DNIPRO UNIVERSITY of TECHNOLOGY 1899

Mining of Mineral Deposits
Volume 14 (2020), Issue 4, 82-89

MMD JOURNAL / MINING.IN.UA

UDC 622.276.53:621.671 (047) https://doi.org/10.33271/mining14.04.082

Dynamic analysis of well equipment to produce oil

Volodymyr Grudz[1*], Yaroslav Grudz[1], Volodymyr Bevz[1], Mykhailo Chernetsky[1]

[1]*Ivano-Frankivsk National Technical University of Oil and Gas, Ivano-Frankivsk, 76019, Ukraine*

**Corresponding author: e-mail michael.chernetskyi@gmail.com, tel. ++380966470016*

Abstract

Purpose is to study dynamics of a technological cycle of well oil production equipment to evaluate the forces acting on its structural components while operating. It is required to improve their reliability and durability owing to the decreased inertia.

Methods. Mathematical modeling of the system relies upon the basic law of motion dynamics of a complex system with the attached mass involving deformation of the components and further implementation of the mathematical model using methods of mathematical analysis. To improve informativeness and reliability of the results, obtained in the process of mathematical modeling, it is proposed to divide the technological cycle into separate stages each of which characterizes a certain motion process as well as changes in the nature of forces influencing the system elements.

Findings. Analysis of results of mathematical modeling of the system operating cycle makes it possible to draw conclusions about the process time as well as about the motion nature of a landing top during operation and a value of inertia acting on the well-drilling equipment demonstrating the ways to decrease them while providing reliability and durability of the facilities. Components of a hydraulic drive have been studied thoroughly while dividing its operation into eight phases of motion cycles. It has been identified that decrease in the inertia influence on the system components results from the following: power hydraulic cylinders are manufactured with the step-up diameter increase in their upper half; hollow rods in their lower half are equipped with a discharge valve dumping a certain share of liquid into a reservoir to decrease the traverser raise velocity.

Originality. Mathematical modeling has helped identify that drastic decrease in the system inertia depends upon its structural and kinematic characteristics; moreover, it may vary broadly.

Practical implications. During the practical operation of well-drilling equipment for oil production, decrease in inertia effect on the system components will help improve its reliability and durability.

Keywords: *drilling equipment, oil production, dynamics, force field, inertia, operating cycle, phase, reliability*

1. Introduction

Implementation of secondary methods to produce liquid carbohydrates involves the use of deep-well pumps; components of well equipment are used for their drives. In this context, travel length and the number of a cycle per time unit are the basic parameters characterizing a well efficiency. Even if increase in one of the parameters is observed, a well productivity is considered as the improved one. Admissible travel length for a standard pumpjack is up to 3 m. Thus, the improved efficiency of a well may be achieved exclusively owing to the increase in the cycle number resulting in the intensified inertia action on the system elements (rods namely) factoring into breakdown rate increase [1], [2]. The problem is especially topical for deep and very deep wells where inertia action first provokes rod deformation which results in the decrease of actual length of a deep-well pump stroke [2]-[5]. Under such conditions, it is useful to apply well equipment with the significant increase in admissible travel length and simultaneous decrease in the cycle frequency. There is such well equipment [6], [7] where the travel length can be regulated almost unrestrictedly; however, that results in the origination of considerable inertia due to huge weight of the facilities. The changes happen when the system varies its motion direction which factors into the excessive fault rate. That is why it is important to analyze dynamic characteristics of well facilities to improve their design aimed at the inertial decrease giving rise to better reliability and longevity of the equipment.

2. Methods

Analysis of numerous designs of drives of sucker-rod pumping units (SRPUs), developed during last two decades, shows that the key tendency is to increase their travel length. Moreover, such kinematic schemes and patterns of balancing devices are under development which would help lessen dimensions (as well as weight) of the drive; to contract the efforts affecting a sub-structure; and to improve the drive reliability. Improvement of the facilities is followed by the increased number of structures where a hydrostatic power drive is applied which depends upon its high power intensity and simplicity of transformation of rotary movement of a high-speed motor into the slow reciprocating motion of rod hanger. The new design is based upon a mission to develop well equipment for hydrocarbon reserve production by means of rational arrangement of the equipment components (Fig. 1) making it possible to minimize the inertia influence on rod tools of deep-well pumps as well as increase durability and efficiency of the facilities and hydrocarbon production.

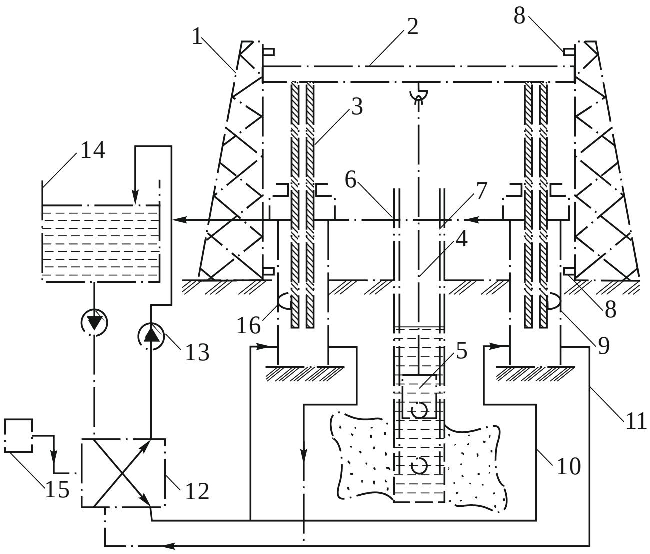

Figure 1. Arrangement scheme of well equipment: 1 – vertical riser; 2 – traverser; 3 – polished connecting rod; 4 – pump rods; 5 – SRP; 6 – production string; 7 – PT string; 8 – limit switch; 9 – hydraulic leg; 11 – blowdown hydraulic line; 12 – spreader; 13 – pump; 14 – tank for liquid; 15 – control desk; 16 – safety valve

The problem is solved as follows: power hydraulic cylinders are manufactured with stepwise increase in the diameter within their upper share; hollow rods in their lower half are equipped with a discharge valve to dump a certain share of liquid into a reservoir through a blowdown hydraulic line to decrease the traverser raise velocity.

It is expedient to start analyzing components of hydraulic drive of the well unit with dividing its operation into phases of motion cycles. To do that, separate eight phases and consider their operating principles separately and in detail.

For the purpose, assume distancing of rod hanger *b* (Fig. 2) from its lower position as the generalized *s* coordinate. It is quite obvious that the *s* coordinate is simultaneously a traverser distancing from the lower switches 8 as well as distancing of the suckers of hydraulic lines (i.e. hydraulic cylinders from their bottom position).

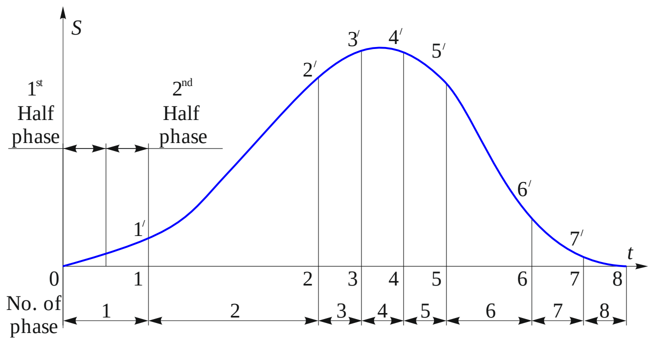

Figure 2. Motion phases of rod hanger point during a cycle

A cycle starts when rod hanger point is transiting from its bottom position. After lower limit switches are activated, a signal is addressed to a spreader (its switching lasts during the certain time period, i.e. 0.05-3.0 seconds) [1], [2]. When the spreader is in its midposition, both hydraulic lines (i.e. pressure line and discharge one) are shut down; velocity of a rod hanger point is equal to zero. In the process of the spreader slide transition from a mid (i.e. neutral) position to the left one, a pressure hydraulic line does open, and the discharge one remains closed. Working liquid goes under the hydraulic cylinder pistons. Traverser as well as rod hanger point starts the upward motion. Phase one is the phase of unsteady motion of point B while its moving up. It lasts from a starting moment of rod hanger point until its steady motion. The phase may be divided into two half phases. Half phase one begins when the rod hanger point starts its motion while finishing at the moment when the velocities of all cross sections of rods become equal.

The phase is stipulated by the fact that the deformations (longitudinal ones in this context) have certain expansion velocity being simultaneously sound velocity within a material.

A moment, within which the efforts in a lower rod section (near a plunger) is such when the plunger starts its upward motion, is the termination of the 1st half stage of rod hanger point motion as well as the initiation of the 2nd stage of its upward motion. Half stage two lasts until the plunger reaches constant velocity; hence, the half stage is the period of SPR plunger acceleration.

Then, phase two (i.e. stable motion phase) approaches. The phase is the longest one. A pump plunger has constant velocity; in this context, well hydrocarbons (i.e. oil) get to the surface. As soon as a traverse activates upper limit switches, a signal is addressed to a control desk and then to a spreader. Switching of the spreader takes a certain time period; spreaders with hydraulic and electrohydraulic control involves a possibility to adjust motion velocity of a slider (reverse time is 0.05-3.0 seconds) [1].

The interval, during which a pressure line becomes closed completely, corresponds to the upward braking period of pistons of hydraulic legs. That is phase three of the plant operation cycle in terms of which the motion velocity of rod hanger point drops from its steady-state value down to zero. Then, the interval takes place when the rod hanger point becomes motionless which corresponds to phase four of the plant operation cycle. The phase duration is minor; it depends upon the spreader design as well as upon its previous adjustment.

After that, a discharge hydraulic leg turns out to be open. Under the weight of a traverse and equipment, connected to it, the discharge hydraulic leg turns out to be open, and downward motion of rod hanger point starts. At the beginning of the downward rod hanger point motion, a pump plunger remains motionless until the extended rods shorten down to their nominal length. The interval between the start of rod hanger point downward motion and start of pump plunger motion is stage five of the plant operation cycle. Phase six continues from the start of downward pump motion until a moment when a traverse activates lower switches. Perhaps, during the stage a motion law of rod hanger point will be neither an interval with constant velocity depending upon the variable resistance of the liquid, pushed out from hydraulic legs, nor the variable resistance of the liquid, moving within the discharge hydraulic line.

When a traverse activates the lower limit switches, a signal is addressed to a control desk and then to a spreader. Its switching takes place. Discharge line shuts down; pressure line is closed too. Pistons within the hydraulic lines start braking along with the braking of rod hanger point. The braking period is rather short. It lasts from the moment when the spreader switching starts until complete shutdown of a discharge line (i.e. the moment when the spreader is in neutral position; in this context, both discharge and pressure hydraulic lines are closed). Braking interval of the rod hanger point is phase seven of the plant operation cycle. Finally, the interval, during which the rod hanger point is motionless in its lower position, is phase eight of the plant operation cycle. The phase period is the time from the moment when the rod hanger point stops till the moment of pressure line opening (i.e. starting point of the plant operation cycle).

3. Results and discussion

Consider the efforts to be studied while analyzing motion law of the rod hanger point in terms of each phase.

Phase one. As it has been mentioned before, the phase is divided into two half stages. During the half stage one, rod reference marks moves up; in this context, the rod string stretches. Tensile force is proportional to the movement of the rod hanger point. A deformation wave spreads down along the rod string at the velocity of sound within the line material. Thus, relative deformation differs depending upon the cross section of the rod string. A deformation process of rod string continues until a plunger of a deep-well pump starts its upward motion (Fig. 3).

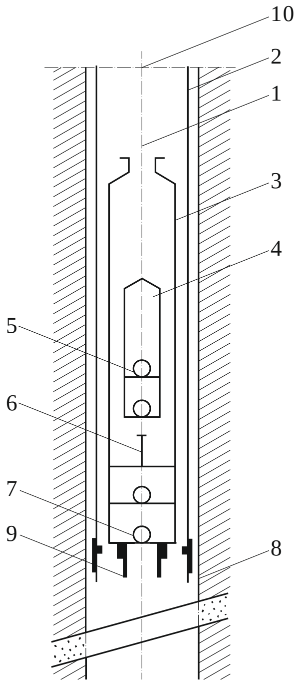

Figure 3. Installation diagram of a rod deep-well pump: 1 – rods; 2 – PT; 3 – sucker rod pump; 4 – plunger; 5 – pressure value; 6 – piston rod of induction valve; 7 – induction valve; 8 – surface casing of a well; 9 – inlet screen; 10 – hanger point of a rod string

During the half phase one, rod hanger point in under the rod weight effect within the formation fluid. When the rod hanger point is moving within the half phase, the force, arising in the cross section of rods in the neighbourhood of the hanger point, depends upon a value of its upward motion from a bottom position. Thus, it depends upon a value of absolute deformation of the rod string; friction force between the formation fluid and F_{T1} rods (the liquid is motionless and cross sections of the rods perform upward motion; and friction force between PT and F_{T2} rods. Paper [8] proposes to consider that during the half phase one, velocity of cross sections of a rod string varies according to linear law along its length. It is v with the hanger point being equal to zero in the neighbourhood of a plunger.

In the context of accurate consideration of motion of the rod hanger points, a wave process, stipulated by upward motion of the rod hanger point at the v velocity, should be regarded. However, a plunger has to be motionless. Moreover, in the process of upward motion of the rod hanger point, the rods get a share of formation fluid weight G_p being:

$$G_p = \frac{sEA_r}{l-h}\,, \tag{1}$$

where:

s – displacement of rod hanger point;

E – elasticity modulus of rod material;

A_r – cross-sectional area of the rods;

l – plunger depth;

h – working submergence under a dynamic level.

Statement (1) ignores finiteness of the deformation wave propagation. It means that relative deformation of the upper and lowers rod cross sections are equal at any time.

It is a significant problem to take into consideration the friction forces between the rods and PT as well as between the rods and formation fluid. It is obvious that friction force F_{T1} between the rods and formation fluid depends heavily upon the formation fluid viscosity. Paper [8] mentions the fact arguing that the force is 200-500 H for low-viscosity oil; as for the viscous oil, the force is close to rod string weight. Moreover, we believe that F_{T1} force should also depend upon the velocity of relative motion between the rods and fluid. Consideration of friction force F_{T2} between the rods and PT is of equal importance. In addition, authors of paper [8] think that the force of mechanical friction between the rod string and PT depends on numerous factors. Consequently, it is impossible to identify the force accurately. Hence, they assume its value as 2-5% of static forces (G_p-G_r') where G_p is weight of fluid column $l-h$. It is also indicated that there is a recommendation to identify the force as follows [9], [10]: $F_{T2} = 0.5f_T$, where f_T – friction coefficient between rods and pipes ($f_{T\max} = 0.25$-0.30); β – inclination of a well axis from a vertical, rad; F_r – rod weight in air.

Consider thoroughly the conditions under which SRP plunger starts its upward motion, i.e. under what conditions deny the half phase one of rod hanger point motion. At the starting point of rod upward motion, both the rod hanger point and a pump plunger are in their lower position. A pressure valve as well as a suction valve is closed.

In the process of the polished connecting rod upward motion, the rod string takes gradually the forces of fluid column weight (Formula 1) starting to be stretched under their effect. Tensile deformation expands towards a plunger. If the plunger is in its lower position, then the fluid column weight is transferred to the SRP string through a plunger and suction valve. Hence, the string turns out to be extended. As soon as the rod hanger point starts its upward motion, increasing force arises within the plunger-fluid contact. The force increases depending upon the rod deformation, i.e. some share of fluid column weight is transferred to the rod string. The abovementioned results in the decrease of the tension force applied to the SRP. That is why SRP tightens, i.e. upward transition of its cross sections, takes place. However, since in this situation a plunger and a unit of the suction valve are in contact, the transition of SRP cross sections involves upward transition of a pump plunger with no plunger transition relative to a cylinder. The process continues until the tensile force, stipulating the rod deformation, becomes equal to the fluid column weight in a well. Upward motion of a plunger and a cylinder reduces the rod deformation. Consequently, the starting point of the plunger transition relative to a cylinder should involve the following rod hanger point transition: $\Delta l_r + \Delta l_{TP}$ (where Δl_r is rod deformation stipulated by force being equal to the fluid column weight above the plunger; and Δl_{TP} is SRP deformation resulting from the same force effect). If transition of s hanger point is: $s = \Delta l_r + \Delta l_{TP}$, then half stage one of the rod hanger point terminates. Δl_r and Δl_{TP} values are determined as follows:

$$\Delta l_r = \frac{G_p l}{EA_r}; \quad \Delta l_{TP} = \frac{G_p l}{EA_{TP}},, \tag{2}$$

where:

A_r and A_{TP} – cross section areas of rods and pipes respectively;

G_p – fluid column weight above the plunger.

Fluid column weight is:

$$G_p = \frac{\pi\left(d_p^{\,2} - d_r^{\,2}\right)(l-h)}{4}\rho g, \tag{3}$$

where:

d_p and d_r – diameters of a plunger and rods;

ρ – density of the formation fluid;

g – acceleration of gravity.

In addition to the abovementioned forces, analysis of law of rod hanger point motion should involve the driving force applied on the part of the polished connecting rod being equal to the total of forces developing in two hydraulic strings (i.e. vertical hydraulic cylinders). Specify the force as F_p. Its value will be:

$$F_p = 2\frac{\pi d_{hc}^2}{4} p, \tag{4}$$

where:

d_{hc} – internal diameter of the hydraulic cylinders;

p – pressure within the hydraulic cylinders.

Hence, during the half stage one, the rod hanger point motion depends upon such forces as G_r', F_{T1}, F_{T2}, G_p', F_p.

While determining a law of the rod hanger point motion, one may do the following: consider conditionally each rod string as strainless considering it also as that one consisting of finiteless number of infinitesimal parts having different velocities and being interconnected by means of elastic elements which weight can be neglected (it has been abovementioned [8] that it is possible to assume approximately a linear law of changes in the velocities of rod cross section from V velocity in a rod hanger point down to zero within a cross section with a pump plunger connection). The same assumption is available in papers [9], [10]. Such an approach will help reduce each force and weight to the rod hanger point (G_r', F_{T1}, F_{T2}, F_p). The only exception is G_p' force applied to a pump plunger since it experiences its 0 up to G_p increase in proportion to the rod string stretching. While reducing, G_p' force will be equal to zero as the plunger velocity amounts to nothing. The force influences only the time of the termination of half phase one of the rod hanger point as well as a value of acceleration with which a pump plunger will start its upward motion.

Resulting from the reduction of forces and weights to the rod hanger point, obtain a differential equation which solution will help obtain $S(t)$ and $F_p(t)$ functions. Next, assume that the force, arising within the rod cross section in the neighbourhood of the plunger, is equal to $F_p(t)$ force within the rod hanger point. Since the formation fluid column presses on the plunger, it is possible to identify the column motion under the effect of $F_p(t)$, force, if one knows $F_p(t)$ function. While solving the differential equation, being formed in this context, obtain t time moment when a pump plunger starts its upward motion, the pump plunger acceleration at the moment, and a value of extra effort within a cross section of a pump plunger-rod string connection.

Another approach is possible. It is as follows: after determination of $S(t)$ function, consider the wave processes within a rod string, and define effort function $F(t)$ within the cross section of rods in the neighbourhood of a plunger. Then, it is required to determine the motion of formation fluid above the pump plunger.

Half phase two. It starts when a pump plunger begins its upward motion terminating when the rod hanger point develops stable (constant) velocity. In terms of the half phase, velocities of all cross sections are almost similar. They are almost similar since at the moment, when a plunger starts its upward motion, significant oscillation processes originate within the rods. The processes depend upon considerable inertia by the fluid column [11]. However, to compare with the displacements, resulting from the motion of rod hanger point, the displacements, stipulated by the oscillations of rod cross sections, are less important. Thus, displacements of cross sections along the whole length of rods can be considered as almost similar ones. It means that the motion of rod string may be regarded as the motion of an absolute solid body.

During the half stage of the motion, following forces are applied for the rod hanger point: rod weight in fluid $G_ø'$; fluid column weight above a plunger G_p; friction force between the formation liquid and rods F_{T1} (the friction force availability is stipulated by the fact that in the process of upward rod motion, formation fluid also performs its upward motion; however, their velocities differ); friction force between rods and SRP F_{T2}; friction force between plunger and cylinder F_{T3}; and pressure force F_{kv} stipulated by the pressure difference within a suction valve (when a pump plunger starts it upward motion, dilution, resulting from pressure difference within a suction valve, originates since the pressure under the suction valve exceeds the pressure over it). Difference of the pressures, multiplied by the plunger area develops F_{kv} force applied for the plunger and directed downwards.

Determination of F_{kv} force should involve the data concerning pressure loss within a suction valve. Papers [11]-[13] analyzed the problem. To determine pressure loss within a suction valve, authors of paper [12] applied the formula:

$$h_{los} = \frac{1}{\mu^2}\frac{A_p^2}{A_k^{\ 2}}\frac{V_p^2}{2g}, \tag{5}$$

where:

μ – loss coefficient of a valve which depends upon the valve design and Reynolds number;

A_p – plunger area;

A_k – valve hole area;

V_p – plunger velocity.

Perform roughly approximate calculation using Formula (5) to identify whether F_{kv} force is a significant factor while analyzing a motion law of the rod hanger point. Obtain F_{kv} =17 H for the averaged parameter values. As it is understood, the force is not important; thus, it may be ignored.

Formula (5) demonstrates that pump plunger acceleration will result in rather fast increase of the force. Papers [12], [14], [15] identify h_{los} to determine the minimum allowable value of the working submergence ($h_{\min}$).

Moreover, a driving force F_p is applied for the rod hanger point. The force is defined according to Formula (4). Inertia of all the moving bodies is taken into consideration in the underwritten equation of motion of rod hanger point.

To identify friction between the rods and pipes F_{T2}, paper [11] applies a formula with the reference to paper [9]:

$$F_{T2} = 0.5 f \beta_l g \rho_r A_r l, \tag{6}$$

where:

ρ_0 – rod material density;

f – rods-pipes friction coefficient;

β_l – angle characterizing curvature of vertical wells (it is mentioned that in the context of the majority of wells, the curvature is not more than 5-6°; β_l = 0.1 may be assumed).

Formula (6) coincides completely with Formula (2). Papers [9], [10] assert that friction coefficient f is not more than 0.2 if dry steel-steel friction takes place or insufficient lubrication is available. However, if the coefficient is applied in excess as f = 0.4, then, when β_l = 0.1, we obtain $F_{T2} = 0.02G_ø$ using (6) where $G_ø$ is the rod weight in air.

Hence, friction force between SRP and rods is not more than 2% of the rod weight in air. Also, paper [11] gives recommendations concerning the selection of a friction coefficient f:

– f = 0 for watered oil which viscosity is 10^{-6}-10^{-5}m^2/s;

– f = 0.2 for light oil which viscosity is less than $3\cdot10^{-5}$ m^2/s;

– f = 0.16 for light oil which viscosity is more than $3\cdot10^{-5}$ m^2/s.

The friction force between a plunger and a cylinder F_{T3}, mentioned in paper [11], is a significant value which may achieve 3 kN. Largely, the force depends upon pump diameter, fluid viscosity, availability of the suspended solid particles in it, and gap between a plunger and a cylinder. Theoretically, it is a great problem to identify the force. The idea is supported by papers [9], [10]. In this connection, following empiric formulas were proposed:

$$F_{T3} = 1.84\frac{D}{\delta} - 137, \tag{7}$$

if water lubrication takes place.

$$F_{T3} = 1.65\frac{D}{\delta} - 127, \tag{8}$$

if lubrication is performed using grease which characteristics are close to circuit-breaker oil,

where:

D – plunger diameter;

δ – gap between a plunger and cylinder.

To consider friction force between the formation fluid and rods, paper [11] applies A.V. Klapan formula. As it is stated in paper [11], the formula has been obtained involving average fluid consumption per one cycle of upward and downward plunger motion. It is as follows:

$$F_{T1} = 2\pi\nu_p \rho l\left(\pm\pi m S_{lp} A_1 - UB_1\right), \tag{9}$$

where:

l – rod string length; and n is plunger stroke frequency:

$$A_1 = \frac{\left(m^2-1\right)+4\dfrac{l_p m}{m^2-1}-2}{\left(m^2+1\right)l_p m-\left(m^2-1\right)}; \quad B = \frac{\left(m^2-1\right)-2l_p m}{\left(m^2+1\right)l_p m-\left(m^2-1\right)};$$

$$U = \frac{8Qp}{\left(m^2+1\right)l_p m-\left(m^2-1\right)}; \quad m = \frac{D_{tv}}{d_r}.$$

Paper (9) proposes to take "+" sign if plunger performs its upward motion and "–" sign if downward motion takes place.

There is one more friction force stipulated by the hydraulic resistance in pipes. Specify it as F_{T4}. The force intensifies fluid pressure on a pump plunger. To determine it, pressure loss in pipes should be multiplied by a plunger area, i.e.:

$$F_{T4} = \lambda\frac{l}{d_T}\frac{\rho V_{av}}{2}A_p, \tag{10}$$

where:

λ – hydraulic friction coefficient in SRP (Darcy factor);

d_T – internal SRP diameter;

V_{av} – average velocity of the formation fluid in SRP in terms of upward plunger motion;

ρ – formation fluid density;

A_p – cross section area of a plunger.

In addition to the abovementioned forces, arising at the starting point of a half phase two when a pump plunger begins its upward motion, a vibration force, developed by the inertia fluid pressure of a plunger, originates. The force value was identified by A.S. Vyrnovskyi [9], [10]. Below you can find the problem statement as well as its solving. Vibration force value will be determined separately from the effort value in rods arising in terms of upward motion of the rod hanger point under the effect of the abovementioned forces.

Hence, following forces should be taken into consideration during the half stage two of the rod hanger point motion to identify its movement: rod weight in fluid $G_ø'$; fluid column weight above plunger G_p; friction force between the formation fluid and rods F_{T1}; friction force between the rods and SRP F_{T1}; friction force between a plunger and a cylinder F_{T3}; pressure force resulting from the pressure difference in a suction valve F_{kv}; friction force stipulated by hydraulic resistance in SRP when the formation fluid performs its upward motion; and drive force F_P stipulated by the fluid pressure on pistons in the hydraulic legs.

Consideration of the abovementioned forces will help compile an equation of the rod hanger point motion within the half phase. An equation to identify vibration force will be compiled separately.

Phase two starts when rod hanger point achieves its maximum and simultaneously constant velocity. As it has been mentioned, the phase is the longest in terms of time. The phase terminates when a traverse activates the upper terminal switches.

The phase involves the same forces as the half two of stage one (i.e. running start of a rod hanger as well as a plunger). Since the rod hanger point velocity is V = const within stage two, efforts in the rods are determined by means of the applied forces mentioned above. Some of them may be considered as the concentrated forces; and other may be considered as those distributed along the rod string. The concentrated forces are: fluid column weight G_p above a plunger; friction force between a plunger and a cylinder; resistance force resulting from the pressure difference in a suction channel F_{kv} (as it has been mentioned above, the force may be ignored); and driving force F_P. In turn, friction forces F_{T1}, F_{T2}, F_{T4} are the forces, distributed along the rod string. While adding vibration force to the listed ones, it is possible to identify a normal force within the arbitrary cross section of the rod string as well as the stress within the cross section.

Phase three involves brakeage of piston motion within the cylinders of hydraulic legs as well as brakeage of the motion of rod hanger point. The phase starts when a traverse activated upper terminal switches. In this context, it is addressed to switch a spreader. It closes down a pressure line; thus, motion velocity of pistons within the cylinders of hydraulic legs drops to zero.

In terms of the phase, motion of the rod hanger point is determined with the help of the following forces:

– driving force F_p applied to the rod hanger point on the side of a traverse;

– weight of the formation fluid G_p lifted by a pump plunger;

– weight of the rod string within the formation fluid G_r';

– friction forces F_{T1}, F_{T2}, F_{T3}; F_{T4}.

The listed forces explain that within the phase, the rod hanger point is influenced by the same forces as within phase two. The only difference is that driving force F_p cannot provide the uniform upward motion of the rod hanger point anymore since the fluid delivery under pistons within the hydraulic legs deceases gradually down to zero resulting in the brakeage of piston motion as well as rod hanger point.

Phase four is a short interval during which both pressure line of the hydraulic system and discharge one are closed down. Balance of forces takes place during the interval.

Phase five lasts from the start of downward motion of the rod hanger point up to the moment when a pump plunger begins its downward motion. The phase involves shortening of the stretched rod string with a plunger. That is why weight of the formation fluid, located above the plunger, transfers gradually through the fluid under the plunger and the closed down suction valve to PT string being stretched. During the opening point of the pump plunger downward motion, forces of inertia masses, starting their movement, develops an impulse, stipulating an oscillation process within the rod string. The process is comparable with that one arising when a pump plunger starts its upward motion. Later, the fluid mass above a plunger is not connected with it. During phase five, no plunger motion relative to a cylinder is available since PT stretch results in the simultaneous rod string elongation in its lower part (PT stretch under a plunger develops discharge resulting in instantaneous pressing of the plunger against fluid under it owing to the weight of rod string and certain share of the fluid column weight). Downward motion of a plunger along a cylinder is only possible if only the total weight of fluid column G_p above the plunger is taken up by PT string. It will become possible when absolute deformation of rod string becomes Δl_r (Formula one in (4)). However, simultaneous compression of the stretched rod string, depending upon the downward displacement of rod hanger, results in rod stretch in the neighbourhood of a plunger, stipulated by PT extension. Thus, to make absolute rod deformation equal to Δl_r, the hanger point should be displaced by $S = \Delta l_r + \Delta l_{TP}$ value.

During the phase, motion law of the rod hanger point depends on the following forces: weight of rod string in formation fluid G_r'; traverse weight together with piston rods and plungers of hydraulic legs G_{Tr}; resistance force of fluid within discharge line F_{op}; and friction forces F_{T1}, F_{T2}.

Phase six. It involves the interval during which the rod hanger point performs its downward motion from the moment when a plunger starts its downward motion up to the moment when a traverse activates the lower switches, i.e. before closing down of a discharge line.

In this context, motion of rod hanger point depends upon the following forces:

– rod string weight in the formation fluid G_r';

– traverse weight together with the piston rods and plungers of hydraulic legs G_{Tr};

– resistance force of the fluid flowing out of the hydraulic legs into a discharge leg F_{op};

– friction forces F_{T1}, F_{T2}, F_{T3}, F_{T4}.

– resistance force resulting from the pressure difference within a pressure valve F_{kn} (the force may be ignored as well as F_{kv} force).

Weight of rod string within the formation fluid is:

$$G_r' = A_r gl\left(\rho_r - \rho\right), \tag{11}$$

where:

ρ_r, ρ – densities of rod material and formation fluid

Phase seven. It involves the interval from the moment when a discharge line of hydraulic line is closed down up to the moment when the rod hanger point stops. During the phase, motion of the rod hanger point depends upon the following forces [16], [17].

– resistance force of the fluid flowing out of the hydraulic legs into a discharge line F_{op};

– friction forces F_{T1}, F_{T2}, F_{T3};

– traverse weight together with the piston rods and plungers of hydraulic legs G_{Tr}.

The difference between the phase forces and forces of phase six is in the following: the force, resisting to the fluid flowing out of the hydraulic legs, increases since a discharge line is closed down.

Phase eight. As it has been mentioned before, the phase involves the interval during which the rod hanger point remains immovable in its lower position. In this context, the mine well pump valves are closed down. The efforts within the rod hanger point is equal to the rod string weight within the formation fluid G_r'. It is obvious that phase eight is not over when opening of a pressure line of a hydraulic drive starts. To initiate upward motion of the rod hanger point under pistons, such pressure should be developed in the hydraulic legs to provide upward motion of a traverse together with the piston rods and plungers [18]-[20].

4. Conclusions

The analyzed dynamics of well facilities producing oil has made it possible to evaluate the values of forces influencing the system which is quite important for the process of the equipment design. The abovementioned has helped correct the available kinematic schemes of well tools for oil production intended to reduce the effect of inertia on the equipment components potentiating development of conditions for the improvement of reliability and durability of the operation of rod well pump plants.

Evaluation of efforts, effecting the rod system while cycle implementing, will make it possible to identify stress value in the rods as well as dynamics of their changes helping develop functional as well as mode and parametric measures to prevent fatigue destruction of rod material; finally, it will provide the improved operational reliability of well facilities for oil production.

Acknowledgements

The authors express thanks to Ye. Kryzhanivskyi, Doctor of Engineering, Professor, Academician of the NASU, Rector of Ivano-Frankivsk National Technical University of Oil and Gas for scientific advice and the material formation.

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Динамічний аналіз свердловинного обладнання для видобутку нафти

Я. Грудз, В. Грудз, В. Бевз, М. Чернецький

Мета. Дослідження динаміки технологічного циклу установки свердловинного нафтовидобувного обладнання з метою оцінки сил, що діють на елементи конструкції в процесі роботи для підвищення їх надійності і довговічності експлуатації за рахунок зниження інерційності.

Методика. В основу математичного моделювання системи покладено основний закон динаміки руху складної системи з приєднаною масою з урахуванням деформації елементів і з подальшою реалізацією математичної моделі методами математичного аналізу. Для підвищення інформативності та достовірності отриманих в процесі аналітичного моделювання результатів весь цикл технологічного процесу запропоновано розбивати на окремі фази, кожна з яких характеризує певний процес руху і зміна характеру дії сил, що впливають на елементи системи.

Результати. Аналіз результатів математичного моделювання робочого циклу системи дозволяє зробити висновки про тривалість процесу, характер руху точки підвісу колони протягом циклу експлуатації та величини інерційних сил, що діють на елементи установки свердловинного обладнання, вказуючи шляхи їх скорочення, забезпечить надійність і довговічність роботи установки. Детально досліджено елементи гідравлічного приводу насосної установки шляхом розбиття його роботи на 8 фаз циклів руху. Встановлено, що зменшення впливу інерційних сил на елементи системи досягається наступним чином: силові гідравлічні циліндри виконуються з ступінчастим збільшенням діаметра у верхній їх частині, а порожнинні штоки в нижній частині обладнані скидним, який здійснює скидання частини рідини в резервуар, зменшуючи при цьому швидкість підйому траверси.

Наукова новизна. Встановлено за допомогою математичного моделювання, що істотне зменшення інерційності системи залежить від конструктивних і кінематичних її характеристик і може змінюватися в широких межах.

Практична значимість. У практиці експлуатації установки свердловинного обладнання для видобутку нафти зниження впливу інерційних сил на елементи системи дозволить підвищити її надійність і довговічність.

Ключові слова: *свердловинне обладнання, видобуток нафти, динаміка, силове поле, інерційність, робочий цикл, фаза, надійність*

Динамический анализ скважинного оборудования для добычи нефти

Я. Грудз, В. Грудз, В. Бевз, М. Чернецкий

Цель. Исследование динамики технологического цикла установки скважинного нефтедобывающего оборудования с целью оценки сил, действующих на элементы конструкции в процессе работы для повышения их надежности и долговечности эксплуатации за счет снижения инерционности.

Методика. В основу математического моделирования системы положен основной закон динамики движения сложной системы с присоединенной массой с учетом деформации элементов и с последующей реализацией математической модели методами математического анализа. Для повышения информативности и достоверности полученных в процессе аналитического моделирования результатов весь цикл технологического процесса предложено разбивать на отдельные фазы, каждая из которых характеризует определенный процесс движения и изменение характера действия сил, влияющих на элементы системы.

Результаты. Анализ результатов математического моделирования рабочего цикла системы позволяет сделать выводы о продолжительности процесса, характер движения точки подвеса колонны в течение цикла эксплуатации и величины инерционных сил, действующих на элементы установки скважинного оборудования, указывая пути их сокращения, обеспечит надежность и долговечность работы установки. Подробно исследованы элементы гидравлического привода насосной установки путем разбития его работы на 8 фаз циклов движения. Установлено, что уменьшение влияния инерционных сил на элементы системы достигается следующим образом: силовые гидравлические цилиндры выполняются с ступенчатым увеличением диаметра в верхней их части, а полостные штоки в нижней части оборудованы сбросным клапаном, осуществляющий сброс части жидкости в резервуар, уменьшая при этом скорость подъема траверсы.

Научная новизна. Установлено при помощи математического моделирования, что существенное уменьшение инерционности системы зависит от конструктивных и кинематических ее характеристик и может изменяться в широких пределах.

Практическая значимость. В практике эксплуатации установки скважинного оборудования для добычи нефти снижение влияния инерционных сил на элементы системы позволит повысить ее надежность и долговечность.

Ключевые слова: *скважинное оборудование, добычи нефти, динамика, силовое поле, инерционность, рабочий цикл, фаза, надежность*

Article info

Received: 12 May 2020
Accepted: 16 November 2020
Available online: 7 December 2020