2-бөлім **Раздел 2** **Section 2**

Механика **Механика** **Mechanics**

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U.T. Abdrakhimov*, **A.A. Akimkhanova**
Almaty Technological University, Kazakhstan, Almaty
*e-mail: 7055478214@mail.ru

KINETOSTATIC CALCULATION OF A HIGH CLASS LIFTING MECHANISM (BY ASSUR CLASSIFICATION) OF A TECHNOLOGICAL LOAD-LIFTING MACHINE

To improve the technical level of mechanical engineering, it is necessary to expand the technical capabilities of equipment and devices, existing structures, as well as equipping them with fundamentally new mechanisms.
The creation of new designs of machines and mechanisms requires the design of lever mechanisms of high classes, the working bodies of which can reproduce complex movements and trajectories.
Due to the presence of movable, variable closed hinge-lever contours, lever mechanisms of high classes, unlike traditional lever mechanisms with lower pairs, perceive heavy loads, have increased rigidity.
On the basis of a high-class rectilinear guiding mechanism, a lever mechanism has been designed that can be used in lifting machines and mechanisms.
An important machine kinetics section is the force analysis of mechanisms by the given laws of motion of the initial elements and known external forces applied to the mechanism segments. The paper proposes a new design of a lifting mechanism, the peculiarity of which is that it belongs to the high-class Assur group by its structure, which causes ease of operation and reliability of the design. For this design, a synthesis of the mechanism was carried out in [1], and now a force analysis of this mechanism is being performed using the Academician U.A. Dzholdasbekov method of introducing auxiliary points.
Key words: high-class mechanisms, kinetostatic analysis, U.A. Dzholdasbekov auxiliary points.

У.Т. Абдрахимов*, А.А. Акимханова
Алматы технологиялық университеті, Қазақстан, Алматы қ.
*e-mail: 7055478214@mail.ru

Технологиялық жүк көтергіш машинаның жоғары класты (Ассур классификациясы бойынша) көтеру механизмінің кинетостатикалық есебі

Машина жасаудың техникалық деңгейін арттыру үшін жабдықтар мен құрылғылардың, қолданыстағы конструкциялардың техникалық мүмкіндіктерін кеңейту, сондай-ақ оларды түбегейлі жаңа механизмдермен жабдықтау қажет.
Машиналар мен механизмдердің жаңа конструкцияларын жасау үшін жұмыс органдары күрделі қозғалыстар мен траекторияларды қайталай алатын жоғары класты тетіктерді жобалау қажет.
Жылжымалы өзгермелі жабық топсалы-рычагтық тізбектердің болуына байланысты жоғары класты рычагтық механизмдер, төменгі жұптары бар дәстүрлі рычагтық механизмдерден айырмашылығы, үлкен жүктемелерді қабылдайды және қаттылығы жоғарылайды.
Жоғары деңгейлі түзу сызықты бағыттаушы механизмнің негізінде жүк көтергіш машиналар мен механизмдерде қолдануға болатын рычаг механизмі жасалған.

Машиналар динамикасының маңызды тарауларының бірі-механизм буындарына қолданылатын бастапқы буындар мен белгілі сыртқы күштердің берілген қозғалыс заңдары бойынша механизмдердің күштік талдауы. Мақала көтеру механизмның жаңа дизайнын ұсынады, оның ерекшелігі құрылымында, ол жоғары класты Ассур тобына жатады, бұл басқарудың қарапайымдылығы мен конструкцияның сенімділігіне әкеледі. Бұл жоба бойынша [1] жұмыста механизмнің синтезі мен талдауы жүргізілді, ал қазір бұл механизмның күштік талдауы академик Ө.А. Жолдасбековтың көмекші нүктелерді енгізу әдісі бойынша жүргізілуде.
Түйін сөздер: жоғары класс механизмдері, кинетостатикалық талдау, Ө.А. Жолдасбековтың көмекші нүктелері.

У.Т. Абдрахимов*, А.А. Акимханова
Алматинский технологический университет, Казахстан, г. Алматы
*e-mail: 7055478214@mail.ru

Кинетостатический расчет механизма подъема высокого класса (по классификации Ассура) технологической грузоподъемной машины

Для повышения технического уровня машиностроения, требуется расширение технических возможностей оборудования и устройств, существующих конструкций, а также оснащение их принципиально новыми механизмами.
Создание новых конструкций машин и механизмов, требует проектирование рычажных механизмов высоких классов, рабочие органы которых могут воспроизводить сложные движения и траектории.
Благодаря наличию подвижных изменяющихся замкнутых шарнирно-рычажных контуров, рычажные механизмы высоких классов в отличие от традиционных рычажных механизмов с низшими парами, воспринимают большие нагрузки, обладают повышенной жесткостью.
На базе прямолинейно-направляющего механизма высокого класса спроектирован рычажный механизм, который может быть использован в грузоподъемных машинах и механизмах.
Одним из важных разделов динамики машин является силовой анализ механизмов по заданным законам движения начальных звеньев и известных внешних сил, приложенных к звеньям механизма. В статье предлагается новая конструкция подъемного механизма, особенностью которого является то, что он по своей структуре относится к группе Ассура высокого класса, что обуславливает простоту в управлении и надежность конструкции. Для данной конструкции в работе [1] производился синтез механизма, а теперь производится силовой анализ данного механизма по методу ввода вспомогательных точек академика У.А. Джолдасбекова.

Ключевые слова: механизмы высокого класса, кинетостатический анализ, вспомогательные точки У.А. Джолдасбекова.

1 Introduction

In order to increase the technical level of mechanical engineering, it is necessary to expand the processing capability of mechanical facilities and devices of existing configurations, as well as equipping them with fundamentally new mechanisms. The development of new configurations of machines and mechanisms, as well as equipment, requires design engineering high class lever mechanisms whose working parts can reproduce complex movements and trajectories. Due to the availability of movable, variable closed hinge-lever contours, high class lever mechanisms, unlike traditional lever mechanisms with lower pairs, stand heavy loads and have increased rigidity [3]. They provide qualitatively diverse spatial movements of two or more output executive parts. The applicability of spatial mechanisms of high classes undoubtedly causes interest in the development of research on these mechanisms. In this regard, the use of high class spatial mechanisms for the purpose of designing lifting machines and mechanisms is highly relevant.

2 Materials and methods

As part of the project on developing domestic universal configurations of technological lifting machines and mechanisms, a new high-class mechanism with a leading element in the form of a slider has been proposed. The paper on "Justification and development of a new mechanism for lifting devices of transport and logistics sectors of the economy" in [1] presents a synthesis of a high-class rectilinear guiding mechanism with a leading slider (Figure 1).

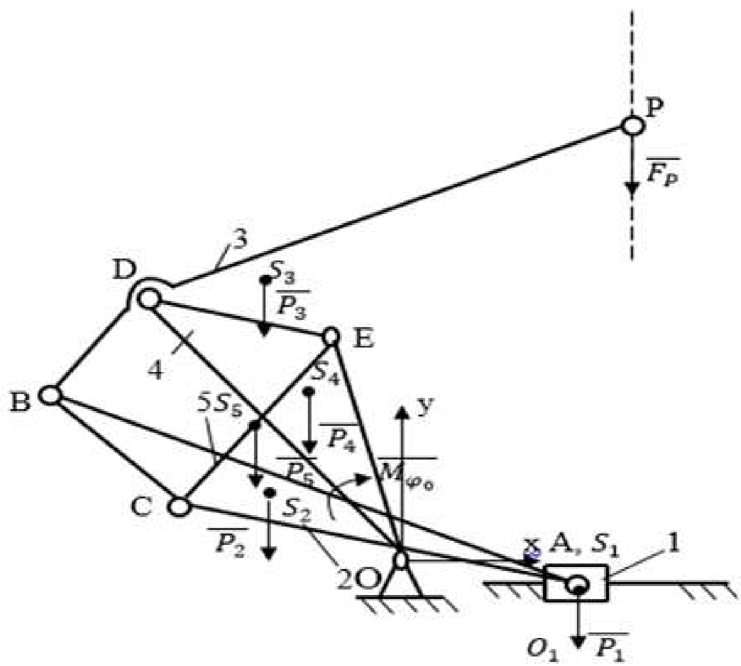

Figure 1: Lifting mechanism

In this construction design, the input element 1 receives rectilinear motion from the hydraulic cylinder. It acts on the three-hinged lever 2, forcing it to turn. Two connecting rods are attached to the three sides of the lever. The first connecting rod 3 interacts with a three-hinged rocker arm 4, while it performs rotational movements relative to the point O_1. The second connecting rod – 5 is the output. The movement of the element 4 around the hinge O_1 causes rectilinear vertical movement of the working point P, according to the parameters of this mechanism determined by the synthesis.

The expressions obtained during kinematic analysis solve the problem of determining the generalized coordinates of the lifting mechanism of the designed equipment according to a given position of the input element (Figure 2). Based on the kinematic dependences obtained, the displacements, velocities and accelerations of the elements of the mechanism under study are determined [9].

An important machine kinetics section is the force analysis of mechanisms by the given laws of motion of the initial elements and known external forces applied to the mechanism segments [4]. There are cases when some of the external forces under which the accepted laws of motion of the initial elements take place are considered unknown. In solving both problems, the principle of kinetostatics is used, according to which the mechanism as a whole or its separate element can be considered as being in equilibrium, if inertia forces are added to all external forces acting on it.

Having found the vectors of the inertia forces of the elements and the points of their application, we will proceed to the force analysis of the projected mechanism (a high-class mechanism, namely the IV$^{\text{th}}$ according to the Assur-Artobolevsky classification), based on the introduction of A. Dzholdasbekov auxiliary points, which will reduce the problem of HCM (high class mechanisms) kinetostatics to solving a set of linear equations with one desired quantity.

3 Results and discussion

The statement that the order of force calculation corresponds to the reverse order of kinematic analysis must be observed only with an unknown balancing force [8]. If in order to determine the reaction forces in kinematic pairs the balancing force is known, it is obvious that the results of the force calculation should not depend on the order of analysis. Such a force will be $\overrightarrow{F}_q$ – the driving force whose line of action is always determined by the engineering design of the transmission mechanism.

The solution of the problem should be carried out in the following sequence:

- determining the accelerations of the centres of gravity of the mechanism elements;
- finding the forces and moments of inertia acting on the mechanism elements according to the known element accelerations;
- determining the desired reactions in the kinematic pairs of the mechanism and the balancing force at the input element.

Let's focus in more detail on the last point of solving the problem. Using the method of replacing the leading element, we transform the mechanism (Figure 1). Taking element 4 as the leading element, we obtain a class III mechanism with the structural formula I (4) $\rightarrow$ III (3, 5, 2, 1)

For such a mechanism, solving force analysis problems explicitly becomes easy.

We set up the equilibrium equations of the moments of forces acting on elements 3 and 5 relative to the corresponding points B and C, and receive

$$\begin{cases} P_3^x \cdot l_{BS_3} \cdot \sin\varphi_3 - P_3^y \cdot l_{BS_3} \cos\varphi_3 + N_{34}^{y_3} \cdot l_{BD} = 0, \\ P_5^x \cdot l_{CS_5} \cdot \sin\varphi_5 - P_5^y \cdot l_{CS_5} \cos\varphi_5 + N_{54}^{y_5} \cdot l_{EC} = 0. \end{cases} \tag{1}$$

These equations are easily solvable with respect to unknowns N_{34}^{y3}, N_{54}^{y5} provided that l_{BD} and l_{EC} are non-zero.

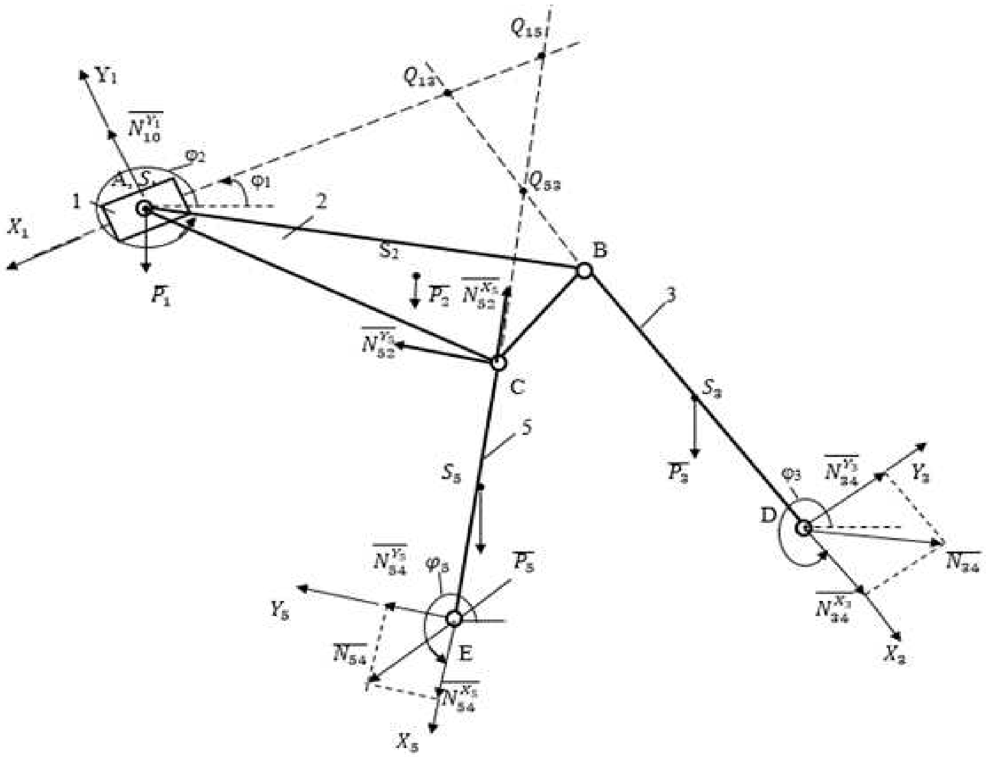

Figure 2: Scheme for the calculation

Then we introduce auxiliary points, the so-called Assur – Dzholdasbekov critical points Q_{15}, Q_{53}, Q_{13} and determine their coordinates $({}_{Q}, {}_{Q})$ as the coordinates of the intersection points of the corresponding right lines passing along the pinion carriers O_1A, EC, BD (Figure 2).

Using the equations of the right lines passing through the given points C and B with known angles of inclination φ_5 and φ_3, we obtain the following system of equations:

$$\begin{cases} Y_{Q_{53}} - X_{Q_{53}} \cdot \operatorname{tg}\varphi_5 = Y_C - \operatorname{tg}\varphi_5, \\ Y_{Q_{53}} - X_{Q_{53}} \cdot \operatorname{tg}\varphi_3 = Y_B - \operatorname{tg}\varphi_3. \end{cases} \tag{2}$$

This system of equations has a unique solution by definition $(X_{Q_{53}},\ Y_{Q_{53}})$, if $\operatorname{tg}_5 - \operatorname{tg}_3 \neq 0$.

The coordinates $X_{Q_{13}},\ Y_{Q_{13}},\ X_{Q_{15}},\ Y_{Q_{15}}$ of the critical points Q_{13} and Q_{15} are determined similarly.

Next, we draw up the equilibrium equations of the moments of all forces acting on the

Assur group as a whole relative to the singular point Q_{53}

$$P_5^x \cdot l_{Q_{53}S_5} \cdot \sin\varphi_5 - P_5^y \cdot l_{Q_{53}S_5} \cdot \cos\varphi_5 - P_2^x \cdot l_{Q_{53}S_2} \cdot \sin\varphi_{S_2} -$$
$$-P_2^y \cdot l_{Q_{53}S_2} \cdot \cos\varphi_{S_2} - P_3^x \cdot l_{Q_{53}S_3} \cdot \sin\varphi_3 +$$
$$+P_3^y \cdot l_{Q_{53}S_3} \cdot \cos\varphi_3 + P_1^x \cdot l_{Q_{53}S_1} \cdot \sin\varphi_{S_1} - P_1^y \cdot l_{Q_{53}S_1} \cdot \cos\varphi_{S_1} + \tag{3}$$
$$+N_{54}^{y_5} \cdot l_{Q_{53}E} - N_{34}^{y_3} \cdot l_{Q_{53}D} + N_{10}^{y_1} \cdot l_{Q_{53}S_1} \cdot \cos\varphi_{s_2} = 0,$$

where $\varphi_{S_i} = \text{arctg}\left(\dfrac{y_i - y_{Q_{53}}}{X_i - X_{Q_{53}}}\right)$ – angular coordinate of vector $\overrightarrow{Q_{53}S_{i=1,2}}$;

$\varphi_{O_1} = \text{arctg}\left(\dfrac{y_{O_1} - y_{Q_{53}}}{X_{O_1} - X_{Q_{53}}}\right)$ – angular coordinate of vector $\overrightarrow{Q_{53}O_1}$;

$\varphi_{A_1} = \text{arctg}\left(\dfrac{y_{A_1} - y_{Q_{53}}}{X_{A_1} - X_{Q_{53}}}\right)$ – angular coordinate of vector $\overrightarrow{Q_{53}A_i}$.

This equation has one unknown $N_{10}^{Y_1}$.

Next, the unknowns $\overline{N}_{34}^{X_3}$ and $\overline{N}_{54}^{X_5}$ are determined similarly.

To determine the reaction in the hinge (Figure 2) we draw up the equilibrium equations of element 5 in projections to coordinate systems E_{X_5,Y_5}

$$\begin{cases} N_{54}^{X_5} + P_5^X \cdot \cos\varphi_5 + P_5^y \cdot \sin\varphi_5 + N_{52}^{X_5} = 0, \\ N_{54}^{Y_5} - P_5^X \cdot \sin\varphi_5 + P_5^Y \cdot \cos\varphi_5 + N_{52}^{Y_5} = 0. \end{cases} \tag{4}$$

These equations are always solvable with respect to the desired $\overline{N}_{52}^{X_5}$ and $\overline{N}_{52}^{Y_5}$. The reactions in other hinges are determined similarly.

For the equilibrium of the conventional leading element 4, the following conditions must be met:

$$N_{40}^X = N_{45}^X + N_{43}^X + P_4^X,$$
$$N_{40}^Y = N_{45}^Y + N_{43}^Y + P_4^Y, \tag{5}$$
$$M_{40} = N_{45}^X(Y_0 - Y_E) + N_{45}^Y(X_0 - X_E) + N_{43}^X(Y_0 - Y_D) + N_{43}^Y(X_0 - X_D) +$$
$$+P_5^X(Y_0 - Y_{S_4}) + P_5^Y(X_0 - X_{S_4}).$$

In this equation, in order to determine the whole system, the moment value M_{40} is introduced.

The following method of solving the problem is proposed:

- a number of specific values of the desired balancing force $\overline{P}_{yp}$ at the input element 1 is set within the range:

$$P_{yp}^{\min} \leq P_{yp \cdot i} \leq P_{yp}^{\max}, \quad i = \overline{0, n}$$

$$P_{yp \cdot i+1} = P_{yp \cdot i} + \Delta P, \quad P_{ypi} = P_{yp}^{\min} + i\Delta P,$$

where $\Delta P = \dfrac{P_{yp}^{\max} - P_{yp}^{\min}}{n-1}$, $P_{yP}^{\max}$, $P_{yP}^{\min}$ – can be determined based on the design features;

- each value P_{ypi} is accepted as set for the transformed mechanism, the force analysis is performed, and the desired reactions in kinematic pairs as well as the conditional moment M_{40} are determined;
- the criteria for the correct result of the power analysis will be meeting the condition $M_{40} = 0$.

4 Conclusion

The kinetostatic analysis of the mechanism was first performed using the critical points suggested by academician U.A. Dzholdasbekov, which allowed solving the problem in an explicit analytical form. The paper scientifically substantiates and practically implements the practice of developing new effective types of auxiliary lifting technological machines and mechanisms. As can be seen from the research solutions, the research is aimed at expanding the technological capabilities of equipment and devices of existing configurations, equipping them with fundamentally new high class mechanisms, whose working parts can reproduce complex movements and trajectories, and also have increased rigidity, due to the availability of movable modifiable closed hinge-lever contours.

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