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SYNTHESIS AND RESEARCH OF THE TUMBLING MACHINE SPATIAL MECHANISM

Purpose. Design engineering of the seven-link spatial mechanism of a tumbling machine without excessive coupling, as well as development and further analytical study of this machine drive unit.

Methodology. An analytical research method based on the geometric and structural synthesis of the seven-link spatial mechanism of the tumbling machine and its drive is used, the design of the machine is modeled in the SolidWorks 2016 computer-aided design system (CAD).

Findings. Based on the structural synthesis, a design of a seven-link spatial articulated mechanism without excessive machine communication with the complex movement of the working capacity is proposed for volumetric processing of parts and mixing of bulk solids; the principle of its operation is described. Analytical investigations of the basic geometric parameters of the machine are conducted. The design of a special machine drive has been developed; a four-link mechanism has been synthesized, which is a part of the drive unit; analytical studies on the main structural parameters of the machine drive have been performed.

Originality. Interconnection between various geometric parameters of the seven-link spatial mechanism is established, which makes it possible to determine rational lengths ratio of the mechanism links; a connection between the geometric parameters of the four-link flat articulated mechanism as a part of the machine drive and the pressure angles in the kinematic pairs of this mechanism is recognized as well.

Practical value. A new design of the seven-link spatial hinge mechanism without excessive (passive) coupling of machines with the complex movement of the working capacity for volumetric processing of parts are developed. Mathematical expressions are obtained for calculating the main structural and geometric parameters of the mechanism. The design of the machine drive is developed, which allows for the rotation of the machine drive shaft with simultaneous reciprocating movements; mathematical dependences for calculating the main structural parameters are obtained in order to ensure the pressure angles in the mechanism kinematic pairs within acceptable limits.

Keywords: *tumbling machine, spatial mechanism, excess (passive) coupling, kinematic pair*

Introduction. For the first time on the territory of the Soviet Union, at the International Chemical Exhibition in Moscow, in 1966, the development of the Swiss firm Willy A. Bachofen (WAB) was presented – the basic design of a machine with a complex spatial movement of the working capacity, which was designed to perform the processes of fine bulk solids mixing. Today, this machine design has a much wider scope and can be used both for mixing [1] and for performing various types of plastering technological operations [2], in particular: grinding and polishing, separation of parts from foundries, grinding, cleaning of metal parts from corrosion products and others. Based on License Agreements (License Agreement No. 6-19 dated 22.10.2019 between Kyiv National University of Technology and Design and “Lightning” PJSC (Baryshevka), License Agreement No. 5-16 dated 28.07.2016 between Kyiv National University of Technology and Design and “Polyplast” LLC (Lviv), License Agreement No. 3-16 dated 06.06.2016 between Kyiv National University of Technology

and Design and “Red October” LLC (Fastiv)), practical implementation of authors development in the industry of Ukraine is known.

The basic design of a machine with a complex movement of the working capacity [2], from a structural point of view, is a six-link spatial mechanism with rotating kinematic pairs. Spatial mechanisms, in particular those with rotary kinematic pairs, are widely used in various industries [3, 4] and are characterized by their structure complexity and considerable laboriousness in their synthesis and analysis [5]. Similarly, a six-link spatial mechanism with rotating kinematic pairs has its own peculiarities – its structure has excess (passive) coupling, which negatively affects the durability and performance of the mechanism. It is established [2] that the mechanism can only operate if the constructive ratios of its link lengths are observed. During operation of the mechanism, even with a slight deformation of one of the links, it can be jammed; as a result, it will immediately fail. In addition, parts must be manufactured on tolerances with increased precision. Because of this, the cost of such equipment is significantly increased while the operational term is reduced.

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Literature review. In general, there are numerous examples of the elimination of static uncertainties (passive couplings) in various hinges [6, 7]. In particular, a considerable number of examples of exemption from passive couplings in spatial mechanisms are presented in the monograph by Kozhevnikov S.N. “Fundamentals of structural synthesis of mechanisms”. However, for the spatial mechanism of a machine with a complex movement of the working capacity, this problem remains relevant today. One of the main tasks when designing this type of tumbling equipment is to create statically defined mechanisms without excess coupling. It is known [6, 7] that under such a loading condition in the links and kinematic pairs of the mechanism stresses will be determined only by forced technological and dynamic interaction. A series of studies on improving of the design of spatial mechanisms of machines with complex movement of working capacities [2], conducted at Kyiv National University of Technology and Design.

Also [3–5] foreign scientists’ investigations of the structure of six-element spatial mechanisms with rotating kinematic pairs are known, which obtained mathematical dependencies for the calculation and correlation of structural parameters of the mechanism with excess coupling, the adherence of which will allow its operation. The design of the mechanism of the machine [8] without excess coupling has been developed.

Unsolved aspects of the problem. There are various ways of freeing up the spatial mechanisms of machines from excess coupling. However, each of them has both advantages and disadvantages.

For example, in the spatial mechanism of a machine with two working capacities, which are interconnected by translational kinematic pair [9], the intensity of movement of the working array in the two capacities will be different, in this connection there will be a condition in which it is impossible to finish the processing at the same time in two capacities with one machine. A similar situation would occur in a machine with two working capacities connected by a rotating kinematic pair [2].

The technical solution using an additional movable link – slider [2] which is kinematically connected to the driven shaft of the machine also has drawbacks. Since the location of the translational kinematic pair will be “at the end” of the kinematic chain, it can lead to the concentration of dynamic uneven loads on the moving links of the machine, to cause their deformation over time; the power from the drive of the machine, which is necessary to allow the slider to slide along the guide, will significantly reduce due to its friction losses in other kinematic pairs of the machine and so on.

Purpose. On the basis of the above, the main purpose of the study is the synthesis of a seven-link spatial mechanism of the tumbling machine with a complex movement of the working capacity, in which the excess (passive) coupling is included in the kinematic chain; analytical studies of the basic geometrical parameters of the given machine design; development of a special design of the drive of the given tumbling machine with further synthesis of the flat four-link hinged mechanism which is part of the drive.

Methodology. When conducting these studies, we used an analytical research method applying additional modeling of the machine design and its drive in SolidWorks CAD.

Results. Structure of a six-link spatial mechanism with rotating kinematic pairs and ways of releasing it from excess coupling. The basic design of a machine with complex spatial motion of the working capacity [2], whose movable links form a spatial six-link mechanism with rotating kinematic pairs, is considered. Due to this arrangement of kinematic pairs, the working body performs complex spatial motion.

Using the Somov-Malyshev formula presented in the work by the author Reshetov L.N. “Designing rational mechanisms”, to determine the degree of mobility of this mechanism, we established that it will be zero.

Thus, the mechanism is incapacitated; however, despite this, it is able to function while providing clear constructive ratios of the lengths of its links, which were obtained in [2]. This fact is explained by the presence in it of excess coupling.

The introduction details the disadvantages of using spatial mechanisms with excess coupling. One way of exempting from the action of excess coupling is to introduce into the kinematic chain the mechanism of an additional movable link – a slider that performs reciprocating movement along a horizontal guide, in whose bearing both leading and driven shaft can be installed with the possibility of rotation.

Technical solution for freeing up the machine mechanism from excess coupling in which the slider is kinematically connected to the driven shaft is introduced in [2]. However, in such a machine design, where the drive shaft together with the slider will perform additional reciprocating movement, in addition to rotational motion, along the guide, can lead to the concentration of dynamic non-uniform loads on the moving links of the machine, over time, cause their deformation. Thus, it may reduce reliability and durability of the machine as a whole. In addition, special requirements should be placed on the friction properties of the reciprocating kinematic pair slider-guide, since even a slight increase in friction in a given kinematic pair can lead to jamming of the mechanism of the machine, because the very location of the kinematic pair “at the end” of the kinematic chain leads to the fact that the power from the machine required to allow the slide to slide along the guide will be partially reduced by its friction losses in other kinematic pairs of the machine, doing useful parts processing work, and so on.

This situation can be solved by kinematically connecting the slider to the drive shaft of the machine. In this case, the reciprocating kinematic slide-guide pair will actually be located “at the beginning” of the kinematic chain of the mechanism, which will create conditions for reducing the dynamic non-uniform loads on the moving links of the machine.

The model of the machine (without drive) implemented with the help of SolidWorks CAD, with an additional movable link slider, in which the bearing shaft of the machine is mounted, is shown in Fig. 1.

This machine contains a frame 1, leading 2 and driven 3 shafts. The leading shaft 2 is mounted in the bearing support of the slider 4, which, in turn, is mounted in a horizontal guide 5. The driven shaft 3 is mounted in the bearing support of the frame. The axes of the leading 2 and driven 3 shafts are parallel to each other. The leading shaft 2 and the driven shaft 3 are pivotally connected at the other ends to the drive 6 and the driven forks 7 respectively, which are diametrically mutually perpendicular to the geometric axes 8 and 9 which are the axes of attachment of the working capacity 10. Thus, unlike traditional spatial hinge schemes, the forks 6 and 7 are arranged

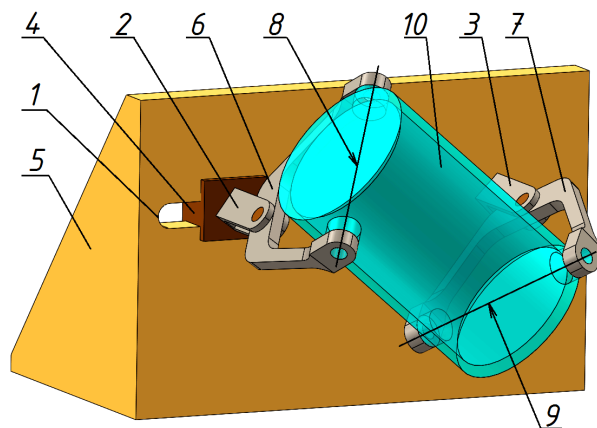


Fig. 1. Model of the machine in which the drive shaft is mounted in the bearing of the slider

perpendicular to each other. The existence of a dual propeller mechanism with a space frame-crosspiece is also reported in the work [7].

Therefore, according to the Somov-Malyshev formula, the mobility ratio of the mechanism will be equal to one. Excess connection is off. The appearance of minor deformations of the links of the mechanism will not affect the loss of working capacity of the machine as a whole. In addition, there is a possibility of varying the longitudinal length of the working capacity in large ranges, which in turn creates conditions for regulating the intensity of the respective technological operations of machining or blending of bulk substances.

However, once released from excess coupling, the drive shaft of this machine design will additionally perform reciprocating movement along the horizontal guide. Thus, it is necessary to develop such a design of the machine drive that will provide a constant transmission of torque to the shaft, which will perform additional reciprocating movement.

Analytical study of mechanism design. The analytical study and synthesis of the given machine mechanism should begin by considering the initial data, namely, certain geometric parameters, which are selected depending on the size and number of workpieces, the type of technological operations performed on the equipment and their intensity of execution. Thus, the axial distance of the driving and driven forks is l_F , the axial distance of the working capacity is l_{WC} . In addition, the design of the machine must be subject to the condition $l_{WC} > l_F$, since it is precisely satisfied, the basic structure of the machine with excess coupling becomes inoperable.

In the course of operation of this mechanism of the machine, due to the formed translational kinematic pair, the distance between the parallel axes of the drive and slave shafts will be cyclically changed from the minimum l_{min} to the maximum l_{max} . The minimum distance l_{min} will occur at such positions of the moving links of the machine (Fig. 2), when one of the axes of fastening of the drive or driven fork will be horizontal and the other vertical. The maximum distance l_{max} will occur when the longitudinal axis of the working capacity is in the vertical plane.

When designing such a mechanism of the machine, there is a need to accurately calculate the amplitude l_a of the drive shaft reciprocating movement. The amplitude l_a can be defined as the difference between the maximum and minimum distances formed between the axes of the drive and the driven shafts

$$l_a = l_{max} - l_{min}. \quad (1)$$

It is known [2] that the maximum distance l_{max} between the axes of the drive and driven shafts can be determined from a right triangle formed in the horizontal projection of the machine as follows

$$l_{max} = \sqrt{(l_F + l_{WC})^2 - l_F^2}. \quad (2)$$

In the position of the moving links, when the lateral planes of the forks are parallel to each other, and the working capac-

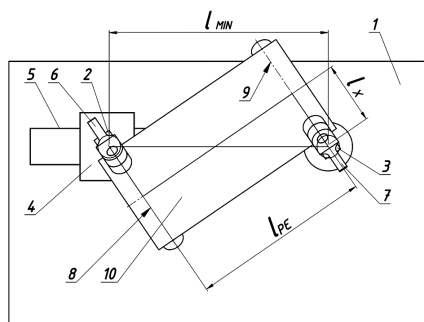


Fig. 2. Machine for processing parts – front view

ity is projected on the vertical plane in full size, the distance between the geometric mutually perpendicular axes of the fork mounting will be minimal l_{min} . The front view, with this position of the moving links of the machine, is shown in Fig. 2 (the designations of the links in Fig. 2 coincide with the designations of the links in Fig. 1)

Thus, based on geometric considerations, l_{min} can be defined as follows

$$l_{min} = \sqrt{l_{WC}^2 + 4l_x^2}, \quad (3)$$

where l_x is the distance in the projection on the vertical plane between the axis of rotation of the driven (drive) shaft and the axis of the working capacity.

In order to obtain an expression for determining the distance l_x , in the position of the moving links of the machine, which is shown in Fig. 2, we make a projection of the machine design on a plane parallel to the upper end of the working capacity.

The frontal projection of the machine, which is projectively connected to the projection of the machine on a plane parallel to the upper end of the working capacity is shown in Fig. 3.

The geometric axes of the forks attachment are mutually perpendicular to each other, the axis of the working capacity, which extends from the “top projection”, divides the right angle between the axes of the forks in half. Thus, on the lower projection an isosceles rectangular triangle with hypotenuse l_F and two catheters l_x was formed. Accordingly, the length l_x will be defined as

$$l_x = l_F \cos 45^\circ. \quad (4)$$

Substitute the value of the expression (4) in equation (3)

$$l_{min} = \sqrt{l_{WC}^2 + 4(l_F \cos 45^\circ)^2}. \quad (5)$$

Substitute expressions (2) and (5) into equation (1)

$$l_a = \sqrt{l_{WC}(2l_F + l_{WC})} - \sqrt{l_{WC}^2 + 4(l_F \cos 45^\circ)^2}. \quad (6)$$

Thus, an expression was obtained to calculate the amplitude l_a of reciprocating movement of the drive shaft along the guide.

Development of a drive that provides the torque transmission to the drive shaft of the machine, performing additional reciprocating movement. Based on the above, the design of the drive of the machine was developed, which provides the transmission of torque to the moving drive shaft. The kinematic scheme of the machine (with a technological drive) is presented in Fig. 4.

The machine for parts processing contains a frame 1, which houses the electric motor 2, on whose shaft the drive

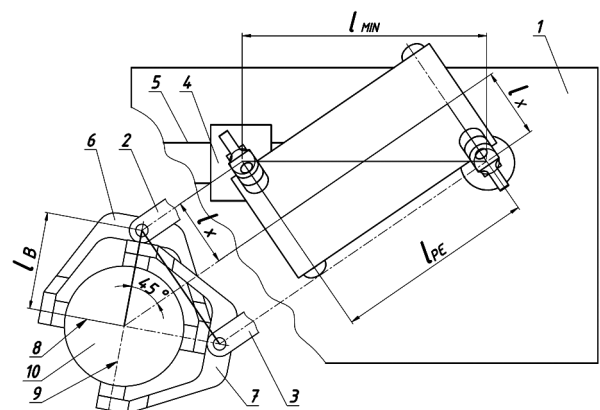


Fig. 3. Frontal projection of the machine in projective connection with the projection of the machine on a plane parallel to the upper end of the working capacity

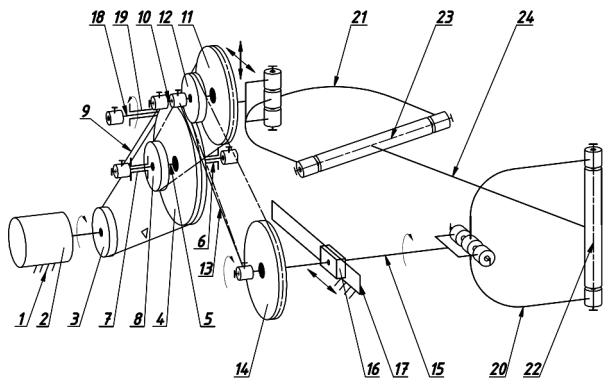


Fig. 4. Kinematic scheme of the drive of the tumbling machine without excess coupling

pulley 3 of belt drive is rigidly fixed, the driven pulley 4 mounted on the shaft of the actuator 5, which is mounted in the bearing supports 6 and 7. On drive shaft 5 the first sprocket 8 of the first chain gear is rigidly fixed to the drive shaft 5; in addition, the drive shaft 5 is kinematically connected to the lower head of the rocker arm 9. Rocker 9, connected through the upper head kinematically to the drive shaft of the actuator 10, on which driven sprocket 11 of the first gear and the drive sprocket 12 of the second gear are rigidly fixed as well as the movable shaft 10 connected to the upper head of the connecting rod 13. The driven sprocket 14 of the second chain gear is rigidly fixed to the drive shaft 15, which is mounted in the bearing support of the slide 16 with the possibility of its reciprocating movement along the horizontal guide 17. In addition, the drive shaft 15 is kinematically connected to the lower head of the connecting rod 13. The driven shaft 18 is mounted in the bearing support 19. The drive 15 and the driven shaft 18 are pivotally connected at the other ends to the drive 20 and driven 21 forks, respectively, whose diametrically mutually perpendicular axes 22 and 23 are the axes of the working capacity 24 attachment.

The machine works as follows. After switching on the motor 2 located in the frame 1, a constant rotational movement through the belt drive formed by the master 3 and the driven 4 pulleys is transmitted to the drive sprocket 8 of the first chain gear. The rotational movement of the drive sprocket 8, by means of the first chain gear, is transmitted to the drive sprocket 11, which will rotate at the same angular velocity together with the driving shaft 10, as well as the drive sprocket 12 of the second sprocket rigidly fixed thereto. The rotational movement of the drive sprocket 12, via the second chain drive, is transmitted to the drive sprocket 14, which is rigidly fixed on the drive shaft 15 mounted in the bearing support of the slider 16, is transmitted to the drive fork 20, which through the axis 22 rotates the working capacity 24, giving it complex spatial movement with rotation about its own axis. Such complex spatial movement of the working capacity 24 is transmitted through the axis 23 to the driven fork 21 and is converted into rotational motion of the driven shaft 18, which is mounted in the bearing support 19.

In addition to rotary motion, the drive shaft 15, which is mounted in the bearing support of the slider 16, performs additional reciprocating movement relative to the horizontal guide 17. For one complete rotation, the drive shaft 15 performs four reciprocating movements along the horizontal guide 17. However, reciprocating movement of the drive shaft 15 will cause rotation of the rocker arm 9 and the plane-parallel movement of the connecting rod 13. The movable shaft 10 together with the upper head of the rocker arm 9 will perform oscillatory movement about the axis of the shaft of the drive 5 along a trajectory with a radius corresponding to the length of the rocker arm 9. Thus, for one complete rotation of the drive shaft 15, the movable shaft 10 performs four oscillatory movements relative to the axis of the shaft of the drive 5.

Confirmation of the possibility of implementing a dual spatial cardan mechanism with reciprocating movement of one of its links is also presented in [7].

In fact, the operation of this actuator is provided by a flat four-link hinged mechanism, which is part of it and consists of the leading link – the slider 16, the connecting rod 13 and the driven link – the rocker arm 9 and the fixed link – the frame 1. With reciprocating movement of the slider 16, there will be a cyclic change in the distance between the center of rotation of the rocker arm 9 and the rotary kinematic pair “connecting rod-slider” from the maximum c_{max} to the minimum c_{min} value.

In designing the drive structure of the machine as a whole, the main task is a synthesis of this hinged mechanism, which is the determination of the rational lengths of the rocker arm 9 and the connecting rod 13 based on the calculated amplitude of the reciprocating movement of the slider 16, which, in general, will ensure the formation of pressure angles in the kinematic pairs of this hinge mechanism within acceptable limits.

Let us represent the hinge component of the drive of the machine at the two extreme positions of the slider 16, which is shown in Fig. 5, for further synthesis.

When synthesizing the mechanism, it is necessary that the maximum values of the pressure angles θ_{max} and α_{max} in the kinematic pairs of the mechanism do not exceed the permissible limits. During the operation of this hinge mechanism, in the position of the movable units corresponding to the formation of the maximum distance c_{max} , in the rotary kinematic pair “connecting rod-slider” the maximum value of the pressure angle θ_{max} will occur. As well as the position of the movable units corresponding to the formation of the minimum distance between the center of rotation of the rocker arm and the kinematic pair in which the connecting rod is connected to the slider – c_{min} , in the rotary kinematic pair “connecting rod – slider” there will be a maximum value of the pressure angle α_{max} .

It is known [10] that in order to avoid jamming of the hinge mechanism, the maximum value of the pressure angle in any of its kinematic pair should not exceed 60° , and for long-term operation of the mechanism it is desirable that the maximum value of the pressure angle does not exceed 40° . So, let us first set the maximum value of the pressure angle θ_{max}

$$\Theta_{max} \leq 40^\circ. \quad (7)$$

Thus, with respect to (7), we write the expression to determine the minimum value of the pressure angle α_{min}

$$\alpha_{min} = \frac{90^\circ - \Theta_{max}}{2}. \quad (8)$$

Next, based on geometric considerations, we write an expression to determine the length b of the rocker arm and connecting rod

$$b = \frac{c_{max} \cdot \sin \alpha_{min}}{\sin \gamma_{max}}, \quad (9)$$

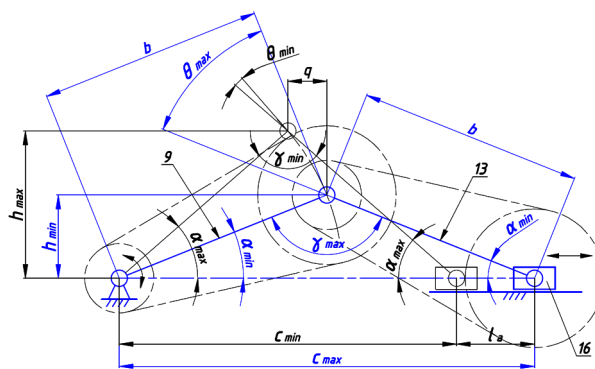


Fig. 5. Kinematic scheme of the four-hinged hinge mechanism

where γ_{\max} is the maximum value of the angle between the rocker arm and the connecting rod.

In turn, the angle γ_{\max} can be defined as follows

$$\gamma = 90^\circ + \Theta. \quad (10)$$

We write expression (9) with respect to equations (8) and (10)

$$b = \frac{c_{\max} \cdot \sin(45^\circ - 0.5 \cdot \Theta_{\max})}{\sin(90^\circ + \Theta_{\max})}. \quad (11)$$

When designing this hinge mechanism, there is a need to calculate the minimum h_{\min} and maximum h_{\max} vertical distance between the center of rotation of the rocker arm and the kinematic pair “connecting rod – slider”, which will occur at the extreme positions of the slider, as well as the calculation of the horizontal component of the amplitude q of kinematic pair “connecting rod – slider”.

Based on geometric considerations, we write down the expressions for determining h_{\min} and h_{\max}

$$h_{\min} = \sqrt{b^2 - (0.5 \cdot c_{\max})^2}; \quad (12)$$

$$h_{\max} = \sqrt{b^2 - (0.5 \cdot c_{\min})^2}. \quad (13)$$

We write expressions (12) and (13) with respect to (11)

$$h_{\min} = \sqrt{\left(\frac{c_{\max} \cdot \sin(45^\circ - 0.5 \cdot \Theta_{\max})}{\sin(90^\circ + \Theta_{\max})} \right)^2 - 0.25 \cdot c_{\max}^2};$$

$$h_{\max} = \sqrt{\left(\frac{c_{\max} \cdot \sin(45^\circ - 0.5 \cdot \Theta_{\max})}{\sin(90^\circ + \Theta_{\max})} \right)^2 - 0.25 \cdot c_{\min}^2}. \quad (14)$$

Next, we write an expression to determine the minimum distance between the center of rotation of the rocker arm and the kinematic pair in which the connecting rod is connected to the slider – c_{\min}

$$c_{\min} = c_{\max} - l_a. \quad (15)$$

Taking into account equation (6), expression (15) will have the form

$$\alpha_{\max} = \arccos\left(\frac{c_{\max} - \sqrt{l_{WC}(2l_F + l_{WC})} + \sqrt{l_{WC}^2 - 4(l_F \cos 45^\circ)^2}}{2b} \right) \leq 40^\circ. \quad (20)$$

If, by expression (20), the maximum value of the pressure angle α_{\max} in the rotary kinematic pair “connecting rod – slider” exceeds the allowable value, it is necessary to increase the distance a_{\max} without changing the maximum allowable value of the pressure angle θ_{\max} .

On the basis of the previously obtained expressions, it is possible to derive an equation for calculating the distance c_{\max} , which will provide the maximum values of the pressure angles θ_{\max} and α_{\max} within acceptable limits in the kinematic pairs of the mechanism. To do this, we express the value c_{\min} from (18)

$$c_{\min} = 2b \cdot \cos \alpha_{\max}. \quad (21)$$

Substituting equations (16) and (11) into expression (21), we obtain

$$c_{\max} - \sqrt{l_{WC}(2l_F + l_{WC})} + \sqrt{l_{WC}^2 - 4(l_F \cos 45^\circ)^2} =$$

$$= 2 \left(\frac{c_{\max} \cdot \sin(45^\circ - 0.5 \cdot \Theta_{\max})}{\sin(90^\circ + \Theta_{\max})} \right) \cdot \cos \alpha_{\max}. \quad (22)$$

After arithmetic transformations and cancellations, expression (22) will take the form

$$c_{\max} = \frac{\sqrt{l_{WC}(2l_F + l_{WC})} + \sqrt{l_{WC}^2 - 4(l_F \cos 45^\circ)^2}}{1 - \frac{2 \cdot \sin(45^\circ - 0.5 \cdot \Theta_{\max})}{\sin(90^\circ + \Theta_{\max})}}. \quad (23)$$

$$c_{\min} = c_{\max} - \sqrt{l_{WC}(2l_F + l_{WC})} + \sqrt{l_{WC}^2 - 4(l_F \cos 45^\circ)^2}. \quad (16)$$

Write the expression (14) with respect to equation (16)

$$h_{\max} = \sqrt{\left(\frac{c_{\max} \cdot \sin(45^\circ - 0.5 \cdot \Theta_{\max})}{\sin(90^\circ + \Theta_{\max})} \right)^2 - 0.25 \cdot \left(c_{\max} - \sqrt{l_{WC}(2l_F + l_{WC})} + \sqrt{l_{WC}^2 - 4(l_F \cos 45^\circ)^2} \right)^2}.$$

We write the expression to calculate the horizontal component of the displacement amplitude q of the kinematic pair “rocker arm-connecting rod”

$$q = 0.5(c_{\max} - c_{\min}). \quad (17)$$

Equation (17) with respect to expression (16) will have the form

$$q = 0.5 \sqrt{l_{WC}(2l_F + l_{WC})} + \sqrt{l_{WC}^2 - 4(l_F \cos 45^\circ)^2}.$$

After taking into account the allowable maximum value of the pressure angle θ_{\max} in the rotary kinematic pair “rocker arm-connecting rod”, expressions are obtained to calculate the basic geometrical parameters of the hinge mechanism, it is necessary to write the equation for the calculation and to check the maximum value of the rotational angle α_{\max} in the rotary kinematic pair “connecting rod – slider”. α_{\max} will occur in this position of the moving links of the mechanism, when the slider will occupy its extreme left position. Write the expression to determine the cosine of the angle α_{\max}

$$\cos \alpha_{\max} = \frac{c_{\min}}{2b}. \quad (18)$$

Next, we write an expression to determine the angle α_{\max}

$$\alpha_{\max} = \arccos\left(\frac{c_{\min}}{2b} \right) \leq 40^\circ. \quad (19)$$

We write the expression (19) taking into account the equation (16)

$$\alpha_{\max} = \arccos\left(\frac{c_{\max} - \sqrt{l_{WC}(2l_F + l_{WC})} + \sqrt{l_{WC}^2 - 4(l_F \cos 45^\circ)^2}}{2b} \right) \leq 40^\circ. \quad (20)$$

Thus, using the dimensions of the lengths of the links of the spatial mechanism of the machine, providing the corresponding amplitude of reciprocating movement of the slider, as well as previously adopting the maximum values of the angles of pressure θ_{\max} and α_{\max} in kinematic pairs of the mechanism within the acceptable limits, using equation (23) it is possible to calculate the rational maximum distance c_{\max} between the center of rotation of the rocker arm and the rotary kinematic pair “connecting rod-slider”, which will become the basic geometric parameter of the mechanism during its further synthesis.

Therefore, it is advisable to start the synthesis of the hinged four-link mechanism, which is part of the drive of the parts processing machine with complex movement of the working capacity from the calculation of the distance c_{\max} by the equation (23).

Conclusions. The paper presents one possible way of freeing up the spatial mechanism of a machine for processing parts from excess coupling. Thus, a synthesis and further analytical study of a seven-link spatial mechanism, with no excess coupling in the kinematic chain, was performed. Based on the synthesized mechanism, modeling of a machine with a complex movement of the working capacity for processing using SolidWorks CAD was performed. Analytical mathematical dependences for determination of basic geometrical ratios of link lengths of the developed machine were obtained. These de-

dependencies can be used for further design of this type of equipment.

The drive design for this machine is developed, which allows realizing the rotation of the drive shaft of the machine with simultaneous reciprocating movement, the four-links hinge mechanism that is part of the drive is synthesized. Mathematical dependences were obtained for the calculation of the basic geometric ratios of the lengths of the links of the four-link hinged mechanism in order to ensure the angles of pressure in the kinematic pairs of the mechanism within acceptable limits. The results obtained allow implementing new designs of the corresponding equipment.

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Синтез і дослідження просторового механізму галтувальної машини

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Мета. Розробка конструкції просторового семиланкового механізму галтувальної машини без надлишково-

го зв'язку, а також розробка й подальше аналітичне дослідження приводу даної машини.

Методика. Використано аналітичний метод дослідження на основі геометричного та структурного синтезу просторового семиланкового механізму галтувальної машини та її приводу, виконане моделювання конструкції машини в системі автоматизованого проектування SolidWorks 2016.

Результати. На основі структурного синтезу запропонована конструкція семиланкового просторового шарнірного механізму без надлишкового зв'язку машини зі складним рухом робочої ємкості, що використовується для об'ємної обробки деталей і змішування сипких речовин, описано принцип її роботи. Проведені аналітичні дослідження основних геометричних параметрів машини. Розроблена конструкція спеціального приводу машини, виконано синтез плоского чотириланкового механізму, що входить до складу приводу, виконані аналітичні дослідження основних конструктивних параметрів приводу машини.

Наукова новизна. Встановлено взаємозв'язок між різними геометричними параметрами просторового семиланкового механізму, що дає можливість визначити раціональні співвідношення довжин ланок механізму, а також встановлено взаємозв'язок між геометричними параметрами плоского чотириланкового шарнірного механізму, який входить до складу приводу машини, та значенням кутів тиску в кінематичних парах цього механізму.

Практична значимість. Розроблена нова конструкція семиланкового просторового шарнірного механізму без надлишкового (пасивного) зв'язку машин зі складним рухом робочої ємкості для об'ємної обробки деталей. Отримані математичні вирази для розрахунку основних конструктивних і геометричних параметрів даного механізму. Розроблена конструкція приводу машини, що дозволяє реалізувати обертання ведучого валу машини з одночасним зворотно-поступальним переміщенням, отримані математичні залежності для розрахунку основних конструктивних параметрів з метою забезпечення кутів тиску в кінематичних парах механізму в допустимих межах.

Ключові слова: галтувальна машина, просторовий механізм, надлишковий (пасивний) зв'язок, кінематична пара

Синтез и исследование пространственного механизма галтовочной машины

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Цель. Разработка конструкции пространственного семизвенного механизма галтовочной машины без избыточной связи, а также разработка и дальнейшее аналитическое исследование привода данной машины.

Методика. Использован аналитический метод исследования на основе геометрического и структурного синтеза пространственного семизвенного механизма галтовочной машины и ее привода, выполнено моделирование конструкции машины в системе автоматизированного проектирования SolidWorks 2016.

Результаты. На основе структурного синтеза предложена конструкция семизвенного пространственного

шарнирного механизма без избыточной связи машины со сложным движением рабочей емкости, используемой для объемной обработки деталей и смешивания сыпучих веществ, описан принцип ее работы. Проведенные аналитические исследования основных геометрических параметров машины. Разработана конструкция специального привода машины, выполнен синтез плоского четырехзвенного механизма, который входит в состав привода, выполнены аналитические исследования основных конструктивных параметров привода машины.

Научная новизна. Установлена взаимосвязь между различными геометрическими параметрами пространственного семизвенного механизма, что дает возможность определять рациональные соотношения длин звеньев механизма, а также установлена взаимосвязь между геометрическими параметрами плоского четырехзвенного шарнирного механизма, который входит в состав привода машины, и значениям углов давления в кинематических парах этого механизма.

Практическая значимость. Разработана новая конструкция семизвенного пространственного шарнирного механизма без избыточной (пассивной) связи машин со сложным движением рабочей емкости для объемной обработки деталей. Получены математические выражения для расчета основных конструктивных и геометрических параметров данного механизма. Разработана конструкция привода машины, что позволяет реализовать вращения ведущего вала машины с одновременным возвратно-поступательным перемещением, получены математические зависимости для расчета основных конструктивных параметров с целью обеспечения углов давления в кинематических парах механизма в допустимых пределах.

Ключевые слова: *галтовочная машина, пространственный механизм, избыточная (пассивная) связь, кинематическая пара*

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