ISSN: 2350-0328
International Journal of Advanced Research in Science, Engineering and Technology
Vol. 7, Issue 2 , February 2020

Synthesis and Kinematic Study of a Vibrator with a Gear-Lever Mechanism

A. Urunov, S.Igamov, A.Nuriddinov

Candidate of Technical Sciences, Department of general technical sciences and technologies, Samarkand State University, Samarkand, Uzbekistan
Assistant, Department of general technical sciences and technologies, Samarkand State University, Samarkand, Uzbekistan
Assistant, Department of general technical sciences and technologies, Samarkand State University, Samarkand, Uzbekistan

ABSTRACT: This article discusses the structure, movement, the degree of excitation determination, the synthesis, and the kinematic analysis of combinational mechanisms.

KEY WORDS:gear-lever mechanism, combined mechanism, degree of mobility, kinematic pairs.

I.INTRODUCTION

Under the guidance of Honored Worker of Uzbekistan, Doctor of Technical Sciences, prof. G.Sh. Zakirov designed and manufactured an experimental bench for the implementation of the technological process and optimization of the parameters of the working bodies of the spindle-free cotton-picking apparatus according to the proposed method for harvesting raw cotton [1].

One of the main units of this apparatus is the vibrator of cotton bushes. It is designed to reduce the strength of the bond and the separation of the lobes from the opened boxes by reporting vibrations to the stem of the cotton bush.

We have developed and manufactured various options for the vibrator, including a vibrator with a gear-lever mechanism and a mechanical drive.

The synthesis task was to determine the size of the links of the mechanism at which the work of the latter is carried out without jamming, and the required trajectory and amplitude of the working body of the vibrator are set.

To carry out the synthesis, the gear-lever mechanism (Fig. 1) is replaced by the given lever mechanism. Since the interlocking gears have the same number of teeth and rotate with equal angular speeds, the gear-link mechanism under consideration will be accepted as two four-link mechanisms having a common output link. Then the synthesis problem will be reduced to determining the dimensions of the crank AB = r (Fig. 2) of the connecting rod BC=$l_{BC}$of the eccentricity axis e - between the axis of movement of the slider and the vertical line passing through the axis of rotation of the crank.

The conditions for eliminating jamming of the mechanism are characterized by the maximum allowable pressure angles $\varphi_x$and $\varphi_p$respectively, at idle and working strokes of the output link [2], i.e. shock organ. This condition is satisfied at φ_x =20° andφ_p =10°.

The required trajectory of the output link, according to the principle of operation of the vibrator described above, was the plane-parallel reciprocating movement of the shock working element in the direction perpendicular to the axis of the row of cotton.

The course of the working body, as follows from the results of experimental work, should be within S = 70 ... 80 mm.

Let's draw a diagram of the reduced crank mechanism in its four positions (Fig. 2):

1. The position of the mechanism with the maximum distance of the slider from the axis of rotation of the crank (position $AB_0\ C_0$).

2. The position of the mechanism with a minimum distance of the slider from the axis of rotation of the crank (position $AB_1\ C_1$).

3. The position of the mechanism during the working stroke (to the bush) of the slider with a maximum pressure angle - φ_w =10°

4. The position of the mechanism at idle (from the bush) of the slider with a maximum pressure angle – φ_i =20°.

ISSN: 2350-0328
International Journal of Advanced Research in Science, Engineering and Technology
Vol. 7, Issue 2 , February 2020

In this case, the distance C_0C_1 between the extreme positions of the slider should be equal to the required stroke size of the impact tool S = 80 mm.

Of $\Delta \acute{A}B_2C_2$ can write

$$sin\varphi_w = \frac{A\acute{B}_2}{B_2C_2} = \frac{r-e}{l_{BC}} \quad (1)$$

and from $\Delta \acute{A}B_3C_3$ you can write

$$sin\varphi_i = \frac{A\acute{B}_3}{B_3C_3} = \frac{r+e}{l_{BC}} \quad (2)$$

We introduce the notation: $\frac{r}{l_{BC}} = \gamma, \frac{e}{l_{BC}} = \varepsilon$, where, γ and ε are dimensionless coefficients of size ratios.

Then
$$\left.\begin{array}{r}\sin\varphi_w = \gamma - \varepsilon\\ \sin\varphi_i = \gamma + \varepsilon\end{array}\right\} \quad (3)$$

Solving these equations together, we obtain:

$$\gamma = 0{,}5(\sin\varphi_i + \sin\varphi_w) = 0{,}235 \quad (4)$$
$$\varepsilon = 0{,}5(\sin\varphi_i - \sin\varphi_w) = 0{,}075 \quad (5)$$

From $\Delta A\acute{A}\acute{C}_0and\Delta A\acute{A}\acute{C}_1$we can write:

$$S = \acute{A}\acute{C}_0 - \acute{A}\acute{C}_1 \quad (6)$$

Where from

$$l_{BC} = \frac{s}{\sqrt{(1+\gamma)^2 - E^2} - \sqrt{(1-\gamma)^2 - E^2}} = 170\ mm \quad (7)$$

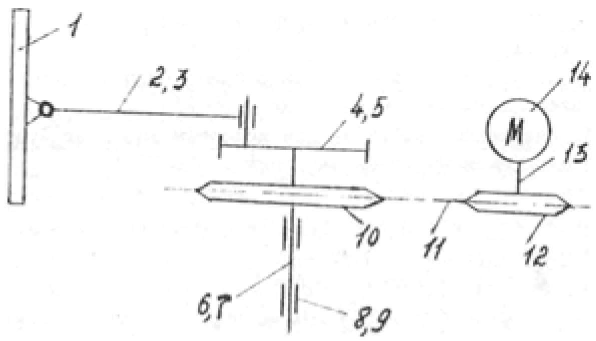

Fig. 1. Schematic diagram of a vibrator of cotton bushes with a drive: 1- working body; 2, 3-rods; 4, 3-gear wheels; 6,7-ball bearings; 8, 9-axis; 10-sprocket sprocket; 11- chain; 12-lead sprocket, 13-shaft, 14-motor

ISSN: 2350-0328
International Journal of Advanced Research in Science, Engineering and Technology
Vol. 7, Issue 2 , February 2020

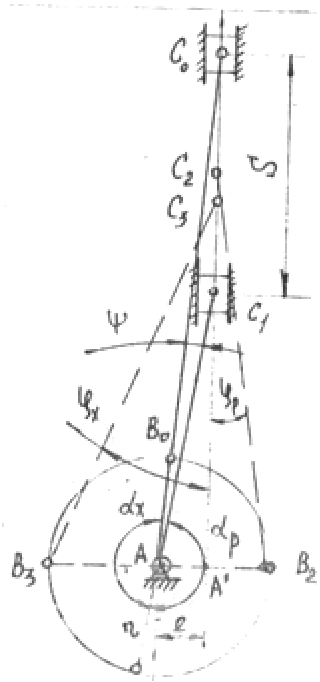

Fig. 2 Scheme for the synthesis of the vibrator mechanism

II. EXPERIMENTAL RESULTS

So the length of the connecting rod should be $l_{BC} = 170$ mm. Then the length of the crank is $r = \gamma \cdot l_{BC} = 0{,}23 \cdot 170\ mm = 40$ mm, and the magnitude of the eccentricity - $e = \varepsilon \cdot l_{BC} = 0{,}075 \cdot 170\ mm = 13\ mm$.

The objective of the kinematic study was to determine the kinematic parameters of the links and their points. The following kinematic parameters were taken:

ω_r-is the angular velocity of the crank, i.e. drive gear;

V_B, V_c- speed of points B and C;

ω_{BC} - angular speed of the connecting rod;

a_B, a_c- acceleration of points B and C;

ε_{BC} - the angular acceleration of the connecting rod.

The coordinates of the crank point B (Fig. 3) are written in the form:

$$\left.\begin{aligned} X_B &= r \cdot \sin\varphi_1 \\ Y_B &= r \cdot \cos\varphi_1 \end{aligned}\right\} \tag{8}$$

Differentiating equation (8) once, we obtain the projection of the velocity of point B, and differentiating the acceleration of the same point twice, we obtain:

$$\left.\begin{aligned} V_{B_x} &= r \cdot \omega_r \cdot \cos\varphi_1 \\ V_{B_y} &= r \cdot \omega_r \cdot \sin\varphi_1 \end{aligned}\right\} \tag{9}$$

$$\left.\begin{aligned} a_{B_x} &= -r \cdot \omega_r^2 \cdot \sin\varphi_1 \\ a_{B_y} &= -r \cdot \omega_r^2 \cdot \cos\varphi_1 \end{aligned}\right\} \tag{10}$$

Then the full velocities and acceleration of point B will be

$$V_B = \sqrt{V_{B_x}^2 + V_{B_y}^2} = r \cdot \omega_r \tag{11}$$

$$a_B = \sqrt{V_{B_x}^2 + V_{B_y}^2} = r \cdot \omega_r^2 \tag{12}$$

ISSN: 2350-0328
International Journal of Advanced Research in Science, Engineering and Technology
Vol. 7, Issue 2 , February 2020

From the condition of closure of the kinematic chain, the coordinates of point C are determined by the equations:

$$\left.\begin{aligned} X_C &= r\cdot\sin\varphi_1 + l_{BC}\cdot sin\varphi_2 \\ Y_C &= r\cdot\cos\varphi_1 + l_{BC}\cdot cos\varphi_2 \end{aligned}\right\} \tag{13}$$

Differentiating equations (13), we obtain expressions for determining the angular velocities and acceleration of the connecting rod BC, as well as the linear speeds and accelerations of its point C:

$$\omega_{BC} = -\frac{r}{l_{BC}}\cdot\omega_r\cdot\frac{cos\varphi_1}{cos\varphi_2} \tag{14}$$

$$\xi_{BC} = \frac{r\cdot\omega r^2\, sin\varphi_1 + l_{BC}\cdot\omega_{BC}^2\cdot sin\varphi_2}{l_{BC}\cdot cos\varphi_2} \tag{15}$$

$$V_{BC} = \frac{r\cdot\omega r\cdot sin(\varphi_2-\varphi_1)}{cos\varphi_2} \tag{16}$$

$$a_C = \frac{r\cdot\omega r^2\cdot cos(\varphi_2-\varphi_1) + l_{BC}\cdot\omega_{BC}^2}{cos\varphi_2} \tag{17}$$

The crank AB rotates at a constant angular speed.

To determine its preliminary value, we use the condition for ensuring the minimum frequency of impact of the impact organ on each cotton bush when moving on laths with a speed of $V_p = 1.04\ m/s$.

If we take the minimum amount of impact of the shock organ in the bush equal to unity, i.e. n = 1, then the amount of time (s) between each working stroke of this body will be equal to:

$$t = \frac{L_p}{2V_p} \tag{18}$$

where $L_p$is the length of the shock of the working body (Fig. 1). Then the required crank speed, i.e. driving gear with a minimum amount of impact of the shock organ in the bush, determined by the formula

$$n_r = \frac{120\ V_p}{L_p} \tag{19}$$

and angular velocity

$$\omega_r = \frac{\pi\cdot n}{30} = \frac{4\pi V_P}{L_P} \tag{20}$$

To a first approximation, we take the force of action of the working body on the cotton bush to be equal to the force of its inertia at the extreme extended working position.

The force of inertia, as you know, is determined by the expression:

$$F_i = -ma_{S_3} \tag{21}$$

where m is the mass of the working body; kg;

a_{S_3} - acceleration of the center of mass of the working body, m / s2.

Since all points of the impact organ reciprocate perpendicular to the axis of the rack, then $a_{S_3} = a_c$

The value of φ_1 can be determined from $\Delta\, A\acute{A}\acute{C}$:

$$\varphi_1 = arcsin\frac{e}{r+l_{BC}} = arcsin0.062 \tag{22}$$

From here

$$\varphi_1 = 3^0 3\acute{0} \tag{23}$$

Given that $\varphi_1 = \varphi_2$, the formula for determining the acceleration of the center of mass S_3 will take the form

$$a_{S_b} = -\frac{r\cdot\omega_r^2 + l_{BC}\cdot\omega_{BC}^2}{cos\varphi_2} \tag{24}$$

The length of the shock part of the working body of the vibrator is chosen structurally equal to the length of the working chamber, i.e. L_P=1,2 m.

Substituting this value in formulas (19), (20), (21), we find

$$\omega_r = \frac{4\pi V_a}{L_P} = 10{,}88\ rad/s \tag{25}$$

Of ω_{BC}=-2,56 rad/s; ε_{BC}=0,12 rad/s^2; ε_{CB}=0,12 rad/s^2; V_C=0; a_c=-5,849 m/s^2

ISSN: 2350-0328

International Journal of Advanced Research in Science, Engineering and Technology

Vol. 7, Issue 2 , February 2020

III.CONCLUSION AND FUTURE WORK

As can be seen, the angular velocity of the connecting rod in the after blowing moment is directed opposite to the angular velocity of the crank, and the acceleration of point C of the working body is against its movement. Therefore, in the extended position of the latter, the inertia force $F_i = -ma_c$can be taken as the impact force and is regulated by the mass of the working body.

REFERENCES

[1]. A.S. 1628919, USSR, MK AO 1D 46/14. Method for harvesting raw cotton / G.Sh. Zakirov et al. (USSR) Declared 07.07.88. Publ. 23.02.91.EI. 7, C4.

[2]. Course design on the Theory of mechanisms and machines / Ed. G.N.Devoyno-Minsk: Higher School, 1986. 283 p.

[3]. A.Djuraev, S.Elmonov, A.Nuriddinov, A.Gadayev Improvement of designs and methods for calculating the parameters of vibrating polyhedral furnace bar // International Journal of Advanced Research in Science, Engineering and TechnologyVol. 6, Issue 9 , September 2019