



THEORETICAL STUDY OF THE CHANGE OF THE TORQUE ON THE SHAFT OF THE FEEDING ROTOR OF THE SHREDDING MACHINE DURING THE CAPTURE AND COMPRESSION OF THE GRASS MASS

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Abstract

The shredding machine is any forage harvester's main and most energy-intensive unit. Its structural design determines the machine's technological scheme and the main units' location. Disc and drum shredders, which are equipped with trailed, haulage, and self-propelled forage harvesters, have become widespread. Biter-knife shredding machines are installed on pick-up trucks and balers.

In general, the energy spent on grinding the grass mass includes components for grinding (compression, cutting, friction), transportation of the leaf-stem mass, as well as costs for the idling of the working bodies. A characteristic feature of the biter-knife apparatus under the research is the use of active disc knives that cut a portion of the leaf-stem mass. Therefore, for an objective assessment of the specific energy intensity of the process of grinding the grass mass, it is relevant to determine the forces and moment on the shaft of the feeding rotor of the shredding machine at the stage of capturing and compressing the grass mass.

The object of the research is the process of grasping and squeezing with the fingers of the feeding rotor a portion of the grass mass in the forming channel of the biter-knife shredding machine.

The purpose of the research is to establish the analytical dependence of the change in the torque on the shaft of the feeding rotor of the shredding machine on the forces of grasping and compressing the grass mass.

The objectives of the work are to perform an analysis of the design features of biter-knife shredding machines, perform an analysis of research and publications on the creation or change of torque (power, energy intensity) on the shaft of the disk, drum, rotor of the shredding machine of forage harvesters, establish the change in effort and torque on the shaft of feeding rotor of the biter-knife shredding machine when capturing and compressing the leaf-stem mass.

The research methodology is based on the analysis of the interaction of the fingers of the feeding rotor with a portion of the captured grass mass in the forming channel of the shredding machine using the methods of theoretical mechanics.

The scientific work considers the process of capturing, compressing the mass, and feeding it into the annular channel by the fingers of the feeding rotor of the shredding machine. The dependence (4) of the horizontal component of the rotation speed of the rotor finger on the angle of its rotation is established. It is noted that the operation of the feeding rotor is possible in three modes, namely, mass loading, intermee, and mass flow stretching. Based on the reliability of the technological process, the mode of operation of the feeding rotor with stretching of the mass flow is more appropriate.

Analytical dependences on the calculation of forces and torques on the shaft of the feeding rotor of the biter-knife shredding machine in such zones as capture, compression of the material (18, 30), its stretching (22, 31), and acceleration into the cutting zone (28, 32) are given. The above dependencies take into account technological, structural, and kinematic factors, as well as the physical and mechanical properties of stem materials. It was found that under the given conditions, the maximum torque on the shaft of the feeding rotor is 132 N·m at an angle of rotation of the beater from 30° to 40°.

Keywords: *forces, torque, grinding, cutting, shredding machine, energy intensity, feeding rotor, finger, biter, mass capture, mass compressing, grass mass.*

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1. Formulation of the problem

The shredding machine is the main and most energy-intensive working unit of the forage harvester. Its structural execution determines its technological scheme and the location of the main nodes.

The majority of high-performance forage harvesters (more than 75%) use a drum shredder, which includes a cutting drum covered by a casing, a counter-cutting plate, and feeding rollers ("Polesse" FS8060, Don 680M, BIG X 500, Jaguar V-Max 930, John Deere 8000). Their advantage is a large moment of inertia, as well as relatively high stability of the work process and increased throughput. Disadvantages include the significant energy consumption of grinding and overspending of energy to create a powerful airflow.

In addition to drum shredders, manufacturers produce machines with disk shredders. Such device is mainly installed on trailed and semi-mounted forage harvesters, as well as machines that are aggregated with universal energy means, where their use is determined by the design of the harvester as a whole. Examples of such machines are combine harvesters of the "Champion" model of the Kemper company, KPK 3000 of the K-G-6 "Polissya-250" complex, and KDP-3000 of the "Gomsilmash" production. Disadvantages of disc shredders include the limitation of the width of the loading mouth of the device by the radius of the disc, which prevents its use on the high-performance machine,s and the uneven load on the disc shaft.

Nowadays, biter-knife shredding machines, which perform multi-plane cutting at a speed of 4-8 m/s, have become widely used. Such machines are used in the construction and technological schemes of balers and pick-up trucks of such well-known foreign companies as Pottinger, Mengele, Taarup (Denmark), Far, Claas, Krone, Deutz Fahr (Germany), New Holland, Case, John Deere (USA), etc. These machines are close to each other in general layout and differ only in the design of individual units.

When evaluating or comparing the shredding machines of stalk fodder of forage harvesting machines, their technical and economic characteristics are of great importance, in which energy indicators are the most informative and objective. Most often, the indicator of specific energy intensity (E_n , kJ·h/kg, kJ·h/m³) is used, which is determined by the ratio of the specific total work (A_{zag} , kJ/kg) spent on cutting the grass mass to the productivity of the shredder (A_{gen} , kJ/kg). The energy intensity of the cutting process depends on many factors, such as physical and mechanical properties, methods of grinding, the condition of the working organs of the machine, etc.

In general, the energy spent on grinding the grass mass includes work expenses for grinding (grabbing, squeezing, cutting, friction), transportation along the channel and feeding the leaf-stem mass into the body or press chamber, as well as work expenses for idling the working organs [1, 2]. A characteristic feature of the biter-knife apparatus [3] is that to capture, compress, cut a portion of the leaf-stem mass and pull it through the forming channel, it is necessary to create a sufficient torque on the feeding rotor of the chopping apparatus. Therefore, for an objective assessment of the specific energy intensity of the grass mass grinding process, it is necessary to establish the moment from all the component forces of the feeding rotor of the shredding machine.

The purpose of the research is to establish the analytical dependence of the change in torque on the shaft of the feeding rotor of the shredder on the forces of grasping and compressing the grass mass.

To achieve the objective, the following tasks must be accomplished: analysis of the design features of biter-knife shredders, analysis of the research and publications on the creation or change of torque (power, energy intensity) on the shaft of the disk, drum, or rotor of the shredding machine of forage harvesters, establish the change in torque of the moment on the shaft of the feeding rotor of the biter-knife shredder when capturing and compressing the leaf-stem mass.

The object of the research is the process of grasping and squeezing a portion of the grass mass in the forming channel of the biter-knife shredding device with the fingers of the feeding rotor.

2. Study Subject Analysis

The main working body of the biter-knife shredding machine of multi-plane cutting is the feeding rotor and the cutting mechanism. The feeding rotor can be made in the form of a chain-finger conveyor (Fig. 1), an eccentric reel with a controled, a controlled rotor,

The use of a chain-finger conveyor (Fig. 1, a), obliquely located above the pick-up in combination with knives with a straight blade, installed at an angle to the direction of movement, allows both to feed the mass to a given height and to cut it at the same time. However, this design, due to the appearance of contradictions between the reduction of the cutting length and the increasing chain pulling effort at the same time, did not find further development.

The rake feeding device (Fig. 1, b) is made in the form of a double eccentric reel with rakes, one of the ends of which contains brackets with rollers that move along guides of a special shape. The

possibility of directional action controlled using the guide track of the rake allows to reliably capture the mass in the pick-up area and push it by pressing to the height of the vehicle body. The curvilinear movement trajectory of the base of the rake to ensure sliding cutting contributed to the use of flat knives with a sickle-shaped blade. In addition, in such a mechanism, it is possible to install a larger number of knives, which allows for reducing the cutting length of the grass mass.

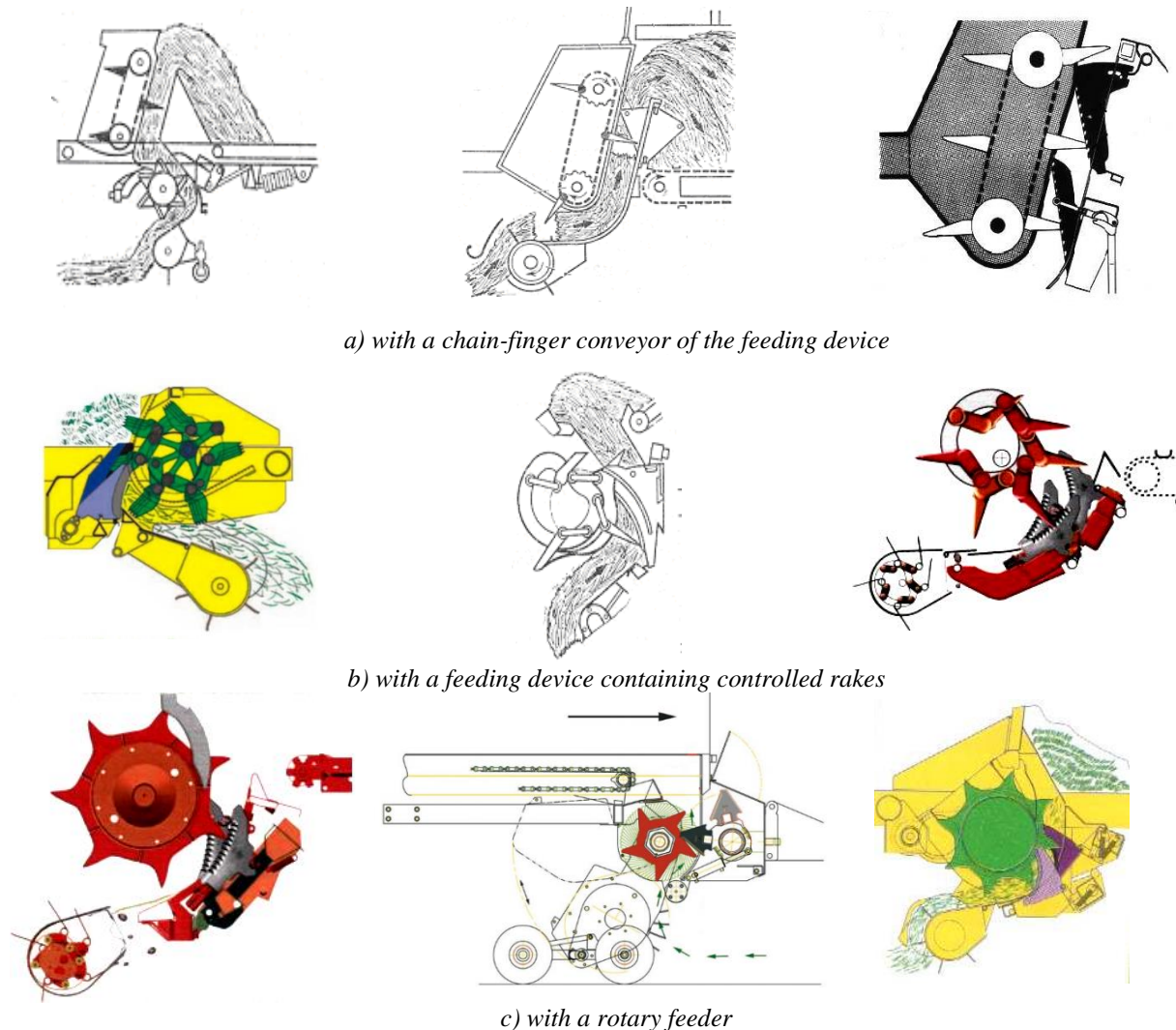


Fig. 1. Design and technological schemes of biter-knife shredding machines

The Krone company uses a rake feeding device (Fig. 2) with a grip width of 1.47 m with 4 and 6 controlled rakes, which are offset relative to each other along the device. Double fingers (teeth) with radial displacement are placed on each of the rakes, forming two sections. The central control system from Krone allows the use of knives number 0, 4, 9, 18, and 35, and therefore quick and easy adjustment of the cutting length. The theoretical cutting length with 18 engaged knives reaches 80 mm, with 35 – 40 mm. The Titan cutting mechanism contains passive plate knives arranged in two rows [4].

Similar in design, shredding machines are used in pick-up trucks of the Boss junior, Boss LT (Fig. 3), Euroboss, and Primo models of Pottinger (Austria) [5].

The rake feeders of the shredding machines of the specified pick-up trucks are high-performance devices, however, taking into account the complexity of its design, metal consumption, pulsating power loads, rapid wear of nodes moving along the treadmills, the rotary feeder has become widely used in modern designs of carts-pickup-trucks (Fig. 1, b).



Fig. 2. Scheme of the shredding machine with controlled rakes of Titan pick-up trucks



Fig. 3. View of the Supermatic feeding device of the Boss junior and Boss LT pick-up trucks

Machines of different foreign companies, based on the same functional purpose, are close to each other in general layout and differ only in the construction of individual nodes and aggregates. They include a frame, a trailer, a chassis, a pickup, a shredder, a body, and an unloading device, which can additionally contain a dosing device for unloading feed in the feeder. Ukraine doesn't manufacture pickup trucks.

In modern models, a simpler and more reliable rotary feeder is used (Fig. 1, b). This is a cylindrical drum on the surface of which curved fingers are fixed, thus forming a star-like shape. The number of fingers around the drum varies from 2 to 9, depending on the diameter. This design of the feeder allows it to be used not only in pick-up trucks but also in balers. In them, the stalk mass is picked up from the swath by a pick-up and fed into the rotation zone of the fingers of the rotary feeder. It is grasped by a pair of fingers and stretched along the forming channel by plate knives. Next, the crushed stem mass is pushed into the pressing chamber of the baler or the trailer of the pick-up truck.

The considered shredders with a rotary feeder of foreign balers and pick-up trucks of companies such as Pottinger, Claas, Krone, Deutz Fahr [5-9] and others are similar in general performance and differ only in the design features of the nodes and working bodies. Thus, their rotary feeder (Fig. 4) differs in length and diameter, the number of rows of fingers, their design (length of the finger and the shape of the line of the working surface), and the method of attachment to the drum. For most models, the rotor contains a built-in reverse mechanism, which is used in case of clogging.

As for the grinding mechanism, in all models, without exception, the knives are lamellar and installed in one or two rows (Fig. 5, 6). Arc-shaped knives have individual protection against breakage in the case of foreign objects [7].

The Claas company (Germany) uses the Roto Cut and Fine Cut shredders (Fig. 4) when harvesting hay and haylage in the Variant, Rollant, and Quadrant series balers, the feed rotor of which contains 4 rows of pairs of spirally installed fingers and works in pairs with 14 and 16 knives (for Variant and Rollant) and 6, 13, 25, and 49 knives for Quadrant. The Fine Cut system, which is equipped with a cutting mechanism with 49 knives, has an estimated cutting length of 20 mm [7].

The feeding rotor of the shredding device of the Quadrant 3400RC baler (Fig. 4) with a length of 1.3 m and a diameter of 86 cm is the largest offered on the market [7]. With 9 rows of fingers, it allows for performing about 28,000 cuts per minute with a cutting length of 45 mm.

The feeding device of models AX, MX, and ZX of Krone pick-up trucks is made in the form of a rotor with a diameter of 760 mm and 880 mm, respectively (Fig. 4), with a working grip length of 1.57 m (for AX models), 1.64 m (for MX models) and 1.84 m (for ZX models), driven by a closed gearbox. For all models (AX, MX, and ZX), the rotor has 8 gripping teeth, each subsequent row of which is offset from the previous one by 2.7°, 2.1°, and 1.9°, respectively, forming a spiral arrangement of them (Fig. 7) [11].

As for Pottinger pick-up trucks, they use Rotomatik, Euromatik, and Powermatik rotary feeders (Fig. 4) [5] on models Faro, Europrofi, Torro, Jumbo, and Jumbo Combiline [12]. In general, they are similar to each other and differ only in structural design (Fig. 8).

The Malone company (Ireland) produces five models of Trojan MT pick-up trucks [13] with trailer capacities from 35 to 62 m³. The rotary feeder, which contains 5 rows of fingers with a thickness of 6 mm, interacts with the cutting mechanism, which includes 35 and 44 plate knives. This design of the shredding machine allows to obtain chopped grass mass of 35 and 40 mm.

The Ros company (Italy) offers pick-up trucks of the CT series with a trailer capacity of 38, 48, and 60 m³, which contain a shredding machine with a rotary feeding device [14]. A distinctive feature of the

device is that the flow (transportation) of the grass mass from the pick-up occurs through the front part of the rotor. The rotor with a working length of 2.0 m contains 5 rows of fingers installed in pairs on the drum with subsequent displacement relative to each other, forming a spiral. The fingers interact with the cutting mechanism, which has 18 plate knives arranged in one row, which have the shape of segments allowing the use of oblique cutting.



View of the feeding rotor of the Roto Cut baler Variant 380 RC by Claas



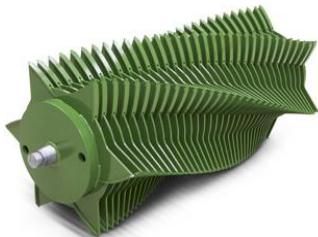
View of the Fine Cut feeder rotor of Quadrant 3200 FC baler by Claas



View of the feeding rotor of the Quadrant 3400 RC baler by Claas



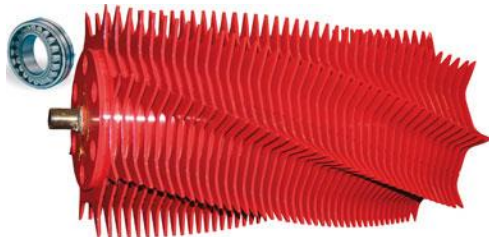
View of the feeding rotor of the Multi Cut baler Round Pack 1250, 1550 by Krone



A view of the AX pick-up truck feeding rotor by Krone



General view of the feeding rotor of the CropCutter of the BigBaler 1290 baler by New Holland



General view of the Powermatic feeding rotor of the Jumbo pick-up truck by Pottinger



View of the feeding rotor of the Cargos 9400 pick-up truck by Claas

Fig. 4. Variants of execution of feeding rotors of balers and pick-up trucks



Fig. 5. Torro 5100 LD pick-up truck cutting mechanism by Pottinger

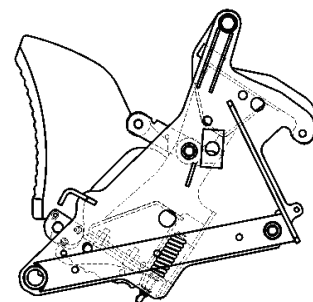


Fig. 6. Scheme of the grinding mechanism

The advantage of the considered constructive and technological scheme of pick-up trucks is the removal of the pick-up truck to the transport position under the trailer body (Fig. 1, b), which facilitates convenient movement and movement of the unit over bumps.

Disadvantages of the shredding machines of these machines are caused by energy losses, which are caused by the size of the cutting force and the friction of the grass mass on the side surfaces of the flat knives. The blades of such knives have a curved shape, and the cutting edge is located on the inner surface of the knife. Such execution of knives makes it difficult to maintain the sharpness of the blade during operation.



Fig. 7. Location of teeth on AX (25 mm) and MX (17 mm) pick-up trucks by Krone

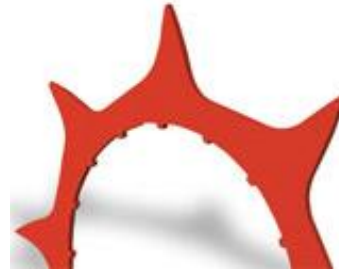


Fig. 8. The toothed disk of the Rotomatic feed rotor of pick-up trucks by Pottinger

The interaction of the working bodies of the machines with the grass mass, the quality, and the energy consumption of the grinding process largely depends on its physical and mechanical properties. Thus, the specific energy consumption for grinding stem fodder with modern grinding devices, depending on their moisture content and the degree of grinding, ranges from 7 to 15 kW per ton of mass.

The theory of blade cutting was developed by academician V.P. Horyachkin. It found its further development in the works of such outstanding scientists as V.A. Zhelihovskiy, N.E. Reznik, S.V. Melnykov, A.I. Zavrashnov, D.I. Nikolaiev, V.A. Ziablov, H.I., Novikov, S.A. Prytchenko, V.I. Osobov, S.F. Kolesnikov, L.O. Afanasiev and others. Based on their theoretical developments, high-performance shredding machines of forage harvesters were created.

It is known that the sliding cutting process can take place at different ratios of normal P_n and tangential P_t cutting forces. This ratio is greatly influenced by the design parameters of the cutting pair and its technical condition during operation. Such parameters include the sharpening angle, blade sharpness, cutting angle, knife installation angle, as well as the gap between the knife blade and the cutting plate.

At the same time, it is difficult to achieve the minimum expenses of specific work A_n , cutting effort P_{cut} or energy intensity of the process [1, 2, 15-17]. If it is necessary to spend the least cutting effort, the work has to be done with a smooth blade, when there is a need for reduced specific work expenses, then the blade should be rough, the author emphasizes, and the cutting effort should not exceed 2/3 of its value for a given sharpening angle. When cutting with sliding, the angle of sharpening should be chosen less than twice the angle of friction of the product against the material of the knife.

The works of N.V. Sablikov, M.P. Horbun, N.Y. Reznik, V.A. Zyablov, P.I. Prokoptsev, D.S. Shvets, V.I. Arnautov, I.L. Gaba. Studies confirm a significant reduction in the net forces of resistance to the cutting of the studied materials with an increase in the sliding angle, and, therefore, the energy intensity of the cutting process.

The most fundamental dependence of determining the specific cutting force P_{cut} was proposed by N. Y. Reznik [1], which combines the relationship between the main physical, mechanical and technological parameters of the stem mass being cut and the design parameters of the cutting pair.

Works [18, 19, 20] are devoted to the study of the energy consumption of the process of cutting thick-stemmed plants. Karp N.A. [20] states that the cutting work decreases when the angle of the knife is reduced, that is when moving from a transverse to a longitudinal cut. In his opinion, an oblique cut makes it possible to cut with small amounts of linear movements of the knife with a small amount of normal force, that is, to have a gain both in the amount of normal force and in the amount of cutting work.

In the process of cutting, specific energy consumption, as noted by V.O. Zhelihovskiy [21], with an increase in the sliding angle τ grows slowly at first, and when it reaches 70-75 °(1,23-1,31 rad), it grows

much more intensively. N.V. Sablikov established that increasing the sliding coefficient at the initial stage is beneficial in terms of energy [22].

Investigating the interaction of the knife blade with the material being cut, it was established that cutting stretched fibrous material significantly reduces energy consumption [23].

In [24], the authors proposed a cutting device that provides sliding longitudinal and transverse cutting of the material flow with minimal dynamic loads, a more stable set particle size of the crushed material, and neutralizes axial loads on the drum supports. A comparative calculation of the drive power of the proposed shredding machine with a traditional drum cutting device is given.

O.F. Hovorov proposed an analytical dependence [25] for determining the power on the drive of the cutting device of the mower-shredder-loader of green fodder, which is considered as the sum of two powers, namely, the power to cut the stems of plants into particles of a given length and the power to overcome the necessary speed of the chopped fodder, which is determined by the height mass line of such a machine.

The reduction of energy consumption is facilitated by the development of shredding machines with an active counter-cutting part [26-29]. Such machines allow cutting stems with an inclined or sliding cut, which, in turn, contributes to reducing energy consumption and improving the quality of cutting the stem layer.

Currently, it is not possible to determine the universal dependence of the cutting force or work on the shredding of leaf stem mass. Therefore, in most cases, energy consumption for each shredder must be determined individually, taking into account the method and degree of grinding, its design features, and material properties.

So, based on the analysis of the results of theoretical and experimental studies of the technological process of grinding by cutting, it is possible to note a sufficiently versatile study of the process of cutting stem fodder. Therefore, the research and establishment of the analytical dependence of the change in the torque on the shaft of the feeding rotor of the biter-knife shredding machine on the efforts of grasping and compressing the grass mass for the further establishment of the energy intensity of the process of grinding by cutting the leaf-stem mass is relevant and deserves further study and improvement.

3. Setting out the Basic Material

Lifting a linear meter of the swath with a mass q (kg/m), the pick-up at a speed V directs the mass with a height H to the receiving mouth of the shredding machine. During the operation of the machine, the fingers of the rotor (beater) capture and partially stretch the mass, as the circular speed of the fingers is greater than the speed of the mass flow. At the same time, the mass is compressed from the thickness H to the size of the annular channel formed by the tray and drum of the feeding rotor. Dragging of the mass in the channel is prevented by frictional forces on the surface of the drum tray. Upon reaching the knives installed in the circular channel in one or more rows, the mass is compressed to a force that allows cutting a portion of the stem mass. After cutting a portion of the mass, the channel smoothly moves away from the center of rotation of the beater, and the fingers of the beater exit from it through slots in the wall of the channel. Thus, the components of the cutting process with a biter-knife shredding machine are:

- capturing, compressing the mass, and feeding it into the ring channel;
- transportation of mass along the channel to disk knives;
- mass cutting;
- pushing the cut mass through the channel.

Consider the process of capturing, compressing the mass, and feeding it into the ring channel with the fingers of the feeding rotor of the grinding machine (Fig. 9). The X -axis of the coordinate system passes parallel to the direction of mass supply (horizontal direction) through the center of the O -axis of the beater drum, and the Y -axis is perpendicular to it and directed vertically upwards.

We assume that:

- the beater of the shredding machine with a drum with a radius of R_b and a disc knife with a radius of r_d rotates uniformly in the directions indicated in Figure 9;
- the flow of stem mass with a thickness of H is supplied parallel to the X axis at a speed of V_f ;
- the finger of the drum captures and moves part of the mass flow with a thickness h_{ch} and a width corresponding to the number of fingers in a row and a length corresponding to the number of rows of fingers on the drum.

The action of adjacent fingers does not affect the operation of a single finger of the feeding rotor.

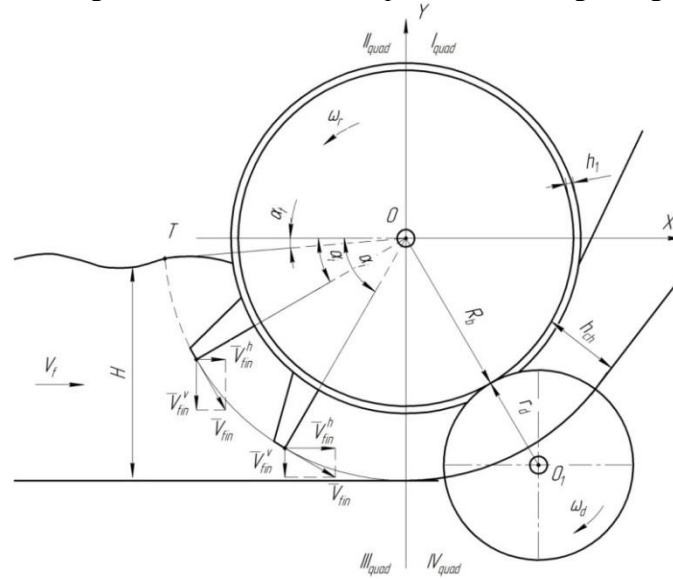


Fig. 9. Scheme of the shredding machine for determining the torque on the shaft of the feeding rotor

Analyzing Figure 9, we can see that the interaction of the fingers with the mass occurs in the *III* and *IV* quadrants of the coordinate system, and in the *III* quadrant, the mass is grasped and compressed, and in *IV*, it is pushed through and directly cut. Taking into account the angle of rotation of the feeding rotor (beater) from the negative branch of the *X* axis and based on the dimensions of the shredding machine and the height of the roll, the value of the angle of the beginning of interaction α_t is determined by the dependence:

$$\sin \alpha_t = \frac{R_b + h_{ch} - H}{R_b + h_{ch}},$$

$$\alpha_t = \arcsin \left(1 - \frac{H}{R_b + h_{ch}} \right), \quad (1)$$

where R_b – radius of the drum, m;
 h_{ch} – the height of the ring channel, m;
 H – the height of the mass layer, m.

After the starting moment (t_b) of the interaction with the stem mass, the rolls with a height of H are compressed to the size h_{ch} - the height of the ring channel. The compression process is completed when the finger is turned to an angle $\alpha = 90^\circ$ (1,58 rad). The relative deformation ε of the material at the same speeds of its feeding V_f and movement by the fingers of the beater is

$$\varepsilon = \frac{H - h_{ch}}{H}. \quad (2)$$

However, because the material enters the shredding machine in a translational way, and the rotor finger performs a rotational movement, the horizontal component of the speed of the end of the finger changes its speed according to a known dependence

$$V_{fin}^h = \omega_r (R_b + h_{ch}) \sin \alpha, \text{ m/s} \quad (3)$$

A graphic illustration of dependence (3) is presented in Figure 10.

We see that at α_t the horizontal speed will have a minimum value $V_{fin}^h = \omega_r (R_b + h_{ch}) \sin \alpha_t$, and at an angle $\alpha = \pi/2$ - a maximum value $V_{fin}^h = \omega_r (R_b + h_{ch})$. The increase in speed occurs according to the sine law, and the average value of the speed will be determined from the dependence

$$V_{fin}^h \text{ mid} = \int_{\alpha_t}^{\pi/2} \omega_r (R_b + h_{ch}) \sin \alpha d\alpha = \omega_r (R_b + h_{ch}) \int_{\alpha_t}^{\pi/2} \sin \alpha d\alpha = \frac{-\omega_r (R_b + h_{ch}) \cos \alpha}{\frac{\pi}{2} - \alpha_t} \Big|_{\alpha_t}^{\pi/2} = \omega_r (R_b + h_{ch}) \frac{\cos \alpha_t}{\frac{\pi}{2} - \alpha_t}, \text{ m/s} \quad (4)$$

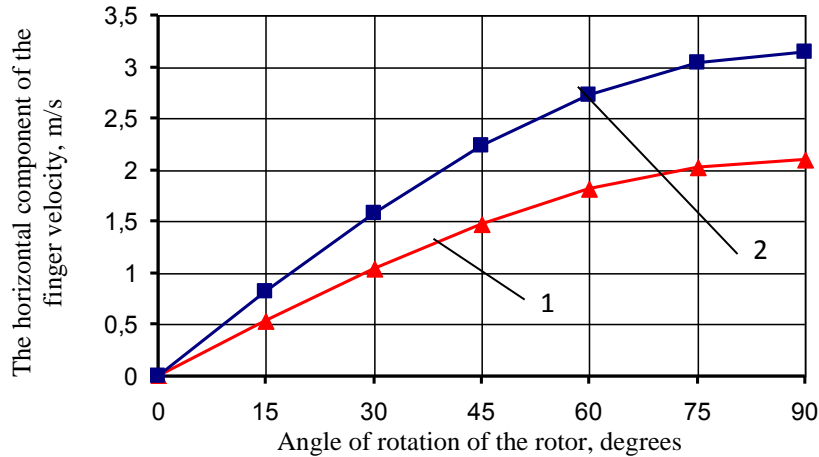


Fig. 10. Dependence of the horizontal component of the rotation speed of the rotor finger on the angle of its rotation at $\omega_r = 10,5 \text{ s}^{-1}$; $h_n = 0,1 \text{ m}$: 1 – drum radius $R_b = 0,1 \text{ m}$; 2 – drum radius $R_b = 0,2 \text{ m}$

Thus, the average speed is determined by the maximum value of the circular speed of the finger $\omega_r(R_b+h_{ch})$ and the angle α_t . The value of the angle α_t depends on the ratio between the dimensions of the drum (R_b+h_{ch}) and the height H of the layer of mass fed into the grinding machine and is determined by dependence (1). For the normal operation of the shredder, it is necessary that the condition $(R_b+h_{ch}) \geq H$ is fulfilled.

The average speed is not directly affected by the value of the angle α_t , but its function $\cos \alpha_t / (\pi/2 - \alpha_t)$, a graphic illustration of which is given below (Fig. 11).

We see that in the range from 0° (0 rad) to 60° (1,05 rad) the function increases from the value of 0,6369 to the value of 0,9555 according to a dependence close to the straight line, and the intensity of the increase decreases with the increase of the angle α_t .

Thus, in the horizontal direction, the stem mass is moved by the feeding rotor (beater) with a variable speed that increases from the minimum value

$$V_{fin\ min}^h = \omega_r(R_b + h_{ch}) \sin \alpha_t \quad (5)$$

to the maximum value

$$V_{fin\ max}^h = \omega_r(R_b + h_{ch}), \quad (6)$$

and the average value of the speed will be determined by dependence (4).

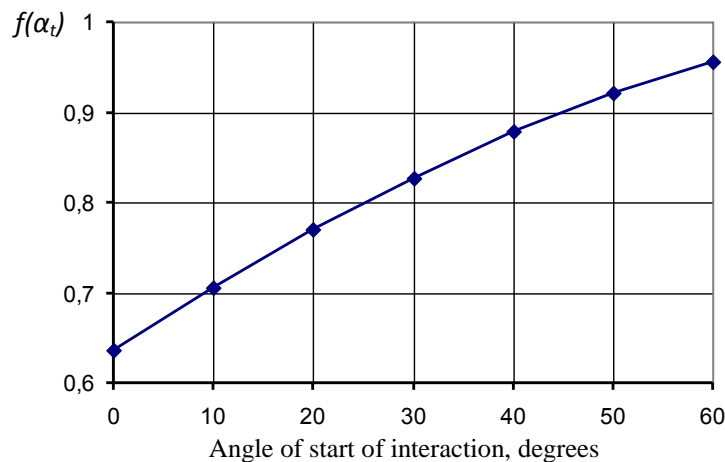


Fig. 11. Dependence of the function change of the interaction start angle on the interaction angle

The value of the average speed the beater's finger will reach when turning to the angle α_{avsp} which is determined by the condition

$$\omega_r(R_b + h_{ch}) \sin \alpha_{avsp} = \omega_r(R_b + h_{ch}) \frac{\cos \alpha_t}{\frac{\pi}{2} - \alpha_t},$$

$$\sin \alpha_{avsp} = \frac{\cos \alpha_t}{\frac{\pi}{2} - \alpha_t}.$$

At the same time, the value of the angle of rotation of the rotor will be

$$\alpha_{avsp} = \arcsin \frac{\cos \alpha_t}{\frac{\pi}{2} - \alpha_t} \quad (7)$$

where α_{avsp} – the angle of rotation of the rotor at which the value of the average horizontal speed is reached.

Graphical interpretation of dependence (7) is presented in Fig. 12.

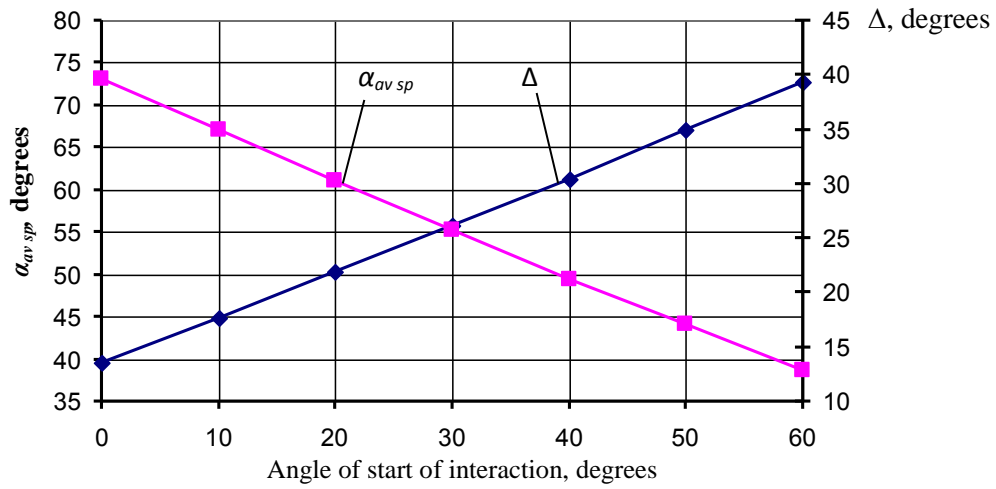


Fig. 12. Dependencies of the angle of reaching the average value of the horizontal component of velocity and the angle of rotation on the angle of the start of interaction: α_{avsp} – the angle of rotor's rotation at which the value of the average horizontal speed is reached, degrees; Δ - the difference between the average value of the angle and the initial value, degrees

The dependence between the starting angle of the interaction with the mass α_t and the angle of the rotor's rotation before reaching the average value of the horizontal component of the velocity α_{avsp} has the character of linear dependence. Having chosen the parameters of the beater in such a way that the ratio is fulfilled, i.e. the equality of the horizontal component of the average speed and the feed speed V_f , uniform movement of the mass will occur. Until this moment is reached, that is $V_f > V_{fin\ av}^h$, when the stem mass will be loaded (accumulated) in front of the feeding rotor, and under the condition, the material will be stretched by it.

Thus, the operation of the feeding rotor (beater) is possible in three modes, the main characteristics of which are presented in Table 1.

Table 1. Modes of operation of the feeding rotor when feeding material

№	Modes of operation	Working condition
1	with mass loading	$V_f \geq V_{fin\ max}^h = \omega_r(R_b + h_{ch})$
2	intermediate	$V_{fin\ min}^h < V_f < V_{fin\ max}^h$
3	with mass flow stretching	$V_f \leq V_{fin\ min}^h = \omega_r(R_b + h_{ch}) \sin \alpha_t$

One of the variants of the intermediate mode of operation is operation under the condition $V_{fin\ av}^h = V_f$. Based on the conditions of the reliability of the technological process, the third option of the feeding rotor operation, with stretching of the mass flow, is more appropriate. The worst option

should be considered the first option, in which the mass is only compressed by the beater, during such work it does not contribute to the uniform passage of the mass, while at the same time creating the prerequisites for slowing down (accumulation) of the mass, the formation of unevenness in its supply.

Not only the speed of movement of the mass flow but also the availability of space for its placement affects the continuous supply of mass. The rotor, rotating with an angular speed ω_r and rotating through an angle $(\pi/2-\alpha_t)$ from a roll with a height close to H_{max} , takes the mass at a distance equal to the length of a finger h_{ch} . We write

$$\frac{V_f(\frac{\pi}{2}-\alpha_t)}{\omega_r} \leq h_{ch}. \quad (8)$$

When working with a smaller thickness of the mass layer, the inequality will look like this

$$\frac{V_f(\frac{\pi}{2}-\alpha_t)}{\omega_r} \leq \frac{h_{ch}}{\cos \alpha_t}. \quad (9)$$

Dependence (9) is determined from the condition of having one finger on the drum sector $(\pi/2-\alpha_t)$. However, having the number of rows of fingers i on the beater, the latter will pass one after the other with the angle $2\pi/i$ and on the sector $(\pi/2-\alpha_t)$ will be located $(\pi/2-\alpha_t)/2\pi/i = i(1/4-\alpha_t/2\pi)$ fingers. In this case, inequality (9) is written down

$$\frac{V_f(\frac{\pi}{2}-\alpha_t)}{\omega_r} \leq \frac{h_{ch} \cdot i \cdot (0.25 - \frac{\alpha_t}{2\pi})}{\cos \alpha_t}, \quad (10)$$

That is,

$$h_{ch} \geq \frac{V_f(\frac{\pi}{2}-\alpha_t) \cos \alpha_t}{\omega_r \cdot i \cdot (0.25 - \frac{\alpha_t}{2\pi})}.$$

Figure 13 illustrates the change in the minimum required finger length h_{ch} depending on the angular speed of the rotor and the mass supply speed V_f for different values of α_t .

Thus, in the zone of mass capture (quadrant III, Fig. 9), both loading and stretching of the mass can occur, depending on the mode of operation. It should be noted that stretching of the captured portion will lead to both an increase in the portion of the captured mass and to its additional acceleration, which will cause a change in the force on the beater's finger. This force acts on the finger in the direction of the X axis (Fig. 9). At the same time, a force acting on the finger is caused by the compression of the mass in the vertical direction, which is applied by the finger.

Let's construct a diagram of the torque M_{top} on the shaft of the feeding rotor from the time of the start of the interaction of its finger with the stem mass. For this, we take the middle of the finger (point A_o) as the place of application of effort (Fig. 14). Let's put the main components of efforts and determine their values.

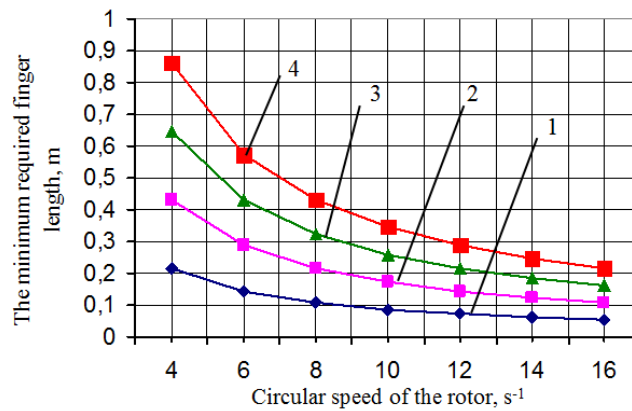


Fig. 13. Change in the minimum required length of the finger depending on the angular speed of the beater at the angle of the start of interaction $\alpha_n = 30^\circ$: the number of rows of fingers of the beater $i = 7$; 1 - mass supply speed of 1,11 m/s; 2 - 2,22 m/s; 3 - 3,33 m/s and 4 - 4,44 m/s

At the middle of the finger (point A_o) at an angle α to the perpendicular to the working radially directed line of the finger (Fig. 14) parallel to the Y axis, the compressive force of the stem mass F_{comp} is directed. To determine the relative deformation ε of the mass (dependency (2)), by which we mean the ratio of the absolute deformation to the initial thickness H of the layer [30], we determine the value of the absolute deformation. At the same time, it should be taken into account that at the initial stage, the stem mass will be compressed by the edge of the finger (point A), and when the rotor is rotated, the finger will sink into the mass and the latter will be compressed by the surface of the rotor drum. Based on this, we calculate relative to point A_1 , which lies at the base of the finger.

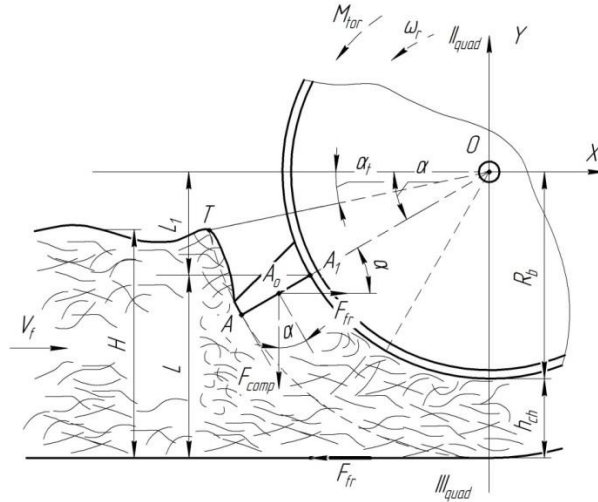


Fig. 14. The scheme for determining the torque on the shaft of the feeding rotor at the stage of mass supply

In the process of turning the rotor, the number of absolute deformation changes from 0 (at $\alpha = \alpha_t$) to $(H-h_{ch})$ at $\alpha = 270^\circ$ (4,73 rad). The amount of deformation is expressed in terms of the angle of the rotor's rotation α , the thickness of the initial layer of mass H , and the dimensions of the rotor R_b and h_{ch} .

The distance L_1 from point A_1 to the horizontal axis X will be determined:

$$L_1 = R_b \sin \alpha, \quad (11)$$

where R_b – radius of the drum, m;
 α - rotor rotation angle, degrees.

Knowing the value of L_1 , we determine the thickness L of the compressed material

$$L = (R_b + h_{ch}) - R_b \sin \alpha = h_{ch} + R_b(1 - \sin \alpha). \quad (12)$$

With the known thickness of the compressed material layer and the initial thickness of the material layer, we determine the relative deformation

$$\varepsilon_1 = \frac{H - (h_{ch} + R_b(1 - \sin \alpha))}{H} = 1 - \frac{h_{ch} + R_b(1 - \sin \alpha)}{H}. \quad (13)$$

Since the feed rate to ensure the technological reliability of the process is lower than the speed of mass transport by the rotor, the deformation value ε of the mass will be slightly lower due to its stretching, i.e.:

$$\varepsilon = \varepsilon_1 \cdot k, \quad (14)$$

where ε - the amount of relative deformation, taking into account the speeds of feeding and transportation;

ε_1 - the amount of relative deformation without taking into account the speeds of feeding and transportation;

k - the coefficient that takes into account the decrease in the value of ε . $k = V_f / V_{fin}^h$.

phenomenon occurs under the condition that the horizontal component of the speed of the finger exceeds the speed of supply V_f of the mass layer, i.e.

$$V_{fin}^h = \omega_r(R_b + 0,5h_{ch}) \sin \alpha \geq V_f. \quad (19)$$

Thus, the angle of rotation of the feeding rotor at which the stretching of the mass begins will be

$$\sin \alpha = \frac{V_f^h}{\omega_r(R_b+0,5h_{ch})}. \quad (20)$$

We see that the size of the angle depends on the ratio of feed speeds V_f and the circular speed of the finger $\omega_r(R_b+0,5h_{ch})$. If $V_f = 0,5\omega_r(R_b+0,5h_{ch})$, then the angle α will be equal to 30° (0,53 rad), if $V_f = 0,5\sqrt{2}\omega_r(R_b + 0,5h_{ch})$, then 45° (0,79 rad).

An interesting option is when $V_f = V_{fin}^h$. In this case, at the initial contact points, the mass is slowed down and loaded, and upon reaching the horizontal component of the beater finger of the feed speed V_f , the mass is stretched, and the flow will be stretched by as much as it was previously loaded.

The force that occurs when the mass is stretched is formed by the adhesion between individual plant elements, and by friction during mutual movement. According to [32], the main factors affecting the amount of force are the thickness of the material layer (with an increase in the layer thickness, the shear force decreases) and the density of the material (an increase in density leads to an increase in the shear force) (Fig. 16).

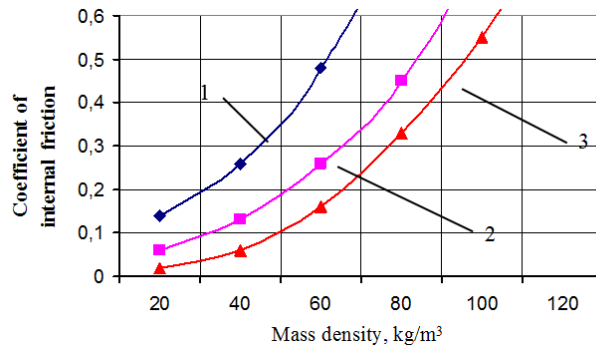


Fig. 16. Dependence of the coefficient of internal friction on mass density and mass layer thickness: 1 – 50 mm; 2 – 100 mm; 3 – 300 mm

Based on the data [32], the dependence of the change in the coefficient of internal friction on the size of the gap between the moving surfaces and the density of the stem material was constructed

$$f_c = a_0 + a_1\gamma + a_2h + a_3\gamma^2 + a_4h^2 + a_5\gamma h. \quad (21)$$

Knowing this dependence and the magnitude of the force that occurs in the vertical direction (dependence (18)) on the plane on which one finger acts, we can determine the magnitude of the horizontal force that occurs on the finger

$F_{st} = F_{comp} \cdot f_c, N$	(22)
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The stretching force, as indicated earlier, begins to arise only if the horizontal component of the finger speed exceeds the speed of the feed of the stem mass (dependency (19)). In this case, the value of the angle of rotation of the beater α is determined by dependence (20), and the thickness of the mass layer L - by dependence (12).

The density γ_0 of the uncut mass fed to the shredding machine increases with the humidity increase and is 20-25 kg/m³ at a humidity of 17-20% and 30-35 kg/m³ at a humidity of 55-60 % [31, 33].

A decrease in the thickness of the mass layer leads to an increase in its density. Based on the invariance of the weight of the material interacting with each of the rotor fingers, the density γ of the material depending on the angle α of the rotor's rotation, its initial density γ_0 , and the height of the layer H will be

$$\gamma = \frac{H\gamma_o}{h_{ch} + R_b(1 - \sin \alpha)}. \quad (23)$$

Knowing the thickness of the layer (12) and its density (23), we can write the expression (22) as

$$F_{st} = F_{comp} \cdot f_c = \frac{E \cdot V_f \cdot l \cdot h_{ch} (H - h_{ch} - R_b(1 - \sin \alpha))}{H\omega_r(R_b + h_{ch}) \tan \alpha} * \\ * \left(a_o + a_1 \frac{H\gamma_o}{h_{ch} + R_b(1 - \sin \alpha)} + a_2 (h_{ch} - R_b(1 - \sin \alpha)) + a_3 \dots \right). \quad (24)$$

The force of friction that occurs during the movement of the flow of mass between the bottom of the grinding apparatus and the stem mass F_{fr} (16) resists the movement of the flow, but it is partially taken into account in the tensile force, so it is not considered in this model.

During the capture and subsequent transportation of the mass, there is an inertial force connected with an increase and change in the direction of the speed of movement.

At the angle of rotation of the rotor α_r , the point A_o in the middle of the finger moves tangentially (perpendicular to the radial finger) with the speed $V_a = \omega_r(R_b + 0,5h_{ch})$. The component of the feed speed V_f in this direction will have the speed $V_f \sin \alpha_r$. Thus, at the moment of contact, the increase in speed will be

$$\Delta V = V_a - V_f \sin \alpha = \omega_r(R_b + 0,5h_{ch}) - V_f \sin \alpha_t \quad (25)$$

The maximum value of the portion of the mass will be determined by the bulk weight of the mass γ_o and the volume with which one finger of the rotor interacts. With the distance along the width of the drum between the fingers l (Fig. 15) and along the length of the drum - a , and the height of the finger h_{ch} , the captured volume will be determined $V = a \cdot l \cdot h_{ch}$. Based on this, the mass of the portion will be determined according to a well-known dependence

$$M = \gamma_o \cdot a \cdot l \cdot h_{ch}, \text{ kg} \quad (26)$$

The time during which the portion will gain speed of the rotor finger will be determined by the time of passing the finger to the place of the previous one. Based on the fact that the number of rows of fingers along the length of the drum circle is i , the rotation time will be:

$$t = \frac{\Delta \alpha}{\omega_r} = \frac{2\pi}{\omega_r \cdot i}, \text{ s} \quad (26)$$

where $\Delta \alpha$ – the angle between two adjacent rows of fingers, degrees;

ω_r - angular speed of the drum (rotor), s^{-1} ;

i - number of rows of fingers along the length of the drum circle, pcs.

Knowing the time of acceleration of the mass and the magnitude of the change in speed, we determine the acceleration a_m

$$a_m = \frac{\Delta V}{t} = \frac{(\omega_r(R_b + 0.5h_{ch}) - V_f \sin \alpha) \cdot \omega_r \cdot i}{2\pi} = \frac{\omega_r \cdot i}{2\pi} * \\ * \left(\omega_r(R_b + 0.5h_{ch}) - V_f \left(1 - \frac{H}{R_b + h_{ch}} \right) \right), \text{ m/s}^2 \quad (27)$$

The force acting on the finger and creating the acceleration of the mass is determined by the formula

$$F_{ac} = M \cdot a_m = \frac{\omega_r \cdot i \cdot \gamma_o \cdot a \cdot l \cdot h_{ch}}{2\pi} \left(\omega_r(R_b + 0.5h_{ch}) - V_f \left(1 - \frac{H}{R_b + h_{ch}} \right) \right), \text{ N} \quad (28)$$

Having established the dependences that determine the magnitude of the forces and the direction of their action, we will construct the dependence of the change in the torque on the shaft of the feeding rotor, which is caused by the action of forces on one finger of the rotor. As an example, we consider a shredding machine with parameters $R_b = 0.20$ m; $h_{ch} = 0.10$ m; $H = 0.25$ m; $V_f = 3.5$ m/s; $\omega_r = 8.37$ c^{-1} ;

$l = 0.10$ m ($i = 7$); $\gamma_o = 30$ kg/m³; $E = 120 \cdot 10^3$ N/m² (according to interpolation) [30] (average value of the deformation modulus at a density of up to 35 kg/m³ and a humidity of 54.2%); $a = 0.180$ m; $\Delta\alpha = 2\pi/7 = 51,4^\circ$.

The torque from the compressive force of mass M_l will be determined by the formula:

$$M_1 = F_{comp}(R_b + 0.5h_{ch}) \cos \alpha, \text{ N}\cdot\text{m} \quad (29)$$

where $(R_b + h_{ch})$ – shoulder of force application, m. In reality, the shoulder of force application is somewhat smaller - $(R_b + 0,5h_{ch})$, however, since the application of force at point A of the finger was considered in the previous calculations, to simplify the dependencies, we slightly overestimate the shoulder of the force application.

By substituting the value of the compression force, which is expressed by the dependence (18), we get

$$M_1 = (R_b + 0.5h_{ch}) \frac{E \cdot V_f \cdot l \cdot h_{ch} (H - h_{ch} - R_b (1 - \sin \alpha))}{H \omega_r (R_b + h_{ch}) \tan \alpha} \cos \alpha = \frac{E \cdot V_f \cdot l \cdot h_{ch} (R_b + 0.5h_{ch}) (H - h_{ch} - R_b (1 - \sin \alpha)) \cos^2 \alpha}{H \omega_r (R_b + h_{ch}) \sin \alpha}. \quad (30)$$

We determine α , for the indicated feed parameters and beater sizes according to dependence (1)

$$\alpha_t = \arcsin \left(1 - \frac{H}{R_b + h_{ch}} \right) = 9,59^\circ (0,17 \text{ rad}).$$

Taking $E = 120 \cdot 10^3$ N/m² [31], we calculate the expression

$$\frac{E \cdot V_f \cdot l \cdot h_{ch}}{H \omega_r} = \frac{120 \cdot 10^3 \cdot 3,5 \cdot 0,10 \cdot 0,10}{8,37 \cdot 0,25} = 2,007 \text{ kN},$$

and taking into account the angle of rotation, we construct the value of the torque M_l from the compression of the mass (Fig. 17).

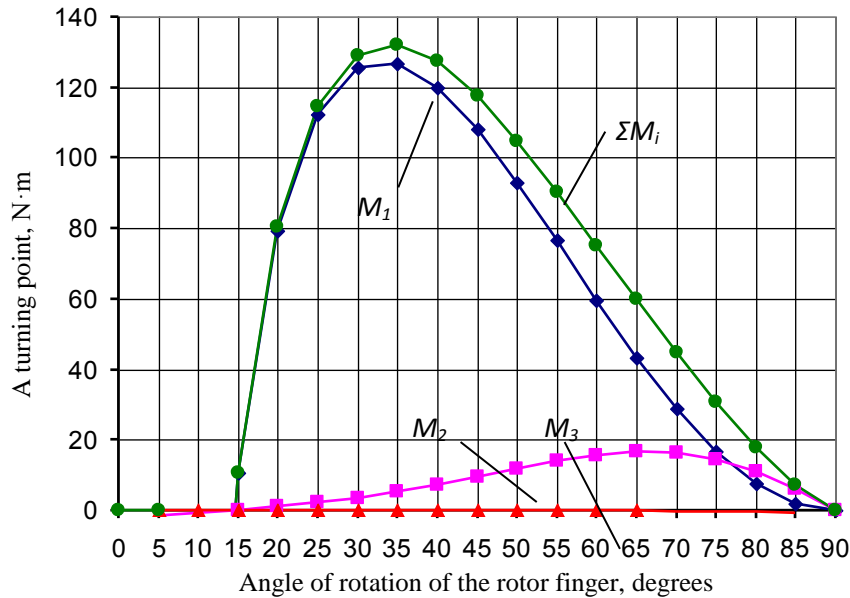


Fig. 17. Diagram of torques on the shaft of the feeding rotor with parameters $R_b = 0.20$ m; $h_{ch} = 0.10$ m; $H = 0.25$ m; $V_f = 3.5$ m/s; $\omega_r = 8.37$ s⁻¹; $l = 0.10$ m ($i = 7$); $\gamma_o = 30$ kg/m³; $E = 120 \cdot 10^3$ N/m²; $a = 0.180$ m; $\Delta\alpha = 2\pi/7 = 51.4^\circ$

The moment M_2 , generated by the force from the stretching of the mass, is defined as the product of dependence (24) on the shoulder, taking into account the direction of the force

$$\begin{aligned}
M_2 &= F_{st}(R_b + 0,5h_{ch}) \sin \alpha = F_{comp}f_c(R_b + 0,5h_{ch}) \sin \alpha = \\
&= f_c \frac{E \cdot V_f \cdot l \cdot h_{ch}(H - h_{ch} - R_b(1 - \sin \alpha)) \cdot (R_b + 0,5h_{ch}) \sin \alpha}{H\omega_r(R_b + h_{ch}) \tan \alpha} = \\
&= f_c \frac{E \cdot V_f \cdot l \cdot h_{ch}(H - h_{ch} - R_b(1 - \sin \alpha)) \cdot (R_b + 0,5h_{ch}) \cos \alpha}{H\omega_r(R_b + h_{ch})}, \text{ N}\cdot\text{m}
\end{aligned} \tag{31}$$

The value of f_c is calculated by the thickness and density of the mass, which are determined by dependencies (12) and (23)

- layer thickness $L = h_{ch} + R_b(1 - \sin \alpha)$;

- density $\gamma = \frac{H\gamma_o}{h_{ch} + R_b(1 - \sin \alpha)}$.

The force created by the acceleration of the mass acts from the beginning of the interaction, which is determined by the angle α_o and during the turn to the angle $\Delta\alpha = 2\pi/i$ and has a maximum value at the moment of turning to the angle $\alpha_o + 0,5\Delta\alpha$.

Let's determine the moment M_3 on the rotor shaft, which is caused by the action of the inertial force

$$\begin{aligned}
M_3 &= F_{ac}(R_b + 0,5h_{ch}) = \\
&= \frac{\omega_r \cdot i \cdot \gamma_o \cdot a \cdot l \cdot h_{ch}(R_b + 0,5h_{ch})}{2\pi} \cdot \left(\omega_r(R_b + 0,5h_{ch}) - V_f \left(1 - \frac{H}{R_b + h_{ch}} \right) \right), \text{ N}\cdot\text{m}
\end{aligned} \tag{32}$$

Under the accepted conditions, its numerical value will be 0,19 N·m. The curve on the diagram (Fig. 17) takes the form of a straight line because it has an order of magnitude smaller values compared to M_1 and M_2 .

The torques on the shaft of the feeding rotor M_1 , M_2 , and M_3 are due to compression forces, mass displacement, and inertia forces, respectively. The total torque $\Sigma M_i = M_1 + M_2 + M_3$ attains a maximum value of 132 N·m at the angle of rotation of the beater from 30°(0,53 rad) to 40°(0,7 rad).

4. Conclusions

The design of the shredding machine determines the technological scheme of the forage harvesting machine and the location of the main nodes. Biter-knife cutting devices are installed on pick-up trucks and balers. Their feeding device can be made in the form of a chain-finger conveyor, an eccentric reel with controlled rakes, or a rotor.

The beater-knife shredding machine, which contains a rotary feeding device and a battery of active disc knives, is considered. Such a structural design of the device allows for to implementation of the cutting process with sliding, which helps to reduce the cutting force, as well as the energy consumption of the grinding process as a whole.

The process of capturing, compressing the mass, and feeding it into the annular channel by the fingers of the feeding rotor of the shredding machine is considered. The dependence (4) of the horizontal component of the rotation speed of the rotor finger on the angle α_t of its rotation was established. The value of the angle α_t depends on the ratio between the dimensions of the drum ($R_b + h_{ch}$) and the height H of the layer of mass fed into the shredding machine and is determined by dependence (1).

The theoretical analysis of physical and mechanical phenomena, which describes the process of cutting leaf-stem materials with a disk knife blade, made it possible to obtain analytical dependences from the calculation of forces and torques on the shaft of the feeding rotor of the biter-knife shredding machine in the zones: capture, compression of the material (18, 30), its stretching (22, 31) and acceleration into the cutting zone (28, 32). Under the given conditions, the maximum torque on the feed rotor shaft becomes 132 N·m at the angle of rotation of the beater from 30°(0,53 rad) to 40°(0,7 rad).

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