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I. M. Matsyuk[1], Cand. Sc. (Tech.), Assoc. Prof.,
orcid.org/0000-0002-0861-0933,
E. M. Shlyahov[1],
orcid.org/0000-0001-5983-9853,
O. I. Yehurnov[2], Cand. Sc. (Tech.)

1 – Dnipro University of Technology, Dnipro, Ukraine, e-mail: shlyahove@nmu.org.ua
2 – Ana-Tems Ltd, Dnipro, Ukraine, e-mail: yegurnov@yahoo.com

ON APPLYING HIGH-CLASS MECHANISMS OF HEAVY-LOADED MACHINES

Purpose. To evaluate the impact of possible variants of assemblies of the fourth class Assur group as a component of a jaw crushing machine on the machine runnability.

Methodology. In this paper, a theoretical study of possible assemblies of a fourth class Assur group, which is part of a jaw crusher, was carried out.

Findings. Based on the conducted research it is found that the number of possible assemblies of the group in this mechanism without its reference to the crushing machine is equal to four. While applying this mechanism in the crushing machine, the maximum number of assemblies equals two and they can be in a close proximity, which can result in emergency.

Originality. The originality in the work involves defining the number of possible assemblies of the fourth class Assur group. For the group under consideration it has been established that the maximum number of assemblies equals four. An algorithm of finding a zone of possible positioning of a driven crank of a crushing mechanism is suggested, which allows influencing the remoteness of two closely-spaced assemblies.

Practical value. The search algorithm of possible assemblies of the fourth class Assur group with the help of Mathcad program can be used when synthesizing similar crushing machines.

Keywords: *jaw crushing machine, mechanism assemblies, Assur group, Mathcad program*

Introduction. The vast majority of planar lever mechanisms applied in current technology belongs to the second class according to Assur-Artobolevsky classification. Mechanisms of the third class feature small application domain. As for the mechanisms of the fourth and higher classes, which are called "high-class mechanisms (HCMs)" in scientific literature, they are rarely used.

With increasing class of the Assur group, the number of its possible configurations and assemblies increases. Every assembly has its own peculiarities and studying them is of certain interest. The issue of choosing an efficient assembly in relation to functions performed by the mechanism is topical.

Literature review. Many researchers (L. V. Assur, I. I. Artobolevsky, U. A. Dzholdasbekov) pointed out that HCMs have considerable kinematic and dynamic possibilities compared to the second class mechanisms, which are traditionally used.

However, these possibilities have not been properly developed up to the present. In many respects this is caused by the lack of engineering methods for their computation and analysis.

For a long time, the issue of high-class mechanisms was studied by U. A. Dzholdasbekov who along with his apprentices developed general methods of structural, kinematic and force analysis of HCMs. Moreover, he suggested a series of devices whose basis is formed by HCMs (hoisting, cargo-handling, lifting gears and others). According to the results of this work, U. A. Dzholdasbekov published the book "The theory of high-class mechanisms" in 2001.

In [1] issues of synthesis of Stephenson six-leverage mechanism which contains the fourth class group, are considered. Article [2] deals with analysis of the fourth-class mechanism. Kinematic analysis of the sixth class mechanism is done in [3]. General issues of finding assemblies in Assur groups of lever mechanisms are considered in [4].

In [5, 6] set forth results of research on mechanisms with alterable contour. Moreover, works regarding the second class mechanisms are published regularly [7].

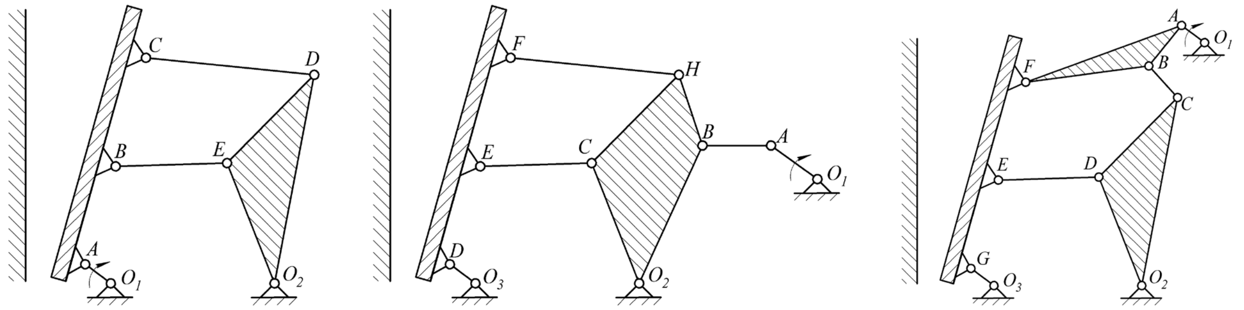
Recently there has been observed a tendency of applying HCMs in crushing machines. Research and development in this field are performed under L. T. Dvornikov's guidance in Russia. He and his apprentices obtained patents of the Russian Federation for a range of crushing machine designs in which mechanisms with closed-loop reconfigurable contours are used (Fig. 1).

Patent No. 2332260 of the Russian Federation is also related to a jaw crusher with a sixth class mechanism.

Crushing machines belong to heavy-loaded machines in which strong forces act and there can be part joint gaps as a result of heavy wear of mated surfaces of the parts. Under such conditions if mechanism assemblies are arranged rather closely, spontaneous transition from one assembly to another can occur, which may cause emergency situation.

Earlier the authors of the article [8] conducted research on design of a jaw crusher in which both jaws move (Fig. 2) which resulted in establishing that in certain cases with one and the same position of the input crank *AB*, two mechanism assemblies can be situated rather closely.

Therefore, studying assemblies in such mechanisms is of great importance at the synthesis stage.



a *b* *c*

Fig. 1. A jaw crusher:

a – with a fourth class mechanism (Patent No.2235594); b – with a fourth class mechanism (Patent No. 2142850); c – with a fifth class mechanism (Patent No. 2232637)

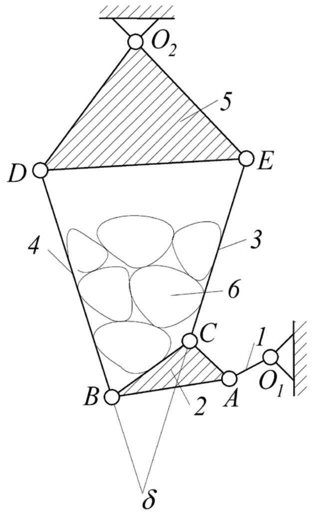


Fig. 2. A jaw crusher in which both jaws move with a fourth class mechanism (Patent No. 2538108)

Purpose. The purpose of the work is to study possible variants of assemblies of the fourth class Assur group when it is used in a heavy loaded jaw crusher.

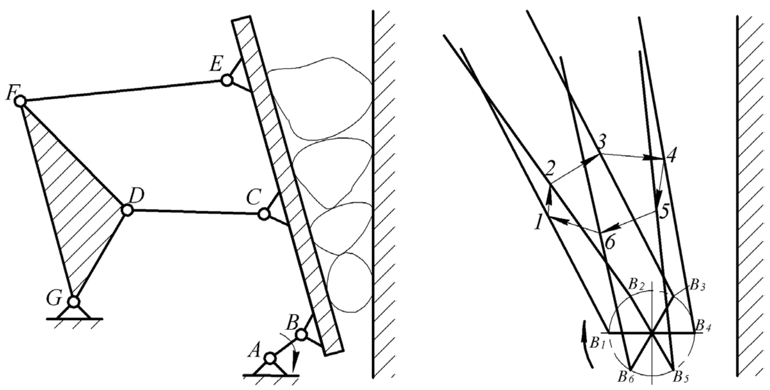
To achieve the set purpose the following tasks were determined:

- to study a fourth class mechanism (Fig. 3, *a*) which is similar to the one in Fig. 1, *a*;
- to define the maximum number of assemblies of this mechanism without reference to its use in the crusher;
- to study possible assemblies of the mechanism when using it in the crusher;
- to define a zone of possible displacement of the crank in order to influence the remoteness of one assembly from the other by changing the place of fixing of a driven crank and its length;
- to solve non-linear equations of mathematical model numerically using the standard procedures of mathematical software package [9, 10];
- evaluate the results achieved taking into account the recommendations of [11, 12].

Results. A jaw crusher (Fig. 3, *a*) with a compound motion of the jaw, which is similar to the crusher in Fig. 1, *a*, is taken as an object of the research.

The crusher is started by the crank *AB*. The complex motion of the jaw *BCE* is ensured by a three-joint rocker *DFG* and two piston rods *CD* and *EF*. The mechanism links form a closed-loop four link contour *CDFE*. The material is crushed between moveable and vertical fixed jaws. When the crank rotates clockwise, the moveable jaw takes the positions (from one to six) shown in Fig. 3, *b*.

Let us consider the mechanism links as vectors (Fig. 4). Let us put the origin of the rectangular axes at point *A*. The modules of the vectors are taken as follows,



a *b*

Fig. 3. Jaw crusher with a fourth class mechanism:

a – diagram; b – displacement of a moveable jaw per one revolution of a crank

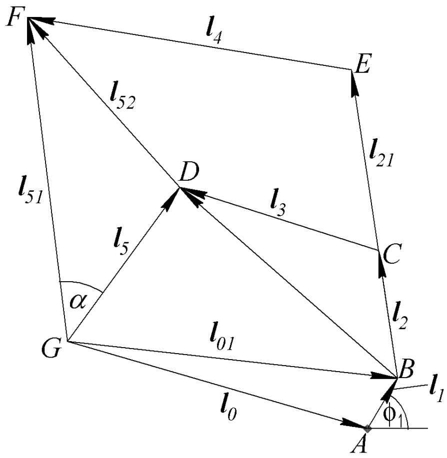


Fig. 4. Vector image of linkage

m: $l_1 = 0.1$; $l_2 = l_{BC} = 0.35$; $l_{21} = l_{BE} = 0.75$; $l_3 = 0.4$; $l_4 = 0.6$; $l_5 = 0.35$; $l_{51} = 0.6$. Position of point G is: $x_G = -0.55$; $y_G = 0.19$. The angle $\alpha = 43°$. The specified geometric parameters of the crusher correspond to the sizes of its model. Since the search for possible assemblies of the mechanism is a geometric problem, any sample with any dimensions proportional to the sizes of a real crusher can be subject to the research.

For the fixed position of point B in the mechanism, two four-link chains $BCDG$ and $BEFG$ can be pointed out. Let us write vector closedness equation for both of them.

$$l_2 + l_3 = l_5 - l_{01} \quad \text{and} \quad l_{21} + l_4 = l_{51} - l_{01}. \qquad (1)$$

We solve the task using the Mathcad-11 application program package, expressing the vectors in terms of exponential form of a complex number.

The system (1) takes on the form

$$l_2 \cdot \exp(i \cdot \phi_2) + l_3 \cdot \exp(i \cdot \phi_3) = l_5 \cdot \exp(i \cdot \phi_5) - l_{01} \cdot \exp(i \cdot \phi_{01})$$

$$l_{21} \cdot \exp(i \cdot \phi_2) + l_4 \cdot \exp(i \cdot \phi_4) = l_{51} \cdot \exp(i \cdot \phi_{51}) - l_{01} \cdot \exp(i \cdot \phi_{01}).$$

For the first four-link chain, let us take link *2* as an input link and link *5* – as an output link. For the second four-link chain, link 2_1 will be an input link and link 5_1 – an output one.

Let us eliminate the angle ϕ_3 from the first equation by assuming the equation as a system (1)

$$l_3 \cdot \exp(1i \cdot \phi_3) = l_5 \cdot \exp(1i \cdot \phi_5) - l_{01} \cdot \exp(1i \cdot \phi_{01}) - l_2 \cdot \exp(1i \cdot \phi_2).$$

By multiplying both sides of the equation by the adjoint vectors, we will obtain l_3^2 on the left, while on the right we will obtain an expression in which the angles ϕ_2 and ϕ_5 are unknown.

Applying the universal trigonometric substitution to this equation (expressing the sines and cosines of the angle ϕ_5 in terms of the tangent of the angle $0.5\phi_5$), we can obtain two formulas for determining the angle ϕ_5 as a function of the angle ϕ_2, which correspond to the upper and lower assemblies of the four-link chain *BCDG*

$$\phi_5(\phi_2) := 2\,\mathrm{atan}\left(\frac{-B(\phi_2) + \sqrt{B(\phi_2)^2 - 4A(\phi_2) \cdot C(\phi_2)}}{2A(\phi_2)}\right)$$

$$\phi_{5a}(\phi_2) := 2\,\mathrm{atan}\left(\frac{-B(\phi_2) - \sqrt{B(\phi_2)^2 - 4A(\phi_2) \cdot C(\phi_2)}}{2A(\phi_2)}\right).$$

In these formulas, $A(\phi_2)$, $B(\phi_2)$, $C(\phi_2)$ are numerical coefficients depending on the value of the angle φ_2.

In a similar way, eliminating the angle ϕ_4 from the second equation of the system (1), we will obtain two formulas for defining the angle ϕ_{51} as a function of the angle ϕ_2, which correspond to the upper and lower assemblies of the four-link chain *BEFG*

$$\phi_{51}(\phi_2) := 2\,\mathrm{atan}\left(\frac{-B_1(\phi_2) + \sqrt{B_1(\phi_2)^2 - 4A_1(\phi_2) \cdot C_1(\phi_2)}}{2A_1(\phi_2)}\right)$$

$$\phi_{51a}(\phi_2) := 2\,\mathrm{atan}\left(\frac{-B_1(\phi_2) - \sqrt{B_1(\phi_2)^2 - 4A_1(\phi_2) \cdot C_1(\phi_2)}}{2A_1(\phi_2)}\right).$$

In these formulas, $A_1(\phi_2)$, $B_1(\phi_2)$, $C_1(\phi_2)$ are numerical coefficients also depending on the value of the angle φ_2.

Let us consider the function

$$\Delta(\phi_2) := \phi_5(\phi_2) - \phi_{51}(\phi_2) + \alpha.$$

If it equals zero, this equates to occurrence of an assembly of the mechanism under consideration. There are only four possible combinations of the angles ϕ_{51} and ϕ_5.

Let us write a small subprogram for enumeration of these combinations changing them by applying parameter m.

$$\Delta(\phi_2) := \left| \begin{array}{ll} \phi_5(\phi_2) - \phi_{51}(\phi_2) + \alpha & \text{if } m = 1 \\ \phi_5(\phi_2) - \phi_{51a}(\phi_2) + \alpha & \text{if } m = 2 \\ \phi_{5a}(\phi_2) - \phi_{51}(\phi_2) + \alpha & \text{if } m = 3 \\ \phi_{5a}(\phi_2) - \phi_{51a}(\phi_2) + \alpha & \text{if } m = 4. \end{array} \right.$$

For example, Fig. 5, *a* shows function graph $\Delta(\phi_2)$ for $m = 3$ at the value $\phi_1 = 120°$.

In the Figure it is apparent that the studied function has the value of zero at three values of the angle ϕ_2. We will find the precise values of the angle ϕ_2 applying the Given-Find resolver. Taking the initial approximation of the angle ϕ_2 as equal to one radian, we obtain

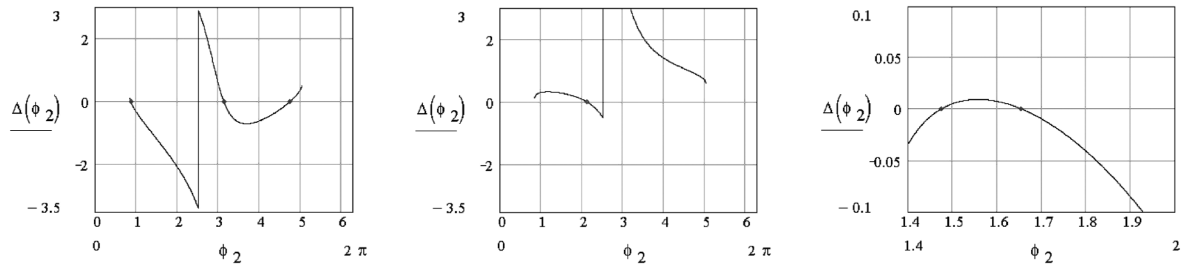
$$\phi_2 := 1 \qquad \text{Given} \qquad \Delta(\phi_2) = 0 \qquad \phi_2 := \mathrm{Find}(\phi_2)$$

$$\phi_2 = 49.211\,\mathrm{deg}.$$

For the initial approximation of $\phi_3 = 3$ radians, we have

$$\phi_2 := 3 \qquad \text{Given} \qquad \Delta(\phi_2) = 0 \qquad \phi_2 := \mathrm{Find}(\phi_2)$$

$$\phi_2 = 179.126\,\mathrm{deg}.$$



a *b* *c*

Fig. 5. Function graph $\Delta(\phi_2)$ for combinations of the angles $\phi5$ and ϕ_{51}, corresponding to: $a - m = 3$; $b - m = 1$; $c - m = 1$ and $\phi_1 = 0°$

For $\phi_2 = 5$ radians, we obtain

$$\phi_2 := 5 \qquad \text{Given} \quad \Delta(\phi_2) = 0 \qquad \phi_2 := \text{Find}(\phi_2)$$

$$\phi_2 = 271.612\,\text{deg}.$$

Fig. 5, *b* shows function graph $\Delta(\phi_2)$ for $m = 1$.

It is easily seen that in this case we have another assembly of the mechanism; to find this assembly we can take $\phi_2 = 2$ radians as the initial approximation. We have

$$\phi_2 := 2 \qquad \text{Given} \quad \Delta(\phi_2) = 0 \qquad \phi_2 := \text{Find}(\phi_2)$$

$$\phi_2 = 121.85\,\text{deg}.$$

For the studied configuration of a fourth class mechanism the total number of assemblies equals four.

Let us see if there are assemblies located in close proximity in this mechanism when being applied in the crushing machine. In this case the variation limits of the angle ϕ_2 can be taken from 1.4 to 2.3 radians. The study on this issue showed that with $\phi_1 = 0°$ and $m = 1$, there are two assemblies which are characterized by the following values of the angle ϕ_2: 84.4 and 94.8° (Fig. 5, *c*). These assemblies are shown in Fig. 6.

Such "aboutness" is rather hazardous since it occurs when a moveable jaw moves rightwards, i.e. when it is affected by peak loads.

One of the efficient ways to influence the variants of assemblies is to change the place of crank positioning and its length. Let us define the zone beyond which the trajectory of point *B* of the crank should not go.

We put the origin of coordinates at point *G* and fix link *5* directing the vector l_{51} upward vertically (Fig. 7). Moveable links *2*, *3*, *4* along with the fifth link make a four-bar linkage.

To solve the task of defining the zone of positioning the driven crank of the lever mechanism of the crusher it is necessary to find extremal values of the modulus of vector $\boldsymbol{R}$. the modulus of this vector characterizes the distance of point $\boldsymbol{B}$ from point $\boldsymbol{G}$.

We get the expression for the vector $\boldsymbol{R}$ as a function of the angle ϕ_3. The given data to the task remain the same, there are preserved directions of the vectors replacing the mechanisms links, only coordinates of point *G* are taken to be equal to zero.

Let us indicate the interval *CE* of link *2* as $\boldsymbol{l}_{22}$. The length of this vector is

$$l_{22} := l_{21} - l_2 \qquad l_{22} = 0.4.$$

The length of the vector $\boldsymbol{l}_{52}$ is

$$l_{52} := \sqrt{l_5^2 + l_{51}^2 - 2l_5 \cdot l_{51} \cdot \cos(\alpha)} \qquad l_{52} = 0.419.$$

With the vertically upward-directed vector $\boldsymbol{l}_{51}$, the angle is

$$\phi_{52} := -\frac{\pi}{2} + \text{asin}\left(\frac{l_5}{l_{52}} \cdot \sin(\alpha)\right) \qquad \phi_{52} = -55.246\,\text{deg}.$$

Taking into account lengths $\boldsymbol{l}_3 = 0.4$ and $\boldsymbol{l}_4 = 0.6$ m we ascertain that for this mechanism the Grashof rule (the sum of crank lengths and any other link is less than the

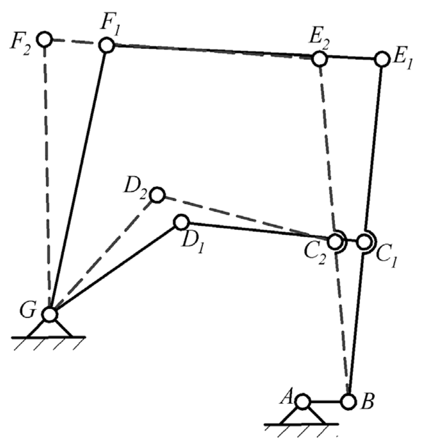


Fig. 6. Two assemblies of a crusher lever mechanism (ϕ_2: 84.4° – index 1; ϕ_2: 94.8° – index 2)

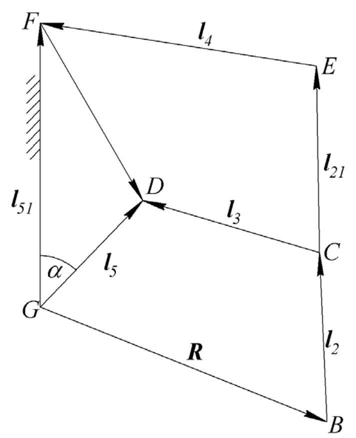


Fig. 7. Vector scheme for defining the zone of positioning the driven crank

sum of the other links) fails; therefore, the *ECDF* mechanism is a double-rocker four-link chain.

The vector closedness equation for this four-link chain is

$$l_4 \cdot \exp(\phi_4 \cdot 1i) - l_{22} \cdot \exp(1i \cdot \phi_2) = l_{52} \cdot \exp(1i \cdot \phi_{52}) + l_3 \cdot \exp(\phi_3 \cdot 1i).$$

Having made mathematical transformations, we obtain the dependence of the angle ϕ_4 on the angle ϕ_3. For the given mechanism assembly it is written as

$$\phi_4(\phi_3) := 2\,\mathrm{atan}\left(\frac{-B(\phi_3) + \sqrt{B(\phi_3)^2 - 4A(\phi_3) \cdot C(\phi_3)}}{2A(\phi_3)}\right).$$

In this formula $A(\phi_3)$, $B(\phi_3)$, $C(\phi_3)$ are numerical coefficients, depending on the value of ϕ_3.

The expression for the vector **R**, as a function of the angle ϕ_3

$$R(\phi_3) := l_5 \cdot \exp\left[1i \cdot \left(\frac{\pi}{2} - \alpha\right)\right] \ldots$$
$$+ l_3 \cdot \exp(1i \cdot \phi_3) - l_2 \cdot \exp(1i \cdot \phi_2(\phi_3)).$$

It is necessary to obtain the value of the minimal and maximal distances of point *B* to point *G*, that is why we test the modulus of this vector for the extreme point, that is

$$R(\phi_3) := |R(\phi_3)|.$$

Let us apply special functions of Mathcad.

$$\phi_3 := 1.5 \qquad \text{Given} \qquad \frac{-\pi}{2} \le \phi_3 < \pi$$

$$\phi_{3min} := \mathrm{Minimize}(R, \phi_3)$$

$$\phi_{3min} = 87.123\,\mathrm{deg} \qquad R(\phi_{3min}) = 0.35659$$

$$\phi_3 := 0 \qquad \text{Given} \qquad \frac{-\pi}{2} \le \phi_3 < \pi$$

$$\phi_{3max} := \mathrm{Maximize}(R, \phi_3)$$

$$\phi_{3max} = -4.999\,\mathrm{deg} \qquad R(\phi_{3max}) = 0.68198\,.$$

Module ***R*** has the minimal value with $\phi_{3\,min} = 87.123°$, and the maximal value with $\phi_{3\,max} = -4.999°$. The mean value of modulus

$$R_m := \frac{R(\phi_{3max}) + R(\phi_{3min})}{2} \qquad R_m = 0.531,$$

is the radius of a circle on which the rotational center of the driven crank of the crusher mechanism should be placed in case of its maximal length.

The maximal length of the crank is

$$l_{AB} := \frac{R(\phi_{3max}) - R(\phi_{3min})}{2} \qquad l_{AB} = 0.151.$$

Fig. 8 shows arcs of the circle of the radiuses R_{min}, R_{max} and R_m, while the dotted line shows one of possible trajectories of point *B* of the crank with its maximal length. If, based on the technological requirements, the

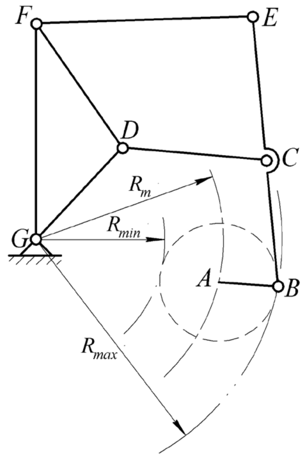


Fig. 8. The zone of possible positioning of the crank for the crusher lever mechanism

crank has to be of a shorter length, the trajectory of its point *B* should not go beyond the bounds of its area between the segments of the arcs R_{min} and R_{max}.

Conclusions. Theoretical research on geometry of the planar fourth class Assur group of a lever mechanism of a jaw crushing machine was conducted.

It has been established that the number of possible assemblies in this mechanism without reference to a crushing machine equals four.

The mechanism being applied in a crusher, the maximum number of assemblies equals two and they can be located in close proximity which may cause an emergency situation.

The synthesis of these mechanisms is to exclude occurrence of assemblies located in close proximity, particularly at the crushing stage when the mechanism links are affected by peak loads. This problem can be solved by a rational choice of positioning a power-driven crank and its length.

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Про застосування механізмів високих класів у важконавантажених машинах

І. М. Мацюк[1], *Е. М. Шляхов*[1], *О. І. Єгурнов*[2]

1 – Національний технічний університет „Дніпровська політехніка“, м. Дніпро, Україна, e-mail: shlyahove@nmu.org.ua

2 – ВАТ „Ана-Темс“, м. Дніпро, Україна, e-mail: yegurnov@yahoo.com

Мета. Оцінити вплив на працездатність машин можливих варіантів складань стрижневої структурної групи четвертого класу у складі щокової дробильної машини.

Методика. У роботі виконане теоретичне дослідження можливих складань структурної групи четвертого класу, що входить до складу щокової дробарки.

Результати. На підставі виконаного дослідження встановлено, що кількість можливих складань групи в цьому механізмі, без прив’язки його до дробильної машини, дорівнює чотирьом. При застосуванні цього механізму у дробарці максимальна кількість складань дорівнює двом і вони можуть перебувати в безпосередній близькості, що може стати причиною виникнення нештатної ситуації.

Наукова новизна. Полягає у визначенні кількості можливих складань структурної групи такого виду. Для розглянутої групи встановлено, що максимальна кількість складань дорівнює чотирьом. Запропоновано алгоритм знаходження зони можливого розташування приводного кривошипу механізму дробарки, що дозволяє впливати на віддаленість двох близько розташованих складань.

Практична значимість. Використаний у роботі алгоритм пошуку можливих складань групи четвертого класу за допомогою програмного продукту Mathcad може бути використаний при синтезі подібних дробильних машин.

Ключові слова: *щокова дробарка, складання механізму, структурна група, програма Mathcad*

О применении механизмов высоких классов в тяжелонагруженных машинах

И. Н. Мацюк[1], *Э. М. Шляхов*[1], *А. И. Егурнов*[2]

1 – Национальный технический университет „Днепровская политехника“, г. Днепр, Украина, e-mail: shlyahove@nmu.org.ua

2 – ООО „Ана-Темс“, г. Днепр, Украина, e-mail: yegurnov@yahoo.com

Цель. Оценить влияние на работоспособность машин возможных вариантов сборок стержневой структурной группы четвертого класса в составе щековой дробильной машины.

Методика. В работе выполнено теоретическое исследование возможных сборок структурной группы четвертого класса, входящей в состав щековой дробилки.

Результаты. На основании выполненного исследования установлено, что число возможных сборок группы в этом механизме, без привязки его к дробильной машине, равно четырем. При применении этого механизма в дробилке максимальное число сборок равно двум и они могут находиться в непосредственной близости, что может явиться причиной возникновения нештатной ситуации.

Научная новизна. Заключается в определении количества возможных сборок структурной группы такого вида. Для рассмотренной группы установлено, что максимальное число сборок равно четырем. Предложен алгоритм нахождения зоны возможного расположения приводного кривошипа механизма дробилки, что позволяет влиять на удаленность двух близкорасположенных сборок.

Практическая значимость. Использованный в работе алгоритм поиска возможных сборок группы четвертого класса с помощью программного продукта Mathcad может быть использован при синтезе подобных дробильных машин.

Ключевые слова: *щековая дробилка, сборки механизма, структурная группа, программа Mathcad*

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