Analysis of the performance indicators of oil well sucker-rod pumps

Zuleykha E. Eyvazova, *Azerbaijan State Oil and Industry University*
Tarlan E. Farajov, *Azerbaijan State Oil and Industry University*

Zuleykha E. Eyvazova

Tarlan E. Farajov

Abstract. *The paper notes that as the depths of operated wells grow, the application of cable and pulley mechanisms becomes preferable as compared to the existing pumpjacks. A generalized theoretical analysis of the kinematics of cable and pulley drives is set forth. The authors present the general theoretical analysis of the kinematics of the above mechanisms, as well as the results of computer calculations based of the developed equations for a number of cases. Further analysis of the results showed that the crank mechanisms of a rope pulley have "smooth" kinematics. The research resulted in a proposed invention of the design of mast-type oil well sucker-rod pump drive with lower steel intensity and power consumption that would allow increasing the performance of sucker-rod pumps. The* **Purpose** *of this article consists in finding a utility model of a pump for the well rod in order to ensure the environmental safety of the equipment. That is achieved by lightening the metal structure of the pump with rotary stem and energy consumption is reduced. In the context of this problem, some calculations were performed in order to prove the system's dependability. Based on the performed calculations it was established that the light structure can be used instead of the old heavy structure being its environmentally safe version. Experimental studies conducted by AzINMASH Research and Design Institute of Petroleum Engineering (Baku, Azerbaijan) indicate the feasibility of normal operation of sucker-rod pumps under the condition that n·S = 54÷60 m/min. The authors examined the dependence between the peak output Q and the number of strokes n for various standard pumpjack sizes. The analysis of the parameters shown that the value of the product n·S in the existing pumpjacks is below the recommendations based on experimental data, i.e. there is a tangible opportunity of increasing the productivity by extending the stroke of the rod hanger center, since well pump barrels may be as long as 6 to 7 meters. Estimates show that while studying the kinematics of long-stroke drives the changes in the length of the rope may be practically disregarded due to the displacement of the rope-to-pulley contact point. This simplifies the formulas that describe the kinematics of this type of long-stroke drives. Using the resulting formulas, comparative computer calculations for various cases were performed. It is shown that cable and pulley mechanisms have "softer" kinematics. The calculations confirmed the advisability of modification of the pump's design that ensured reduced pollution of environment and energy savings. The future world will need renewable sources of energy, more power-efficient oil and gas production, minimal or zero pollution of the environment, thus the proposed solution appears to be of relevance. The authors propose a more productive design of sucker-rod pump that is easy to install and maintain at oil and gas production facilities. That can be achieved based on the calculations mentioned above.*

Keywords: *pumpjack, cable and pulley mechanism, deep well sucker-rod pump drive, long-stroke drive, drive link, crank, kinematic calculation, rod hanger center, displacement, speed, acceleration.*

Introduction. As it is known, one of the mechanical methods of oil extraction is the use of sucker-rod pumps (SRP). Currently, the most commonly used deep well sucker-rod pump drive (DWSRPD) in the world are balanced pumpjacks. Over the past few years, the design of the pumpjack has remained almost unchanged due to its relative simplicity, ease of maintenance, operational characteristics, and reliability. However, in recent years, there was significant progress in the implementation of new design solutions of DWSRPD drives that is due to the intense development of oil production equipment and technology. The aim of this paper is to research the improvement of the DWSRPD drive design, aimed at reducing steel intensity of the existing drives and increasing SRP efficiency.

The main quality of SRP is its efficiency that depends on the following basic standard parameters:

1) stroke length of the rod hanger center (plunger);

2) number of double strokes per minute of the rod hanger center;

3) diameter of plunger.

In order to find the most advantageous pumping mode, it is required to choose the correct combination of these indicators.

The first two indicators depend on the design of the deep well sucker-rod pump drive. The field experience of deep well pumps operation has shown that increasing the pump performance by increasing the number of plunger strokes is impractical, since with a number of strokes greater than 15 per minute the frequency of sucker rod breakage increases significantly.

The formula of the daily capacity of a pumping plant is as follows:

$$Q = 1440 K_p f_a nS \left(\frac{1}{\cos \dfrac{\pi nH}{30a}} - \frac{\lambda}{S} \right).$$

where Q is the plant capacity, m³/day; K_p is the pump delivery coefficient; n is the number of double strokes of the rod hanger center (RHC); S is the length of RHC stroke, m; H is the pump running depth, m; λ is the total static deformation of pumps and rods under the load of the liquid

mass, m; f_a is the cross sectional area of the plunger, m²; a is the sound speed in rods, m/s.

The results of experimental studies conducted by AzINMASH Research and Design Institute of Petroleum Engineering (Baku, Azerbaijan) indicate the feasibility of normal operation of sucker-rod pumps under the condition that $n{\cdot}S = 54{\div}60$ m/min.

Let us examine the dependence between the peak output Q and the number of strokes n for various standard pumpjack sizes (Table 1).

Analyzing the table parameters we can see that the value of the product $n{\cdot}S$ in the existing pumpjacks is below the recommendations based on experimental data, i.e. there is a tangible opportunity of increasing the productivity by extending the RHC stroke, since well pump barrels may be as long as 6 to 7 meters.

Let us examine the following example to prove the statement regarding the applicability of the long stroke in increasing the plant capacity.

Figure 1 shows two dependences of the sucker-rod pump capacity on the number of RHC strokes – $Q = f(n)$.
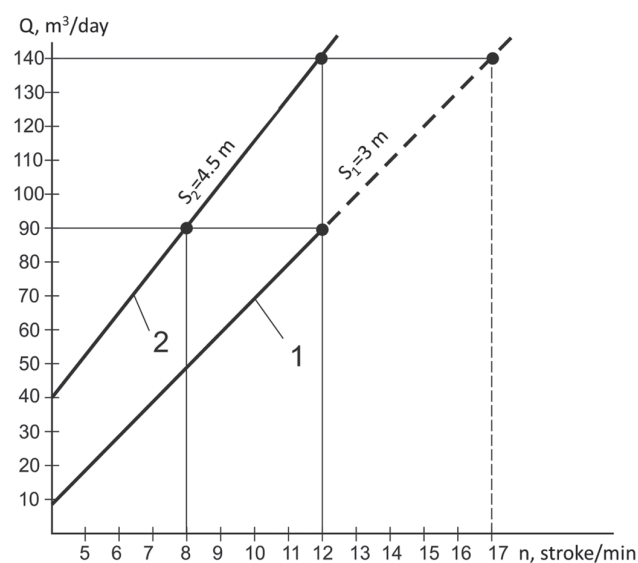

Figure 1. Performance to number of RHC strokes dependence graph

For the first dependence, the following standard parameters of the SKD8-3-4000 pumpjack are chosen: the

Table 1. SKD-type pumpjack technical parameters

Pumpjack standard size	Parameters				
	Diameter of pump d_n, mm	Length of motion S, m	Number of RHC double strokes, n	Pump output Q, m³/day	n·S
SKD 3-1.5-710	68	1.5	15	84.9	22.5
SKD 4-2.1-1400	93	2.1	15	225.8	31.5
SKD 6-2.5-2800	93	2.5	14	245.8	35
SKD 8-3-4000	93	3	12	250	36
SKD 10-3.5-5600	93	3.5	12	291	42
SKD 12-3-5600	93	3	12	236.7	36

number of double strokes per minute of the rod hanger center is $n = 12$; the stroke length of RHC is $S_1 = 3$ m; the deep well pump with the diameter of $d_p = 57$ mm is hung at a depth of $H = 980$ m; pumps diemeter is 73 mm; the structure of stem rod is three-stage and consists of rods with diameters of 19 mm, 25%, 22 mm, 40% and 25 mm, 35%. The capacity is determined on the basis of computer calculations (Appendix 1) and is as follows: $Q_1 = 90$ m^3/day; $n·S = 36$.

The second dependence corresponds to the stroke length $S_2 = 4.5$ m, the capacity of $Q_2 = 140$ m^3/day (Appendix 2), $n·S = 54$ and other equal parameters.

Considering the fact that the dependence of the capacity on the number of strokes is close to the straight line law, the diagram shows that if capacity S_1 increases for the depended 1 to the value of $S_2 = 140$ m^3/day, the number of double strokes per minute of the rod hanger center increases to 17.

Such number of double strokes per minute of the rod hanger center is unacceptable, since in addition to the increasied number of sucker-rod brackage, the number of strokes cannot provide the required capacity. The positive aspect of increasing the RHC stroke length to 4.5m is that in order to obtain the capacity of $S_2 = 140$ m^3/day, the number of double strokes per minute of the rod hanger center decreases up to 8, which should reduce the frequency of the sucker-rod brackage, as the diagram shows. In this case, the real possible capacity due to the increase of the stroke length will be at $S_2 - S_1 = 140 - 90 = 50$ m^3/day greater.

If the long-stroke DWSRPD drive is developed on the basis of the kinematic parameters of the SKD-type pumpjack, it must be taken into account that the overal dimension as well as the pumpjack mass will increase, since the length of all parts – k_1, k, l, r; pole distance p, and, consequently, column height and frame length will increase. Additionally, the electric motor power N_{em} also grows (Table 2). The mechanism's dynamics deteriorate as well.

Estimates show that while studying the kinematics of long-stroke drives the changes in the length of the rope may be practically disregarded due to the displacement of the rope-to-pulley contact point. This simplifies the formulas that describe the kinematics of this type of long-stroke drives. For this purpose, let us assume in the calculation scheme in the Figure 2 that the contact point is fixed, i.e. the point A_o coincides with the point A. Then, in the initial state of the machine (when the rod hanger center D starts to move upwards), the cable length from the crank to the pulley (minimum length) will be as follows:

$$l_o = AO_1 - R, \text{ where } AO_1 = \sqrt{p^2 - r^2} \text{ or } l_o = \sqrt{x_o^2 + y_o^2 - r^2} - R.$$

After the crank rotation by some angle φ, the current length of this cable section will be:

$$l(\phi) = \sqrt{AC^2 + CB^2},$$
$$\text{where } AC = l_o + R(1 - \cos\phi); CB = R\sin\phi,$$
$$l(\phi) = \sqrt{\left[l_o + R\left(1 - \cos\phi\right)\right]^2 + R^2\sin^2\phi}.$$

The displacement amount of point D is determined in accordance with the following formula:

$$S(\phi) = l(\phi) - l_o =$$
$$= \sqrt{\left[l_o + R\left(1 - \cos\phi\right)\right]^2 + R^2\sin^2\phi} - \sqrt{p^2 - r^2} + R.$$
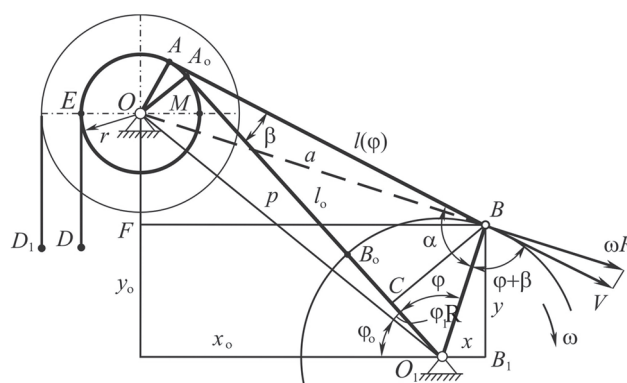

Figure 2. The calculation scheme of the crank cable and pulley drive of DWSRPD

The movement speed of this point is determined as the line speed projection of the cutpoint of the cable and crank on the cable direction:

$$V(\phi) = \omega R\cos(\pi/2 - \phi - \beta) = \omega R\sin(\phi + \beta).$$

The acceleration of this point is determined as the derivative of the speed found with respect to time:

$$W(\phi) = \frac{dV(\phi)}{dt} = \frac{d\phi}{dt} \cdot \frac{dV(\phi)}{d\phi} = \omega\frac{dV(\phi)}{d\phi} = \omega^2 R\frac{d}{d\phi}\cos\left(\phi + \beta\right),$$
$$W(\phi) = \omega^2 R\left(1 + \frac{d\beta}{d\phi}\right)\cos\left(\phi + \beta\right).$$

Let us calculate the angle $\beta(\phi)$:

$$\beta(\phi) = \text{arctg}\frac{CB}{AC} = \text{arctg}\frac{R\sin\phi}{l_i + R\left(1 - \cos\phi\right)}.$$

To simplify calculations, let us expand the function arctg$x$ by formal power series provided that $|x| < 1$:

Table 2. Comparative table of SKD-type pumpjack and assumed long-stroke DWSRPD drive performance indicators

Type of DWSRPD drive	Indicators							
	S_o, mm	k_1, mm	k, mm	l, mm	r, mm	p, mm	N_{em}, kW	Q, m^3/day
SKD 8-3-4000	3	2	2.29	3	1.2	3,62	23.5	90
Long-stroke DWSRPD drive	4.5	3.43	3	4.52	1.8	5.43	35	140

$$\text{arctg}x = \sum_{n=0}^{\infty}(-1)^n\frac{x^{2n+1}}{2n+1} = x - \frac{x^3}{3} + ...$$

Assuming that $\left|\dfrac{R\sin\phi}{l_o + R(1-\cos\phi)}\right| < 1$ and, using just the

first two expansion term with a high degree of accuracy, we obtain:

$$\beta(\phi) = \frac{R\sin\phi}{l_o + R(1-\cos\phi)} - \frac{1}{3}\left[\frac{R\sin\phi}{l_o + R(1-\cos\phi)}\right]^3.$$

The final formula for the acceleration $W(\varphi)$ will be:

$$W(\phi) = \omega_2 R\left\{1 + \frac{R(l_o+R)\cos\phi - R^2}{[l_o + R(1-\cos\phi)]^2}\left\{1 + \left[\frac{R\sin\phi}{l_o+R(1-\cos\phi)}\right]^2\right\}\right\} \times$$

$$\times \cos\left\{\phi + arctg\left[\frac{R\sin\phi}{l_o + R(1-\cos\phi)}\right]\right\}.$$

For the differential pulley, in order to obtain values $S(\varphi)$, $V(\varphi)$ and $W(\varphi)$ for the point D_1, the abovementioned formulas should be multiplied by gear ratio $\lambda = r_1/r$.

On the basis of the obtained formulas, the computer simulation was performed and comparative computer calculations for the long-stroke drives were made based on a conventional double-arm balanced pumpjack with the stroke length of the
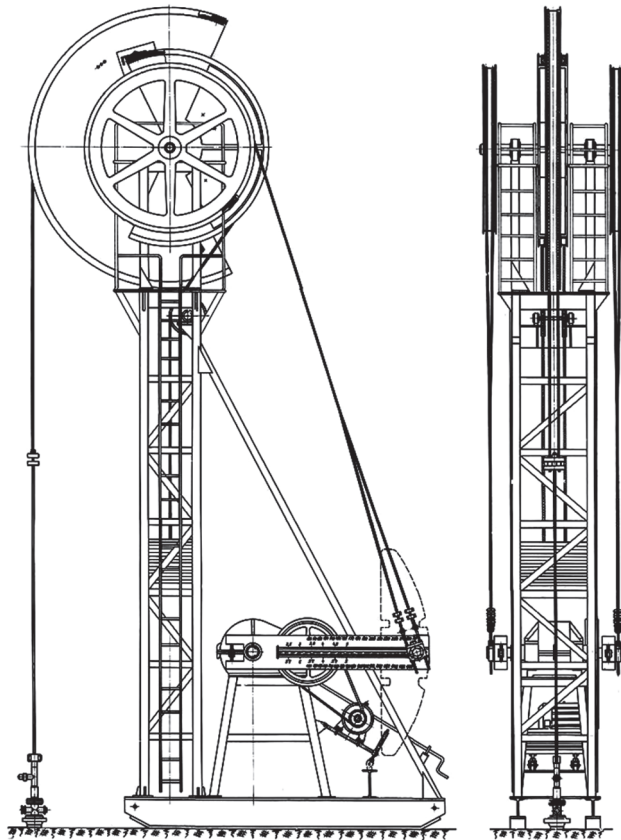

Figure 3. Ultra-long-stroke cable and pulley drive of DWSRPD (Azerbaijan)
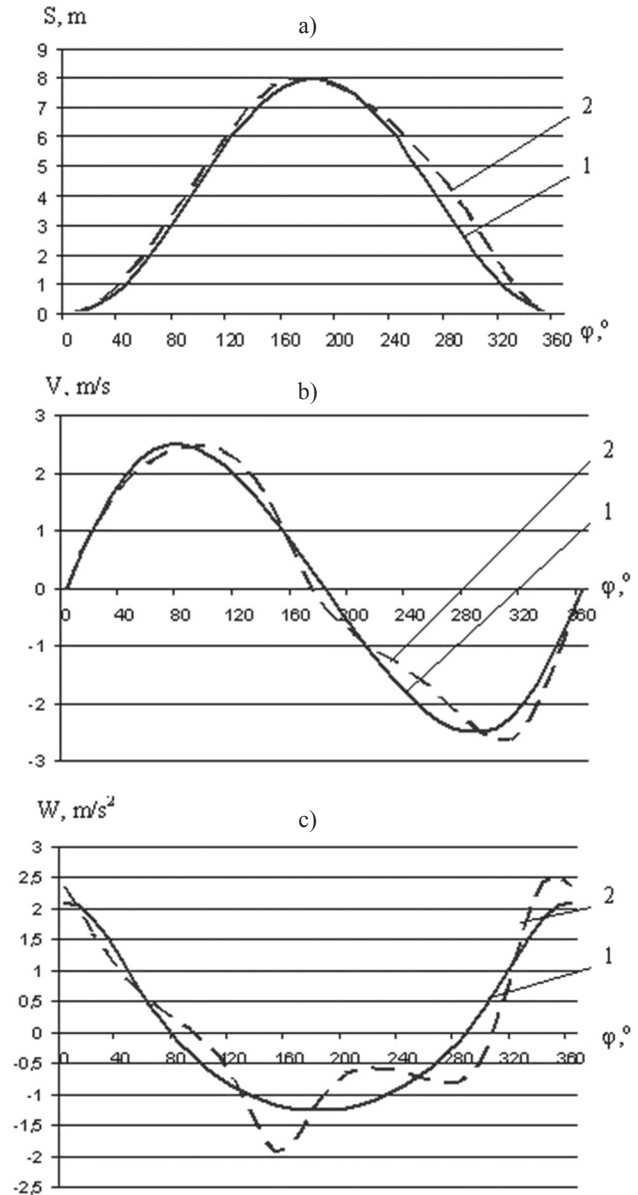

Figure 4. The diagrams of change of displacement (a), speed (b) and acceleration (c) of RHC SRP:
The 1-st is the cable and pulley drive of DWSRPD;
the 2d is the off-center pumpjack

rod hanger center $S = 8$ m and drive on the scheme in Figure 3 with the same stroke length.

The calculation of the pumpjack was carried out according to the well-known conventional formulas for a modern off-center SKD-type pumpjack produced with the off-center angle $\theta = 9°$. The ratio of the crank radius r to the lengths of the tail half of walking beam k and rocker l were equal to $r/k = 0,6$ and $r/l = 0,4$ respectively, in accordance with the moderm design practice. The ratio of the length of the front arm of the walking beam k_1 to the length of tail half of walking beam k was equal to $k_1/k = 1.4$. In accordance with the stroke length $S = 8$ m, the obtained dimensions of the pumpjack are as follows: the length of the front arm of the walking beam is $k_1 = 6.1$ m; the length of the tail half of the walking beam is $k = x_o = 4.36$ m; the length of the

rocker is $l = y_o = 6.54$ m; the length of the pole distance is $p = 7.86$ m; the crank radius is $R = 2.62$ m.

The dimensions of the machine according to the diagram in the Figure 2 are as follows: the radius of the large pulley is $r_1 = 2.5$ m; the radius of the small pulley is $r = 1.5$ m; the radius of the crank is $R = 2.4$ m; the radius of the length $x_o = 1.5$ m; $y_o = 9.2$ м; the radius of the pole distance is $p = 9.3$ m. The calculation of this option was carried out in accordance with the abovementioned formulas for the differential pulley.

Figure 4 shows the results of the comparative kinematic calculation, where the curves 1 correspond to the cable and pulley drive, and curves 2 correspond to the conventional double-arm off-center pumpjack.

Since the dynamics of the downhole equipment operation and, consequently, the strength load of the drive and transmission largely depend on the machine kinematics and mainly on acceleration, the presented comparative schemes allow concluding that the main performance indicators of the crank cable and pulley mechanism of the SRP drive for the oil extraction are better.

To create long-stroke DWSRPD drives that allow increasing the capacity of SRP, reducing steel intensity and power consumption in comparison with the existing long-stroke drives, the design of mast-type oil well sucker-rod pump drive was proposed as an invention [2].

Conclusion

Comparative calculations of the performance of SRPS allowed making a conclusion regarding the applicability of long-stroke SRP.

The paper sets forth basic dependences of the kinematics of long-stroke drives, on which comparative calculations and graphs of movement, speed and acceleration of the rod

hanger center of SRP rope and pulley drive and eccentric pumpjack were based showing the superior basic performance of the rope and pulley mechanism.

The research resulted in a proposed invention of the design of mast-type SRP drive with lower steel intensity power consumption that would allow increasing the SRPS performance.

References

[1]. Vagidov MA, Eyvazova ZE. Krivoshipnye kanatno-shkivnye mekhanizmy [Crank-driven cable and pulley mechanisms]. Mekhanika i mashinostroyenie 2006;2:40-42 [in Russian].

[2]. Application for invention A 2016 9973. Eyvazova ZE, Farajov TE. Deep well pump drive. Priority as of 17.06.2016.

[3]. Burovye kompleksy [Drilling systems]. Porozhsky KP, editor. Yekaterinburg: UrSMU publishing; 2013 [in Russian].

[4]. Bagramov RA. Burovye mashiny i kompleksy: Ouchebnik dla vuzov [Drilling machines and systems: Textbook for higher educational institutions]. Moscow: Nedra; 1988 [in Russian].

About the authors

Zuleykha E. Eyvazova, Associate Professor, educator, Azerbaijan State Oil and Industry University, Baku, Azerbaijan, e-mail: eyvazovaze@mail.ru

Tarlan E. Farajov, Master of Chair, Azerbaijan State Oil and Industry University, Baku, Azerbaijan, e-mail: tarlan.farajov@hotmail.com