ω = 70, 80 and rad/sec, the maximum knife acceleration reaches 350, 500 and 600 m/sec^2.

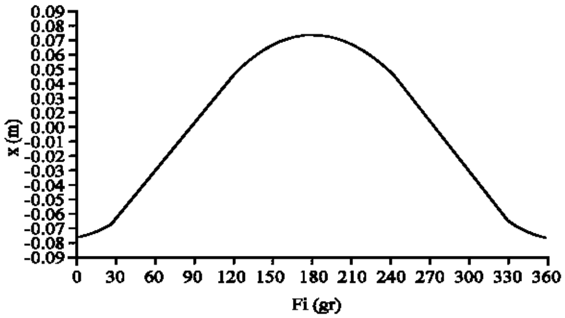

Fig. 5: Diagram of moving of knife with a double run in dependence on rotation angle of crank

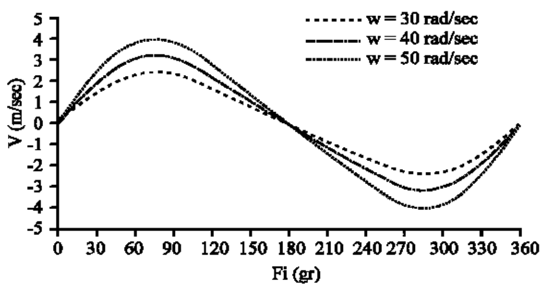

Fig. 6: Charts of change of knife speed with double run in dependence on rotation angle of crank

Therefore, in order to reduce the inertial forces of the knife, it is advisable to mow at a cranking speed of 70 rad/sec.

Figure 5 shows the movement of the knife in dependence on the rotation angleof crank for the cutting device with a double blade run. Figure 6 shows the graphs of the change of knife speed with a double run at different rotational speeds of crooked spike, depending on angle of crank.

As the speed of rotation increases, the speed of blade increases too. At rotational speed of ω = 30 rad/sec, the speed at the beginning of cutting is 2.30 m/sec and at the end of cutting it is 1.75 m/sec what is below the minimum permissible value (2.5 m/sec). At a speed of ω = 40 rad/sec, the maximum blade speed is 3.1 m/sec. The maximum values of speed in this mechanism are shifted compared to the crank-conrod mechanism at 18°, ..., 20°. Knife velocities at the beginning and at the end of cutting are not equal (respectively, 3.05 and 2.6 m/sec). The speeds of forward and reverse strokes of knife are equal to each other. Thus, for a favorable performance of the mowing process, the rotation frequency of crank must be at least 40 rad/sec.

The graphs of changes of acceleration of knife with a double run in dependence on angle of rotation of the crank at different speeds are shown in Fig. 7.

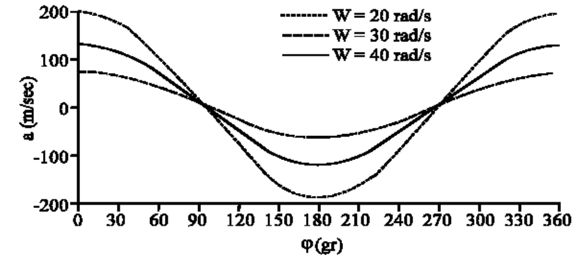

Fig. 7: Graph of the changes of acceleration of knife with a double run in dependence on angle of rotation of crank

In order to determine the correctness of results in theoretical calculations in determining the stroke (movement) of the knife, we determine the stroke of the knife by graphical method constructing in 12 positions the crank of mechanism.

For the vector contour DEFF'D of the non-axial rocker-slider mechanism, we write the Eq. 10 (Klenin and Sakun, 1980):

$$\bar{l}_{DE}+l_n = \bar{x}_F+\bar{e}+\bar{a} \quad (10)$$

Projecting it on the x and y axis gives:

$$\begin{aligned} \bar{l}_{DE}\times\cos(180\text{-}\gamma)+\bar{l}_n\times\cos\delta &= x_F \\ \bar{l}_{DE}\times\sin(180\text{-}\gamma)+\bar{l}_n\times\sin\delta &= e+a \end{aligned} \quad (11)$$

where δ is the angle between the positive direction of the x axis and the longitudinal axis of lead.

Moving the end of knife head vertically is determined by eqation:

$$\Delta h = R(1\text{-}\cos\theta)$$

where $\theta = \gamma$-90°C is the angle of inclination of rocker to the vertical axis.

As a result of comparison, we are convinced of the correctness of analytical method for determining the stroke pattern of knife. It coincides with the graph which is constructed by graph-analytical method. The discrepancies constitute not more than 0.5÷0.7% what is within the permissible values.

Determination of power required for driver of cutting machine of mower. The resistance force P is determined by Eq. 12 (Popov, 1986):

$$P = P_{avr} + P_j + F \quad (12)$$

Where:

P_{avr} = The average value of the shearing force (H)
P_j = The inertia force of the knife mass (H)
F = The friction force of the knife (H)

$$P_{avr} = P^1_{avr} + P^{"}_{avr} \quad (13)$$

The value of the average is not the same with shear and it is determined by Eq. 14:

$$P_{avr} = P^1_{avr} + P^{"}_{avr} = \frac{eF_{H_1} \times Z}{?_{f_1}} + \frac{eF_{H_2} \times Z}{x_{f_2}} \quad (14)$$

Where:

Z = The number of segments

ε = The coefficient that takes into account the number of stems and their characteristics for grasses, $\varepsilon = (2, ..., 3) \times 10^{-2}$J

In the apparatus of normal cutting with a double run of knife of load area F_{H1} and F_{H2} and for one and the other fingers, near which the plants are cut are different (F_{H1} = 0.32 LS, F_{H2} = 0.18 LS).

X_{f1} and X_{f2}, the movement of knife from the beginning to the end of the cutting, respectively, at middle and last fingers.

The inertia force is determined by the value of the mass of knife m_H and the acceleration α:

$$P_j = m_H a$$

$$\text{as; } a_n = \frac{R}{R_1} r\omega^2 \cos\omega t = \frac{R}{R_1} r\omega^2 \left(1 - \frac{x}{r}\right), \quad (15)$$

$$\text{To } P_j = m_H \frac{R}{R_1} r\omega^2 \left(1 - \frac{x}{r}\right)$$

The change in the inertia force in dependence withthe movement of knife will be represented by a straight line 2 while the maximum values $P_{jmax} = R/R_1 r\omega^2$ corresponde to the beginning and the end of the blade stroke and $P_j = 0$ for x = r.

The gravity force of the knife in mowers and reapers on a length of 1 m is equal.

In mowers with a crank-rocker mechanism, the frictional force of the knife on elements of the toe bar is composed of the frictional force F_1 which is caused by the force of gravity of knife and by the force F_2 from the action of the leash:

$$F = F_1 + F_2 \quad (16)$$

The friction force from the gravity of the knife on elements of the toe bar is equal to (Klenin and Sakun, 1980):

$$F_1 = G_H f \quad (17)$$

Where:

G_H = The force of gravity of knife (H)

F = The coefficient of friction

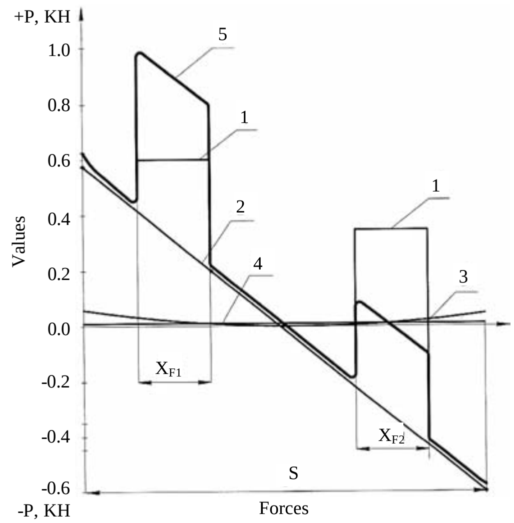

Fig. 8: The graph of the change in forces that act on the double-edged knife in dependence on its movement: 1) Cutting force; 2) Inertia force; 3) Friction force from the weight of the knife; 4) Frictional force from knife pressing by leash and 5) Resultant force

Considering that the rubbing pair of knife-finger bar works without lubrication, in abrasive medium the coefficient of friction can be taken as f = 0.25, ..., 0.3. The force of the leash:

$$F_2 = \frac{\left(P_{avr} + P_j + fG_H\right) tg\delta}{1 - ftg\delta} \times f \quad (18)$$

The force required for movement of the knife P will be:

$$P = \frac{\varepsilon F_{H_1} \times z}{x_{f_1}} + \frac{\varepsilon F_{H_2} \times z}{x_{f_2}} = m_H a + fG_H + \frac{\left(P_{cp} + P_j + fG_H\right) tg\delta}{1 - ftg\delta} \times f \quad (19)$$

In the crank-conrod mechanism δ angle of inclination of connecting rod to the plane of knife varies within the limits of 25-35° (tgδ = 0.47-0.70) (Osobov and Vasiliev, 1983) and in crank-beam mechanism the angle of inclination of the lead to the plane of the knife δ varies within the range of 0-5° (tgδ = 0.0-0.0875). It follows that the frictional force F_2 in the crank-beam rocker mechanism is reduced by 5-8 in comparison with that in the crank-conrod mechanism.

The graph of the change in forces which act on the double-edged knife in dependence on its movement is shown in Fig. 8.

(a)

(b)

Fig. 9: a) Crank-rocker mechanisms for driver of the cutting device with a double stroke which are used in mowers and b) In mower-conditioner KP-3.0

The power required to overcome the knife resistance forces to motion is determined from the expression (Osobov and Vasiliev, 1983):

$$N = P \times v_{average}$$

Where:

P = The knife resistance to motion (N)

$v_{average}$ = The average speed of the knife (m/sec)

As a result of the calculation of power which is required to drive the cutting machines of the mower with a working width of 2.1 m, it is established that the power required for the drive of crank-rocker mechanism with a double knife stroke when mowing grass is 2.93 W. The power required to drive a crank-crank mechanism with a single blade run is 4.8 kW (Osobov and Vasiliev, 1983). Consequently, the required power for the drive of the crank-rocker mechanism is reduced by 1.6.

The developed crank-beam mechanism for driving the cutting device with a double run of the knife is used in KS-2,1Z mower, KP-3.0 mower-conditioner and two-branched KDP-4.0 mower (Fig. 9) which passed production tests with positive results.

The power parameters of KS-2.1Z mower, KDP-4.0 two-bridle mower and KP-3.0 conditioner mower were determined in the field conditions by the method of tensometry of the torque and the PTO speed of the tractor. The power required to mow the grass by KDP-4.0 two-branched mower with two mower modules with a working width of 4.0 m at a machine speed of 2.0 m/sec is 6.0 kW and the idling speed of the cutting machines is 3.1 kW. The productivity at mowing was 2.8, ..., 3.6 ha/h. As a result of the tests, it has been established that on uneven haymaking relief the technological processes of mowing grass with stowage are carried out reliably and qualitatively. Mowers are able to work in conditions of uneven terrain, natural and sowing hayfields as well as in large and small areas of sown and natural grasses.

CONCLUSION

By the method of vector contours, analytical equations are obtained for determining the analogs of knife displacement, speed and acceleration of motion of the knife with a single and double strokes. The parameters of the crank-rocker drive mechanism of cutting device with single and double strokes of knife are justified: r = 0.038 m; ω = 40 rad/sec.

The developed crank-beam drive mechanism of cutting device is used in mower KS-2.1 G, two-sided mower KDP-4.0, mower-conditioner KP-3.0 which passed production tests with positive results. The use of new driving mechanism of cutting device in construction of the mower KS-2.1J and two-bridged trailed mower KPD-4.0 ensures reliable operation of mowers on the uneven terrain of natural and sow hayfields, even when the outer shoe is more than 0.2 m above the inner shoe. The design of the drive will reduce the frictional force from the action of the lead and eliminate the pinching of the knife, thereby liquidating the breakage of details in cutting machine and increasing the operational reliability and extending the service life. The use of new crank-rocker mechanism with double stroke of the knife allows to reduce the crank rotation frequency by 1.5 times, increase the knife supply or productivity by 1.6 times, reduce the inertial forces by 1.2, ..., 1.3 times and respectively reduce the energy intensity of process of mowing herbs.

RECOMMENDATIONS

Possible prospects for further research on the subject are to carry out work to reduce the oscillations in the double stroke of the knife; a partial reduction in oscillations was achieved by installing balancing counter weights.

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