Research Concerning the Identification of Some Parameters of a Sucker Rod Pumping Unit

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One of the solutions to reduce the production and maintenance costs of the sucker rod pumping installations is to develop automated systems for regulating and controlling their operations. The development of these automated systems requires an attentive modeling of the dynamics of the mechanism of the pumping unit, process in which the identification of the values of the parameters involved in the calculations plays an essential role. The paper presents the manner of determining the values of some parameters of the mechanism of a C-320D-256-100 pumping unit starting from the variation on a cinematic cycle of the motor torque at the crank shaft. Simulations were performed with a computer program developed by the authors, and the experimental records were processed with the program Total Well Management.

Keywords: sucker rod pumping unit, parameters identification, dynamics, motor torque

Reducing production costs and increasing the efficiency of the sucker rod pumping installations are major objectives for the exploitation of any oil well. Considering that most of the oil production is extracted by pumping [1], special attention has been paid in the last period to the functioning automating of these installations. One of the key points in the development of these regulating and control systems of the operation of these installations is the modeling of the dynamics of the mechanism of the pumping unit and of the dynamics of the sucker rod column. In the development of dynamic models in these two situations the identification of the values of the parameters involved in the calculations plays an essential role.

Some of the first significant results regarding the behavior of the sucker rod pumping systems during operation, the dynamics of the sucker rod column and the kinematics and dynamics of the mechanism of the pumping units are presented in [2-5]. Also, a number of more recent results that have strongly helped to the achievement of the research from this paper are presented in [6-9].

In this paper is presented a way of identifying the values of some parameters of the mechanism of a C-320D-256-100 pumping unit using the variation on a cinematic cycle of the motor torque at the crank shaft. In establishing the motor torque variation were considered the inertial forces and the inertial moments acting on the components of the pumping unit mechanism along with their corresponding weight forces and the force at the polished rod. Simulations were performed with a computer program developed by the authors using *Maple* programming environment. The identification of the parameters values has been accomplished by searching an optimum regarding the proximity between the experimental records and the values obtained with the simulation program. In this scope has been used *NLPSolve* function included in the *Optimization* package of Maple.

Experimental part

The experimental records have been processed with the program Total Well Management (TWM) [10]. The analyzed well is serviced by a C-320D-256-100 pumping unit manufactured by Lufkin (fig. 1). The driving motor is of type Toshiba NEMA D.

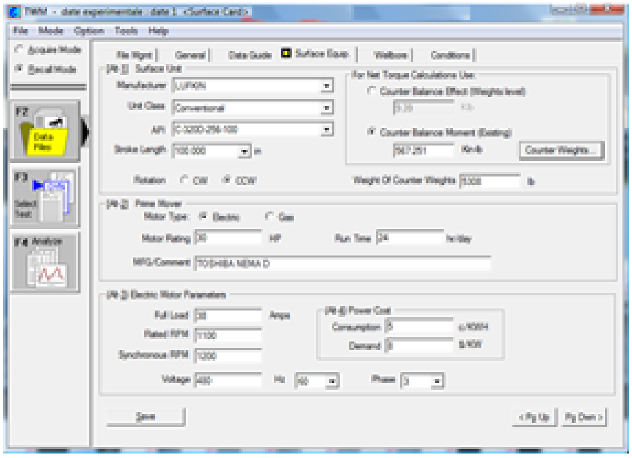

Fig. 1. Data on the surface equipment of the analyzed well

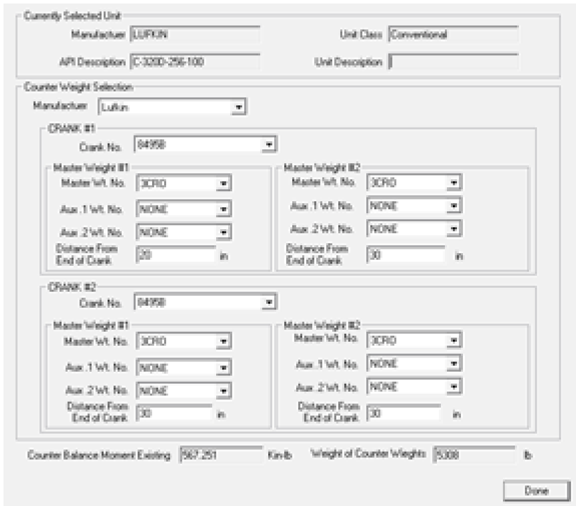

Fig. 2. Data on the counterweights

On the cranks of the pumping unit are mounted four identical counterweights (fig. 2). One of them is mounted at 20 in. (0.508 m) from the end of the crank and the other three are at 30 in. (0.762 m) from the end of the crank. The total weight of the four counterweights is of 5308 lb (23611 N).

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The variation on a cinematic cycle of the motor torque at the crank shaft M_m was obtained starting from the records concerning the variation of the power P [kW] of the driving motor, using the following relation given in [10]:

$$M_m = \frac{84.5 \cdot P \cdot EFF}{SPM \cdot SV} \qquad (1)$$

where:

-EFF is a parameter that consider the efficiency of power conversion by the motor and of the power transmission through the belt drive and the gear reducer, whose value for a normally loaded and properly installed system has been estimated at 0.8 [10];

-SPM is the instantaneous cranks speed. In the calculation SPM is assumed to be constant and in the analyzed case has the value of 8.6 rot/min;

-SV is a factor that takes into account the motor's speed variation during the stroke and it is calculated based on the motor's performance characteristics as entered in the well file [10]. In the analyzed case SV has the value of 0.91.

In figures 3 and 4 are presented the variation of the power of the driving motor for two consecutive strokes (6 and 7) of the sucker rod column.

In establishing the variation on a cinematic cycle of the motor torque at the crank shaft using the simulation program mentioned before were used the records concerning the variation of the force at the polished rod for the two consecutive strokes 6 and 7 (fig. 5 and 6).

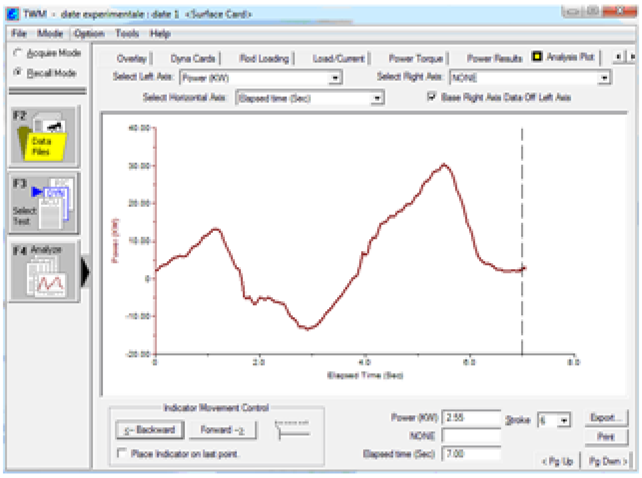

Fig. 3. The variation of the power of the driving motor during the stroke 6

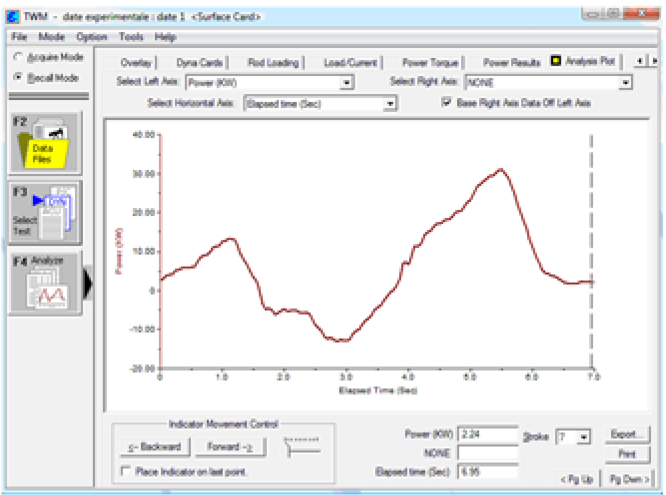

Fig. 4. The variation of the power of the driving motor during the stroke 7

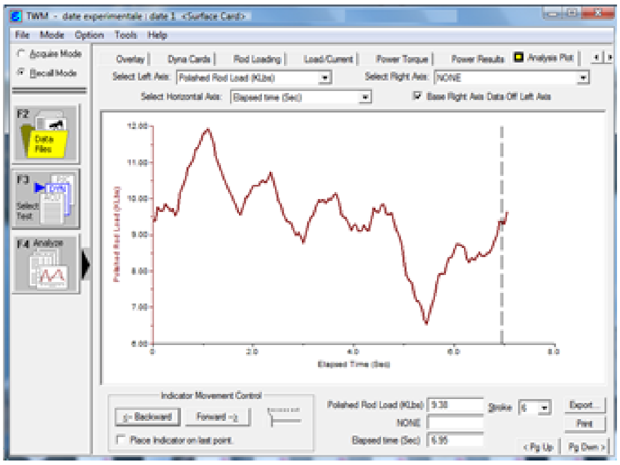

Fig. 5. The variation of the force at the polished rod during the stroke 6

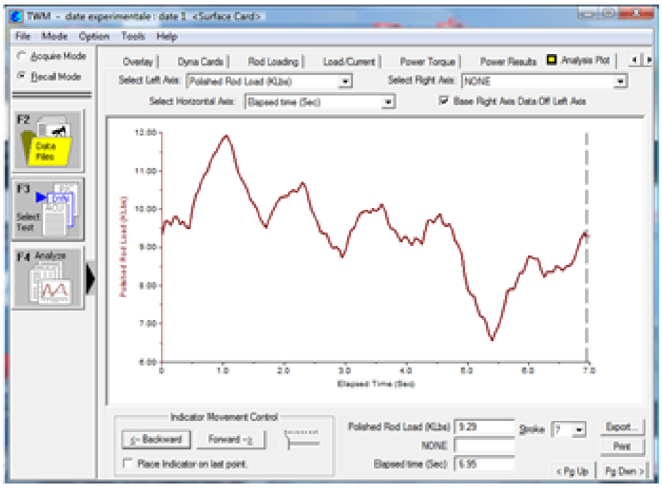

Fig. 6. The variation of the force at the polished rod during the stroke 7

Establishing by calculation of the motor torque at the crank shaft

In figure 7 it is represented the mechanism of a sucker rod pumping unit with conventional type geometry. With C_1, C_2 and C_3 are noted the mass centers of the cranks, connecting rods and of the rocker, respectively. m_{CGRi}, $i=1,4$, are the masses of the balancing counterweights, m_{L1} is the total mass of the connecting bearings between the cranks and the connecting rods, m_{L2} is the mass of the spherical connecting bearing between the connecting rods and the rocker, m_{tr} is the mass of the equalizer traverse. With m_{CB} is noted the mass of the rocker head considered to have the mass center in point *N*.

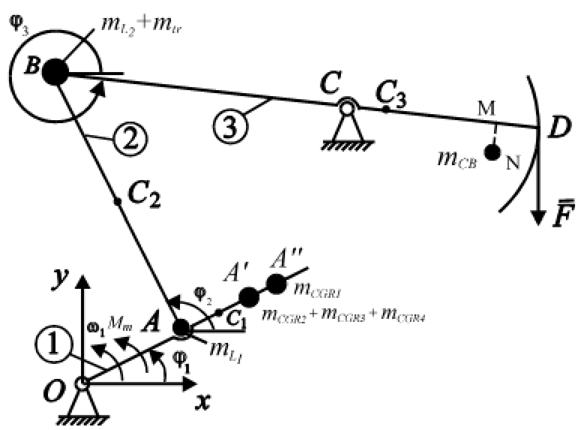

Fig. 7. The mechanism of a pumping unit with conventional type geometry

The motor moment at the cranks shaft M_m is calculated by expressing the dynamic equilibrium in instantaneous powers of all external and inertial forces and moments that work on the mechanism [6]:

$$P_m + P_g + P_i + P_F = 0 \tag{2}$$

where:

$$P_m = \overline{M}_m \cdot \overline{\omega}_1 \tag{3}$$

is the instantaneous power developed by the driving motor, ω_1 being the angular speed of the cranks expressed in rad/s;

$$P_g = \sum_{j=1}^{3} \overline{G}_j \cdot \overline{v}_{C_j} + \overline{G}_{CGR1} \cdot \overline{v}_{A''} + (\overline{G}_{CGR2} + \overline{G}_{CGR3} + \overline{G}_{CGR4}) \cdot \overline{v}_{A'} + \\ + \overline{G}_{L1} \cdot \overline{v}_A + (\overline{G}_{L2} + \overline{G}_{tr}) \cdot \overline{v}_B + \overline{G}_{CB} \cdot \overline{v}_N \tag{4}$$

represents the instantaneous power corresponding to the weight of the components that are part of the pumping unit mechanism: $\overline{G}_j = m_j \cdot \overline{g}$; $j=\overline{1,3}$ in which m_1, m_2 and m_3 are the masses of the cranks, connecting rods and of the rocker, respectively, and $\overline{g}$ is the gravitational acceleration vector that has the same direction as (Oy) axis, but to the contrary of this axis (fig. 7); $\overline{v}_{C_j}$; $j = \overline{1,3}$, are the speeds of the mass centers of the cranks, connecting rods and of the rocker, respectively; $\overline{G}_{CGRi} = m_{CGRi} \cdot \overline{g}; i = \overline{1,4}$; $\overline{v}_{A'}$ and $\overline{v}_{A''}$ are the speeds of the points where the mass of the counterweights are concentrated; $\overline{G}_{L1} = m_{L1} \cdot \overline{g}$; $\overline{G}_{L2} = m_{L2} \cdot \overline{g}$; $\overline{G}_{tr} = m_{tr} \cdot \overline{g}$; $\overline{G}_{CB} = m_{CB} \cdot \overline{g}$ and $\overline{v}_A$, $\overline{v}_B$ and $\overline{v}_N$ are the speeds of the points where these weight forces acting;

$$P_i = \sum_{j=1}^{3} (\overline{F}_{ij} \cdot \overline{v}_{C_j} + \overline{M}_{ij} \cdot \overline{\omega}_j) + \overline{F}_{iCGR1} \cdot \overline{v}_{A''} + (\overline{F}_{iCGR2} + \overline{F}_{iCGR3} + \overline{F}_{iCGR4}) \cdot \overline{v}_{A'} + \\ + \overline{F}_{iL1} \cdot \overline{v}_A + (\overline{F}_{iL2} + \overline{F}_{itr}) \cdot \overline{v}_B + \overline{F}_{iCB} \cdot \overline{v}_N \tag{5}$$

is the instantaneous power corresponding to the inertial forces and moments that work on the mechanism components: $\overline{F}_{ij} = -m_j \cdot \overline{a}_{C_j}; j = \overline{1,3}$, are the inertial forces corresponding to the cranks, connecting rods and to the rocker, respectively, where: $\overline{a}_{C_j}; j = \overline{1,3}$, are the accelerations of their mass centers; $\overline{M}_{ij} = -J_{Cj} \cdot \overline{\varepsilon}_j; j = \overline{1,3}$, are the inertial moments corresponding to the cranks, connecting rods and to the rocker, respectively, where: $J_{C_j}; j = \overline{1,3}$, represent their mass moments of inertia and $\overline{\omega}_j, \overline{\varepsilon}_j; j = \overline{1,3}$, are their angular speed and accelerations; $\overline{F}_{iCGR1} = -m_{CGR1} \cdot \overline{a}_{A''}$, $\overline{F}_{iCGRi} = -m_{CGRi} \cdot \overline{a}_{A'}, i = \overline{2,4}$, where $\overline{a}_{A'}$ and $\overline{a}_{A''}$ are the accelerations of the points where the mass of these counterweights are concentrated; $\overline{F}_{iL1} = -m_{L1} \cdot \overline{a}_A$, $\overline{F}_{iL2} = -m_{L2} \cdot \overline{a}_B$, $\overline{F}_{itr} = -m_{tr} \cdot \overline{a}_B$ and $\overline{F}_{iCB} = -m_{CB} \cdot \overline{a}_N$, where: $\overline{a}_A$, $\overline{a}_B$ and $\overline{a}_N$ are the accelerations of the points where these inertial forces acting;

$$P_F = \overline{F} \cdot \overline{v}_D \tag{6}$$

is the instantaneous power corresponding to the force $\overline{F}$ at the polished rod, $\overline{v}_D$ is the speed of the point where this force acting (fig. 7).

The manner of determining the positional and kinematical parameters (the angles φ_2 and φ_3 (fig. 7), the coordinates of the points where acting the forces and moments occurring in relations 3÷6 and the speeds and accelerations of these points) depending on the dimensions of the component elements of the pumping unit, the crank angle φ_1 and the angular speed of the cranks ω_1 is presented in [8,9]. In [9] it is also presented the manner of determining the surface stroke of the pumping unit and of the crank angles φ_{1d} and φ_{1a} corresponding to the beginning of the upward and downward movements of the sucker rod column.

Simulation results and discussions

The motor torque M_m calculation methodology presented before has been transposed by the authors into a computer program using Maple programming environment [11].

The dimensions of the component elements of a C-320D-256-100 pumping unit (fig. 7) produced by *Lufkin* [12] are: *OA* = 42 in. (1.0668 m); *AB* = 132 in. (3.3528 m); *BC* = 111.07 in. (2.8211 m); *CD* = 129 in. (3.2766 m). The coordinates of the point *C* (fig. 7) are [12]: x_C=111in (2.8194 m) and y_C =136in. (3.4544 m). The values of the crank angles φ_{1d} and φ_{1a} are: 87.83° and 264.02°, respectively.

The simulations have been accomplished by considering the following values of the other parameters involved in motor torque calculation: *BM* = 210 in. (5.334 m); *MN* =40 in. (1.016 m); m_{L1} = 55 kg; m_{L2} = 125 kg; m_{tr} = 400 kg; m_{CB} = 443 kg; q_1 = 566 kg/m; q_2 = 25.7 kg/m; q_3 = 217.5 kg/m (q_1, q_2 and q_3 are the linear masses of the cranks, connecting rods and of the rocker, respectively). The results obtained for the two strokes 6 and 7 (curves *2* in figs. 8 and 9) show relatively large differences compared to the results established starting from the records concerning the variation of the power of the driving motor (curves *1*).

So, it has been sought finding an optimum regarding the proximity between the experimental results and the values obtained with the simulation program by identifying of some parameters values involved in motor torque calculation. In this scope has been used *NLPSolve* function included in the *Optimization* package of *Maple* [11].

It has been considered that: *BM* varies between 145 in. (3.683 m) and 230 in. (5.842 m); *MN* varies between 10 in. (0.254 m) and 48 in. (1.2192 m); m_{CB} varies between 415 kg and 535 kg; q_1 varies between 275 kg/m and 600 kg/m and q_3 varies between 165 kg/m and 245 kg/m.

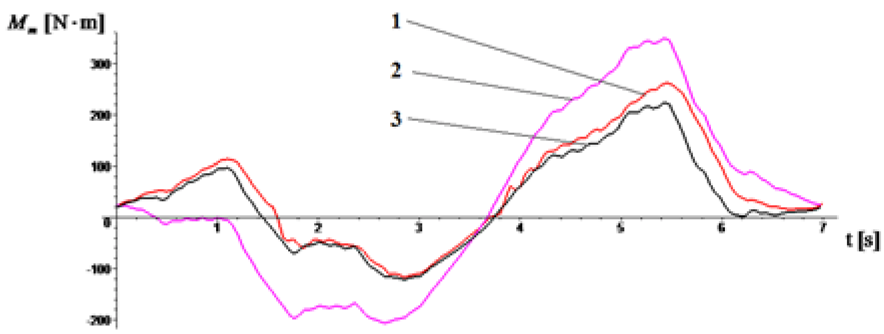

Fig. 8. The variation of the motor torque during the stroke 6 (experimental - curve *1*; initial simulation - curve *2*; simulation after parameter identification - curve *3*)

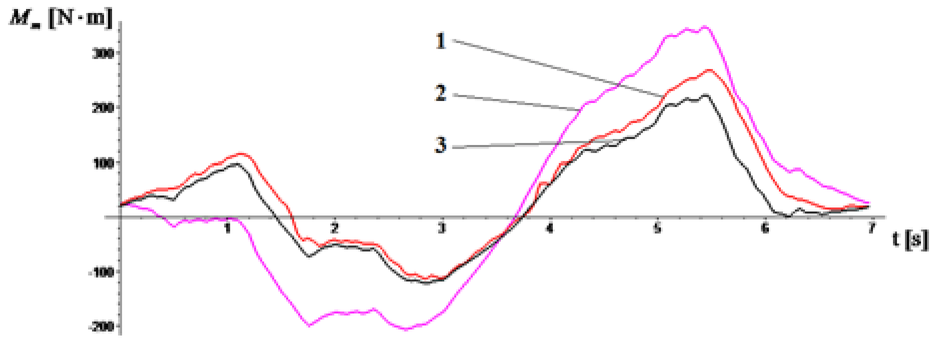

Fig. 9. The variation of the motor torque during the stroke 7 (experimental - curve *1*; initial simulation - curve *2*; simulation after parameter identification - curve *3*)

After performing calculation, the values of the above parameters that ensure the best proximity between the experimental results and those obtained with the simulation program are the following: BM = 4.569 m; MN = 0.698 m; m_{CB} = 499.7 kg; q_1 = 299.89 kg/m; q_3 = 214.9 kg/m. The simulation results obtained for the two strokes 6 and 7 by considering these values of BM, MN, m_{CB}, q_1 and q_3 are presented in figures 8 and 9 (curves *3*).

Figures 8 and 9 highlight a good concordance between the variation of the motor torque established starting from the records concerning the variation of the power of the driving motor and the results obtained with the simulation program after identification of some parameter. The differences that appear, especially towards the end of the strokes, are largely due to the values of the parameters used for the calculus of the variation of the motor torque starting from the records concerning the variation of the power of the driving motor with relation (1) and to the assumption that the angular speed of the cranks is constant during the operation cycles.

Conclusions

This paper presents a way of identifying the values of some parameters of the mechanism of a C-320D-256-100 pumping unit using the variation on a cinematic cycle of the motor torque at the crank shaft. Simulations were performed with a computer program developed by the authors using *Maple* programming environment. For identification purposes has been used the *NLPSolve* function included in the *Optimization* package of *Maple.* The results obtained after identification show a good accordance with those obtained from the records concerning the variation of the power of the driving motor.

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