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ABSTRACT

of the dissertation for the degree of Doctor of Philosophy

INVESTIGATION AND IMPROVEMENT OF DRIVE MECHANISMS OF MACHINES OPERATED WITH MECHANIZED EXPLORATION METHOD

Speciality: 3313.02 - Machines, equipment and processes

Field of science: Technical sciences

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GENERAL CHARACTERISTICS OF THE WORK

The actuality and study degree of the topic. Various types of oilfield equipment (stationary installations and machines) are widely used in the development and operation of oil and gas fields. The machines and equipment used in oil production are regularly improved and their rational use is increasingly ensured. Since there is a lot of demand for energy saving, the use of centrifugal and screw pumps from oil-field equipment and machinery has not found wide application. Currently, in the world, 70% of the oil produced per year by 5.5 million wells fund falls on the share of rod pumps. The number of wells operated by rod pumps in Azerbaijan exceeds 4000.

Efficient exploitation of the beam pumping system of wells operated with rod pumps can ensure operating coefficient to be at the required level for a long time and the overall production performance stable. In the beam pumping system, elements often causing troubles are considered engine-reducer (first contour) and reducer-beam-pumping unit (second contour).

From this point of view, on the basis of new theoretical-experimental studies improvement of contours of engine-reducer and beam-pumping of drive system of the wells operating by rod pump has been considered an actual issue. Since such an approach is scientifically important, it will always attract the attention of researchers. However, the analysis of technical literature shows that the problem involved in the study of dissertation has not been investigated by other researchers in recent years.

The dissertation work has been carried out with the aim of conducting research in the mentioned direction.

The object and subject of the study. The object of research – Contour elements of drive mechanisms of machines operated by a mechanized method of work; the subject of research – Technological optimization of operational and power characteristics of machines operated by a mechanized method of work, based on the study of contour elements of drive mechanisms.

The purpose and main objectives of the study.

- the aim of the study is to create new efficient bases of drive mechanism by considering the drive mechanism as the associated engine-reducer and reducer-beam-pumping unit contours system and optimizing the elements in the contour composition in terms of design and use in order to achieve the efficient work of the machines operated by the rod pump method.

The following developments have been carried out in the research work:

-Technological optimization of operational and power characteristics of the chain of drive mechanisms of machines;

-Creation of appropriate effective structural clusters, assuming that the drive mechanism of the beam-pumping unit consists of connected contours of the engine-reducer and reducer- beam-pumping unit;

-Determination of the criteria for the similarity of the belt drive using the theory of similarity and dimensions and according to these criteria development of the optimal design method based on computer simulation of the belt drive;

-Solving the problem of optimizing the replacement of used reducers with a two-stage chain reducer.

Research methods. Research methods and theories used: systematic analysis, fuzzy logic, theories of similarity and dimension, mathematical methods related to individual sections of mathematics.

Basic provisions for defense:

1.Assessment of operational quality based on the study of various combinations of engine-reducer contour (1-contour) structural clusters of drive mechanisms of a machines operated by the rod pump method;

2.Determination of the kinematic and dynamic modes of action required for the study of the kinematic and dynamic models of the reducer-beam-pumping unit (2-contour) of the drive mechanisms chain;

3.Creation of constructive cluster assembly, which compensates for the required structural, kinematic and dynamic characteristics of the reducer-beam-pumping unit of drive mechanisms.

Scientific novelty of the research:

1.With the relevant constructive clusters created by examining the engine-reducer and reducer-beam-pumping unit contoures of drive mechanism, the optimal constructive components have been determined.

2.Using the theory of similarity and dimensions, the similarity criteria of the belt transmission have been determined, and according to these criteria the practical design method of the belt transmission based on computer simulation and allowing to obtain the optimal power consumption has been developed;

3.The angle between the straight and the horizontal lines connecting the engine axis and the axis of the reducer valve has been found intervals.

4.As a result of the solution of the optimization problem by substantiating the application of the two-stage chain reducer, possibility of replacing of the three-stage reducer with a two-stage chain reducer has been shown.

Theoretical and practical significance of research. The theoretical and practical base of the results obtained in the dissertation is the development of methodological bases for the improvement of the constructive chain of the drive mechanisms, justification of the range of parameters to determine the effectiveness of the functioning of the contours, and the creation of new constructive clusters. The practical significance of the work is determined by the proposals made to improve the engine-reducer and reducer-beam-pumping unit contours of the drives of the machines in order to increase the technical efficiency of the mechanized operation machines.

Theoretical and practical significance of the work can be set by the results of the study on the development of new optimal elements of drive contours obtained applying a research method that can be used in carrying out similar studies, as well as the result of software analysis based on visual computer simulation.

The results obtained in the dissertation work can also be used to optimize the energy indicators of the operation of oil-field equipment in the oil extraction process.

Approbation and application. The main provisions of the dissertation have been presented at the following conferences:

1.V International Scientific and Practical Conference “Theory and Practice of Modern Science” (Moscow, 2012);

2.XVII Republican Scientific Conference of doctoral students and young researchers (Baku, 2013);

3.The 1st International Scientific Conference of young researchers dedicated to the 90th anniversary of the national leader of the Azerbaijani people Heydar Aliyev (Baku, Gafgaz University, 2013);

4.Scientific Practical Conference “Azerbaijan-2020: prospects for the development of oil and gas industry” dedicated to the 90th anniversary of national leader of the Azerbaijani people Heydar Aliyev (Baku, Azerbaijan State Oil Academy, 2013);

5.II International Scientific and Practical Conference "Bulatov Readings" (Krasnodar, 2018);

6.10^{th} International Conference on Theory and Application of Soft Computing. Computing with Worlds and Perceptions (ICSCW-2019).

The results of the work have also been discussed at the general meetings of the Department of Mechanics of ASOIU and included in the department's annual research reports.

The results of the work have been confirmed by the relevant acts of field testing in the wells 3073, 3207 and 3262 of “Bibiheybatneft” OGPD.

16 scientific works on the topic of the dissertation, including 10 articles (4 of them abroad) and 6 Republican and international conference materials and theses have been published.

Name of the organization where the dissertation is performed: Azerbaijan State Oil and Industry University.

The total volume of the dissertation with the indication of the separate volume of the structural units of the dissertation: The dissertation work consists of 126 titles, including introduction, 4 chapters, conclusion, 7 internet sites, literature list and appendix. The volume of the work is 153 pages, including 15 figures, 11 diagrams, 15 tables. The volume of dissertation is 231840 symbols.

CONTENT OF WORK

In the introduction the subject actuality has been interpreted, the purpose of the work, the protected provisions, the scientific novelty of the results, the practical importance of the work, information on the approbation and application of the results have been given.

In recent years, research on the drive of beam-pumping unit used in methods mechanized operation has been weakening. The research work on the drive is carried out in connection with the process from belt transmission to chain transmission. Various variants of chain transmission have been tested in field conditions, and the transition from belt transmission to chain one has been determined to be technically and technologically efficient and energy efficient.

In chapter one, one of the factors causing the deterioration of the technical and economic performance of machines operated by mechanized exploration method is related to the work of the drive mechanisms of these machines consisting of two independent contours has been noted. Each unit of the constructive chain of drive mechanisms is based on the efficiency criterion of the device separately, the management of technological parameters of mechanized operation machines is characterized as a major theoretical and practical direction of research and the construction of machines operated with mechanized exploration method and their related problems have been investigated, as well as the directions and prospects of improvement of the engine-reducer part of the constructive chain of drive and the contour of the reducer-beam pumping unit have been commented [42][121].

It is noted that the beam pumping unit with a balance is distinguished from the usual structures of the balanced beam pumping unit produced in foreign countries according to the following indicators:

- exclusion of the negative effects of errors during the

[12] Calculation methodics of beam-pumping units. Gasanov R.A., Karimova I.M.

preparation and installation of the inverter;

- ensurance of the most profitable transmission angle in any position of crank-shaft (β=90°);
- reduction of K_F form coefficient of torsion moment curve;
- reduction of inverter refusal;
- reduction of metal capacity and requiring a lot of effort during preparation;
- decrease in the volume of the unit base.

It has been found that when the circle center of the back arc of the beam balance is located at different points, various kinematic schemes of the conversion mechanism of beam pumping unit are obtained, and at the hanging points of the sucker-rods the K_1/K ratio is less than when the center of the front and back arcs of the beam balance coincides with the center of its swing. This reduces the forces acting on the joints, the torque of the reducer in the shaft, the shape coefficient of the torsion moment curve, and increases the longevity of the unit.

In this chapter, as well as the necessary notes on the implementation of the issues improving the constructive chain of drive machines operated by mechanized exploration method have been reflected[5,6][7][9].

Chapter two is devoted to the interpretation of the results of the research on improving drive mechanisms of a machines operated with mechanized exploration method. In this section, proposals allowing to improve drive mechanisms have been developed to justify the kinematic scheme, metric dimensions, kinematic and dynamic characteristics of the conversion mechanism of the contour of reducer – beam – pumping unit.

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[5] About the expediency of the study of belt transmission in beam pumping unit. Karimova I.M.

[6] About the possibility of the application of gear belt transmission in beam pumping unit drives. Karimova I.M.

[9] Choice of rational type of electric engine for beam pumping unit drive. Karimova I.M.

In the chapter based on the analysis of dimensional and dimensionless parameters, the following functional dependence has been identified to ensure efficient operation of belt transmission:

$$f(\delta_{A1}, V_{A1}, P_{A1}, P_{B1}, g, R_1, R_2, EF, f, \alpha, \varphi) = 0$$

For the first time, the α angle formed by a horizontal line with a straight one connecting the center of the pulley of the engine and the center of the pulley of the reducer has been included in this function.

In this function there are 8 mechanical (force, length and time) size quantity, 3 mechanical sizes quantity. According to the π - theorem, from these 8 parameters, we can choose three, for example, P_{B1}, R_2 and V_{A1}, so that power, length and unit of time can be obtained from their combination. Then, $[P_{B1}] = N$, $[R_2] = m$, $\left[\frac{R_2}{V_{A1}}\right] = t$ is obtained.

Based on the π - theorem, the number of undefined parameters will be 8-3=5. Dimensionless parameters will be:

$$\Pi_1 = \frac{\delta_{A_1} V_{A1}^2}{EF}; \ \Pi_2 = \frac{P_{A1}}{EF}; \ \Pi_3 = \frac{P_{B1}}{P_{A1}}; \Pi_4 = \frac{g\delta_{A1} R_2}{EF};$$

$$\Pi_5 = \frac{gR_2}{V_{A1}^2}; \ \Pi_6 = f; \ \Pi_7 = \varphi; \ \Pi_8 = \alpha .$$

The mechanical essence of the criteria Π_1, Π_2, Π_3... Π_8 is as follows:

- Π_1 –belt inertia criterion – determines the inertia force of the belt compared with the hardness of the cross-section;

- Π_2 – belt tension criterion – compares the stiffness of the cross section of the tension force on the tip of the pulley;

- Π_3 – belt forces criterion – characterizes the ratio of forces generated in the cross-sections of the belt arc on the pulley;

- Π_4 – single weight criterion of the belt – gives a comparison of the uniform weight force of the belt with the stiffness of the cross-section;

- $П_5$ – utility criterion of the belt drive – gives a comparison of the belt weight force per unit length with the belt inertia force;

- $П_6, П_7, П_8$ – similarity criteria characterizing friction, angle of friction and the location of the centers of the cranks.

Therefore, the functionality that provides belt drive control can be as follows:

$$f(П_1, П_2, П_3, П_4, П_5, П_6, П_7, П_8) = 0$$

Since $П_1 \cdot П_5 = П_4$, then the defining parameters will be; $П_1$, $П_2$, $П_3$, $П_4$, $П_5$, $П_6$, $П_7$, $П_8$.

An appropriate algorithm for computer simulation has been developed based on the corrected analytical dependence of the angle α on the indicators, which takes into account the criteria for determining the tensile strength of the belt.

The angle α formed by the horizontal line with the line connecting the engine axis and the axis of the driving shaft to the reducer, that is, the choice of the position of the line mentioned above, the balance of forces between the engine-reducer contour of the belt, the forces of harmful resistance brought to the shaft, etc.

The results of the calculations based on the developed algorithm are given in Graph 1 as an example.

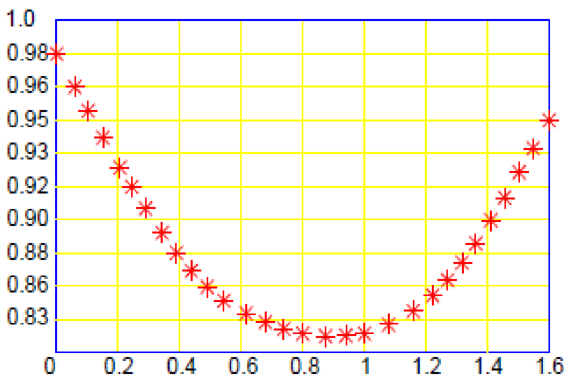

Diagram 1. Dependence of power consumption (resistance forces brought to the driven shaft) from the angle α

The optimal calculation result shown in Graph 1 is that the angle between the axis of the engine and the horizontal line connecting the axis of the shaft of the reducer has been established within 45-57 º.

Number of engine cycles, number of transmission, distance between the pulley axles, accuracy of the elements preparation, change limit of forces affecting the belt, tension of the belts, etc. the presence of factors influence has been adopted in the technical literature, but the influence of the angle α has not been taken into account.

The *XOY* coordinate system is selected so that the abscess to be passed through the O_1 and O_2 centers of the driven and driving pulleys of a engine, and the ordinate axis passes through the contact point "*O*" with the center of the master pulley of the *X* axis and the sliding angle φ_T contains part of the full coverage angle. The flexible joint in this angle gets additional deformation and some change along the curve of the coverage angle (Figure 1) and it is accepted that for the normal operation of the belt drive, the pulley should not exceed the sliding limit (φ_T') at which the belt should slide.

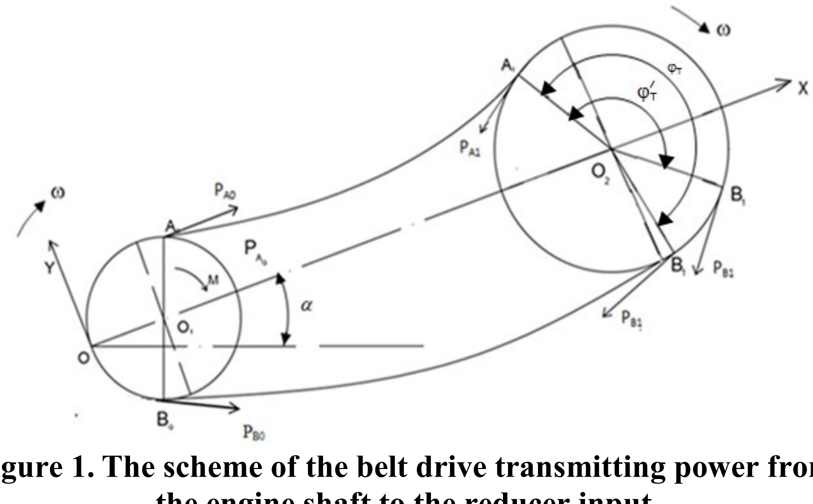

Figure 1. The scheme of the belt drive transmitting power from the engine shaft to the reducer input

In this chapter the task of determining the tension of a flexible belt, taking into account the inertia forces at the point of coverage of the driven and driving shaft through the flexible belt has also been studied. For this purpose, differential equation for tension force *P* of flexible belt has been obtained and tension force *P* was determined depending on the angle φ at the point of coverage of the driven pulley. As a result, dependence of flexible belt tension on the slope angle φ_T is defined [10][8]:

$$A(1-\frac{P_{A1}}{EF})(P_{B1}-P_{A1})+\frac{A}{2EF}(P_{B1}^2-P_{A1}^2)-\frac{SG}{f}(e^{f\varphi_T}-1)-\frac{ST}{2f}(e^{2f\varphi_T}-1)-$$
$$-fg\delta_{A1}R_2[\cos(\varphi_T+\alpha]-\cos\alpha]-g\delta_{A1}R_2[\sin(\varphi_T+\alpha]-\sin\alpha]=0$$

where $A=1-\dfrac{\delta_{A_1}V_{A1}^2}{EF}$, $B=\delta_{A1}V_{A1}^2\left(1-\dfrac{P_{A1}}{EF}\right)$,

$$S=f(P_{A1}A-B),\ G=1-\frac{1}{EF}\left(P_{A_1}-\frac{B}{A}\right),\ T=\frac{1}{EF}\left(P_{A_1}-\frac{B}{A}\right),$$

P_{A_1} and P_{B_1} are respectively, the tension forces of the belt at points A_1 and B_1, which are the beginning and end of the coverage curve of the pulley;

E – the elastic modulus of the belt;

F – the cross-sectional area of the belt;

f – friction coefficient between the belt and the pulley that touches it;

δ_{A_1} – mass coefficient of the belt length at the point A_1;

R_2 – radius of the pulley;

α – the angle formed by the line connecting the centers of the driven and driving pulleys with horizon;

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8 Mechanical parameters and similarity criteria of belt transmission. Gurbanov R.S., Bakhshaliyev V.I., Karimova I.M.

V_{A_1} – the speed of the point A_1.

Using the obtained relationship, calculations of certain values of the parameters have been made for the initial tension force of the driven belt arm and the elastic modulus of a specific transmission at different values.

The regression equation has been obtained for dependence curves of the tension ratio of the belt arms (λ_p), depending on the coverage angle φ_T at different values of the initial tension force [113][14]:

$$\lambda_p = 1 + a_1\lambda_\varphi + a_2\lambda_\varphi^2 + a_3\lambda_\varphi^3,$$

where λ_φ –coverage angle of the tension of the pulley arms $\lambda_\varphi = \varphi_T / \varphi'_T$; $a_1 = 0{,}2294\pi$; $a_2 = 0{,}0538\pi^2$; $a_3 = -0{,}003335\pi^3$.

From the analysis of the curves (diagram 2) on the dependence expressed by this equation, it becomes clear that with the increase in the angle of coverage λ_φ of the tension of the belt arms λ_p increases.

We come to such an experimental conclusion that the tension criterion for the normal operation of the flexible connection should be less than 1,986, i.e. $\lambda_p \le 1{,}986$. In addition, it appears from these curves that for smaller values the elasticity module the belt loses its ability. For example, if $E = 5 \cdot 10^7\ N/m^2$, $\varphi_T > \frac{3\pi}{4}$, then an empty rotation of the belt occurs.

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14 On Possibility of Using a Toothed Belt Drive in a Drive to Rocker-Machines. Karimova I.M.

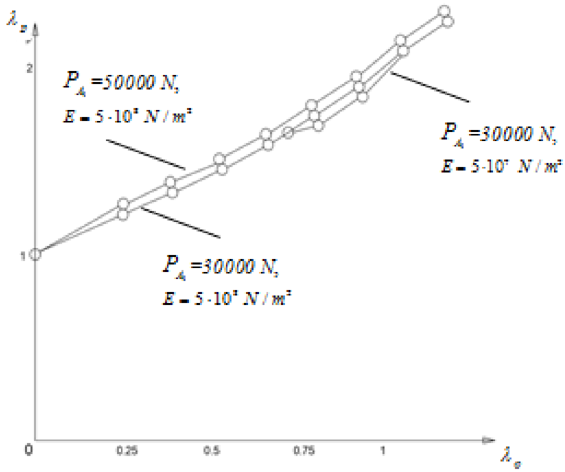

Diagram 2. Dependence curves of λ_p on the coverage angle $\lambda_\varphi = \varphi_T / \varphi'_T$.

In this chapter also the reducer loading in the engine-reducer contour of the drive mechanisms also has been investigated.

On the basis of the theoretical and practical analysis, the optimal options of the structures of the belt drive in the 1st contour of the drive mechanisms have been selected and the accepted criteria (goal function) have been developed and the parameters of the construction have been determined.

However, the traction capacity, longevity, instability and limitation of the transmission number of the belt drive do not fully meet the demand for the optimal construction of drive mechanism of the beam pumping unit[4].

To meet these conditions, the possibility of the use of various types of (flat, multiwedge and gear) belts have been analyzed. Gear belts are considered to be perspective type of flexible connection due to their large traction capacity, efficiency factor and stability of the

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[4] Choice of rational type of the belt used in beam pumping unit drive. Karimova I.M.

transmission number. This type transmissions work without lubrication, have the ability to synchronize the motion of the inlet and outlet units through they are subjected to the influence of abrasive and polluted environment, their operation is simple, and therefore the application of these elements in drive mechanisms can be considered as the basis for reducing metal grip. According to the passport data of this type belt drive mechanisms, the guaranteed working life is higher than 2000 hours (<600 hours in other types of belts). A comparative analysis of the applied belt drive mechanisms that the latter are superior to other belts due to their value, production and operation-technological characteristics, and the application of this type of flexible transmissions in their drive is competitive.

The dissertation also proposes to equalize the load of the teeth of the gear wheel on the drive shaft of the reducer of the 1st contour of the drive mechanisms, varying depending on its position from the values of “0” at the “dead” points of the beam pumping unit in one cycle of the deep pumping unit to the maximum value by providing periodic changes in the drive shaft of the slider for rotating moment transmission (Graph 3) [67, 80][7,3]:

$$T_{B_1} = r\frac{F'_m}{2}\sin\frac{\varphi}{z}; \;\; T_{H_1} = -r\frac{F'_j}{2}\sin\frac{\varphi}{z}.$$

where z is the number of teeth in the slave gear wheel of the reducer (for the reducer of the beam pumping unit, based on SS 5866-66 is z=85); $\frac{\varphi}{z}$ – the rotation angle of the driven shaft tooth.

On the basis of the studies, it has been determined that Novikov-mounted two- and three-stage cylindrical reducers used to transfer the rotating moment from the electric engine to the beam balance of the beam pumping unit equivalent touching and normal tension in the teeth of the gear wheel is the function of the rotating moment and decreases by 20% depending on the relative rotation angle (φ').

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[7] Reduce of gear load of the reducer wheel of beam-pumping unit. Karimova I.M.

[3] Increase of longevility of beam-pumping unit. Mamedzade O.A., Karimova I.M.

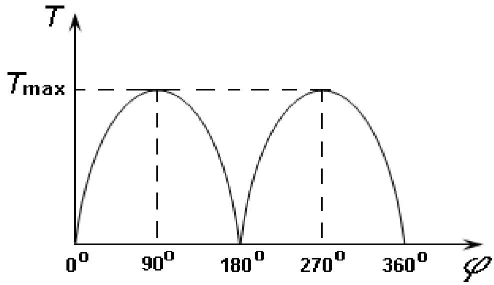

Diagram 3. Change of rotating moment formed in the shaft of the reducer

In this chapter, the task of optimization of the currently used three-stage cylindrical gear wheel reducer has been considered, since the use of a chain of drive mechanisms in the engine-reducer contour has been thought promising.

As a goal function, the minimum value of the contact tension of the gear wheel of the reducers has been adopted. A minimum value is required to meet this condition.

Based on the problem solution, the following expressions have been found for the slow, intermediate and sharp transmission numbers of the three-stage cylindrical reducer [9][1]:

In this chapter, simultaneously, the kinematic scheme of the reverse mechanism of the 2nd (reducer–beam pumping unit) contours of the chain of the drive mechanism has been substantiated, the kinematic and dynamic characteristics of the hanging point have been investigated.

The problem of determination the dynamic characteristics of the chain of the drive mechanism of the reducer-beam-pumping unit contour has been solved. It has been determined that after initial deformation of the rods and pipes, with an increase in the speed and momentum of the hanging point of the rods on the ascending and descending strokes, the additional inertial load increases and

[1] Application of three-stage gear reducers in beam-pumping unit and optimization of the construction. Gurbanov H.Y., Karimova I.M., Allahverdiyeva A.T.

accordingly, the maximum load on the hanging point points increases, and the minimum load decreases.

In the third chapter, the methods of determining the shape, mass and metric characteristics of the executive bodies of both contours in the chain and assessing the operational reliability of both contours, the selection of the optimal variant of belt drive and the determination of the optimum dimensions of the drive mechanism in the reducer-beam-pumping unit contours have been developed. In order to select the optimal parameter variant of the belt drive, which is intended to be applied in the first contour of the chain of a drive mechanisms, based on linear programming requires a minimum of transmission running time (*t*), a minimum of transmission cost (*Q*), i.e. [5][2]:

$$\left.\begin{aligned} t &\to max \\ Q &\to min \end{aligned}\right\}$$

If we replace the two goal functions with one function, then:

$$g_0 = \frac{Q}{t} \to min$$

or
$$g_0 = \frac{(a_i z + Q_2 + Q_3 + Q_4 + Q_5)\sigma_{max}^m \cdot 3600 \cdot U \cdot z_2 \cdot v_2}{\sigma_N^m \cdot N_0 \cdot v_1} \to min ,$$

where $a_i z = Q_1 -$ the cost of belts;

Q_2 – the value of pulleys;

Q_3 – the value of the electric engine;

Q_4 – the value of the tension forming screw device;

Q_5 – the value of the spent electricity,

z – the number of wedge-formed belts in the transmission;

a_i – the value of one belt.

Then: $Q_2 = Q_{2,1} + Q_{2,2}$, $Q_{2,1} = A_1 \xi_1 \rho \frac{\pi D_1^2}{4} \cdot \left(2S + \frac{cz}{2}\right)$ is the value of the driving pulley, $Q_{2,2} = A_2 \xi_2 \rho \frac{\pi D_2^2}{4} \cdot \left(2S + \frac{cz}{2}\right)$ is the value of the

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[2] Choice of the optimal version of belt transmission of beam pumping unit. Karimov Z.H., Mammadzade O. A., Karimova I.M.

driven pulley.

A_1 and A_2 are the values of the unit weight of the driving and driven pulleys accordingly; ξ_1 and ξ_2 – the coefficients that consider the gaps in the driving and driven pulleys; ρ – the density of the material of the pulley; D_1 and D_2 – the diameters of the driving and driven pulleys. (The values of *S* and *C* are given in tables).

The value of electricity $Q_5 = q \cdot P \cdot t.$

Here, *q* is the price per kilowatts of electricity;

P – nominal power of the engine – kilowatt hours;

t – the working time of the belt drive, hours.

The following restrictions are imposed on the parameters of drive:

1.Number of runs on the belts:

$$U = \frac{l}{v} \leq [U]$$

where l – the length of the pulley;

v – speed of the belt;

$[U]$ – the allowed number of runs of the belt.

2.A transmission number of belts must be $z \leq [z]$, where $[z]$– the allowed number of belts.

3.Transmission number must be $u \leq [u]$, where $[u]$ – the allowed value of the transmission number.

4.Coverage angle must be:

$$\alpha_0 = 180^0 - 57{,}3^0 \cdot \frac{D_2 - D_1}{a} \geq [\alpha_0] \text{ ,}$$

where [α_0]- the allowed value of the coverage angle;

a - the distance between the centers of the pulleys.

$$2(D_1 + D_2) \geq a \geq 0{,}55(D_1 + D_2) + h$$

where h- the height of the cross-sectional area of the belt.

The sequence of selection of the optimal variant of the wedge-formed belt drive with the power on the input shaft P_1 (kvt), the angular velocity and the transmission number has been developed.

In this chapter the results of determining the optimal ratio of the rotating mechanism units in the 2nd (reducer-beam-pumping unit) outline of the chain of drive mechanisms.

The following parameters are taken into account for the optimal synthesis of the rotary mechanism of the beam balanced pumping unit: as the input parameters S_0 – the nominal length of the strokes of wellhead stock and n – the number of strokes of the hanging points of rods; as the output parameters – r/p, r/e, δ_{max} and γ_0.

The following is accepted to

characterize the column of rod pumps:-reduced σ_{gt} tension criterion at the hanging point of the rods; -for convers

ion mechanism is the maintenance and release of wellhead during lifting and lowering operations;

-maximum torsion moment T_{max} and torsion moment curve coefficient K_F in the driven shaft of the reducer.

In general, it should be taken into account that machines with high r/p, $K_1/(e+R)$ values should be considered more efficient from the point of view of reduction of bulk sizes and weight of the conversion mechanism, consequently, mounting and operating (K_1 is the length of the forearm of the beam balance).

Knowing the radius r=1,2 m of the crank shaft, the maximum stroke of the wellhead piston rod S_0=3,5 m, while the centers of the arches of the front and the back should coincide at one point, we get the following:

$$\frac{K_1}{K} = \frac{S_0}{2r} = 1{,}46\,,$$

where K is length of the back arm of beam balance.

In order to ensure the safe and comfortable work of the worker in the wellhead, during well lifting and lowering operations, when opening of rods from the tubings, the distance from the axis of the wellhead to the base of the foundation should be l=1200 mm. The nominal distance from the axis of the beam to the base of the foundation is constructively l_1=475 mm. Then the minimum length of the front arm of the beam balance is:

$$K_1 = l + l_1 = 1200 + 475 = 1675\,mm.$$

Maximum amplitude angle of beam balance is:

$$\delta_{\max} = \frac{S_0}{K_1} \cdot \frac{180}{\pi} = \frac{3{,}3}{1{,}675} \cdot \frac{180}{3{,}14} = 120°,$$

length of the back arm is:

$$K = \frac{K_1}{1{,}46} = \frac{1{,}68}{1{,}46} = 1{,}15 \text{ m},$$

where K_1-K=1,675-1,15=0,525>0,5 m, that is, the requirements of the safety technique are met.

By placing the center of the back arc arm at different points in the swing plane of the beam balance, the radius of the back arc of the balance can be determined by the following formula:

$$R = 2r/(tg \arccos \frac{R}{A_1} + \delta_{\max} + \arccos \frac{p - e\cos\gamma_0}{A_1} -$$

$$- \arccos \frac{R}{A_1} - tg \arccos \frac{R}{A_2} - \arccos \frac{p - e\cos\gamma_1}{A_2} + \arccos \frac{R}{A_2})$$

$$A_1 = \sqrt{p^2 + e^2 - 2pe\cos\gamma_0}$$

$$A_2 = \sqrt{p^2 + e^2 - 2pe\cos\gamma_1}$$

$$\gamma_1 = \gamma_0 - \delta_{\max}$$

The results of the calculations show that the radius of the back arc arm decreases as the center of the circle of the back arm of balance beam increases sliding from its swing axis. This can reduce the durability of the wing, which connects the traverse with the balance beam. The conducted production experiments showed that during the reduction of the arc radius, the regularity and fluidity of the working surface of the back arc of the balance deteriorates, which can reduce the durability of the wing. According to the results of the production experiments, the arc radius R=900 mm is acceptable for beam №24.

When limitations for the radius of the circle are R>900 mm and difference between the arms is K_1-K≥525 mm, during releasing the wellhead, the studies showed that the defined area of the center of the circle of the back arm is as follows: e=0÷300 mm and γ_0>108°. As a result of calculations for the randomly taken values of the output parameters (e and γ_0), it was determined that the values of the reduced tension in the hanging point of the rods almost do not change, and the values of the form coefficient of the torsion moment curve differ from each other, so we can assume K_F as a purpose function.

At the same time, for several variants of e and γ_0 parameters, the motion characteristics of the rods hanging point have been studied.

The analysis revealed that as the back arc moved away from the center axis, the velocity and urgency of the rods decrease during the upward movement of the rods, the downward movement increases and the ratio $\frac{K_1}{K}$ of the belts to the maximum load at the hanging point decreases; as the starting position of the beam balance increases with respect to the polar axis, its velocity and urgency increase as the pole axis moves upward, while its downward movement decreases and the ratio $\frac{K_1}{K}$ of the belts to the maximum load at the hanging point increases.

When the center of the back arc of the beam balance coincides with its center of gravity, the ratio $\frac{K_1}{K}$ is 1,46. At $r/e = 4$ and $\gamma_0 =$ 110°, the maximum load at the point of the beam balanced pump $\frac{K_1}{K}$ ration will be 1,38÷1,39, this will reduce the forces acting on joints and in places of connection. This will also lead to a significant reduction in the energy consumed by the units and an increase in its working time.

In the fourth chapter, the task of justification and optimization of the use of a two-stage chain reducer has been solved [8][10,11]. The mathematical model of the problem of the two-stage

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10 Study of the influence of centrifugal forces of chain transmission rods. Karimova I.M.

chain reducer parameters has been developed.

The mathematical implication of the problem of optimization of the two-stage chain reducer parameters is described as follows:

$$g_0 = \left\{ \frac{\pi \overline{K} e_1 \rho}{4} \left\{ ib_1 \left[\left(\frac{p_1}{\sin \frac{180^0}{z_1}} \right)^2 + \left(\frac{p_1}{\sin \frac{180^0}{z_2}} \right)^2 \right] + ib_2 \left[\left(\frac{p_2}{\sin \frac{180^0}{z_3}} \right)^2 + \left(\frac{p_2}{\sin \frac{180^0}{z_4}} \right)^2 \right] \right\} + \right.$$

$$\left. + p_1 \frac{z_1 + z_2}{2} + 80 p_1 + \frac{p_1}{40} \left(\frac{z_2 - z_1}{2\pi} \right)^2 + p_1 \frac{z_1 + z_2}{2} + 80 p_2 + \frac{p_2}{40} \left(\frac{z_4 - z_3}{2\pi} \right)^2 \right\} \rightarrow min$$

$$g_1 = \frac{20 i_1 e^{0,051 p_1}}{2K_e T_1 \sin \beta_1 + 40 k_f q_1 i_1 p_1^2 (0,385 p_1 - 7,98) + i_1 (0,385 p_1 - 7,98) \cdot \left[\frac{\pi n_1 p_1}{30 \sin \beta_1} \right]^2} > 1$$

$$g_2 = \frac{20 i_2 e^{0,051 p_2}}{2K_e T_e \sin \beta_3 + 40 k_f q_2 i_2 p_2^2 (0,385 p_2 - 7,98) + i_2 (0,385 p_2 - 7,98) \cdot \left[\frac{\pi n_3 p_2}{30 \sin \beta_3} \right]^2} > 1$$

$$g_3 = \frac{2,85}{p_1} \cdot \sqrt[3]{\frac{T_1 K_e}{(31 - 2u_1) K_1 p_1}} > 1$$

$$g_4 = \frac{2,85}{p_2} \cdot \sqrt[3]{\frac{T_3 K_e}{(31 - 2u_2) K_1 p_2}} > 1$$

$$g_5 = \frac{(31 - z_1^2) \cdot (31 - z_3^2)}{4 z_1 z_3} \geq 1$$

where g_0 is a function of determining the minimum cost of two-stage chain transfer;

g_1, g_2, g_3, g_4 and g_5 are coefficients considering

[11] Analysis of the calculation of kinematic scheme of conic planetary transmission. Aliyev S.Ya., Karimova I.M.

strength, reliability, wear resistance of joints and kinetics of chain transmission;

$\overline{K}$ – costs of chain transfer stars;

e_1 – coefficient of star size overflow, $e_1 = 0,7 \div 0,8$;

ρ – the densities of the star material;

p_1, p_2 – stages of the chain drive of the first and second stages;

b_1, b_2 – width of single-rank and p_1 and p_2 stages chains;

z_1, z_3 – number of the teeth of the driving stars to the first and third stage;

z_2, z_4 – number of the teeth of driven stars to the second and fourth stage;

i_1, i_2 – chain order number of first and second stages;

K_e – the coefficient, dependent on the line, the chain adjustment method, the lubrication character, the dynamic loading, the chain length along the horizontal distance of the center;

T_1, T_3 – torsion moments in the belts leading to the first and third stage;

β_1, β_3 –parameters that characterize the angle stage of stars;

q_1, q_2 – weight of 1m of the first and second stage chains drive.

The load on the chain twigs varies widely depending on the cours speed of the chain and the pre-twisting of the branches. Our research has revealed that:

-Centrifugal forces in chain branches do not balance and impact the stars, mainly their rod;

-The value of the force (load) on the star support depends on the previous tension of the branches and the speed of the chain.

-On more tightly-held branches, centrifugal forces release the star's support and shaft from the load.

As an example of a two-stage chain transmission construction, the chain transmission box of the Model 65 Winch mobile hoist

Assembly of Wilson Products has been considered. The results of chain optimization are shown in Table.

In chapter four the possibility of application of wedge-shaped belt transmission in the contour of the drive mechanism chain, the tasks of evaluation of operational and quality indicators of the relevant transmission, calculation of mechanical losses and tension in the field of flexible transmission element pulley coverage of the wedge-shaped belt transmission in the contour of the constructive chain of the drive mechanism have also been investigated [6][13].

Table

Parameters	Existing construction		Optimal construction	
	I stage	II stage	I stage	II stage
Number of teeth of driving stars, z_1, z_3	24	30	24	30
Number of teeth of driven stars, z_2, z_4	52	58	52	58
Stage of chain of transmission, p mm	38,1	38,1	25,4	31,75
Number of chain queues, i	3	3	6	4
Distance between centers, a mm	1200	1800	1200	1800
The length of the chain, L mm	3,8	5,3	3,4	5,0
Diameters of driving stars, d_1, d_3 mm	291,9	364,5	194	303,7
Diameters of driven stars, d_2, d_4 mm	631	703,7	420,7	586,4
Theoretical weights of the stars, Q kg	2630	3420	1460	2450
Chain transfer values, AZN	636	850	460	700

[13] Study of transmission system of two multi-wedge belts. Karimova I.M.

Price of chain drive reducer, AZN	1485	1157
The weight of the chain drive reducer, kg	775	532

The calculation of tension and mechanical losses in the field of flexible transmission element gaskets coverage of the wedge belt transmission in the constructive chain contour of the transmission belt has been carried out considering the weight and the rigidity of the wedge-shape belt, which is of practical importance, especially at high transmission speeds.

The part of the driving and driven pulleys and the bent flexible belt are shown in figure 2. The abscissa of the selected coordinate system *xOy* passes through the centers of these pulleys, and the ordinate axis *y* passes through the point of intersection of the pulley leading to the seal with axis *x*.

Let's separate the $C'C''$ element with the length of the flexible belt ds and look at the forces that impact this element. Here, it has been pointed out: $d\varphi$ is center angle of element $C'C''$; φ_s' − sliding cover angle; φ_c − full angle of belt coverage of the pulley; α − the angle at which the axis of the abscess is formed by the horizon. The following forces will impact the viewed element of the belt with it on the shell of the pulley: *P* is the tension force of the left thrown part of the belt and directed on touching the point C' in the circle of the pulley; $P+dP$ -the right side of the belt is the tensile force of the thrown part and aimed at touching the circle of the

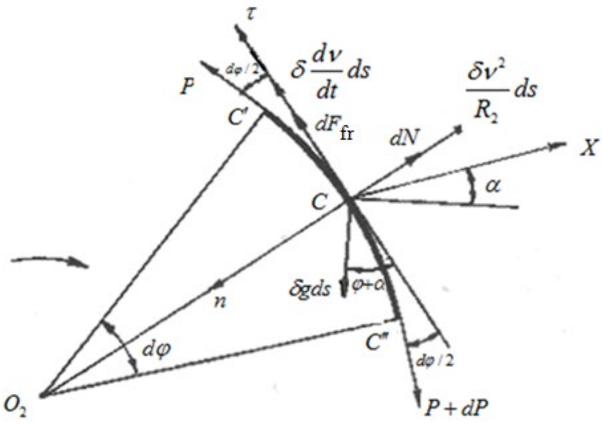

Figure 2. Flexible belt element in the cover area of the pulley

pulley at the C'' point; $dF_n^p = \frac{\delta v^2}{R_2} ds$ is the centrifugal inertia force of the element ds and applied in the center of the element at the point C in the direction of the head norm of the circle of the pulley (δ − mass of the unit length of the belt; v − speed of point C; R_2 − radius of the carrying out pulley); $dF_\tau = \delta \frac{dv}{dt} ds$ is the touch force of inertia of the element and focuses on touching at the point C of the circle; $dG = \delta g ds$ − weight force of the belt element ds; dN − reaction force per viewed element $C'C''$ of the belt; $dF_{fr} = fdN$ − the force of friction between the belt element ds and the touching surface of the pulley is directed opposite the direction of sliding on the pulley itself, which touches the circle of the pulley; f − it is a friction coefficient between the belt and the surface of the pulley touched by it [68][15][66][16].

It is accepted that according to the principle of Dalamber, the forces acting on the element of the belt are formed by balanced forces.

Mathematical relation has been obtained for determination of the tensile strength of the flexible belt depending on the sliding angle in the given section of the belt cover arc.

In this chapter, the task of determining the force spent on the removal of the friction force caused by the sliding of the belt elements on this surface between the belt and the contact surface has been resolved. To determine the strength lost in the sliding process on the surface of the flexible belt pulley, we take the following formula:

$$
\begin{aligned}
N_{fr} = {} & \frac{aV_A}{2EF}\left(P_B - P_A\right)^2 - g\delta_A R_2 V_A c(1+c)\left[\sin\left(\varphi_s' + \alpha\right) - \sin\alpha\right] - \\
& - g\delta_A R_2 V_A c(1+2c)\sin\alpha\left[\frac{e^{f\varphi_s'}}{\tilde{f}^2+1}\left(\tilde{f}\sin\varphi_s' - \cos\varphi_s'\right) + \frac{1}{\tilde{f}^2+1}\right] + \\
& + g\delta_A R_2 V_A c(1+2c)\cos\alpha\left[\frac{e^{f\varphi_s'}}{\tilde{f}^2+1}\left(\tilde{f}\cos\varphi_s' + \sin\varphi_s'\right) - \frac{\tilde{f}}{\tilde{f}^2+1}\right] + \\
& + g\delta_A R_2 V_A c^2 \sin\alpha\left[\frac{e^{f\varphi_s'}}{4\tilde{f}^2+1}\left(2\tilde{f}\sin\varphi_s' - \cos\varphi_s'\right) + \frac{1}{4\tilde{f}^2+1}\right] - \\
& - g\delta_A R_2 v_A c^2 \cos\alpha\left[\frac{e^{2\tilde{f}\varphi_c'}}{4\tilde{f}^2+1}\left(2\tilde{f}\cos\varphi_s' + \sin\varphi_s'\right) - \frac{2\tilde{f}}{4\tilde{f}^2+1}\right].
\end{aligned}
$$

where $a = \frac{\delta V_A^2}{EF} - 1$; $c = \frac{1}{EF}\left(P_A - \frac{b}{a_1}\right)$; $\tilde{f} = -f\,\frac{a_1}{a}$;

$$a_1 = 1 - \frac{\delta_A V_A^2}{EF};\quad b = \delta_A V_A^2\left(1 - \frac{P_A}{EF}\right);$$

[15] Methodics of the calculation of mechanical loses wedge-shape belt transmissions. Karimova I.M.

[16] Calculation model of the stress of wedge-shaped belt in the coverage area of flexible transmission pulleys. Karimova I.M.

δ_A – the mass coefficient of the length of the belt at point A.

CONCLUSION

1.In contrast to the known researches on the drive of the beam-pumping unit, the two-related units (engine-reducer and reducer-beam-pumping unit) have been considered, the selection possibilities of the sale cost characteristic associated with the number of engine power and cycles used in the engine-reducer contour have been given, the dynamic mode of the belt transmission has been studied, the executive similarity criteria have been defined and on the basis of this, the impact of the axles in the engine-reducer contour, the state of the straight line connecting the drive shaft of the engine and the center of the driven shaft of the reducer on the transmission process, the balance of forces and new technologies related to the determination of some indicators were developed to design the belt transmission, on the basis of the obtained dependence, the angle between the axles is considered to be optimal 45-57^{o} by carrying out computer simulation, which saves 10-20% on energy consumption compared to horizontal situation.

2.It is based on the possibility of replacing the multi-stage reducer with the double-stage chain reducer, which is used on the basis of the solution of the displacement problem, which ensures this replacement.

3.For the first time, on the basis of analytical studies, instead of the flat belt transmission of the drive mechanisms, the possibility of chain transmission has been confirmed and the rigidity and metric characteristics of the transmission have been substantiated.

4.Availability of application of wedge-shaped belt transmission in the contour of the chain of the drive mechanism, and the tasks of evaluation of operational and quality indicators of the relevant transmission have been substantiated, and calculation of mechanical losses and tension in the field of flexible transmission element pulley coverage of the wedge-shaped belt transmission in the contour of the chain of the drive mechanism have been solved.

5.It has been confirmed that with the proposed cylindrical layout calculations for maximum energy performance, it is possible to sharp reductions of metric and weight characteristics of reducer-beam-pumping unit.

6.Geometric, kinematic and dynamic characteristics of the inverter mechanism of the reducer-beam-pumping unit contour have been studied, and depending on these characteristics, the determination of forces in the head of the beam balance method has been developed and the change ranges that ensure the efficient operation of the pump assembly of the mentioned characteristics have been substantiated.

The main contents of the dissertation are reflected in the following published scientific works:

1.Gurbanov H.Y., Karimova I.M. Application and optimization of the design of three-stage gear reducers on the beam pumping units// - Baku: Scientific Works of the Azerbaijan State Marine Academy, - 2009. №1, - p. 72-75.

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Personal contribution of the applicant in the scientific works published in connection with the conducted research:

1.[1,8,11]-giving an idea, putting the issues under study, solution and getting results;

2.[2,3]-solution of the problem and analysis of the results;

3.[12]-making, solving and obtaining the results of the studied issues;

4.[4,5,6,7,9,10,13,14,15,16] - independently executed.

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