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IMPROVEMENT OF THE DESIGN OF A HIGH-SPEED SHEEP SHEARING MACHINE

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This study presents a comprehensive substantiation of the rational design parameters of the pressure mechanism, lever system, and eccentric drive of a high-speed sheep shearing machine. The analysis was performed considering the kinematic, dynamic, and operational characteristics of shearing devices operating under increased oscillatory and impact loads. Particular attention was paid to optimizing the geometric parameters and relative arrangement of the mechanism components to improve reliability, reduce vibration, and ensure the stability of the sheep shearing process.

The kinematic analysis of the working elements made it possible to establish the relationships between the parameters of the eccentric mechanism, the oscillation amplitude of the cutter block, and the magnitude of the contact forces in the cutting zone. Rational relationships between the geometric and force parameters were determined to reduce energy losses, minimize component wear, and improve wool cutting efficiency.

It was established that deviations of the oscillation centers of the pressure rod and the lever from their optimal positions lead to additional dynamic loads, misalignment of the cutter assembly, and non-uniform distribution of contact pressure. These factors adversely affect shearing quality and reduce the service life of the machine components.

To eliminate these drawbacks, it is proposed to improve the manufacturing accuracy requirements for shearing head housings by introducing positional tolerances for the relative locations of the holes accommodating the lever oscillation center and the pressure mechanism. Ensuring the coaxial alignment of these elements minimizes misalignment and improves the operational performance of the shearing machine.

The obtained results provide a foundation for the further implementation of mechatronic control systems, sensor-based monitoring, and robotic technologies in sheep shearing. The integration of vibration, load, and temperature sensors enables adaptive real-time control of the machine operating modes, thereby enhancing operational reliability, efficiency, and equipment durability.

Keywords, *sheep shearing, shearing machine, pressure mechanism, eccentric mechanism, mechatronics, robotics, automated control.*

Eq. 14. Fig. 3. Ref. 10.

1. Problem formulation

An analysis of the design features of existing sheep shearing machines reveals several shortcomings in the pressure mechanism, lever system, and eccentric drive. The structural imperfections of these assemblies limit the possibility of increasing the cutting speed, complicate the achievement of stable and high-quality wool shearing, and lead to increased energy consumption. In addition, intensified heat generation occurs within the contact zone of the working elements, adversely affecting the service life of the cutter assembly and reducing the operational reliability of the equipment during prolonged operation.

Insufficient stiffness and inadequate positional accuracy of the pressure mechanism components result in a non-uniform distribution of contact forces between the movable cutter and the comb. Consequently, friction increases, local overloads occur, and the specific power required for the cutting process rises. Furthermore, the non-optimal design parameters of the eccentric mechanism restrict the attainable oscillation frequency of the cutter due to undesirable vibration and impact loading.

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In view of these limitations, a comprehensive research program was developed to improve the design of the principal functional assemblies of the shearing machine. The program addressed the following objectives:

1. To provide a scientific justification for the rational design of the pressure mechanism by considering the kinematic and force-related factors governing the stability of the clamping force and the uniformity of contact within the cutter assembly.

2. To develop and substantiate the design parameters of a high-speed shearing head capable of providing an increased cutter oscillation frequency without compromising its service life.

3. To optimize the geometric and mass-inertia parameters of the eccentric mechanism in order to minimize dynamic loads, reduce vibration, and improve the efficiency of the drive system.

The investigations were carried out taking into account the operational characteristics of these assemblies within a high-frequency electrically driven shearing machine. Such a drive is characterized by a high rotational speed and specific dynamic interactions between moving components, requiring refinement of the analytical models and consideration of inertial effects at elevated operating speeds.

The primary object of the investigation was a sheep shearing machine equipped with a 400 Hz high-frequency electric drive [1]. This machine enabled an investigation of the operating characteristics of the mechanisms under high-speed conditions. The selection of this particular machine was motivated by its potential for increasing shearing productivity and by the possibility of implementing fundamentally new design solutions for the motion transmission system and the pressure-generating mechanism.

At the same time, current trends in the development of livestock machinery increasingly emphasize the integration of mechatronic systems, automation technologies, and robotic process control. In the case of sheep shearing equipment, this involves integrating high-speed electric drives with vibration, temperature, and load sensors, together with intelligent control systems.

The application of a mechatronic approach enables continuous real-time monitoring of the technical condition of the shearing machine and automatic adjustment of the drive operating parameters according to wool density, the contact force between the cutter and the comb, and the prevailing dynamic loads. This approach creates opportunities for developing adaptive and robotic sheep shearing systems with enhanced productivity, energy efficiency, and operational reliability.

2. Analysis of recent research and publications

Recent research on sheep shearing machines is primarily focused on improving the kinematic and dynamic characteristics of the cutting mechanism, increasing shearing productivity, reducing energy consumption, and improving the ergonomic properties of the equipment. A number of studies consider the traditional design of a shearing machine as a spatial four-bar mechanism in which the rotational motion of the drive is transformed into reciprocating motion of the cutter relative to the comb. Computer-aided modeling using ADAMS and analytical methods has made it possible to determine the displacement, velocity, acceleration and torque characteristics of the mechanism and to assess their influence on the machine's power requirements [2-5].

An important research direction is the synthesis and optimization of new mechanisms for converting the rotational motion of the drive into the reciprocating motion required for shearing. Researchers have compared alternative mechanisms according to their mass, cost, wear resistance and strength, followed by dimensional synthesis and kinematic analysis. The obtained results indicate the possibility of improving the existing design through optimization of the mechanism geometry and operating parameters. Dynamic modeling using modern simulation environments, including RecurDyn, has also been applied to investigate the interaction between the crank, fork, cutter and comb, as well as the influence of cutter pressure on cutting efficiency and power consumption [6-8].

Considerable attention is given to the design and operating parameters of the cutting unit. Studies of rotary shearing machines have shown that the width of the comb, machine feed speed and traction force significantly affect the productivity of the shearing process. Narrower combs can reduce resistance during movement, while appropriate selection of the working width and feed speed makes it possible to increase productivity without a corresponding increase in the physical load on the operator. Modern recommendations also emphasize the importance of correct selection and adjustment of the comb and cutter, their maintenance, and proper tension, since these factors directly influence cutting quality, equipment service life and safe operation.

Another important area concerns the materials and manufacturing technology of the comb and cutter. Research into domestically produced sheep-shearing machines has demonstrated the need for high hardness, wear resistance and sufficient toughness of these components because they are subjected to intensive friction and repeated loading. The use of appropriate tool steels and heat treatment has been considered as a means of increasing the durability of the cutting elements and reducing maintenance requirements. At the same time, ergonomic and operational aspects remain relevant, particularly the reduction of vibration, noise, mass and operator fatigue [5].

Despite the considerable amount of research devoted to the kinematics, dynamics and individual components of sheep shearing machines, the problem of creating a high-speed design that simultaneously provides increased productivity, stable cutting quality, low vibration, reduced power consumption and convenient operation remains insufficiently solved. Therefore, further improvement of the machine should be aimed at optimizing the cutting mechanism, selecting rational operating parameters, improving the interaction between the comb and cutter, and reducing dynamic loads in the drive system. This creates the prerequisites for the development of an improved high-speed sheep shearing machine with enhanced technical and operational characteristics [9-10].

3. The purpose of the article

The objective of this study is to improve the technical and operational performance of a sheep shearing machine by enhancing the design of its working components through the scientifically substantiated determination of the rational parameters of the pressure mechanism. Achieving this objective requires an analysis of the kinematic and force interactions in the cutter–comb contact zone, evaluation of the influence of the pressure mechanism parameters on cutting stability and energy consumption, and optimization of the contact conditions. Particular attention is focused on minimizing dynamic loading, reducing wear intensity of the machine components, and ensuring a uniform distribution of contact pressure.

4. Results and discussion

To ensure an efficient wool shearing process, the cutter teeth must maintain uniform contact pressure against the comb throughout the entire reciprocating motion of the cutter along the comb surface, regardless of the instantaneous position of the cutter.

To substantiate the parameters of the pressing mechanism that will ensure uniform pressing of the knife to the comb, Figure 1 shows a diagram of the pressing mechanism [2-4].

The notation on the diagram is as follows:

O is the centre of oscillation of the lever;

A, *C* – points of position of the thrust rod on the lever at the greatest deviations of the lever from the symmetry axis of the machine.

O_1 is the centre of vibration of the thrust rod;

AO_1, CO_1 - the position of the thrust rod projection on the horizontal plane at the greatest deviations of the lever from the symmetry axis of the machine.

For uniform pressing of the knife to the comb at points *ABC*, the distance between these points and the centre of oscillation O_1 must be the same [2, 3].

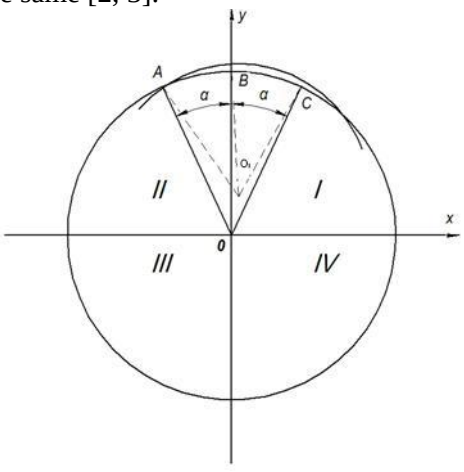


Fig. 1. Diagram of the clipper's presser mechanism

This is possible only when the centre of vibration [3] of the thrust rod O_1 is aligned with the centre of vibration of the lever O, which is evident from the diagram and is also confirmed by mathematical calculation, where the distance between the two points (Fig. 1) is determined by the formula

$$d = \sqrt{(x_2 - x_1)^2 + (y_2 - y_1)^2} \quad (1)$$

where x_2, x_1, y_2, y_1 are the coordinates of the corresponding points.

Thus, to ensure that the knife is pressed evenly against the comb, it is necessary that the centre of vibration of the lever and the centre of vibration of the stop rod are on a common vertical to the plane of contact of the comb.

The displacement of the oscillation centre of the stop rod relative to the oscillation centre of the lever along the *Y*-axis (Fig. 1) depends on both the length of the stop rod and the degree of wear of the knife and stop rod of the shearing machine.

The displacement of the oscillation centres of the lever and the stop rod relative to each other along the *X*-axis depends on the accuracy class of the shearing machine parts.

In this regard, we have investigated the degree of influence of the deviation of the oscillation centres separately along both the *Y* and *X* axes on the unevenness of pressing the knife against the comb.

It is known that in clippers with a high-frequency electric drive, the inevitable

deviation of the centres of vibration of the lever and the stop rod relative to each other along the *Y*-axis is about 1 mm [4, 6]. It depends on the wear of the knife and the thrust rod. We have found that the change in the distance between the thrust rod thrust bearings is 0.0132 mm.

Measurements of the parameters of serial shearing machines have shown that the displacement along the *X*-axis of the centre of vibration of the stop rod relative to the axis passing through the centre of vibration of the lever perpendicular to the plane of contact of the comb reaches ± 0.5 mm.

The displacement of the oscillation centres of the lever and the stop rod occurs as a result of the fact that in the manufacture of shearing machine parts, the inclined hole for the pressure mechanism and the hole for the lever foot are machined on different devices when the shearing machine bodies are based on casting surfaces. In addition, after machining the inclined bore and milling the comb fit plane in the shearer body, the axis of the inclined bore deviates from the plane passing through the symmetry axis of the shearer perpendicular to the comb fit plane. As a result, the centre of oscillation of the thrust rod receives an additional displacement from the axis of the centre of oscillation of the lever.

Calculations have shown that a 0.5 mm deviation of the oscillation centres of the lever and the thrust rod relative to each other along the *X*-axis results in a 0.221 mm change in the distance between the thrust rod's bearings.

This leads to a significant unevenness of the knife pressing against the comb in different positions of the lever during its movement.

Comparing the values of the change in the distance between the thrust rod bearings of 0.0132 mm and 0.221 mm, it can be concluded that, while maintaining the manufacturing technology of the shearing head housing, a smaller deviation of the oscillation centres of the thrust rod and lever is possible.

To substantiate the design parameters of the pressure mechanism, it is necessary to introduce tolerances for the relative position of the hole for the centre of vibration of the lever and the inclined hole in the drawings for the shearing head bodies. The axis of the hole for the centre of oscillation of the lever and the axis of the inclined hole for the pressure mechanism must be manufactured to a high accuracy class on a common plane perpendicular to the plane of contact of the comb. This is easily achieved by using special conductors when machining the holes for the centre of oscillation of the lever and the pressure mechanism.

In clippers, levers of the first and second kind are used.

The front support of the levers is a knife. Therefore, the forces applied to the lever also act on the knife [6].

In Fig. 2. shows a lever of the first kind assembled with the pressure feet, the lever foot and the thrust rod foot. The same figure shows a diagram of the movement of the eccentric mechanism roller in a circle with a force F arising from the action of the eccentric mechanism and creating a bending moment relative to the *AB* axis, which connects the lever support points.

The axis AB is displaced relative to the axis of the eccentric mechanism *CD* in the plane of motion of its roller by an amount a. The points of application of force *F* to the lever in one of the positions of the roller above the axis *CD* and in one of the positions of the roller below the axis *CD* are marked *b* and b_1.

To determine the vertical forces arising from the action of the eccentric mechanism and transmitted to the teeth of the knife, we add the sum of the acting moments about the axis *AB*:

$$\sum \left[M(AB) \cdot Fh - \left(F_{fr} + F_{in}\right) \cdot k - M_{in} - Q \cdot \frac{m}{2} = 0\right] \quad (2)$$

where h – is the shoulder of application of the F force relative to the AB axis; F_{fr} – is the horizontal component of the friction force arising from the movement of the thrust rod; F_{in} – is the force of inertia from the mass of the thrust rod; k – the shoulder of application of the forces F_{fr} and Fin relative to the AB axis; M_{in} – moment relative to the AB axis from the lever inertia forces; Q – is the force perceived by the presser feet and opposing the moments on the lever relative to the AB axis; m – is the arm of the force Q.

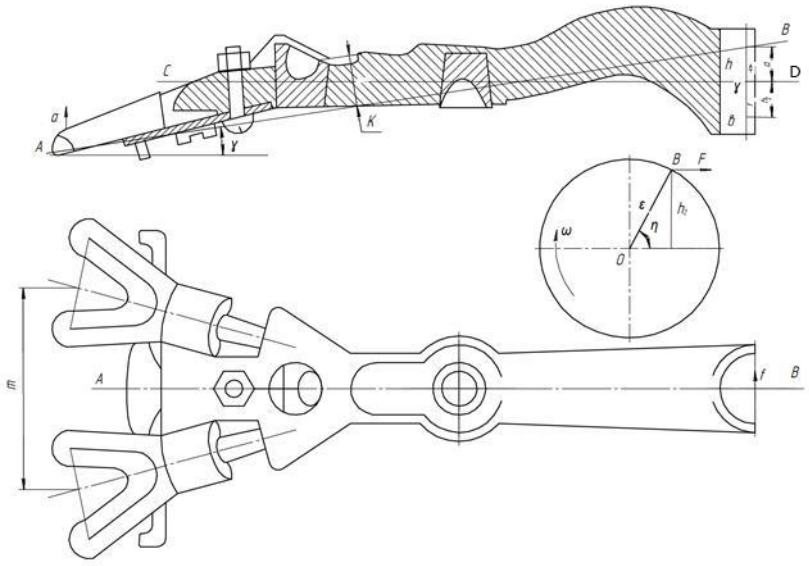


Fig. 2. Lever of the first type of clipper

$$h = h_1 \cos\gamma,\ h = a \pm h_2$$
$$h_2 = \varepsilon \cdot \sin\eta, h_2 = (a \pm \varepsilon \cdot \sin\eta)\cos\gamma, \quad (3)$$

where η is the angle that determines the position of the eccentric roller.

Substituting the value of h into Equation 2, we obtain:

$$F\cos\gamma\,(a \pm \varepsilon \cdot \sin\eta) - \left(F_{\text{тр}} + F_{\text{ін}}\right) \cdot k - M_{in} - Q\frac{m}{2} = 0 \quad (4)$$

$$Q = \frac{2}{m}\left[F\cos\gamma\,(a \pm \varepsilon \cdot \sin\eta) - \left(F_{fr} + F_{in}\right) \cdot k - M_{in}\right], \quad (5)$$

The above equation shows that the knife is subjected to a vertical variable force Q, which arises not from the action of the pushing mechanism, but as a result of the eccentric application of driving forces with respect to the line connecting the points of support of the lever.

To determine the vertical forces arising from the action of the eccentric mechanism and transmitted to the teeth of the knife when using a second-type lever, consider a second-type lever complete with presser feet, lever foot and thrust rod foot (Fig. 3).

In a second-type lever for a clipper with a high-frequency electric drive, the AB axis, which passes through the lever support points, passes through the centre O of rotation of the eccentric mechanism roller. The radius of the crank is .

The points of application of the force F from the eccentric mechanism to the lever in one of the roller positions above the CD axis and in one of the roller positions below the CD axis are denoted by b and b_1.

The sum of the acting moments about the axis AB:

$$\sum M(AB) \cdot Fh - \left(F_{fr} + F_{in}\right) \cdot k - M_{in} - Q \cdot \frac{m}{2} = 0, \quad (6)$$

in this equation is similar to formula 2:

$$h = h_1 \cos\gamma,\ h_1 = \varepsilon \cdot \sin\eta, h = \varepsilon \cdot \sin\cdot\cos\gamma, \quad (7)$$
$$F \cdot \varepsilon \cdot \sin\cdot\cos\gamma - \left(F_{fr} + F_{in}\right)k - M_{in}, \quad (8)$$

Where from:

$$Q = \frac{2}{m}\left[F\sin\eta\cos\gamma\,\varepsilon - \left(F_{fr} + F_{in}\right) \cdot k - M_{in}\right], \quad (9)$$

The points of application of the force *F* from the eccentric mechanism to the lever in one and the force *Q* when using the first and second type of lever are variable in magnitude and are not perceived by all the teeth of the knife, but only by the left or right pair of knife teeth, as they are transmitted to the knife by one of the pressing feet.

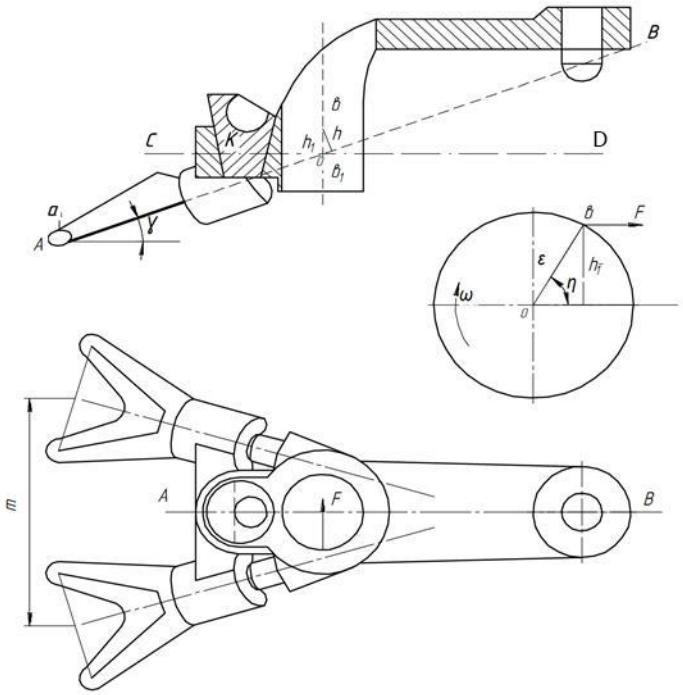


Fig. 3. Lever of the second type of clipper

Comparing formulas 5 and 9, which define the values of the forces *Q*, it is clear that, along with the general components of these forces, the force *Q* when using a lever of the first kind, compared to the force *Q* when using a lever of the second kind, is supplemented by an additional component, the value of which depends on the amount of displacement ***a*** of the axis connecting the points of the lever's supports relative to the axis of the eccentric mechanism (Fig. 3).

$$\frac{2}{m} cos\,\gamma\,\alpha, \tag{10}$$

From the above, it can be assumed that when using the second type of lever, the variable force *Q* is smaller than when using the first type of lever, mainly as a result of the fact that in the second type of lever, the centre of rotation of the eccentric mechanism finger is located on the axis connecting the points of the lever supports.

In addition, the second type of lever is more compact compared to the first type, which makes the shearing machine more compact.

As a result of the above advantages, the second type of lever should be used in a shearing machine with a high-frequency electric drive.

To select the shape of the second type of lever and the shearing machine body mated to it, the dependence of the force of pressing the knife against the comb on the force generated by the pushing mechanism and the position of the lever supports was derived.

$$R_A \frac{P\,cos\left(\frac{\pi}{2}+\gamma-\beta\right) l\,cos\,\gamma}{p}, \tag{11}$$

where *P* – is the force transmitted by the thrust rod of the pushing mechanism to the lever; β – the angle of inclination of the thrust rod to the plane of the comb; γ – is the angle formed by the line connecting the points of support with the lever and the plane of contact of the comb; *l* – the distance from the centre of oscillation of the lever to the thrust rod foot; *p* – is the distance from the centre of oscillation of the lever to its front support.

In the lever of the second type of shearing machine with a high-frequency electric drive, this is achieved by setting the lever's foot in the required position, the spherical part of which is made eccentrically to the conical seating surface (Fig. 3). As a result of rotating the lever arm around the axis when folding the

clipper, the required movement of the knife relative to the comb is achieved. As a result, no additional adjustment is required when changing the cutting pairs.

Justification of the Rational Design Parameters of the Eccentric Drive Mechanism

The performance of clippers largely depends on the size of the knife stroke, and the knife stroke is determined by the radius of the eccentric mechanism crank [5-7].

The obtained results are of considerable importance not only for improving the mechanical design of the sheep shearing machine but also for the further integration of mechatronic control modules. The established relationships governing the clamping force, dynamic loading, and vibration characteristics can serve as fundamental input parameters for the development of automatic control algorithms.

In particular, the integration of acceleration sensors, strain gauges, and temperature sensors enables automatic regulation of the drive rotational speed, cutter oscillation amplitude, and contact force between the cutter and the comb. This approach reduces dynamic overloads, minimizes energy losses, and improves the quality and efficiency of wool shearing.

The ends of the knife teeth are located and move along arcs with fixed radii. At the same time, the chords of the arcs described by the tops of the knife teeth have different angles with the symmetry axis of the machine. Based on this, and taking into account the fact that the lengths of the arcs of the trajectories of all the ends of the teeth are equal to each other, it can be assumed that in the cutting apparatus of clippers, the strokes S_1 of different knife teeth, determined in the direction perpendicular to the axis of symmetry of the machine, are not equal to each other and depend on the position of the tooth on the knife.

Based on the foregoing, the stroke of the shearing knife will be as follows:

$$S = \frac{m_1 t_0 + \Delta S}{\cos \phi} \tag{12}$$

where t_0 – is the comb pitch; m_1 – the number of comb pitches t0 overlapped by the knife tooth during its stroke; ΔS – is the increase in the stroke of the knife tooth end, which takes into account the wear of the lever and parts of the eccentric mechanism during operation of the clipper; ϕ – is the angle between the perpendicular to the symmetry axis of the machine and the chord of the arc described by the top of the outermost knife tooth.

As a result of the analysis of the effect on the cutting process of both the transition of the ends of the knife teeth to the comb teeth,

and the incomplete overlap of the cutting edges of the knife teeth with the cutting edges of the comb teeth at the end of the knife stroke, we can assume:

$$\Delta S = b_1 + b_2, \tag{13}$$

where b_1 – is the width of the working part of the comb tooth at the level of the knife tooth ends; b_2 – width of the knife tooth end.

The radius of the crank of the eccentric mechanism, which ensures the optimal stroke of the knife, can be determined by formula (14)

$$\varepsilon = \frac{R(m_1)t_0 + b_1 + b_2}{2p\cos\left(\arcsin\frac{3t}{2p_1}\right)}, \tag{14}$$

where R – is the distance from the axis of the lever oscillation centre to the axis of the lever cylindrical groove;p_1 – dista ce from the axis of the centre of vibration of the lever to the ends of the knife teeth at maximum wear of the knife; t – knife pitch.

5. Conclusion

Stable and uniform contact between the movable cutter and the comb throughout the reciprocating motion of the cutter can only be achieved by ensuring the geometric alignment of the pressure mechanism components. In particular, the oscillation center of the pressure rod and the oscillation center of the lever must be located on the same vertical line with respect to the contact plane of the comb. Failure to satisfy this condition results in misalignment of the cutter assembly and a non-uniform distribution of contact pressure within the cutting zone.

The most significant factor responsible for the non-uniform contact between the cutter and the comb in sheep shearing machines is the deviation of the oscillation centers of the pressure rod and the lever from their common vertical axis in the direction perpendicular to the machine symmetry axis. Such deviations generate additional dynamic loads and localized overloads within the cutting pair.

During operation, the cutter is subjected to a cyclic vertical force that is generated not directly by the pressure mechanism but by the eccentric application of forces transmitted from the eccentric drive mechanism relative to the line connecting the lever support points. This phenomenon gives rise to parasitic load components that adversely affect the wool shearing process, increase the power consumption of the drive system, and intensify heat generation at the cutter–comb interface.

To minimize these undesirable effects, an improved lever design incorporating an eccentrically positioned contact pad is proposed. The contact pad is installed in such a manner that the optimum relative position of the cutter and the comb is achieved during assembly. The proposed design reduces misalignment of the cutter assembly, stabilizes the clamping force, and improves the kinematic compatibility of the mechanism components.

A promising direction for the further development of sheep shearing equipment is the implementation of mechatronic control systems and robotic technological platforms for automating the shearing process. The integration of improved mechanical assemblies with modern sensor systems, electronic control units, and adaptive control algorithms will enable intelligent regulation of the machine operating parameters according to the actual operating conditions.

The robotization of the sheep shearing process will reduce the influence of human factors, increase productivity, improve shearing quality, and decrease energy consumption, thereby meeting the current requirements of digital and intelligent livestock production.

The obtained results establish a scientifically substantiated foundation for the further modernization of high-speed sheep shearing machines. The proposed engineering solutions are aimed at improving the efficiency of the shearing process, reducing energy consumption, and enhancing the operational performance of the equipment. Their implementation is expected to increase productivity, improve operational reliability, and extend the service life of sheep shearing machines operating under intensive conditions.

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УДОСКОНАЛЕННЯ КОНСТРУКЦІЇ ВИСОКОШВИДКІСНОЇ МАШИНИ ДЛЯ СТРИЖКИ ОВЕЦЬ

У цьому дослідженні представлено комплексне обґрунтування раціональних конструктивних параметрів притискного механізму, системи важелів та ексцентрикового приводу високошвидкісної стрижки овець. Аналіз проводився з урахуванням кінематичних, динамічних та експлуатаційних характеристик стрижальних пристроїв, що працюють в умовах підвищених коливальних та ударних навантажень. Особлива увага приділялася оптимізації геометричних параметрів та взаємного розташування компонентів механізму для підвищення надійності, зменшення вібрації та забезпечення стабільності процесу стрижки овець.

Кінематичний аналіз робочих елементів дозволив встановити взаємозв'язки між параметрами ексцентрикового механізму, амплітудою коливань ріжучого блоку та величиною контактних сил у зоні різання. Визначено раціональні взаємозв'язки між геометричними та силовими параметрами для зменшення втрат енергії, мінімізації зносу компонентів та підвищення ефективності різання вовни.

Встановлено, що відхилення центрів коливань притискного штока та важеля від їх оптимальних положень призводять до додаткових динамічних навантажень, перекосу ріжучого вузла та нерівномірного розподілу контактного тиску. Ці фактори негативно впливають на якість стрижки та скорочують термін служби компонентів машини.

Щоб усунути ці недоліки, пропонується покращити вимоги до точності виготовлення корпусів стрижальних головок шляхом введення допусків положення для відносного розташування отворів, що вміщують центр коливань важеля та механізм тиску. Забезпечення співвісного вирівнювання цих елементів мінімізує перекіс та покращує експлуатаційні характеристики стрижальної машини.

Отримані результати створюють основу для подальшого впровадження мехатронних систем керування, сенсорного моніторингу та роботизованих технологій у стрижці овець. Інтеграція датчиків вібрації, навантаження та температури дозволяє адаптивно керувати режимами роботи машини в режимі реального часу, тим самим підвищуючи експлуатаційну надійність, ефективність та довговічність обладнання.

Ключові слова: *стрижка овець, стригальна машина, механізм тиску, ексцентриковий механізм, мехатроніка, робототехніка, автоматизоване керування.*

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