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DETERMINATION OF THE BALANCING MOMENT OF THE SIX-LINK STRAIGHT-LINE CONVERSION MECHANISM OF THE BEAMLESS ROD PUMP DRIVE

The paper considers a six-link straight-line conversion scissor mechanism, which is used as a new design of the conversion mechanism of the beamless rod pump drive.
The purpose of balancing the conversion mechanism of rod pump drive (RPD) is to reduce the required engine power and its uniform load per cycle of movement. The task of optimal dynamic balancing of the conversion mechanism of the RPD is to determine the optimal values of the weight of the counterweight G_{CW} and the distance $l = OL$ from the crank axis at which the minimum peak value of the balancing moment on the crank shaft is provided. In practice, the determination of these values is carried out empirically by comparing two peak values – the torque on the crank shaft for the cycle of the mechanism movement. The result kinetostatics analysis, solving the equilibrium equations of the six-link scissor mechanism, determined reactions of mechanism hinges and values – the torque on the shaft of the crank shaft per cycle of movement of the mechanism. Also, for the reliability of the results, according to the principle of possible movements through the power of the acting forces, values – the torque on the crank shaft were determined.
Key words: Drive, transforming mechanism, crank, connecting rod, balancer, poise, analysis.

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Штангалы сорғыш қондырғыларының теңгерімсіз жетегінің алтызвенолы түзу-сызықты бағыттауыш түрлендіргіш механизмінің теңдестіру моментін анықтау

Бұл мақалада алтызвенолы түзу-сызықты бағыттауыш түрлендіргіш механизмі туралы айтылады, оны біз штангалық сорғы қондырғыларының теңгерімсіз жетегінің түрлендіргіш механизмінің жаңа конструкциясы ретінде қолданамыз.
Штангалы сорғыш қондырғыларының (ШСҚ) түрлендіргіш механизмін теңестірудің мақсаты қозғалтқыштың қажетті қуатын және оның қозғалыс циклі кезінде оның біркелкі жүктемесін азайту болып табылады. $G_{\prod}$ түрлендіру механизмін оңтайлы динамикалық теңдестірудің міндеті қарсы салмақтың GP оңтайлы мәндерін және иінді $l_{\prod} = OL$ білікте теңдестіру сәтінің минималды мәні қамтамасыз етілетін иінді біліктен қашықтықты анықтау болып табылады. Іс жүзінде бұл шамаларды анықтау $_{\text{T}}$ екі мәні – механизмнің бір қозғалыс циклына иінді біліктің айналу моментін салыстыру арқылы эмпирикалық түрде жүзеге асырылады.Кинетостатикалық талдау нәтижесінде алты звенолы топсалы-иінтіректі механизмнің буындарының тепе-теңдік теңдеулерін, сонымен қатар алғанда механизм ілмектерінің реакция күш және $_{\text{T}}$ мәнінің – механизмнің қозғалу циклі үшін иінді біліктің айналу моментінің бірлесіп шешілуі нәтижесінде анықталды.

Сондай-ақ, нәтижелердің сенімділігі тексеру үшін әрекет етуші күштердің қуаттары арқылы ықтимал орын ауыстыру қағидасына сәйкес $_{\text{т}}$ мәндері – иінді біліктің айналу моменті анықталды.
Түйін сөздер: Жетек, түрлендіруші механизм, айналшақ, бұлғақ, балансир, тепе-теңдік, талдау.

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Определение уравновешивающего момента шестизвенного прямолинейно-направляющего преобразующего механизма безбалансирного привода штанговых насосных установок

В данной статье рассматривается шестизвенный прямолинейно-направляющий шарнирно-рычажный механизма, которого используем в качестве новой конструкции перобразующего механизма безбалансирного привода штанговых насосных установок.
Цель уравновешивания преобразующего механизма штанговых насосных установок (ШНУ) заключается в уменьшении необходимой мощности двигателя и равномерной его нагрузки за цикл движения. Задача оптимального динамического уравновешивания преобразующего механизма ШНУ заключается в определении оптимальных значений веса противовеса $G_{\prod}$ и расстояние $l_{\prod} = OL$ от оси кривошипа при котором обеспечивается минимальное пиковое значение уравновешивающего момента на валу кривошипа. На практике определение этих величин осуществляется эмпирически путем сравнения двух пиковых значений $_{\text{ур}}$ – крутящего момента на валу кривошипа за цикл движения механизма. В результате кинетостатического анализа, решая совместно уравнения равновесия звеньев шестизвенного шарнирно-рычажного механизма, определены силы реакции шарниров механизма и значения $_{\text{ур}}$ – крутящего момента на валу кривошипа за цикл движения механизма. Так же для проверки достоверности результатов, по принципу возможных перемещений через мощности действующих сил определили значения $_{\text{ур}}$ – крутящего момента на валу кривошипа

Ключевые слова: Привод, преобразующий механизм, кривошип, шатун, балансир, уравновешенность, анализ.

1 Introduction

Consider the dynamic modes of operation of the rod pump drive. When rod string goes up, the balancing devices give the power body the energy accumulated during the motion of the rods down. In the device under consideration, the counterweights are attached by special devices to the 3rd link of the mechanism. It worth to note that the weight of the 3rd link also plays the role of a counterweight, i.e. performs useful work.

In the problem of dynamic synthesis of mechanisms, the structural diagram of the mechanism and, as a rule, metric parameters (lengths of links) are considered known. When designing the mechanism, the given dynamic characteristics and mass-inertial characteristics are found, and in some cases constant geometric parameters are sought, at which the required dynamic characteristics of the movement are provided. As a rule, the problem of dynamic balancing of mechanisms is considered for mechanisms for which $a_{\max}/g > 1$. The following tasks are most common [4, 5, 6].

1. balance (balance the masses) the impact of the mechanism on the support (the foundation on which the mechanism is located);

2. balance (balance the moment or power) the impact on the engine of the mechanism (or on the prime mover);

3. balance the effect on the kinematic pairs.

It is clear that to reduce the value of the inertia forces of moving links, it is necessary to reduce the mass of these links. Due to the complexity of solving the problem using this feature, such problems are solved by special methods [1, 5, 7].

It is customary to distinguish between static and dynamic balancing of the mechanism; their elimination in the designed mechanism will correspond to its static and dynamic balancing [1]. In this case, according to the degree of equilibrium, you can get an exact or approximate solution (balancing).

Traditionally, the criterion for accurate static balancing of the mechanism is the condition that the main vector of inertia forces of its links is equal to zero.

$$\overline{F}^{IN} = 0 \tag{1}$$

which corresponds to the immobility of the general center of mass of the mechanism. With precise dynamic balancing, simultaneously with the specified condition, it is still necessary to zero out the main moment of the inertia forces of the links, i.e.

$$\overline{F}^{IN} = 0, \quad M_0 = 0. \tag{2}$$

If the mechanism has managed to achieve the exact balancing conditions in any way, then these conditions will be preserved under any law of motion of the input link and, consequently, the balance of the mechanism (both static and dynamic) becomes an integral quality of the mechanism.

Approximate balancing of the mechanism can be considered as an approximation to the exact one, when in solving a specific problem some minor conditions can be neglected.

2 Analysis of literature data and problem statement

As a drive for rod pumps, beam-pumping units are traditionally used, which have a simple studied scheme, and in comparison with other drives, an economical, repair-suitable design [1, 2, 3-7, 8, 9,10].

Figure 6a shows the straight-line lambda-shaped Chebyshev mechanism, and Figures 1b-e show its modifications and related mechanisms, which can be obtained by applying the Roberts-Chebyshev theorem. The mechanisms shown in Figures 1a and 1b provide a fairly high accuracy of reproducing a rectilinear trajectory with equal lengths of the connecting rod-rocker group links and with strict observance of symmetry in the size of the connecting rod (the connecting rod is an isosceles triangle).

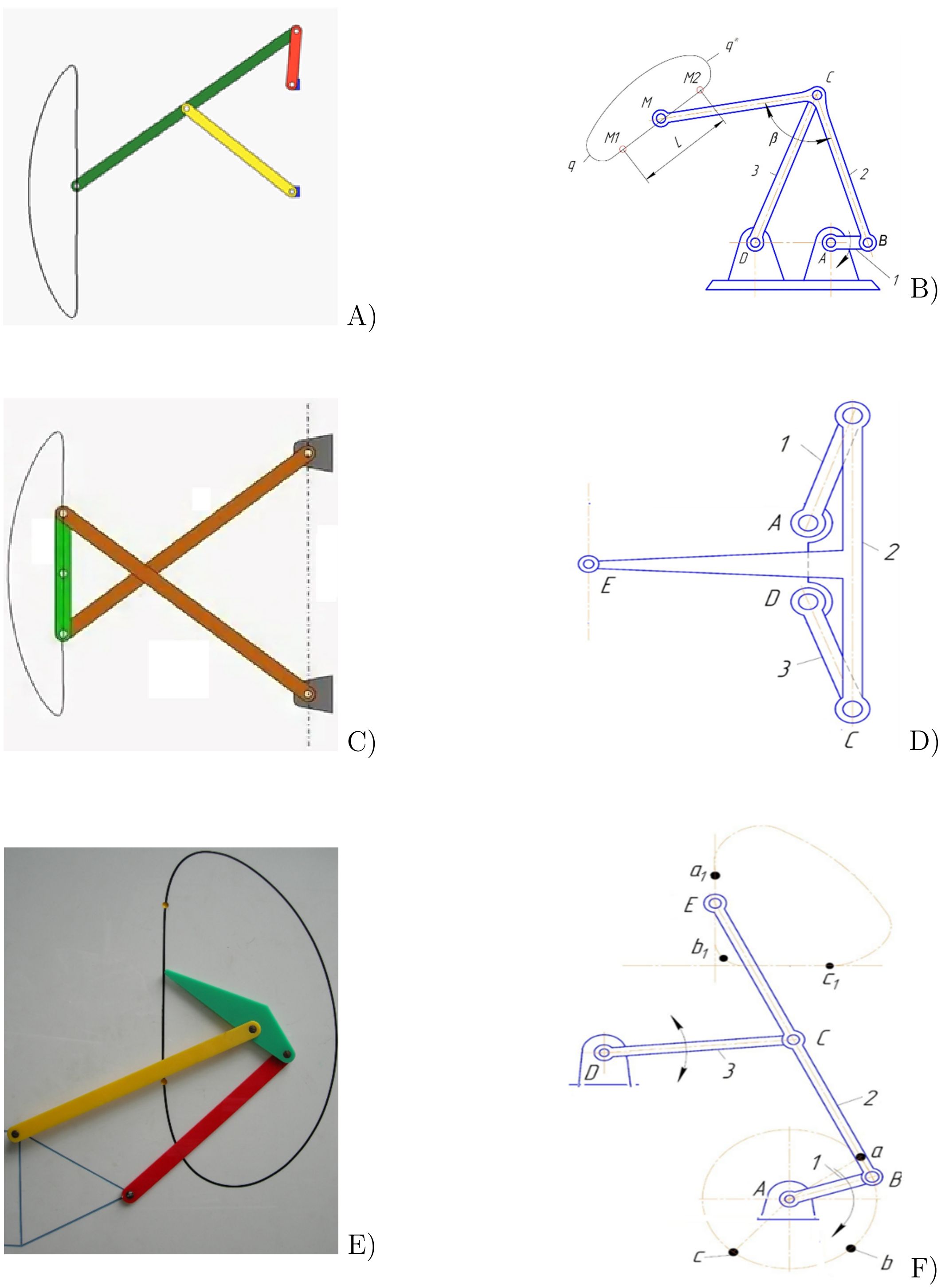


Figure 1: Chebyshev Mechanisms

For further research, we selected the mechanisms shown in Figures 1a and 1g, since the use of the remaining schemes is difficult due to the lack of access to the wellhead. However, a more detailed study of the functionality of the selected schemes showed that it is not possible to achieve a reduction in overall dimensions here. Thus, the gain is only in eliminating the arched horsehead.

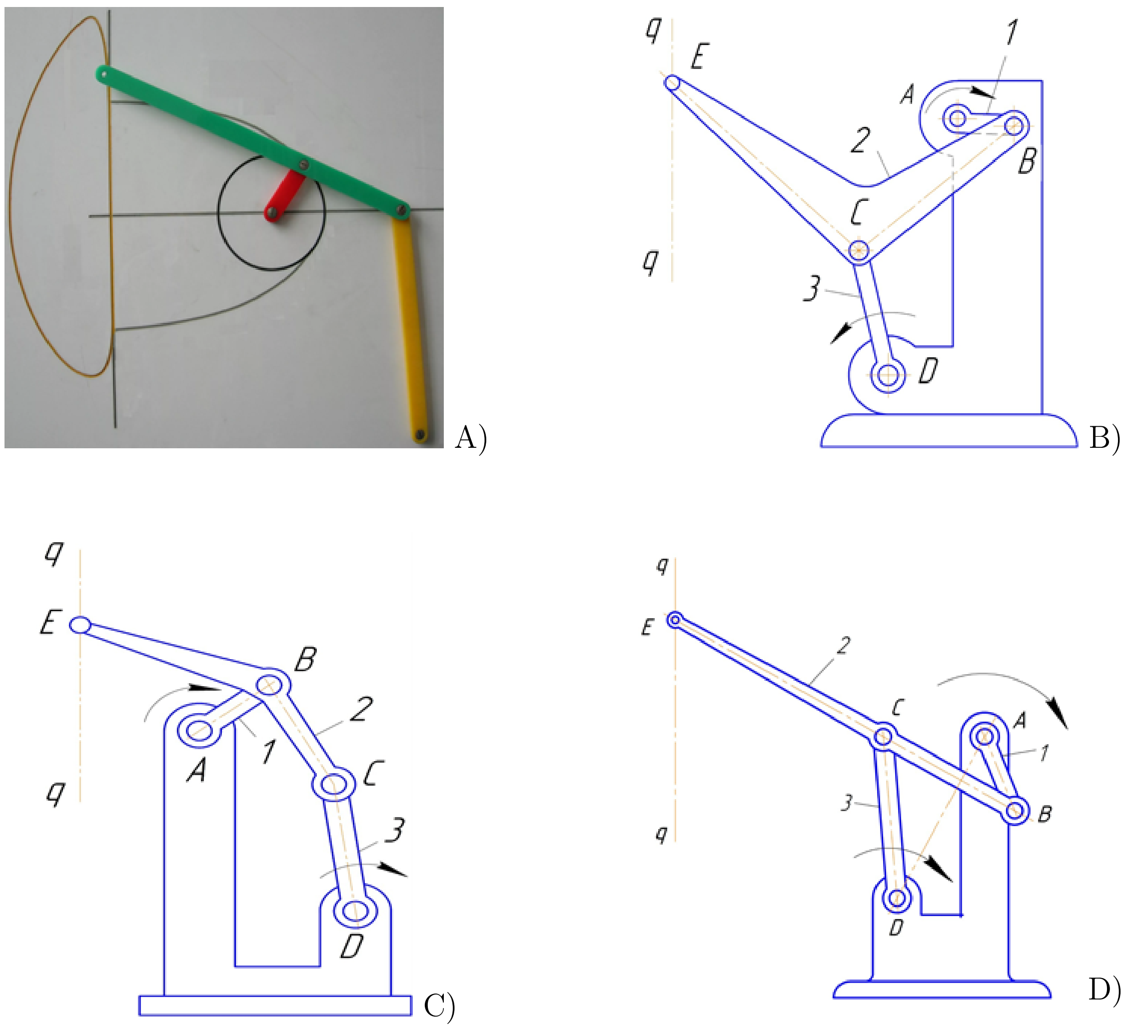


Figure 2: The Evans mechanism and its variants

Much more interesting were the mechanisms of the Evans type (Figure 2) and Evans – de Jonge type (Figure 3), which are characterized by good access to the well, as well as a small ratio of the length of the straight section of the rod curve trajectory to the lengths of the mechanism links. Upon closer examination, the schemes in Figures 2 were the most promising. The diagram in Figure 7a is the most compact.

However, to ensure the maximum ratio of the length of the straight segment to the length of the connecting rod, an additional synthesis must be carried out taking into account this additional restriction. This mechanism, therefore, is a competing scheme with respect to the Roberts scheme, and the position of the upper rack was significantly lower than in the Roberts mechanism.

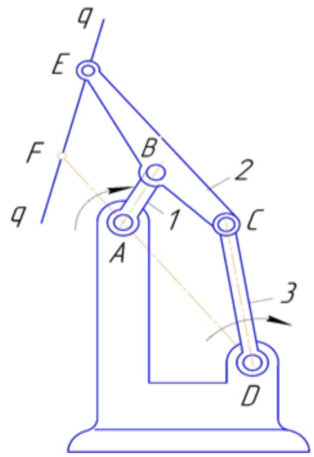


Figure 3: Evans-de Jonge Mechanisms

In [11, 12], a methodology, algorithms, and software for the kinematic and kinetostatic calculation and optimal balancing of the conversion mechanisms of the RPD with a two-arm beam and rotary equalizer were developed. The methodology and software were used to calculate the pumping units RPD6-2,5-3500 and RPD8-3-4500 with maximum loads in the wellhead gland of 6t and 8t.

An alternative option is to use straight-line mechanisms as a conversion mechanism. Thus, the advantages of the PU (pumping unit) "with a floating beam" were confirmed by the experience of developing and operating the 2CKM7 type, created on the basis of the CKH70-3012 pumping unit [2, 11]. The Evans lemniscate straight line is used here as a converting mechanism. Another example of the use of straight-line mechanisms is the recent "Minnesota" development [13], in which the reciprocating motion of the rod suspension is provided by the Roberts mechanism. The goal of the development is initially to eliminate the massive and complex head ("horse head") in typical installations [6]. Moreover, the overall dimensions in both cases were almost two times smaller than the prototypes.

3 Solution of the problem

There are other types of straight-line mechanisms that could also be used effectively [11, 12, 13, 14, 15]. But a systematic study of them in relation to the problem under consideration has never been conducted.

Next, we will conduct a kinetostatic analysis of the six-link scissor mechanism of the RPD. The mechanism is affected by the load in the wellhead oil seal and the gravity of the links and loads (Figure 4). The forces of friction in the joints and the forces of inertia are not taken into account, since the values of these forces are insignificant.

For the equilibrium of the system of forces, the main vector of the force and the main moment of the force are equal to zero

$$\sum \vec{F_i} = 0 \tag{3}$$

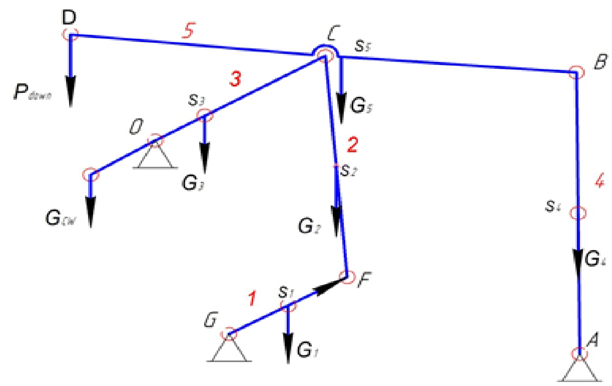


Figure 4: Power analysis of the six-link scissor mechanism conversion mechanism of pumping units

and

$$\sum \vec{M}_i = 0 \tag{4}$$

We consider the equilibrium of each link. G_5 – weight of the fifth link operates on the 5th link at the center of mass, as well as P load (the weight of the rod string and pumped liquids) in the suspension point of the D rod string, and reactive power R_{54} and R_{52}. Then the equilibrium equations of the 5th link

$$\begin{gathered} -R^x_{54} - R^x_{52} = 0 \\ -G_5 - R^y_{54} - R^y_{52} - P = 0 \\ G_5(X_{S5} - X_B) + R^y_{52}(X_C - X_B) + R^x_{52}(Y_B - Y_C) + P(X_D - X_B) = 0 \end{gathered} \tag{5}$$

Consider the equilibrium of the 4th link, the connecting rod. The connecting rod is affected at the center of mass of the connecting rod by G_4 – the weight of the connecting rod and R_{04}, R_{54} reaction forces. Then we make the equilibrium equations of the connecting rod – the 4th link, taking into account $R_{45} = -R_{54}$.

$$\begin{gathered} R^x_{04} + R^x_{54} = 0 \\ -G_4 + R^y_{04} + R^y_{54} = 0 \\ G_4(X_{S4} - X_A) + R^y_{54}(X_B - X_A) + R^x_{54}(Y_A - Y_B) = 0 \end{gathered} \tag{6}$$

Now consider the equilibrium of the 3rd link. The third link is affected at the point S_3 by G_3 force – the weight of the third link, at point by G_{CW} force – the weight of the counterweight and R_{03} and R_{32} reaction forces.

Then for the 3rd link

$$\begin{gathered} R^x_{04} - R^x_{32} = 0 \\ -G_3 + R^y_{03} - R^y_{32} - G_{CW} = 0 \\ G_3(X_{S3} - X_O) + R^y_{32}(X_C - X_O) + R^x_{32}(Y_O - Y_C) + G_{CW}(X_{CW} - X_O) = 0 \end{gathered} \tag{7}$$

Now consider the balance of the 2nd link, the connecting rod. The connecting rod is affected in the center of mass of the connecting rod by G_2 – the weight of the connecting rod and R_{21}, R_{32}, R_{52} reaction forces.

Considering that $R_{52} = -R_{25}$ and $R_{32} = -R_{23}$, we compose the equilibrium equations of the connecting rod - the 2nd link

$$\begin{gathered} -R_{21}^x + R_{32}^x + R_{52}^x = 0 \\ -G_2 - R_{21}^y + R_{32}^y + R_{52}^y = 0 \\ G_2(X_{S2} - X_F) + R_{32}^y(X_C - X_F) + R_{32}^x(Y_F - Y_C) + R_{52}^y(X_C - X_F) + R_{52}^x(Y_F - Y_C) = 0 \end{gathered} \tag{8}$$

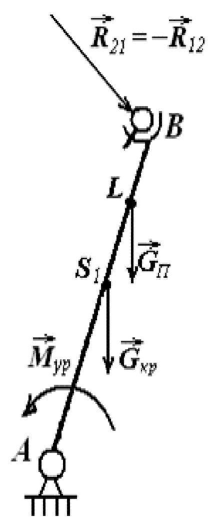


Figure 5: The equilibrium of the crank

The crank is affected at S_1 point in the center of mass of the crank by G_1 – the weight of the crank and $_{et}$ engine torque, also at F point R_{21} reaction and at G point reaction. We compose the equilibrium equations for the 1st link, where $R_{21} = -R_{12}$.

$$\begin{gathered} R_{01}^x + R_{21}^x = 0 \\ -G_1 + R_{01}^y + R_{21}^y = 0 \\ G_1(X_{S1} - X_A) + R_{21}^y(X_F - X_G) + R_{21}^x(Y_F - Y_G) + M_{et} = 0 \end{gathered} \tag{9}$$

By solving equilibrium equations simultaneously, we find unknown components of reactions and an unknown moment.

The purpose of the balancing task is to minimize input torque M on the crank shaft. To do this, one needs to properly pick up the mass counterbalance and the distance of the center counterweight from the axis of rotation. In the case of rotary balancing, the counterweight is set on the crank. And here the counterweight is set on the 3rd link.

The distance of the center of the mass counterweight from the axis of the rotation of the link is defined in the first approximation as [15]:

$$OL = k \cdot H_s \left(P_{up} + P_{down}\right)/4 \cdot F_{cw} \tag{10}$$

where H_s – is the length of the rod string, P_{up}, P_{down} are loads at the point of rod suspension at the motion up and down, F_{cw} is the total weight of counterweights, the correcting coefficient that is manually entered by the user until the two peaks of values - torque on the shaft of the crank will not be equal.

Let's use the well-known principle of possible movements

$$\sum \delta_i = 0 \quad or \quad \sum N_i = 0. \tag{11}$$

According to the principle of possible movements, power of these forces should be zero. Let's write this down for our problem:

$$F\hat{V}_{S_1} + \hat{F}_2\hat{V}_{S_2} + \hat{F}_3\hat{V}_{S_3} + \hat{F}_4\hat{V}_{S_4} + \hat{F}_5\hat{V}_{S_5} + F_{CW}\hat{V}_L + +\hat{F}_P\hat{V}_D + M_{GF} = 0 \tag{12}$$

Here, are the velocities of the corresponding points of gravity forces application;
ω_{GF} – is angular speed of the crank;
M – is the torque on the crank shaft.

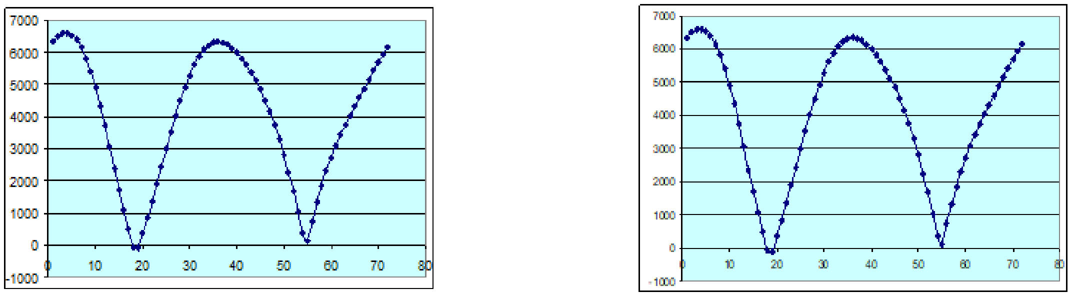
4 Discussion of experimental results

The results of the study of the balancing modes are presented in table 1. The last two columns show and values, which were found in such a way as to minimize torque on the crank shaft. In the calculation program, this is achieved by manually specifying correcting coefficient in formula (10).

It was also found that when the direction of the crank rotation is changed (counterclockwise), the moment on the crank shaft practically does not change, but the balancing mode is somewhat worse (the counterweights are removed from the axis of rotation).

Table 1 – Results of the study of balancing modes

Pumping mode				The maximum torque on the crank shaft	Weight of the counterweight with optimal balancing	Counterweight distance from the crank rotation axis
P_{up}	P_{down}	Angular velocity	The length of the crank			
kN	kN	rpm	mm	kNm	kg	mm
60	30	6,9	550	10,7	478	0,525m
60	40	4,3	1000	8,412	478	0,753m
60	40	6,9	1000	8,365	478	0,753m



A) Graph of the change in torque as a result of kinetostatic analysis

B) Graph of torque changes based on the principle of possible displacements

Figure 6: The Graphs of torque changes

5 Conclusion

According to the results of the kinetostatic analysis, a graph of the change in the torque is obtained. Figure (5) shows the graphs of the torque change obtained in various ways.

The results of the work carried out show that the goal of the study has been achieved. Since a detailed kinetostatic analysis confirms the possibility of using the studied six-link straight-line mechanism as a converting mechanism for the rod pumping drive.

1. A kinetostatic analysis was performed and a mathematical model of the kinetostatic analysis of the six-link conversion mechanism in the Maple environment was developed in order to test the performance of the new design.

2. Numerical results obtained by various methods confirm that the results are reliable.

6 Acknowledgments

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