Analysis of Operational Load of Flat-Parallel Moving Movable Frame of Cotton Harvesting Apparatus

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Abstract: The article provides a brief description and a basic diagram of a new flat-parallel moving movable frame of a vertical-spindle cotton harvesting apparatus. The nature of the forces acting on the frame is analyzed taking into account changes in the design of the suspension of the movable sections on the frame by the suspension and the drive of the spindle drums and brush pullers. Calculation schemes of the new design of the movable frame under the action of reaction forces from the interaction of the frame with the surfaces of the bed, with elements of cotton bushes, inertial and load forces of the drive elements of the working bodies are developed. Based on the developed calculation schemes, equations of the forces acting on the flat-parallel moving movable frame with a new design of the suspension are compiled.

1 INTRODUCTION

The process of economic modernization in the Republic of Uzbekistan shows the further advancement of society towards liberalization, expressed in the transformation of economic relations and the formation of an innovative model for the development of the country's economy. Today, as a result of the rapid development of the "new economy", the strengthening of the connection between the capital market and new technologies, the increase in their mobility and the growth in the scale of the creation and use of knowledge, technologies, products, services, real conditions have arisen for solving the tasks set in the Development Strategy for five priority areas of development of the Republic of Uzbekistan in 2017-2021 (Decree of the President of the Republic of Uzbekistan PF-4947, 2017). The place of a given country in the world economic community, its level of competitiveness in the world arena of countries significantly depends on the system of formation of new knowledge and technologies. It should be noted that 80-95% of the growth of GDP of developed countries of the world falls on knowledge-intensive industries implementing innovations for their development, in other words, in these countries the innovative economy is widely developing (Decree of the President of the Republic of Uzbekistan UP 3416, 2017).

Agricultural engineering of the Republic of Uzbekistan is one of the key industries and is focused mainly on the production of tractors and machines for cultivating and harvesting cotton and has great potential for exporting them to neighboring and distant countries. Currently, there are 18 enterprises operating in it, including 5 joint-stock companies (JSC), 13 joint ventures (JV) with leading global companies such as CNH, CLAAS, Lemken, John Deere, etc. (Matchanov, 2023). Domestic semi-mounted vertical-spindle (VS) cotton harvesting machines (CHM) on a tractor, manufactured by JSC TTP, are inferior in productivity and completeness of harvesting to American self-propelled horizontal-spindle (HS) machines. But it is not profitable for farms to purchase an expensive self-propelled CHM, which operates only 20-30 days a year. They find it convenient and profitable to use a mounted or semi-mounted CHM, the power source (tractor) of which can be used during the year for other agricultural work. According to calculations by Research Institute of Agricultural Mechanization Uzbekistan, the efficiency of a mounted CHM is twice as high as that

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of a self-propelled one (Khojiev et al., 1994). The comparatively low cost and operating costs of the semi-mounted VS CHM, high selective capacity for crop maturity and the ability to harvest at 55-60% boll opening, as well as the ability to quickly (in 2-3 hours) mount it on a tractor and dismount it after the end of the harvesting season for use in other agricultural work, determine their potential not only for Uzbekistan and the countries of Central Asia, but also for other cotton-producing countries in the northern belt, starting to harvest at a low boll opening (Abdazimov et al., 2011).

One of the factors reducing the technical level of serial semi-mounted tractors VS CHM series MX is the insufficient stability of technological adjustments (width of the working gap, staggered arrangement of the spindles of adjacent drums) of the harvesting apparatus (HA), resulting from the failure to improve the design of the frame and drive of the spindle drums of the movable section (Abdazimov et al., 2014), is that when changing the width of the working gap HA, the staggered arrangement of the spindles of adjacent drums is disrupted, leading to deterioration of the agrotechnical indicators (ATI) of the CHM. In this case, the magnitude of the disruption of the staggered arrangement of the spindles of adjacent drums consists of two components - from a change in the position of the frame of the movable section in space and from an additional rotation of the spindle drum of the movable section due to the presence of a dog (tooth-lever) mechanism for driving the drums of the movable section. That is, the division of the HA into movable and fixed sections and the presence of a gear-lever (drive) mechanism in the drive design and ensuring the width of the working gap by moving only one outer section, during which additional turns of the spindle drum of the movable section occur, lead to a violation of the staggered arrangement of the spindles of adjacent drums and, as a consequence, to a deterioration in the quality and reliability of the machine.

The above-mentioned shortcomings have been eliminated in the new design of the cotton harvesting apparatus movable frame developed at the Department of Ground Transport Systems at Tash STU (Abdazimov et al., 2021), the diagram of which is shown in Figure 1, a-front view, b-top view.

In the new design of the movable frame of sections (Abdazimov et al., 2021), the drive of the spindle drums 5 and strippers 6 (see Fig. 1, a and b) of the movable frames of sections 7 and 8 is carried out using bevel 3 and cylindrical gears 4 located in the reducer 2. The reducers of the left 7 and right 8 movable sections are connected to each other by an inter-section splined cardan shaft 1.

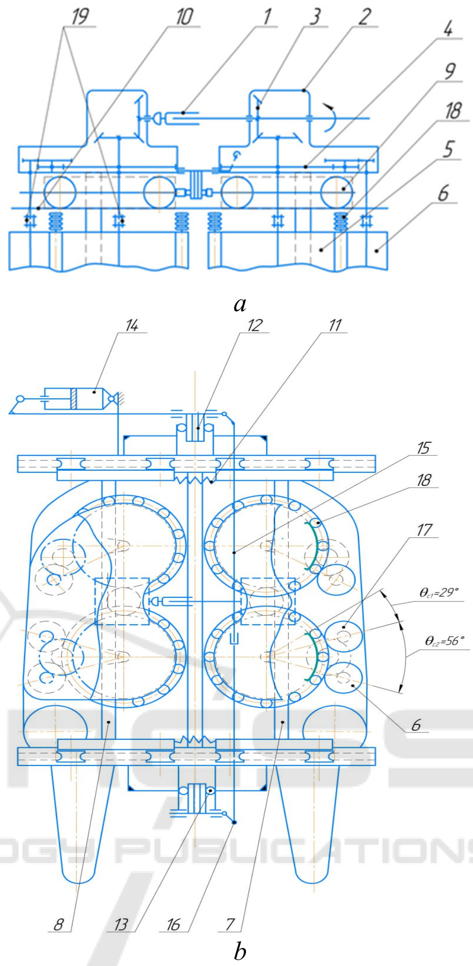


Figure 1: Schematic diagram of a cotton harvesting apparatus with plane-parallel moving movable frames.

The movable frames of the section contain rollers 9, by means of which they can move on the transverse bars of the guides 10 of the apparatus frame, are pulled together by springs 11 and symmetrically plane-parallel move apart relative to the axis of the cotton row when their projections with rollers 13 of the cams 12 of the working gap adjustment mechanism act on them, containing a hydraulic cylinder 14, kinematically connected by a rod 15 and a lever 16 of the axis of the cams 12. To increase reliability and reduce energy consumption, the bevel and cylindrical transmissions of the reducer of the movable section are made helical. To ensure maintainability and technical maintenance, the shafts in the working members - spindle drums 5 and strippers 6 are made composite - part in the reducer with drive gears, the other part in the working members and are connected to each other by sleeve couplings 19. If technical maintenance or repair is necessary, the said working members can be dismantled from the movable section by removing the couplings.

Due to the changes made in the design of the suspension of the movable frames on the frame and the drive of the working bodies (the exclusion from the design of the vertical hinged suspension of the movable frame to the frame and the toothed-lever mechanism of the drive), the stability and reliability of the technological adjustments of the HA are ensured, which contributes to an increase in the ATI and productivity of the CHM. The purpose of this work is to develop theoretical and experimental foundations for substantiating the main parameters of the new design of the movable section of the cotton harvesting apparatus, taking into account its operational load in real conditions. For this purpose, an analysis of the forces acting on the movable frame during the operation of the machine in cotton rows is necessary.

2 MATERIALS AND METHODS

When the HA is operating in the field, the following forces act on the movable frame (Fig. 2):

1. Inertial forces - $m_b\,\ddot{x}$, $m_b\,\ddot{y}$, $m_b\,\ddot{z}$ and $m_c\,\ddot{x}$, $m_c\,\ddot{y}$, $m_c\,\ddot{z}$, which act on the frame through the upper and lower supports of the spindle and removable drums, and the forces $m_b\,\ddot{z}$ and $m_c\,\ddot{z}$ only through the upper supports (since the outer races of the bearings of the drum shaft supports are seated in supports with a sliding fit, which allows the lower ends of the drum shafts to move relative to the lower rods of the frame, and the upper part of the drum shafts is stationary in the vertical direction relative to the upper beam of the frame). These forces arise at the center of gravity of the spindle and removable drums during oscillations of the cleaning apparatus on the suspension mechanisms.

2. The weights of the spindle G_b and removable G_c drums, which are applied to the upper supports of the drums (the weight forces of the gear blocks were neglected due to their insignificance).

3. Bush pressures $N_k(t)$, directed perpendicular to the *xoz* plane and transmitted to the upper and lower supports of the spindle drums.

4. Compression springs P_{n1}^{x}, P_{n1}^{y}, P_{n1}^{z} and P_{n2}^{x}, P_{n2}^{y}, P_{n2}^{z} transmitted to the frames.

5. Soil pressures P_{n2}^{x} and P_{n2}^{z} (Fig. 3, a), acting on the lower front part of the frame. These forces arise due to the unevenness of the furrow, when driving over ditch-irrigations with lowered devices, etc.

6. Soil pressure on the lower frame rod. These forces arise due to the interaction of the lower surface of the frame with the soil during the process of harvesting raw cotton (Fig. 3, b).

7. Soil pressure on the lateral surface of the lower frame rod (not shown in Fig. 2 and 3). Lateral forces arise due to the interaction of the lateral surface of the lower frame with the bed when the CHM exits onto the headland with lowered devices.

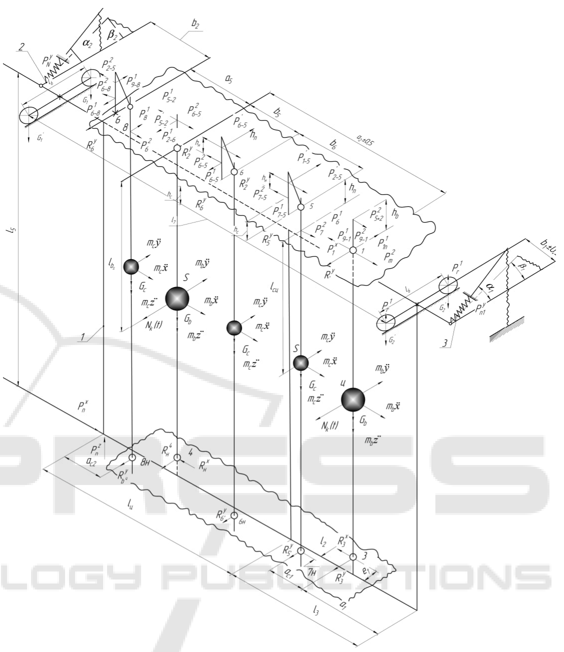


Figure 2: Diagram of forces acting on the movable frame of the HA.

To determine the inertial forces $m_b\,\ddot{x}$, $m_b\,\ddot{y}$, $m_b\,\ddot{z}$ and $m_c\,\ddot{x}$, $m_c\,\ddot{y}$, $m_c\,\ddot{z}$, it is necessary to experimentally establish the values of the accelerations $\ddot{x}$, $\ddot{y}$, $\ddot{z}$ of the center of gravity of the apparatus during its oscillations. Then the actions of the latter on the supports are determined from the following relationship (see Fig. 2).

$$P_{b.o}^{x} = \pm m_i\ddot{x}\left(1-\frac{l_{ц}}{l}\right),\quad P_{н.o}^{x} = \pm m_i\ddot{x}\frac{l_{ц}}{l},$$

$$P_{b.o}^{y} = \pm m_i\ddot{y}\left(1-\frac{l_{ц}}{l}\right),$$

$$P_{н.o}^{y} = \pm m_i\ddot{y}\frac{l_{ц}}{l},\quad P_{b.o}^{z} = \pm m_i\ddot{z} + G_i l_n$$

where l_n is the distance between the supports of the spindle drums and pullers.

The bush pressure is determined by the formula (Glushchenko, 1985):

$$N_k(t) = \sum_{i=0,1}^{i_k} N_{ki} \, cos i\omega_k t, \tag{1}$$

where N_{ki} − amplitude of the "i" harmonic;

$$\omega_k = \frac{2\pi V_m}{S_k}$$

i_k − number of the last harmonic taken into account;

V_m − driving speed of the CHM;

S_k − average distance between adjacent cotton nests (S_k depends on the planting pattern of the bushes).

The forces of the tension springs are equal

$$P_{n1} = k_1 \Delta l_1 \text{ и } P_{n2} = k_2 \Delta l_2,$$

where k_1 and k_2 − spring stiffness coefficients, respectively, Δl_1 and Δl_2 − deformations of the springs accordingly.

The projections of the forces of the tension springs on the *x*, *y*, and *z* axes will be equal to (Fig. 2) (Abdazimov et al., 2022):

$$P_{n1}^{x} = P_{n1} sin\beta_1, \;\; P_{n1}^{y} = P_{n1} cos\alpha_1, \;\; P_{n1}^{z} = P_{n1} sin\alpha_1$$

$$P_{n2}^{x} = P_{n2} sin\beta_2, \;\; P_{n2}^{y} = P_{n2} cos\alpha_2,$$

$$P_{n2}^{z} = P_{n2} cos\alpha_2. \tag{2}$$

The force of soil pressure on the front part of the lower frame is determined by the formula (Handbook, 1968):

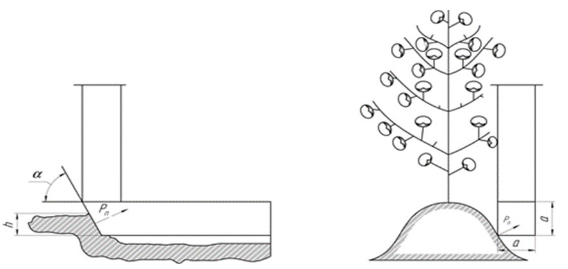
$$P_n = \varepsilon \rho a h v^2 \tag{3}$$

where ε – dimensionless coefficient depending on the frame shape and soil properties;
ρ – soil density, kg/m^3;
υ – driving speed of the cotton harvesting apparatus, m/s;
h – height of soil unevenness;
a – rod width;
α − rod elevation angle.

The projections of the soil pressure force P_n on the *x* and *z* axes are equal to

$$P_n^{x} = P_n sin\alpha \qquad P_{п}^{z} = P_n cos\alpha, \tag{4}$$

The pressure of the soil on the lower and lateral surfaces of the lower frame rod can be replaced by a uniformly distributed load applied along the length of the lower frame rod, i.e.



a *b*

Figure 3: Scheme of the action of soil pressure on frames when moving a CHM across an arrow ditch with lowered devices

$$N = f \cdot P_n \tag{5}$$

where f – coefficient of friction;
a – height of the lower rod (see Fig. 2).

To determine the forces from the gear transmissions acting on the frame, let us consider the drive diagram of the spindle and removable drums of the HA with a new design of the movable section (Abdazimov et al., 2021).

To determine the load on the shaft or on the cantilever axis, it is necessary to draw a diagram for each drive gear. Fig. 4 shows a diagram for determining the loads on the spindle and removable drums and on cantilever axles of parasitic gears.

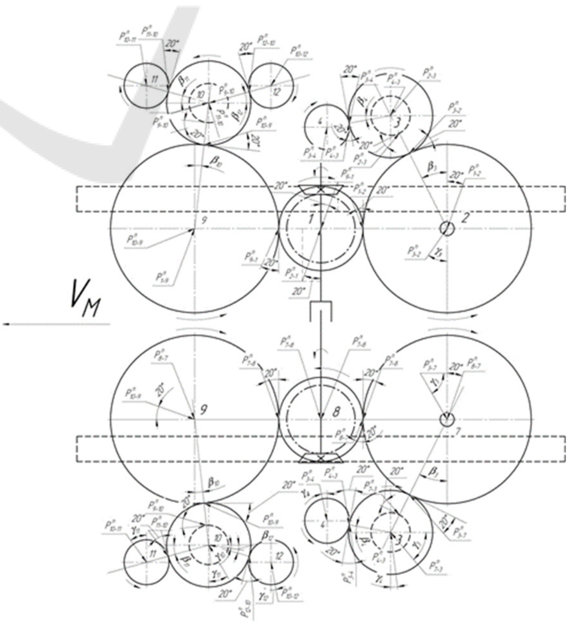


Figure 4: Scheme for determining loads from gear transmissions.

Considering that the values of torque on the spindle shafts and removable drums are known, we determine the values of torque on the drive gears:

$$M_{Z_8} = M_C, \quad M_{Z_6} = \frac{M_{Z_8}Z_6'}{\eta Z_8}, \quad M_{Z_2} = \frac{M_{Z_6}Z_2}{\eta Z_6} + M_{ш}, \quad M_{Z_5} = \frac{1}{\eta}\left(\frac{M_{Z_2}Z_5}{Z_2} + \frac{M_{Z_7}Z_5'}{Z_7}\right), \quad M_{Z_1} = \frac{M_{Z_5}Z_1}{Z_5} + M_{ш}, \tag{7}$$

$$M_{Z_7} = M_C, \quad M_{Z_9} = \frac{M_{Z_1}Z_{10}}{\eta Z_1}, \quad M_{Z_{10}} = \frac{M_{Z_9}Z_{10}}{\eta Z_9},$$

where M_C and $M_{ш}$ are the torques on the shafts of the removable and spindle drums.

Let us consider the action of the Z_8 puller gear on the Z_6' gear block gear. The action of the Z_8 gear on the Z_6' gear is replaced by the force $P_{8\cdot 6}^n$, directed along the engagement line "*ab*" (see Fig. 4.).

The total pressure on the tooth is determined by the formula

$$P_{8\cdot 6}^n \frac{2\,M_{Z_6}}{D_{Z_8}cos\alpha}, \tag{8}$$

where $M_{Z_8} -$ torque on puller shaft 8;

$D_{Z_8} -$ diameter of the pitch circle of the gear Z_8;

$\alpha = 20° -$ engagement angle.

Since gear Z_6' of the gear block rests on axis 6, the action of gear Z_6' on this axis can also be replaced by force $P_{8\cdot 6}^n$.

Similarly, we get

$$P_{6\cdot 2}^n = \frac{2\,M_{Z_6}}{D_{Z_6}cos\alpha}, \quad P_{2\cdot 5}^n = \frac{2\,M_{Z_2}}{D_{Z_2}cos\alpha}, \quad P_{7\cdot 5}^n = \frac{2\,M_{Z_7}}{D_{Z_7}cos\alpha}, \tag{9}$$

$$P_{S\cdot 1}^n = \frac{2\,M_{Z_5}}{D_{Z_5}cos\alpha}, \quad P_{1\cdot 9}^n = \frac{2\,M_{Z_1}}{D_{Z_1}cos\alpha}, \quad P_{9\cdot 10}^n = \frac{2\,M_{Z_9}}{D_{Z_9}cos\alpha}, \quad P_{10\cdot 11}^n = \frac{2\,M_{Z_{10}}}{D_{Z_{10}}cos\alpha}.$$

The projections of these forces along the *Y* and *X* axes will be equal to

$$P_{8\cdot 6}^1 = P_{8\cdot 6}^n sin\gamma_6, \quad P_{8\cdot 6}^2 = P_{8\cdot 6}^n cos\gamma_6, \quad P_{6\cdot 2}^2 = P_{6\cdot 2}^n tg\alpha,$$

$$P_{6\cdot 2}^1 = P_{6\cdot 2}^n cos\alpha, \quad P_{2\cdot 5}^2 = P_{2\cdot 5}^n tg\alpha, \quad P_{2\cdot 5}^1 = P_{2\cdot 5}^n cos\alpha,$$

$$P_{7\cdot 5}^1 = P_{7\cdot 5}^n cos\gamma_5, \quad P_{7\cdot 5}^2 = P_{7\cdot 5}^n sin\gamma_5, \quad P_{5\cdot 1}^1 = P_{5\cdot 1}^n cos\alpha, \tag{10}$$

$$P_{5\cdot 1}^2 = P_{5\cdot 1}^n tg\alpha, \quad P_{1\cdot 9}^1 = P_{1\cdot 9}^n cos\gamma_9, \quad P_{1\cdot 9}^2 = P_{1\cdot 9}^n sin\gamma_9,$$

$$P_{9\cdot 10}^1 = P_{9\cdot 10}^n tg\alpha, \quad P_{9\cdot 10}^2 = P_{9\cdot 10}^n cos\alpha, \quad P_{10\cdot 11}^1 = P_{10\cdot 11}^n cos\gamma_{10},$$

$$P_{10\cdot 11}^2 = P_{10\cdot 11}^n sin\gamma_9,$$

where $\gamma_6 = \beta_6 - \alpha$; $\gamma_5 = \beta_5 - \alpha$; $\gamma_9 = \beta_1 - \alpha$; $\gamma_{10} = 90° - (\beta_6 + \alpha)$.

The obtained expressions of forces acting on the movable frame are typical for the static mode. In the dynamic mode, the load of the movable frame of the cotton harvesting apparatus section as such has not been studied as a whole. Although the dynamics of its working elements, such as spindles, spindle and stripper drums, as rotation units were studied in sufficient detail in the works (Glushchenko, 1985; Glushchenko, 1990; Turanov, 1989). Analytical expressions of bending, torsional, pendulum and axial oscillations of the spindle, bending and torsional oscillations of the spindle drum shaft and the stripper shaft were obtained, approximate methods for solving systems of equations describing the specified processes were proposed. Calculation and experimental studies have established that the working elements of the drum-type CHM themselves are sources of excitation of dynamic loads, leading to the failure of individual connections of the parts of these working elements due to their design imperfections (Turanov, 1989). The specified loads are transferred to some extent by the frames that carry them, but their values in relation to technological ones (impacts from the surfaces of the ridges of the bed and the bush mass in the row) are significantly less (Turanov, 1989). Consequently, it is of great importance to determine the values of external impacts on the movable frame under operating conditions.

3 RESULTS AND DISCUSSION

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4 CONCLUSIONS

The conducted studies substantiated the methodology for drawing up calculation schemes for determining the loads acting on the elements of the new HA structures with plane-parallel moving movable frames. The developed calculation methodology allows determining the values of bending and torque moments in the sections of movable frames taking into account individual values determined based on the results of planned experiments.

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