
RESEARCH ARTICLE

Determination of the kinematic parameters of the new constructive solution of the beamless sucker-rod pump

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Abstract

The petrol industry has very big impact on the modern world economy and politics. With any options and scenarios for the development of the economy in the coming years, natural resources, especially fossil oil products, will be the main factor of world economic growth. The sucker-rod pumps are one of the important aggregates for the mechanized oil recovery on land. However, the low efficiency, the big expenditure of the electric power, an incomplete construction of driving gear and reformative mechanisms, impossibility of use in existing aggregates of electric motors with a rotation high frequency lead to research of more perfect construction of a mechanical drive pumps. The present study describes a new design solution for a beamless sucker-rod pump for oil production, characterized by small dimensions and metal consumption, less energy consumption, better dynamic characteristics. The results of analytical kinematical studies and numerical calculations of the new beamless sucker-rod pump are presented.

Keywords

Beamless sucker-rod pump; Kinematics; Analytical solution; Duration

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1. Introduction

At present, various pump installations are used for the extraction of oil by a mechanized method. In the open literature there are some studies about the history of pumps, production technology and kinematics. Laine et al. [1] presented information on harmonic polished bar motion and production technology of pumping unit. Byrd et al. [2] discussed the issue of history, background and reasons of the oilfield pumping unit. Beck and Peterson [3] compared the performance of linear actuator according to walking pumping systems. Takacs [4] presented full kinematics and torsion analysis of rotaflex pumping units. The main

disadvantages of pumps are the large metal capacity of the structure, low efficiency, poor balance, the need for a massive foundation, etc [5]. Of the existing mechanized methods of oil production, the most common is the sucker-rod pump with the balanced individual drives of mechanical action.

Beamless sucker-rod pumps have smaller metal capacity and dimensions compared to balancing. On the other hand, compared to a balancing sucker-rod pumps, the beamless has a slightly different movement characteristic of the rod suspension point and a different picture of the forces acting. One of essential deficiencies of known mechanical drives of sucker-rod pumps (beam pumping unit) is

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that at the big traverse stroke, except increase in a dimension of the machine tool; raise as well the sizes of a beam head. At the usual beam with application of an arc beam head the span arrow in any position of the mechanism is organized.

Increasing the efficiency of sucker-rod pumps is carried out with the use of more advanced designs and mechanisms. Since the complex approach to the process of improving the sucker-rod pumps is somewhat complicated, it is usually enhanced by the flexibility of the transforming mechanism as well as from its energy saving point of view. One of the important issues when designing the kinematic scheme of the transforming mechanism is to ensure that the suspension point of the shaft is closer to the harmonic law [5].

Operation of the deep pump at each movement plunger up and downwards is accompanied by change of a direction of movement of pump rods and transition through some positions (dead points) with zero speed that should vary during one cycle of operation of machine on magnitude and a direction. At that dynamic loads on rods depend on magnitude of speed and acceleration which are drastic shown at descent of pumps to the big depths [6].

As rods are set in motion by sucker-rod pump, operation of the deep pump essentially depends on the law of movement of a suspension bracket polished rod. Thus, an important task at the oilfields today is to find ways to increase the efficiency and reliability of the sucker-rod pumps in order to reduce operating costs and increase the inter-repair period of well operation. Therefore, the development of a new design solution of beamless mechanical drives for sucker-rod pumps, which eliminates the above drawbacks, is of great practical importance for modern oil engineering. In this study, the solution of the problem is discussed.

2. Formulation of the problem

2.1. A new design of a beamless sucker-rod pump

The proposed design of the oil pumping unit has an original design and provides the following

advantages over other types of oil production equipment:

1. It consumes about 1.5-1.7 times less electricity than conventional sucker-rod pumps.
2. Increases the service life of the gearbox, due to the absence of negative torque on the output shaft.
3. Does not require a solid and high foundation.
4. Less sensitive to the uneven draft of the foundation.
5. There is no massive beam and a bulky rotary head.
6. Due to the work on the symmetrical cycle, the service life of the rod string increases, since dynamic loads and vibrations are greatly reduced.
7. Has small dimensions.
8. It is possible to free space around the wellhead during installation, repair and adjustment of the trajectory of the rod suspension point along a vertical line.
9. It is possible to fully unfold the front and rear racks during the transportation of the rocking machine.

Fig. 1 shows a diagram of a new solution of the beamless sucker-rod pump.

The new solution of the beamless sucker-rod pump contains two cranks 7, rigidly fixed on both sides at the output ends of the driven shaft of the multi-stage AN-reducer 4, which has only two shafts and gear ratio 1: 125 [7]. At one end of the drive shaft of the AN-reducer, a stepped driven pulley 6 of the V-belt drive 3 is installed, and at the other output end- a two-disk brake 5 is located. The mechanical drive comprises a frame 1 made of profile rolling in the form of two longitudinal cross-connects, two brackets for connecting the front pillars 18, two brackets for connecting the rear pillars 19, and two brackets for connecting the front rods 16 and two brackets for connecting rear rods 17. On the frame together with an AN-reducer is installed a three-phase asynchronous electric motor 2.

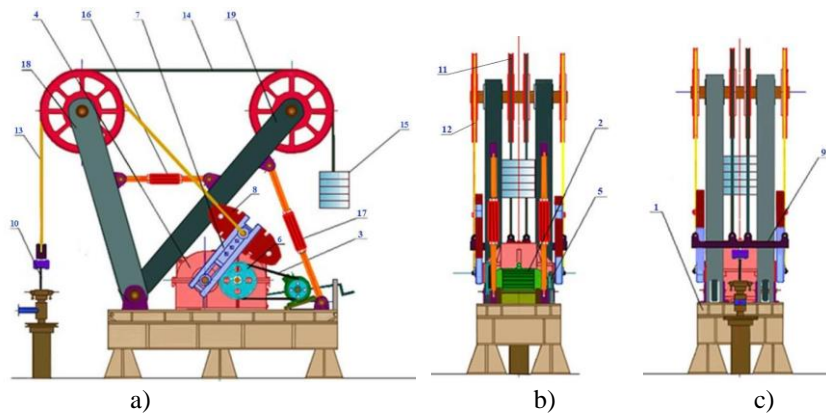


Fig. 1. A diagram of a new solution of the beamless sucker-rod pump

As part of the transforming mechanism of mechanical drive, there are also ropes 13, 14 and blocks 11, 12. At one end of the parallel ropes, a crosspiece 9 is suspended from which the column of the rods 10 is fixed and at the other end – cranks with a counterweight 8 are fixed. On the other hand, with the parallel ropes 14, the traverse is connected to a movable counterweight 15 giving a gain in strength. The lower end of the rods column 10, rigidly connected to the traverse, is connected to the pump piston. The front and rear drive racks, pivotally connected to the frame brackets, are interconnected by front traction bars with right and left threads each.

In addition, the rear pillars with the help of the rear traction rods are also pivotally connected to the frame brackets. Front and rear traction rods provide a change in the angles of inclination of the front and rear racks. These racks can be made telescopic to adjust the height of the stand depending on the stroke of the rod suspension. The lower ends (base) of the front and rear pillars are attached to the bracket on the hinged support with the possibility of deflecting to the right or left with the front rods to free space around the wellhead during repair and to precisely control the trajectory of the rods suspension point along the vertical line. In addition, the connection of the pillars to the bracket on the hinged support makes it possible to completely unfold the front and rear racks of mechanical drive when transported to the installation site. To reduce the load on the elements of the transforming

mechanism by means of a flexible link to the traverse, a movable counterweight is additionally secured.

2.2. Kinematic study of a new design of a beamless sucker-rod pump

It is known that the laws of motion of the rod suspension point depend on the kinematic scheme of the used mechanical drive for sucker-rod pumps. Therefore, the main goal of the kinematic analysis of any hinged mechanism is to determine the positions, velocities and accelerations of its various nodes and points [8, 10, 11]. If the length of the crank r_1 is known, then the displacement of the rod suspension point will be: $S = 2r_1$ as seen in Fig. 2.

As can be seen from the diagram, the movement of the suspension point of the rod will be equal to the movement of the corresponding point of the rope. In the kinematic study, the length of the crank $AB = r_1$, the angle of its rotation φ_1 , the radius of the block $DC = R_b$, the coordinates of the points $A(0,0)$, $D(a,b)$ and the length of the rope L are specified. The purpose of the kinematic study is to determine the displacements, velocities and accelerations of the suspension point of the rod E, depending on the angle of the crank (as a function of time). As can be seen from the scheme of the installation, the displacement of the point E uniquely depends on the distance BC and the arc

length of the block CF , covered by the rope, i.e.,
 $S_E = f(BC + CF)$.

3. Solution of the problem

To solve the problem, first need to determine the distance between the points B and C , that is, from the point of contact of the rope with the crank to the point of contact of the rope with the block, and then the arc length of the block covered by the rope from point C to point F .

Since the angle φ_1 is known:

$$x_1 = r_1 \cos \varphi_1, \quad y_1 = r_1 \sin \varphi_1 \quad (1)$$

The length of the straight line BC tangent to the radius R_b from the given point $B(x_1, y_1)$ will be [9]:

$$BC = \sqrt{(x_2 - x_1)^2 + (y_2 - y_1)^2} \quad (2)$$

To determine the unknown coordinates x_2 and y_2 of the point C , which is formed by touching the line BC with a circle of radius R_b , we write the equation of the circle:

$$(x - a)^2 + (y - b)^2 = R_b^2 \quad (3)$$

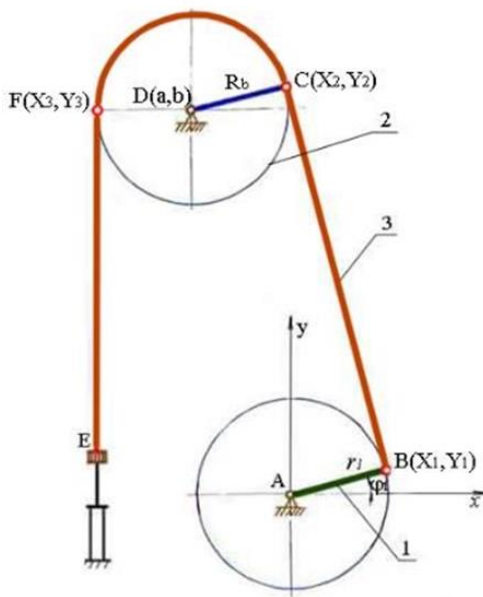


Fig. 2. Kinematic scheme of the transforming mechanism of a new constructive solution of a beamless sucker-rod pump

where a and b are the coordinates of the center of the circle of radius R_b . The equation of the line BC passing through the point:

$$y - y_1 = k(x - x_1) \quad (4)$$

Since this line passes through the point $C(x_2, y_2)$, the correlations x_2 and y_2 must satisfy Eq. (4):

$$y_2 - y_1 = k(x_2 - x_1) \quad (5)$$

where k is coefficient of angle. To find this coefficient, we find the derivative of Eq. (3) in an implicit form:

$$y' = \frac{a - x}{y - b} \quad (6)$$

From Eq. (5)

$$k = \frac{y_2 - y_1}{x_2 - x_1} \quad (7)$$

On the other hand

$$k = f'(x_2) = \frac{a - x_2}{y_2 - b} \quad (8)$$

Hence:

$$\frac{a - x_2}{y_2 - b} = \frac{y_2 - y_1}{x_2 - x_1} \quad (9)$$

Considering

$$(x_2 - a)^2 + (y_2 - b)^2 = R_b^2 \quad (10)$$

We find solving Eq. (9) from the system of equations:

$$\begin{cases} x_2^2 + y_2^2 - x_2(a + x_1) - y_2(b + y_1) + ax_1 + by_1 = 0 \\ x_2^2 + y_2^2 - 2x_2a - 2y_2b + a^2 + b^2 - R_b^2 = 0 \end{cases} \quad (11)$$

Using the program MATLAB, we determine the coordinates x_2 and y_2 , then, depending on the angle of rotation of the crank φ_1 , we determine the required distance BC and the displacement of the point E

$$S_E = \left(BC_{\max} + CF(\varphi_1) \right) - \left(BC(\varphi_1) + CF(\varphi_1) \right) \quad (12)$$

where CF is the arc length of the block covered by the rope

$$CF = \int_{x_2}^{x_3} \sqrt{1 - (f'(x))^2} dx \quad (13)$$

To find $y'(x)$, we take $f'(x) = y'(x)$ and solve the equation of the circle (3) with respect to y

$$y = b \pm \sqrt{R_b^2 - x^2 + 2ax - a^2} \quad (14)$$

Differentiating this equation with respect to x , we obtain.

Then Eq. (13) takes the form

$$\begin{aligned} CF &= \int_{x_2}^{x_3} \sqrt{1 + \frac{(a-x)^2}{R_b^2 - x^2 + 2ax - a^2}} dx \\ &= \int_{x_2}^{x_3} \frac{1}{\sqrt{1 - \left(\frac{x-a}{R_b}\right)^2}} dx \end{aligned} \quad (15)$$

Assuming $t = \frac{x-a}{R_b}$, $dt = \frac{dx}{R_b}$ and $dx = R_b dt$, we get:

$$\begin{aligned} CF &= \int_{x_2}^{x_3} \frac{1}{\sqrt{1-t^2}} R_b dt = R_b \arcsin t \Big|_{x_2}^{x_3} \\ &= R_b \arcsin \frac{(x-a)}{R_b} \Big|_{x_2}^{x_3} \\ &= R_b \left[\arcsin \frac{(x_3-a)}{R_b} - \arcsin \frac{(x_2-a)}{R_b} \right] \end{aligned} \quad (16)$$

4. Numerical computations

To confirm the obtained analytical dependencies, a numerical experiment was performed with the following data: $r_1 = 0.5$ m, $R_b = 1$ m, $b = 3.5$ m, $a = 1.8$ m, and $\omega_1 = 1.256$ rad/s. The results are given in Table 1 and Figs. 3-5. To determine the movement, velocity and acceleration of the rod suspension point depending on the crank angle φ_1 , the point of calculation is taken to be $\varphi_1 = 240^\circ$. As can be seen from the deviation graphs of actual movements, velocities and accelerations, the rod

suspension points, depending on the crank angle, are close to the parameters of ideal harmonic laws.

When the rod suspension point moves up and down, the periods of displacements are subject to harmonic laws, however, the amplitude of displacements when moving upwards is approximately 8-15% greater than when moving downward.

When the crank is turned through the angle of $\varphi_1 = 90^\circ$, the point of suspension of the rod passes more than half of its stroke ($S = 506.7669$ mm) and when the crank is turned to the second quarter of revolution by $r/2$ ($S = 456.9165$ mm). This is explained by the fact that this mechanism is a positive de-axial, that is, in the de-axial mechanism, the duration of the forward and reverse stroke is different. Therefore, when the crank is rotated clockwise (the wellhead on the left), the upward stroke is faster than the down stroke, and when the crank is rotated counterclockwise, the upstream travel takes longer than the down stroke.

From the graphic image in Fig. 4, it is seen that the maximum value of the rod suspension point velocity corresponds to the angle of $\varphi_1 = 90^\circ$, thus, $V = 0.699725$ m/s. For $\varphi_1 = 0^\circ$ and $\varphi_1 = 180^\circ$, the speed of the rod suspension point is zero, because at these points the direction of travel of the suspension point of the rod changes.

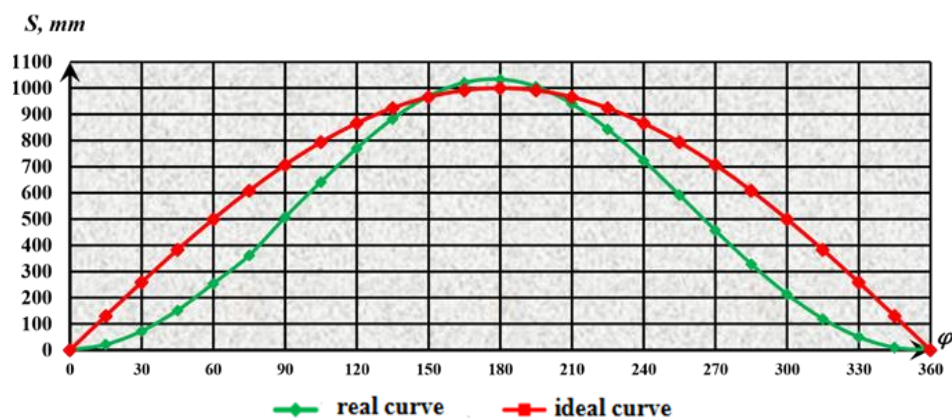
As can be seen from the graphical image in Fig. 5, the acceleration of the rod suspension points is harmonic functions of the angle φ_1 . The crank angle φ_1 , at which the acceleration of the rod suspension point is zero, is equal to the angle at which the piston speed has the maximum value.

5. Conclusions

The aim of the present study is to develop a new design solution for a beamless sucker-rod pump for oil production, characterized by small dimensions and metal consumption, less energy consumption, better dynamic characteristics. The results of the analytical kinematical studies and numerical calculations of the new beamless sucker-rod pump are presented.

Table 1. Determination of the movement, velocity and acceleration of the rod suspension point, depending on the crank angle φ_1

φ_1	x_1	y_1	x_2	y_2	x_3	BC	CF	S , mm	V , m/s	a , m/s ²
0	500	0	803.6	3584	2800	4321.025	3056.714	0	0	1.172692
15	482.96	129.4	804.3	3592	2800	4299.701	3048.823	21.32368	0.102615	1.062023
30	433.01	250	806.1	3610	2800	4248.363	3031.083	72.66168	0.247052	1.028730
45	353.55	353.55	809.5	3638	2800	4169.419	3003.643	151.6063	0.409902	0.984300
60	250	433.01	814.9	3672	2800	4066.962	2968.751	254.0628	0.553047	0.865198
75	129.4	482.96	2496.8	2783	2800	3959.662	799.8699	361.3623	0.656353	0.624402
90	0	500	2474.7	2762	2800	3814.258	830.2382	506.7669	0.699725	0.262145
105	-129.4	482.96	2458.5	2747	2800	3679.374	851.9724	641.6513	0.675099	-0.148840
120	-250	433.01	2449.6	2740	2800	3550.867	863.7381	770.1575	0.618405	-0.342670
135	-353.6	353.55	2448.2	2739	2800	3438.737	865.5781	882.2874	0.539599	-0.576302
150	-433	250	2453.6	2743	2800	3352.660	858.465	968.3644	0.414225	-0.757780
165	-483	129.4	2474.8	2762	2800	3300.513	830.1027	1020.512	0.250946	-0.986890
180	-500	0	2480.4	2767	2800	3287.685	822.4880	1033.34	0.061732	-1.143650
190	-483	-129.4	2498.9	2785	2800	3315.451	796.9380	1005.573	-0.133620	-1.180740
210	-433	-250	2519.1	2805	2800	3381.048	768.2900	939.977	-0.315670	-1.100330
225	-353.6	-353.6	2539.7	2827	2800	3477.877	738.1719	843.1478	-0.465970	-0.908440
240	-250	-433	2559.7	2850	2800	3597.235	707.9447	723.7903	-0.574380	-0.655270
255	-129.4	-483	824.8	3721	2800	3729.219	2918.419	591.8062	-0.635140	-0.367260
270	0	-500	818.5	3692	2800	3864.108	2948.941	456.9165	-0.649120	-0.084520
285	129.4	-483	813.3	3663	2800	3992.725	2978.316	328.2994	-0.618940	0.1824520
300	250	-433	809.3	3636	2800	4107.467	3005.105	213.5577	-0.552170	0.4035780
315	353.55	-353.6	806.5	3613	2800	4201.791	3027.513	119.2341	-0.453910	0.593890
330	433.01	-250	804.6	3596	2800	4270.865	3045.639	50.15945	-0.332410	0.734395
345	482.96	-129.4	803.7	3586	2800	4311.183	3055.543	9.841723	-0.194020	0.836432
360	500	0	803.6	3584	2800	4321.025	3056.714	0	0	1.172692

**Fig. 3.** The change of movement of the rod suspension point depending on the crank angle

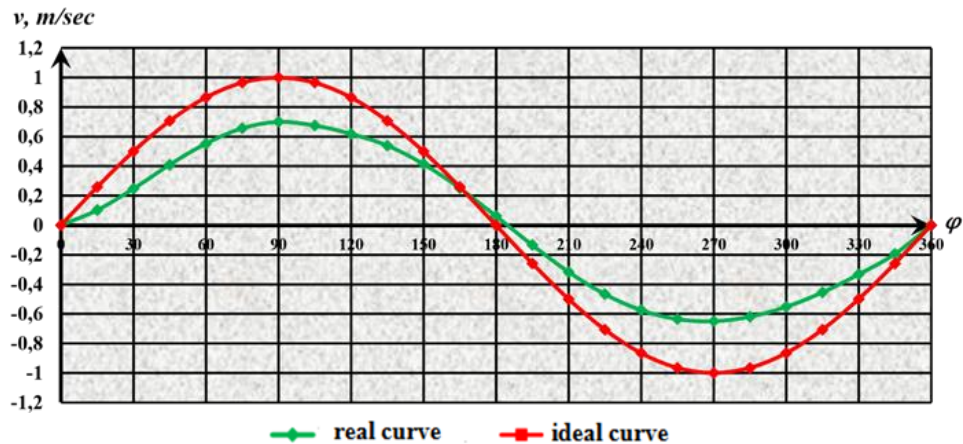


Fig. 4. The change of speed of the rod suspension point depending on the crank angle



Fig. 5. The change of acceleration of the rod suspension point depending on the crank angle

The results support the following conclusions:

- Analytical dependencies are obtained to determine the basic kinematic characteristics of the de-axial transforming mechanism of a new design of a beamless sucker-rod pump.
- Using the proposed mechanism with a positive de-axial when the crank is rotated counter-clockwise helps reduce the inertial component of the load on the suspension point of the rods during the up stroke.
- In the de-axial mechanism, the duration of the forward and reverse stroke is different: by changing the duration of the idling or working stroke, it is possible to regulate the performance or provide more favorable operating conditions for the drive.

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