Запропоновано метод кінематичного аналізу механізмів хлопкопереробних машин, в тому числі механічного бунтоукладчика, та розроблено алгоритм його комп'ютерної реалізації. Метод дозволяє автоматизувати процес отримання і рішення рівняння для визначення кінематичних параметрів усіх елементів механізму при заданому законі руху ведучої ланки. Експериментальним шляхом вивчена можливість формування верхньої частини бунту з необхідною щільністю бавовни, що забезпечує стійкість формованого бунту

Ключові слова: бавовна-сирець, бунт бавовни, форма бунту, верхня частина бунту, кулачок, щільність бунту

Предложен метод кинематического анализа механизмов хлопкоперерабатывающих машин, в том числе механического бунтоукладчика, и разработан алгоритм его компьютерной реализации. Метод позволяет автоматизировать процесс получения и решения уравнения для определения кинематических параметров всех элементов механизма при заданном законе движения ведущего звена. Экспериментальным путем изучена возможность формирования верхней части бунта с необходимой плотностью хлопка, что обеспечивает устойчивость формируемого бунта

Ключевые слова: хлопок-сырец, бунт хлопка, форма бунта, верхняя часть бунта, кулачок, плотность бунта

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RESEARCH INTO THE PROCESS OF MECHANICAL FORMATION OF THE UPPER PART OF A RAW COTTON BUNDLE

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1. Introduction

Cam-lever mechanisms are widely used in the drives of cotton-pressing and drying-cleaning mechanisms, etc. [1]. This is a mechanism for the formation of a cotton bundle at the cotton ginning plants. The shape of a bundle largely affects quality storage of raw cotton [2, 3]. A bundle formation process is extremely complicated, consisting of a number of labor-intensive operations.

Issues in the search for rational techniques and devices for the mechanized laying of raw cotton were addressed by many scientific and industrial organizations of Azerbaijan.

Thus, a rectangular frame was manufactured and tested, made of the angular steel 40×40, for laying cotton. The density of cotton in the formed bundle, however, did not reach the required level. As a result, upon completion of laying the cotton weighing 135 tons, the bundle collapsed at a height of 3.0–3.2 m [4].

Harvesting raw cotton, depending on weather conditions, lasts for a few weeks. During favorable harvesting, the procuring points may daily receive raw cotton in a volume equal to 3–4 to 7–8 % of annual harvest. About 20–30 % of harvested raw cotton are processed at cotton ginning plants over a harvesting season, but the bulk of the raw cotton still has to be laid for long-term storage, in order to process it in the coming months. That is why stability of a bundle is of great importance for high-quality cotton storage [5].

2. Literature review and problem statement

Tashkent State Specialized Design Bureau (TGSKB, Republic of Uzbekistan) developed and tested a metal frame to form a round bundle with a diameter of 13.0 m. However, when completing the bundle, two upper tiers did not resist lateral pressures, which resulted in the destruction of the frame and collapse of the bundle [6].

Reinforcement of the bundle of raw cotton was proposed by the Central Scientific-Research Institute of Cotton Industry (CNIIHProm, Republic of Uzbekistan). Tests demonstrated that reinforcement reduces the likelihood of a bundle collapse but it increases labor intensity due to the introduction of additional operations. In this case, one should also consider sufficient enough complexity of the process of bundle formation [7].

Article [8] describes results of the studies conducted under industrial conditions. Underlying them was the implementation of an idea about compressing raw cotton for the purpose to mechanize the bundling process. However, after a three-month storing, bottom piles deformed and the bundle collapsed. The HASKI modular method for storing cotton also failed because its application under conditions of Azerbaijan is impossible due to the lack of specialized sites and specifics in the procurement conditions.

Experiments confirmed that the creation of a mechanism for bundling cotton is only possible while combining mechanical distribution and compression of cotton. In order to solve this task, it is a promising mechanism to form the upper part of a bundle using a metal shield. The mechanism performs simultaneously the work of a sealant and renders shape to the upper part of the bundle. Given that the shape affects stability and storage of the bundle of raw cotton, employing the cam-lever mechanisms for the drive of a mechanical bundling press is feasible.

A confirmation of the expediency of using exactly the cam-lever mechanisms is theoretical interest in this type of mechanisms from different researchers [9, 10] and its utilization in the applied directions [11, 12]. In particular, as far as theoretical studies are concerned, it is interesting to note that there is a significant number of cam mechanisms that can be applied to ensure an accurate trajectory, or to direct the body through the precisely-defined infinite positions [9]. The authors of the given work argue that by using the Gruebler's mobility criterion, it is possible to perform a structural synthesis in order to obtain several mechanisms of cam connection. In this case, the proposed synthesis method employs a method of the cycle complex closure and the envelope theory to find the axial line and contact points of the track. Paper [10] proposed an adjustable constant force mechanism (ACFM), which consists mainly of a linear spring and a cam mechanism for passive control over the contact force. It was noted that the proposed model of ACFM is simple and accurate because, in particular, it lacks complex flexible elements. Ensuring precise trajectories in a combination with achieving the specified operational indicators depending on the purpose of a machine predetermines scientific and practical interest towards the application of a cam-lever mechanism in complex technological processes. The latter include, in particular, the process of forming the upper part of a bundle by mechanical raw cotton bundling presses.

3. Research goal and objectives

The goal of present work is to study the process of forming the upper part of abundle. This will make it possible to manufacture a high-quality raw cotton bundle using mechanical bundling presses.

To accomplish the goal, the following tasks have been set:

– to define a method for kinematic analysis of a cam-lever mechanism, which is used in the drives of a mechanical raw cotton bundling press, and to explore the possibility of its application to form the upper part of a raw cotton bundle;

– to examine raw cotton density depending on time and height of the bundle;

– to investigate the effect of bundle density on the adhesion between wastes and cotton and the influence of waste adhesion on the ginning effect.

4. Results of research into the process of mechanical formation of a raw cotton upper bundle

When running a kinematic analysis of cam-lever mechanisms, a traditional method of constructing a replacement mechanism, as well as the motion reversal method, are not applicable for a number of reasons to employ a computer. We shall consider using an automated method of kinematic analysis of flat lever mechanisms based on the state equations for a kinematic analysis of the cam-lever mechanism of a bundling press whose kinematic schematic is shown in Fig. 1.

Information on the structure of the mechanism is entered into the software in the form of specialized characteristic arrays, for example, $n=5$ is the number of movable links; $k=6$ is the number of kinematic pairs (hereafter referred to as nodes); $T(2,n)$ is the topology matrix.

Link of the mechanism can be of variable length (rocking mechanisms, mechanisms with sliders) [13]. In this case, this is the first link that represents a radius-vector of the theoretical cam profile. The variation of links length is described by array $L(n)$ whose elements are $l_1 = 1$ if the i-th link is of variable length, and $l_1 = 0$ at a constant length of this link. In addition, we introduce numerical values for the coordinates of the fixed nodes, links length, as well as motion parameters of the cam, which is the leading link. The law of motion of a cam-lever mechanism is determined by the cam profile, which is assigned in the form of a tabular function of the cam's radius-vector depending on the angle of its turning. Such a function is introduced in the form of array $R(N,2)$. In this case, $R(i,1)=\varphi i$ is the cam's turning angle in the i-th position, while $R(i,2)$ is the corresponding value of radius-vector of the theoretical cam profile.

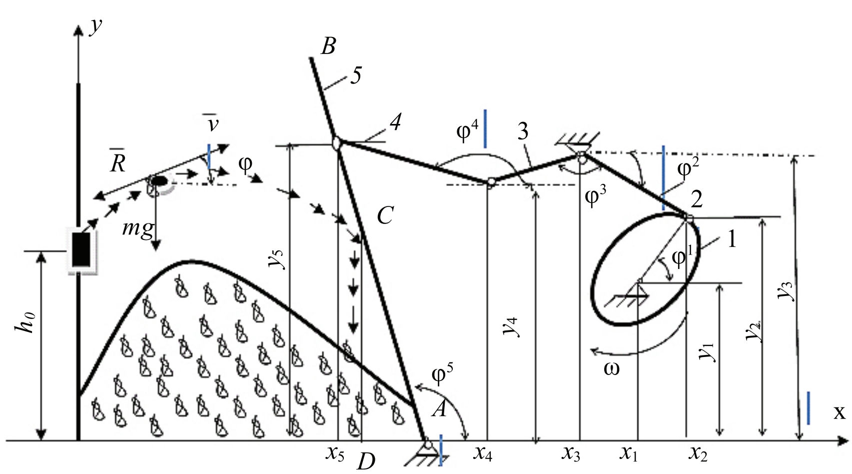

Fig. 1. Kinematic schematic of a cam-lever mechanism

Equation of state of the mechanism positions takes the form (1) [14].

$$Y = AZ, \qquad (1)$$

where $Y(2n)$ is the vector-column that contains projection components of the links length:

$$y_{2i-1} = l^{(i)} \cos\varphi^{(i)};\ y_{2i} = l^{(i)} \sin\varphi^{(i)};\ i = 1,2,\ldots,n;$$

$Z(2k)$ is the vector-column that contains coordinates of the mechanism nodes:

$$z_{2j-1} = x_j;\ z_{2j} = y_j;\ j = 1,2,\ldots,k;$$

$A(2n,2k)$ is the matrix whose non-zero elements take values under condition when $p = 2i-1$ and $q = 2j-1$, where i and j is the number of link and node number, respectively, $a_{pq} = -1$ at $j = t_{2i}$, in all other cases $a_{pq} = 0$ (t_1 and t_{2i}) are the elements of a topology matrix (T).

Equation of position state (1) contains unknowns of various forms, which are the angles of link orientation and coordinates of the movable nodes. Solving the equation of state involves two stages, at each of them we shall determine the unknowns of one of the specified form.

Let k be the total number of mechanism nodes and k_o is the number of fixed nodes. Then $k_1 = k - k_o$ is the number of movable nodes. Reduce (1) to the form

$$\begin{cases} E_1 Y = A_{11} Z_o + A_{12} Z_1, \\ E_2 Y = A_{21} Z_o + A_{22} Z_1, \end{cases} \qquad (2)$$

where $Z_o(2k_o)$ is the vector-column of known coordinates of the fixed nodes; $Z_1(2k_o)$ is the vector-column of unknown coordinates of the fixed nodes;

$$A_{22}(2n-2k_1,2k_1),\ A_{12}(2k_1),\ A_{21}(2n-2k_1,2k_o),$$

$A_{22}(2n-2k_1,2k_1)$ are the matrices derived from $A(2n,2k)$; $E_1(2k,2n), E_2(2n-2k_1,2n)$ are the singular transition matrices.

The algorithm of a matrix equation system (2) is constructed under condition of a square and not a special matrix $A_{12}(2k_1,2k_1)$, for which one can find an inverse matrix $A_{12^{-1}}$.

Representing vector-column Z_1 from the first equation of system (2):

$$Z_1 = A_{12}^{-1} E_1 Y - A_{12}^{-1} A_{11} Z_o \qquad (3)$$

and substituting it in the second equation, we obtain

$$CY = D, \qquad (4)$$

where $c = (E_2 - A_{22} A_{12}^{-1} E_1)$; $D = (A_{21} - A_{22} A_{12}^{-1} A_{11}) Z_0$.

Matrix equation (4) is a system of equations of the projections of vector contours of the mechanism's kinematic circuit. It is obvious that obtaining it is connected with the operations on matrices and can be performed by computer. For a bundle of the forming mechanism, a system of nonlinear transcendental equations (4) takes the form

$$\begin{cases} l^{(1)} \cos\varphi^{(1)} - l^{(2)} \cos\varphi^{(2)} = x_3 - x_1; \\ l^{(1)} \sin\varphi^{(1)} - l^{(2)} \sin\varphi^{(2)} = y_3 - y_1; \\ l^{(3)} \cos\varphi^{(3)} + l^{(4)} \cos\varphi^{(4)} - l^{(5)} \cos\varphi^{(5)} = x_6 - x_3; \\ l^{(3)} \sin\varphi^{(3)} + l^{(4)} \sin\varphi^{(4)} - l^{(5)} \sin\varphi^{(5)} = y_6 - y_3. \end{cases} \qquad (5)$$

Solving this system, we find $\varphi^{(1)}, \varphi^{(2)}, \varphi^{(3)}, \varphi^{(4)}, \varphi^{(5)}$. Once we know components of vector Y, we determine, according to (3), coordinates of the movable nodes.

Equation of velocity states, derived from (1) by the differentiation in time, after transforms, leads to a system of linear algebraic equations of the form

$$B = A_o V, \qquad (6)$$

where $B(2n)$ is the vector-column of free terms; $V(2n)$ is the vector-column of unknown velocities; $A_o(2n,2n)$ is the matrix obtained from $A(1)$.

For the given mechanism (Fig. 1), the terms of system of equations (6)

$$B^T = \left[l^{(1)} \cos\varphi^{(1)}, \sin\varphi^{(1)}, 0,0,0,0,0,0,0,0 \right], \qquad (7)$$

$$V^T = \left[x_2, y_2, x_4, x_5, y_5, \omega^{(1)}, \omega^{(2)}, \omega^{(4)}, \omega^{(5)} \right],$$

where $l^{(1)}$ is the time derivative (or the cam turning angle) of function of radius-vector $R(N,2)$; x_j, y_j are the components of linear velocities of the movable nodes; $\omega^{(i)}$ are the angular velocities of the mechanism's links.

Equation of acceleration state is derived from the differentiation in time of the velocity state equation and, in this case, is written in the form

$$B_1 = A_o W, \qquad (8)$$

where $B_1^T = \left[b_1, b_2, \ldots, b_{2i-1}, b_{2i}, \ldots, b_{2n} \right]$:

$$b_1 = -l^{(1)} \cos\varphi^{(1)} (\omega^{(1)})^2 - 2l^{(1)} \sin\varphi^{(1)} \omega^{(1)} + l^{(1)} \cos\varphi^{(1)};$$

$$b_2 = -l^{(1)} \sin\varphi^{(1)} (\omega^{(1)})^2 + 2l^{(1)} \cos\varphi^{(1)} \omega^{(1)} + l^{(1)} \sin\varphi^{(1)};$$

$$b_{2i-1} = -l^{(i)} \cos^{(i)} (\omega^{(i)})^2;$$

$\ddot{x}_j, \ddot{y}_j$ are the components of linear acceleration of the j-th node; $k^{(i)}$ is the angular acceleration of the i-th link.

Matrix A_o in (8) takes the form (7), B^T, V^T in equation (7) and B_1^T and W_1^T in equation (8) are the transposed vector-columns.

Thus, the velocities and accelerations of all elements of the cam-lever mechanism is determined by solving the systems of linear algebraic equations (6) and (8) in line with the standard program.

Numerical simulation was employed to calculate a sloped shield that forms the upper part of a bundle. Kinematical parameters of the sloped shield (link (5), Fig. 1) for a turning angle from 0 to 160° are shown on chart (Fig. 2).

After conducting theoretical studies, we experimentally formed a bundle and determined the raw cotton density depending on the height of the bundle and storage time of the raw cotton.

For the raw cotton, selection S3038, grade 1, harvested manually, volumetric mass is 65 kg per cub. m., for grade 4 – 60 kg per cub. m. Volumetric mass of the raw cotton pressed in bundles at the lower layers after a 5-month storage, for first varieties, is 255 kg per cub. m., for low grades – 220 kg per cub. m. When compressing the raw cotton, the force of its adhesion with wastes increases, which adversely affects effectiveness of its ginning in the technological process.

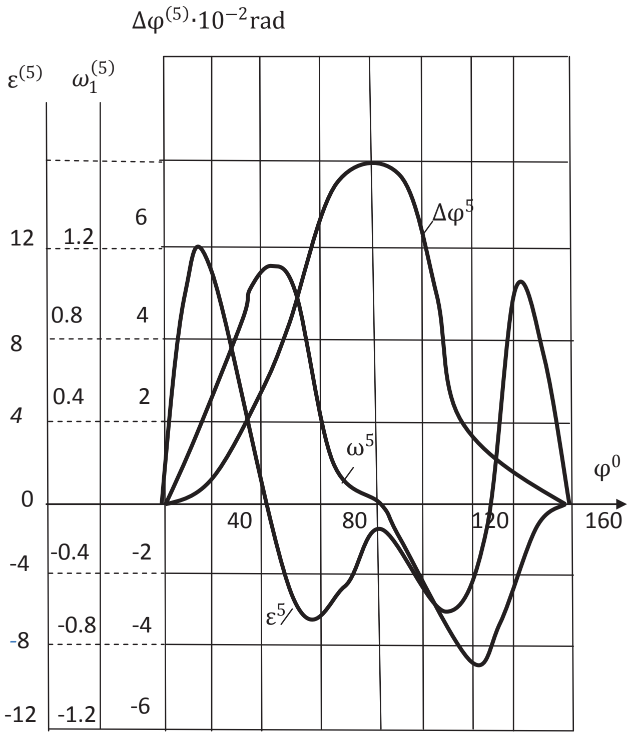

Fig. 2. Kinematic parameters of a sloped shield

Given the fact that the influence of a volumetric mass of raw cotton on the effectiveness of its ginning by different types of purifiers was not investigated, further study was aimed at determining the volumetric mass of raw cotton by the stages of technological process at a cotton plant and its effect on the ginning efficiency.

As is known, all the cotton cleaners for removing large weed impurities, applied industrially, contain one, two or more serrated sections depending on the type of a ginner, for example, the ginners 1HP (RH-1), CH-3M2 «Mehnat», UHK. The way these sections work ultimately determines the quality of ginning raw cotton, as well as fiber. That is why improving the raw cotton ginning depends primarily on the optimization of operation of the serrated section.

The serrated sections currently in use in the cotton industry employ serrated drums of a 480-mm diameter, which rotate at a linear speed of 7–7.5 m/s. The grates that are applied as working bodies for separating large weed inpurities and raw cotton (valves, partd of stalks, etc.) consist of circular 20-mm diameter grates, installed with a gap of 40 mm between them, and a gap within 15÷18 mm from the serrated drum.

There are installed rubbing brushes at the beginning of the grates for levelling the fed cotton on a serrated drum and fixing the removed parts on the teeth of the serrated gear.

The basic parameters that influence efficiency of the ginning by a serrated section are a diameter of the serrated drum, its circumferential speed, type of grates, and the gaps between them. Diameter of the serrated drum and the circumferential speed were investigated in a wide range at the end of 2012 on the raw cotton of different industrial and selection varieties, which is why a reconsideration of these parameters will not improve efficiency of the serrated section. As far as the type of a grate and the gaps between them are concerned, then there is a possibility to improve the ginning efficiency by examining the trihedral grates, applied in the textile industry, with a cross section of 25×19×12 mm.

The basic process in the ginning technology of raw cotton from large weedy impurities and from small waste is the impact interaction between cotton flying detachments and the grates. This very process requires additional detailed analysis. The main task in this case is a theoretical consideration of the dynamics of impact interaction between raw cotton and the grates, described by a system of linear and non-linear models.

An analysis of the process is based on the model of a raw cotton flying detachment, a transition to which is implemented under the following assumptions:

– all the forces acting on a flying detachment are in the same plane;

– actual properties of the flying detachment and fiber locks that connecting the gear and the flying detachment are modeled with elastic elements – linear or nonlinear;

– we assume the deformation of elastic elements to be small;

– we consider mass of the flying detachment to be concentrated in the center of gravity.

Then at impact moment $(t>0)$, the process schematic can be represented in the form shown in Fig. 3, where m and m_c denote masses of the flying detachment and the weed, l_1 is the length of elastic elements that connect the seed with a point of contact between the grate and the weed, l_2 is the length of connection between the tooth and the flying detachment center. Angles α, ψ, γ and ψ_1 coordinate elastic elements and velocities $\vartheta_1=\omega R_1$ and $\vartheta_0=\omega R$. Elastic elements possess rigidity characteristics C_1, C_2, C_3 and C_4 at linear $(n_i \neq I)$ dependence $P(y)$. We shall denote the length of elastic element C_4 as e_4.

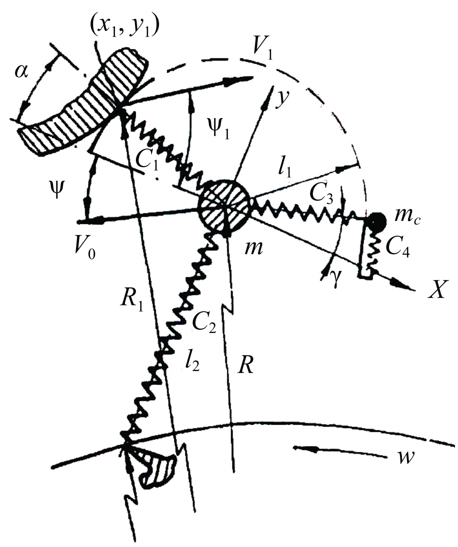

Fig. 3. Schematic of the impact moment in the ginning process

At the first stage, we shall explore behavior of the model without a mass of the weed, assuming it has little effect on the dynamics of the system.

Consider the motion of a closed linear-elastic system that executes an oscillatory process without breaking a contact between the flying detachment and the grate. We shall denote reactions of elastic elements as e_1 and e_2, functions of displacement as x and y, masses m, respectively, as

$$\begin{cases} S_1=-c_1(-x\cos\alpha+y\sin\alpha); \\ S_2=c_2 y, \end{cases} \tag{9}$$

and obtain dirrerential equations of motion

$$\begin{cases} m\ddot{x}+c_1x\cos^2\alpha-c_1y\sin\alpha\cos\alpha=0; \\ m\ddot{y}-c_1x\sin\alpha\cos\alpha+\left(c_1\sin^2\alpha+c_2\right)y=0. \end{cases} \quad (10)$$

By introducing, as accepted:

$$\begin{cases} a_{11}=\frac{c_1}{m}\cos^2\alpha;\ a_{12}=a_{21}=-\frac{c_1}{m}\sin\alpha\cos\alpha, \\ a_{22}=\frac{1}{m}\left(c_1\sin^2\alpha+c_2\right), \end{cases} \quad (11)$$

we shall obtain a canonical system of free oscillations in direct form:

$$\begin{cases} \ddot{x}+a_{11}x+a_{21}y=0, \\ \ddot{y}+a_{12}x+a_{22}y=0, \end{cases} \quad (12)$$

whose particular solution takes the form

$$\begin{cases} x=Axk\sin\left(P_kt+\varphi_k\right), \\ y=Ayk\sin\left(P_kt+\varphi_k\right). \end{cases} \quad (13)$$

Here A_k and P_k are, respectively, amplitude and frequency of the k-th form of natural oscillations of the system ($k=1,2$ – with two degrees of freedom). By substituting (13) in (12) and reducing by sine, we shall have a system of equations relative to unknown amplitudes, A_{xk} and A_{yk}

$$\begin{cases} A_{xk}\left(a_{11}-P_k^2\right)+A_{yk}a_{21}=0, \\ A_{xk}a_{12}+A_{yk}\left(a_{22}-P_k^2\right)=0. \end{cases} \quad (14)$$

System (14) is a homogeneous joint because the rank of its matrix is equal to the number of unknowns ($r=n=2$). Its solution $A_{xk}=A_{yk}=0$ corresponds to the rest of the system and is trivial. For it to have a non-zero solution, it is necessary that the determinant of the system is equal to zero

$$\det A=\begin{vmatrix} a_{11}-P_k^2; & a_{21} \\ a_{12}; & a_{22}-P_k^2 \end{vmatrix}=0. \quad (15)$$

Hence we have the condition imposed on frequency P_k so that $A_{xk}\neq 0$, $A_{yk}\neq 0$:

$$\left(a_{11}-P_k^2\right)\left(a_{22}-P_k^2\right)-a_{12}^2=0. \quad (16)$$

This yields two solutions (the first with a minus sign, the second with a plus sign). As we accepted $P_1\leq P_2$, the given solutions considering (14) take the form

$$P_{1,2}^2=\frac{\left(c_1+c_2\right)\pm\sqrt{\left[c_2+c_1\left(\sin^2\alpha-\cos^2\alpha\right)\right]+4c_1^2\sin^2\alpha-\cos^2\alpha}}{2m}. \quad (17)$$

At a change in $[\alpha]=\left[-\frac{\pi}{2};\frac{\pi}{2}\right]$, always $P_1\geq 0$; $P_2>0$ (at $\alpha=\frac{\pi}{2}$ and $\alpha=-\frac{\pi}{2}$, that is, on the boundaries of the interval, $P_1=0$, $P_2>0$ and the system has a single degree of freedom).

There is an interesting case when the elastic elements are mutually perpendicular ($\alpha=0$). Then $a_{12}=0$; $a_{12}=\frac{c_1}{m}$; $a_{22}=\frac{c_2}{m}$ and in systems (10) and (12) the equations are independent from each other (orthogonal) and their separate solution is possible:

$$P_1=\sqrt{\frac{c_1}{m}};\ P_2=\sqrt{\frac{c_2}{m}} \quad (18)$$

and mass m executes small oscillations, independent of each other, along the Ox and Oy axes.

For each frequency (P_1 and P_2), there is a certain correlation between A_{xk} and A_{yk}, called a form of oscillations for a system with 2 degrees of freedom. There are two such forms:

$$A_1=\frac{Ay_1}{Ax_1}=\frac{P_1^2-a_{11}}{a_{12}}=\frac{a_{12}}{P_1^2-a_{22}}=\frac{c_1\cos^2\alpha-mP_1^2}{c_1\sin\alpha-\cos\alpha}; \quad (19)$$

$$A_2=\frac{Ay_2}{Ax_2}=\frac{c_1\cos^2\alpha-mP_2^2}{c_1\sin\alpha-\cos\alpha}=\frac{c_1\sin\alpha-\cos\alpha}{c_1\sin^2\alpha+c_2-mP_2^2}. \quad (20)$$

It is easy to show that $P_1^2-a_{11}^2<0$, and $P_2^2-a_{11}>0$ and $\lambda_1>0$; $\lambda_2<0$ at $a_{12}<0$. This means that in the first case mass m moves along the Ox and Oy axes «synchronously» (in quadrants 1 and 111), and the accepted direction of axes allows us to conclude that elastic elements c_1 and c_2 are stretched and compressed at the same time; in this case, the oscillation frequency is equal to P_1 ($P_1<P_2$) and matches the most stable form of the motion of masses in one phase. It should be noted at once that this is exactly why the lowest frequency P_1 is the magnitude that defines the time of impact interaction between the flying detachment and the grate.

In the second case, elastic elements execute antiphase oscillations – stretching of spring C_1 is matched with compression C_2; in this case, oscillation frequency is higher $P_2>P_1$ and such a motion is less stable and more difficult to excite.

The total solution of system (10) is composed of a linear combination of particular solutions to (13) with four unknown parameters A_{x1}, A_{x2}, φ_1, φ_2 determined from the initial conditions of values of displacements and velocities m at moment $t=0$.

Hence, it follows that for a full analysis of the process it would suffice if the grate and a tooth of the saw did not change mutual position, or such change were insignificant during the impact. Therefore, for the further study we shall apply a method of the reverse motion. A full analysis of the process of impact interaction between the flying detachment and the grate of cleaner of large waste can be given either by the method of reverse motion, or by applying, in the point of contact between the flying detachment and the grate, a forcing effort in the form of a kinetic momentum. Both methods are equally valid and produce one result. We shall confine ourselves to the forst one, where it is more convenient to take into account initial conditions of the process.

We shall use a schematic in Fig. 3 and, in addition to the accepted designations, denote a displacement of the contact point between the flying detachment and the grate as magnitudes x, y_1, grate mass as μ, with $\mu\gg m$, and hereafter we shall consider it to be infinitely large. System (9) in this case will be rewritten in the form:

$$\begin{cases} S_1=-c_1\left[\left(x_1-x\right)\cos\alpha-\left(y_1-y\right)\sin\alpha\right], \\ S_2=c_2y, \end{cases} \quad (21)$$

while the motion will be described by four differential equations (22):

$$\begin{cases} M\ddot{x}+C_1(x_1-x)\cos^2\alpha-c_1(y_1-y)\sin\alpha-\cos\alpha=0,\\ m\ddot{x}-c_1(x_1-x)\cos^2\alpha+c_1(y_1-y)\sin\alpha-\cos\alpha=0,\\ M\ddot{y}_1-c_1(x_1-x)\sin\alpha\cos\alpha+c_1(y_1-y)\sin^2\alpha=0,\\ m\ddot{y}+c_1(x_1-x)\sin\alpha\cos\alpha-c_1(y_1-y)\sin^2\alpha+c_2y=0. \end{cases} \quad (22)$$

A system of four equations can be simplified by introducing new variables and establishing a link between the derivatives:

$$\begin{cases} \xi=x_1-x;\ \dot{\xi}=\dot{x}_1-\dot{x};\ \ddot{\xi}=\ddot{x}_1-\ddot{x},\\ z=y_1-y;\ \dot{z}=\dot{y}_1-\dot{y};\ \ddot{z}=\ddot{y}_1-\ddot{y}. \end{cases} \quad (23)$$

It is obvious that the new variables have the following meaning – ξ and z are, respectively, the deformations of element e_1 along the Ox, Oy axes; the first is negative because an increase in x_1 shortens the elastic element, while x – extends; the second is positive.

In the new coordinates, we shall have three equations.

$$\begin{cases} m\ddot{\xi}+c_1\xi\cos^2\alpha-c_1z\sin\alpha\cos\alpha=0,\\ m\ddot{z}-c_1\xi\sin\alpha\cos\alpha+c_1z\sin^2\alpha-c_2y=0,\\ m\ddot{y}+c_1\xi\sin\alpha\cos\alpha-c_1z\sin^2\alpha+c_2y=0, \end{cases} \quad (24)$$

that make it possible to fully describe the examined process, taking into account relative motion of the grate and the serrated surface.

It is obvious that if the grate is motionless relative to the saw $(x_1=0,\ y_1=0)$, the two last equations of systes (24) are identical, and it transforms to the form (10).

We shall obtain from the first and third equations (22) at $\mu\to\infty$

$$\begin{cases} \ddot{x}_1=0;\ \dot{x}_1=c_x;\ x_1=c_xt+c_{x1},\\ \ddot{y}_1=0;\ \dot{y}_1=c_y;\ y_1=c_yt+c_{y1}, \end{cases} \quad (25)$$

where constants C_x, C_{x1}, C_y, C_{y1} are determined from the initial conditions of the process (in reversed motion) – at $t=0$

$$c_x=\vartheta_1\cos\psi,\ y_1=C_y=\vartheta_1\sin\psi,\ c_{x1}=c_{y1}=0,$$
$$x_1=\vartheta_1t\cos\psi,\ y_1=\vartheta_1t\sin\psi. \quad (26)$$

That is, motion of the grate is executed in accordance with the law of the portable motion. Having x_1 and y_1, and having received a solution to the system relative to ξ, z, it is easy to find from (23) a natural motion (displacement) of mass m.

Denote, as previously,

$$\begin{cases} \dfrac{c_1\cos^2\alpha}{m}=a_{11};\ -\dfrac{c_1\sin\alpha\cos\alpha}{m}=a_{21}=a_{12},\\ \dfrac{c_1\sin^2\alpha}{m}=a_{22};\ \dfrac{c_2}{m}=a_{33}, \end{cases} \quad (27)$$

and reduce system (24) to the canonical form.

$$\begin{cases} \ddot{\xi}+a_{11}\xi+a_{21}z=0,\\ \ddot{z}+a_{12}\xi+a_{22}z-a_{33}y=0,\\ \ddot{y}-a_{12}\xi-a_{22}z+a_{33}y=0, \end{cases} \quad (28)$$

We shall search for a solution to system (28) in the form

$$\begin{cases} \xi=A_{\xi k}\sin(P_kt+\varphi_k),\\ z=A_{zk}\sin(P_kt+\varphi_k),\\ y=A_{yk}\sin(P_kt+\varphi_k). \end{cases} \quad (29)$$

Here $k=1,\ 2,\ 3$, but, as will be shown below, there are two such frequencies. Substituting (29) in (28) and reducing by the trigonometric function, we shall have the matrix of equation coefficients.

$$A=\begin{pmatrix} (a_{12}-P^2); & a_{21}; & 0\\ a_{12}; & (a_{22}-P^2); & -a_{33}\\ a_{12}; & -a_{22}; & (a_{33}-P^2) \end{pmatrix}. \quad (30)$$

whose rank is $r=3$, since $\det A\neq 0$.

Simplify (30) by adding elements of line 2 to line 3, then we subtract from the elements of column 2 the elements of the third, which, in line with a rule of matrix transform, does not change their magnitude. Then we have

$$A=\begin{pmatrix} (a_{12}-P^2); & a_{21}; & 0\\ a_{12}; & (a_{22}+a_{33}-P^2); & -a_{33}\\ 0; & 0; & P^2 \end{pmatrix}; \quad (31)$$

$$\det A=-P^2\left[(a_{11}-P^2)(a_{22}+a_{33}-P^2)-a_{12}^2\right]=0. \quad (32)$$

Because the analyzed system of equations relative to the unknown amplitudes is homogeneous, (32) is a condition for the existence of a nontrivial solution. Here we immediately have to consider (27)

$$\begin{cases} P_0=0;\\ P_{1,2}^2=\dfrac{(c_1+c_2)\pm\sqrt{\left[c_2+c_1(\sin^2\alpha-\cos^2\alpha)\right]+4c_1^2\sin^2\alpha\cos^2\alpha}}{2m}, \end{cases} \quad (33)$$

that for P_1 and P_2 coincides with (17). Root $P_0=0$ means a non-periodic motion M and motion m that corresponds to it.

The total solution to the system will be written in the form:

$$\begin{cases} \xi=A_{\xi1}\sin(P_1t+\varphi_1)+A_{\xi2}\sin(P_2t+\varphi_2),\\ z=A_{z1}\sin(P_1t+\varphi_1)+A_{z2}\sin(P_2t+\varphi_2),\\ y=A_{y1}\sin(P_1t+\varphi_1)+A_{y2}\sin(P_2t+\varphi_2). \end{cases} \quad (34)$$

Two of the three solutions in (34) are identical. Substitute their particular values in (28) and find the ratio of amplitudes:

$$A_k=\frac{A_{zk}}{A_{\xi k}}=\frac{a_{12}}{p^2-a_{33}-a_{22}};\ \frac{A_{yk}}{A_{\xi k}}=-\frac{a_{12}}{p^2-a_{33}-a_{22}}. \quad (35)$$

Given the multidirectionality of coordinates z and y, dividing the first equation (35) by the second, we obtain

$$\frac{A_{zk}}{A_{yk}}=1. \quad (36)$$

Analysis of the form of oscillations, considered above, holds here for the periodic component of the overall motion of the system. Initial parameters of variables z and ξ are easily found from (23) and (26):

$$\begin{cases} z_0 = 0;\ \dot{z}_0 = v_1 \cos\psi_1, \\ \xi_0 = 0;\ \dot{\xi}_0 = v_1 \sin\xi_1. \end{cases} \tag{37}$$

It is possible to obtain equivalent to it from system (34) by replacing the arbitrary constants with the new ones.

$$A_{ij} = \sqrt{Bi^2 j + ci^2 j}, \tag{38}$$

$$\varphi_{ij} = \operatorname{arctg}\frac{c_{ij}}{B_{ij}}. \tag{39}$$

Here $i = \xi VZ;\ j = IV2$ and consider in this case (39).

Constants are to be found from initial conditions (37):

$$C_{\xi 1} = C_{\xi 2} = 0 \ (\text{at } P_1 \neq P_2), \tag{40}$$

in this case, x and y can be found from (23), (26), (35), (39), (40).

Add to this that

$$\begin{cases} B\xi_1 A_1 = \dfrac{c_1 v_1 (m p_2^2 \cos\psi_1 - c_1 \cos^2 a \cos\psi_1 + c_1 \sin a \cos a \sin\psi_1) \sin a \cos a}{m p_1 (p_2^2 - p_1^2)(c_1 \sin^2 a + c_2 - m P_1^2)}, \\ B\xi_2 A_2 = \dfrac{c_1 v_1 (m p_1^2 \cos\psi_1 - c_1 \cos^2 a \cos\psi_1 + c_1 \sin a \cos a \sin\psi_1) \sin a \cos a}{m p_1 (p_2^2 - p_1^2)(c_1 \sin^2 a + c_2 - m P_2^2)}. \end{cases} \tag{41}$$

It is obvious that if a change in ξ and z is of a periodic character, then displacements x and y contain both harmonic motion and aperiodic, proportional to time.

$$x = v_1 t \cos\psi_1 - \xi,\ y = v_1 t \sin\psi_1 - z. \tag{42}$$

If $\psi_1 = 0$, then only a periodic motion occurs along the direction of axis. At $\psi_1 > 0$ tension of element C_2 subsides, and, because it represents a flexible connection, with a decrease in tension to zero, it is possible that the flying detachment falls off the gear of the serrated drum. Total displacements along Ox might be missing at $\psi = \frac{\pi}{2}$ and $\psi_1 = -\frac{\pi}{2}$ in boundary variants that are not applicable in the process of removing large waste.

At $\alpha = 0$ from (39), we have considering (40) and (41):

$$\begin{cases} \xi = \dfrac{\vartheta_1}{P_1} \cos\psi_1 \sin P_1 t, \\ z = -y = \dfrac{\vartheta_1}{P_2} \sin\psi_1 \sin P_2 t. \end{cases} \tag{43}$$

In addition, at $\psi_1 = \operatorname{arctg} A_{1,2}$ only one form of natural oscillations of the system is possible. At $\alpha = 0$ and $\psi_1 = 0$ oscillations of the system are possible only along the Ox axis (at $\alpha = 0$). At $\psi_1 = \pi/2$ oscillations occur along the Oy axis.

5. Discussion of results of studying the process of formation of the upper part of a bundle and the influence of density on the adhesion between waste and raw cotton

We studied theoretically and experimentally the process of throwing and fixing the particles of cotton on the serrated gear. Based on the elastic multi-link model, we analyzed energy of the process and investigated transformation of the potential energy of flying detachment deformation into kinetic energy.

A comprehensive theoretical and experimental study was conducted of the impact interaction between the flying detachments of raw cotton and the ginning grate. We examined an elastic model of the flying detachment that interacts with the grate, and performed an analysis of natural frequencies and the form of the system's oscillations. It was demonstrated that it is possible, based on this, to define the laws of motion of the flying detachment, to calculate the time, maximal deformation and the impact force between the flying detachment and the grate.

A complex experimental study into an impact interaction between the flying detachment and the grate was carried out. It was established that at an increase in the velocity speed of the shell of serrated drum from 5 to 9 m/s, the impact force changes from 0.39 to 0.72 n. Results of experimental research into the impact process were approximated by linear and non-linear models of elastic properties of fibrous particles. It is shown that the impact force depends on the deformation magnitude in a degree of 1.39.

Research results make it possible to mechanize the process of formation of bundles of raw cotton. An application of mechanical bundling press in the cotton ginning industry eliminates manual labor. Present work is a continuation of scientific studies [15–17].

6. Conclusions

1. The proposed method of kinematic analysis of the cam-lever mechanisms makes it possible to automate the process of obtaining and solving an equation to determine kinematic parameters of all elements of the mechanism at the assigned law of motion of the drive link, which is the cam.

2. The proposed calculation algorithm allows performing an analysis of a sloped shield, forming the upper layer of a cotton bundle. A special feature is that changing kinematic parameters of the shield can alter the shape of the upper part of the bundle depending on the humidity and density of raw cotton that is packed.

3. We studied theoretically and experimentally the process of cotton ginning, taking into account the impact interaction between the flying detachments and the grate of the cleaner. It is shown that the higher the density of raw cotton in a bundle, the lower the ginning effect. It was established that at ginning the impact force depends on the deformation magnitude in a degree of 1.39.

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