ASPECTS REGARDING THE ANALYSIS OF A CRUDE OIL EXTRACTION SYSTEM WITH HELICAL PROGRESSIVE CAVITY PUMPS

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Abstract: Increased global crude oil demand, correlated with the escalation of commercial disputes, political issues in the key-emerging economies of the major exporting countries, inevitably lead to a reduction in production. This calls for the implementation of more stringent markets, increased prices for crude oil and derivatives, which may be merged with increased production through technical solutions that maintain the extraction quota, one of which is increasing the recovery factor. Stimulating the productivity of the probes, conjugated with the extraction of high flow rates when the reaction probes are flooded, contributes to the increasing of the recovery factor. In this respect, the paper analyzes problems regarding crude oil extraction with helical progressive cavity pumps. An analysis of the helical pump extraction system is carried out, showing the pumping plant, which includes bottom equipment and surface equipment, a mechanical calculation of the pumping unit depending on a series of initial data. Also, an analysis of stability loss is presented by comparative calculation using two finite element modelling softwares, Ansys and Cosmos, for a sucking rod pumping unit, which was loaded with a torque and, in turn, with three compression forces.

Keywords: Crude oil extraction, helical PCP, sucker-rod pumps, finite element analysis (FEA)

1. Introduction

The growing energy requirements of the future, especially those related to the hydrocarbon extractive industry, require the best technical management decisions, as well as the development and use of state-of-the-art technologies for the best possible global exploitation quota to reduce energy consumption and without affecting the environment [1, 2], which is also achievable by increasing the crude oil recovery factor from deposits [3].

Current crude oil extraction technologies involve multiple pumping technologies, the most common ones being: piston pumping, centrifugal pumping, pumping with hydraulic jet pumps, and helical pumping [4, 5, 6]. Piston pumping is carried out using surface pumping units with or without beam (pneumatic, hydraulic or mechanical) inserted into the probe and driven from the surface through the pump seal, Figure 1 [7].

Centrifugal pumps are built for a wide range of flows, ranging from 30-40 m³/day to 6000-8000 m³/day, depending on the diameter of the column into which it is inserted. Submersible centrifugal pumps are especially used for large fluid flows and recommended for highly-viscous crude oil [8, 9]. The jet pump is a hydraulic pump without a rotating piston or other moving parts that works on the principle of the ejector - pumps the fluid produced by the probe to the surface due to an energy transfer from the drilling fluid to the fluid produced by the probe. The high-pressure drilling fluid enters the pump at its top and then passes through a nozzle, where the entire pressure of the fluid is converted into kinetic energy, the fluid gaining a very high velocity and consequently a low static pressure. The aspiration of the fluid in the probe is produced due to this depression, Figure 2 [10].
The helical pump plant comprises bottom equipment and surface equipment. Bottom equipment consists of the submersible helical pump, extraction pipes and sucker-rod pumps, and the surface equipment comprises the actuation system of the sucker-rod pumps, the rotor of the pump, the coupling between the actuation system and the drive-head, the drive-head and support system for all bottom equipment [11, 12, 13, 14].

2. The helical PCP plant

The helical PCP is composed of an elastomeric stator and a PCP rotor, usually made of stainless steel or high alloy.

The rotors of these pumps are bodies of work whose composite profiles include circular arcs, line segments, cycloidal curves, and often non-analytical curves.

Such a geometry of the profiles confer the gear processes superior characteristics, to reduce friction and improve sealing conditions.

The efficiency of the helical pumps requires a specific profile for the rotors, allowing a maximum flow section, the shortest gear line, and a leakage section as small as possible, all to achieve a high flow rate for the same PCP rotor and speed.

This results in a high volumetric yield, which also leads to an adiabatic yield increase, because the power losses are lower during the compression process through internal recirculation.

Most of the time, the helical pumps are provided with two helical channels, hence it is double-headed, and the PCP rotor is provided with a single helical channel, hence it is single-headed, the length of the PCP stator step being twice the length of the rotor pitch, Figure 3 [15].
3. The helical pumping plant

In order to provide a correct definition of a helical pump extraction plant, the following steps shall be taken: defining the positioning depth of the pump inside the probe, \( H_p \), based on the dynamic liquid level in the probe, which depends on the bottom pressure and the \( Q \) - flow rate, expected to be extracted; defining the dynamic \( H_d \) level in the probe and calculating the pressure losses in the \( H_{fric} \) extraction pipes due to friction; defining the pressure in the pumping head \( H_{cp} \) and calculating the total lifting height, \( H \); establishing the pump model, according to the total lifting dynamic height, the expected \( Q \) - flow rate to be extracted; defining the reduction of the speed ratio: \( i = n_{motor} / n_{pump} \) and choosing the pump characteristics: length of the PCP stator; the PCP rotor length; number of levels; determining, by calculation, the resistance to the stability of the sucker-rod pump.

By comparison, it shall be submitted the way to choose the helical pump for the extraction system of the two probes (S1 and S2), starting from different dynamic levels, \( H_d = 1700 \) m and \( H_d = 2000 \) m. The primary calculation elements for the two probes are presented in Table 1.

### Table 1: Initial data for S1 and S2 probes

<table>
<thead>
<tr>
<th>Date</th>
<th>Probe S1</th>
<th>Probe S2</th>
<th>Date</th>
<th>Probe S1</th>
<th>Probe S2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter of the fluid [mm]</td>
<td>127 + 114,3</td>
<td>177,8 + 114,3</td>
<td>Water density, ( \rho_w ) [kg/m(^3)]</td>
<td>1040</td>
<td>1040</td>
</tr>
<tr>
<td>At-rest bottom-hole pressure (BHP) [bar]</td>
<td>252</td>
<td>272</td>
<td>Oil viscosity, cP</td>
<td>2,5</td>
<td>2,5</td>
</tr>
<tr>
<td>Ram bottom-hole pressure (BHP) [bar]</td>
<td>205</td>
<td>172</td>
<td>Water viscosity, cP</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Pressure in the swivel head flow ( \rho_{cp} ) [bar]</td>
<td>4</td>
<td>4</td>
<td>Percent impurities [%]</td>
<td>20</td>
<td>28</td>
</tr>
<tr>
<td>Pump flow, ( Q ) [m(^3)/day]</td>
<td>25</td>
<td>16</td>
<td>Difference in level probe-ground [m]</td>
<td>10</td>
<td>15</td>
</tr>
<tr>
<td>Oil density, ( \rho_t ) [kg/m(^3)]</td>
<td>815,5</td>
<td>850</td>
<td>Nominal diameter of the extraction pumps [mm]</td>
<td>73,02</td>
<td>73,02</td>
</tr>
</tbody>
</table>

For S1 Probe, there shall be determined:

- The dynamic level: \( H_d = 1700 \) m;
- Horsehead: \( H_{fix} = H_{d} + h_{sub} = 1700 + 100 = 1800 \) m;
  where \( h_{sub} \) - pump submergence;
- Pumping height: \( H_p = H_{fix} + H_{fric} + H_{cp} = 1800 + 0,22 + 47,4 = 1848 \) m;
- Pressure into the pumping head:

  \[
  H_{cp} = \frac{p_{cp}}{\rho_t \cdot g} = \frac{4 \cdot 10^5}{860,4 \cdot 9,81} = 47,5 \text{ m}
  \] (1)

- Density of the fluid:

  \[
  \rho_t = (1 - i) \cdot \rho_t + i \cdot \rho_w = (1 - 0,2) \cdot 815,5 + 0,2 \cdot 1040 = 860,4 \text{ kg/m}^3
  \] (2)

- Pressure loss:

  \[
  H_{fric} = \frac{\Delta p_{fric}}{\rho_t \cdot g} = \frac{0,019}{860,4 \cdot 9,81} = 0,22 \text{ m}
  \] (3)
\[ \Delta p_f = \frac{\lambda \cdot H_{fix} \cdot v^2}{2 \cdot d_i} \cdot \rho_i = 0.037 \cdot \frac{1800 \cdot 0.069^2}{2 \cdot 73 \cdot 10^{-3}} \cdot 860.4 = 0.019 \text{ bar} \] (4)

- Hydraulic loss coefficients:

\[ \lambda = \frac{64}{Re} = \frac{64}{1737} = 0.037 \] (5)

\[ \text{Re} = \frac{\rho_i \cdot v \cdot d_i}{\mu} = 1737 \] (6)

- Flow velocity:

\[ v = \frac{4 \cdot Q}{\pi \cdot d_i^2} = \frac{4 \cdot 25}{3.14 \cdot (73 \cdot 10^{-3})^2 \cdot 86400} = 0.069 \text{ m/s} \] (7)

KUDU 60TP2000 shall be chosen (rotational speed: \( n = 231 \text{ rpm} \)); drive power: \( N = 10 \text{ kW} \). Its characteristics are extracted from the pump data sheet: the number of levels: 43 levels; the PCP rotor length: 5.7 m; the PCP stator length: 5.55 m; the rotor thread: 33 mm; the outer diameter of the pump: 89 mm. When calculating the resistance of the sucker-rod pumps, proceed as follows:

- There shall be determined the weight of the column of the fluid:

\[ P_i = (\lambda_i - a_p) \cdot H_{fix} \cdot g \cdot \rho_{am} \cdot N \]

\[ P_i = (4,185 \cdot 10^{-3} - 6,424 \cdot 10^{-4}) \cdot 1800 \cdot 9.81 \cdot 860.4 = 53810 \text{ N} \] (8)

- There shall be determined the buoyancy factor:

\[ b = 1 - \frac{\rho_{am}}{\rho_{ol}} = 1 - \frac{860.4}{7850} = 0.89 \] (9)

- There shall be determined the weight of the sucker-rod pumps in the air:

\[ P_p = q_p \cdot H_{fix} = 49,455 \cdot 1800 = 89019 \text{ N} \] (10)

- There shall be determined the normal stress, by means of the formula:

\[ \sigma_i = \frac{P_i + b \cdot P_p}{a_p} = \frac{53810 + 0.89 \cdot 89019}{6.424 \cdot 10^{-4}} = 207 \text{ N/mm}^2 \] (11)

- There shall be determined the torque:

\[ M_i = 9550 \frac{N}{n} = 9550 \frac{10}{231} = 413.4 \text{ N} \cdot \text{m} \] (12)

- There shall be determined polar modulus of strength:

\[ W_i = \frac{\pi \cdot d_p^3}{16} = \frac{\pi \cdot 0.0286^3}{16} = 4593.3 \text{ mm}^3 \] (13)
- There shall be determined tangential stress:

\[
\tau_t = \frac{M_t}{W_t} = \frac{413.4 \times 10^3}{4593.3} = 90 \text{ N/mm}^2
\]  

(14)

- There shall be determined the equivalent stress in the case of compound demand:

\[
\sigma_{eh1} = \frac{1}{2} \left( \sigma_t + \sqrt{\sigma_t^2 + 4 \tau_t^2} \right); \quad \sigma_{eh1} = \frac{1}{2} \left( 207 + \sqrt{207^2 + 4 \cdot 90^2} \right) = 241 \text{ N/mm}^2
\]  

\[
\sigma_{eh2} = 0.35 \cdot \sigma_t + 0.65 \cdot \sqrt{\sigma_t^2 + 4 \cdot \tau_t^2}; \quad \sigma_{eh2} = 0.35 \cdot 207 + 0.65 \cdot \sqrt{207^2 + 4 \cdot 90^2} = 251 \text{ N/mm}^2
\]  

(15)

- There shall be determined the withstand stress, corresponding to the 1530M-steel:

\[
\sigma_a = \frac{\sigma_c}{c_s} = \frac{744}{1.9} = 391.6 \text{ N/mm}^2
\]  

(17)

The dimensioning is correct because the resistance condition is met.

For S2 Probe, there shall be determined:

- The dynamic level: \(H_d = 2000 \text{ m}\);
- Horsehead: \(H_{fix} = 2000 + 100 = 2100 \text{ m}\);
- Pumping height: \(H_p = 2100 + 0.016 + 45.2 = 2146 \text{ m}\);
- Pressure into the pumping head:

\[
H_{cp} = \frac{4 \times 10^5}{903.2 \cdot 9.81} = 45.2 \text{ m}
\]  

(18)

- Fluid density:

\[
\rho_f = (1-0.28) \cdot 850 + 0.28 \cdot 1040 = 903.2 \text{ kg/m}^3
\]  

(19)

- Pressure loss:

\[
H_{f,pc} = \frac{0.014}{903.2 \cdot 9.81} = 0.157 \text{ m}
\]  

(20)

\[
\Delta p_f = 0.037 \cdot \frac{2100 \cdot 0.044^2}{2 \cdot 73 \cdot 10^{-3}} \cdot 903.2 = 0.014 \text{ bar}
\]  

(21)

- Hydraulic loss coefficients:

\[
\lambda = \frac{64}{Re} = \frac{64}{1167} = 0.055; \quad \Re = \frac{\rho_f \cdot v \cdot d_i}{\mu} = 1737
\]  

(22)

- Flow velocity:

\[
v = \frac{4 \cdot 16}{3.14 \cdot (73 \cdot 10^{-3})^2 \cdot 86400} = 0.044 \text{ m/s}
\]  

(23)

KUDU 180TP3000 pump shall be chosen (\(n = 110 \text{ rpm}\)); drive power: \(N = 15 \text{ kW}\) [15]. Its characteristics will be extracted from the pump data sheet: the number of levels: 47.4 levels; the PCP rotor length: 9.25 m; the PCP stator length: 8.73 m; the rotor thread: 33 mm; the outer diameter of the pump: 120 mm.
When calculating the resistance of the sucker-rod pump gasket, proceed as follows:

- There shall be determined the weight of the fluid:
  \[ P_f = (4.185 \cdot 10^{-3} - 6.424 \cdot 10^{-4}) \cdot 2100 \cdot 9.81 \cdot 903.2 = 65901 \text{N} \]  
  \[ (24) \]

- There shall be determined the buoyancy factor:
  \[ b = 1 - \frac{903.2}{7850} = 0.885 \]  
  \[ (25) \]

- There shall be determined the weight of the sucker-rod pumps in the air:
  \[ P_p = 49,455 \cdot 2100 = 103856 \text{N} \]  
  \[ (26) \]

- There shall be determined the unitary tensile stress, by means of the formula:
  \[ \sigma_{t} = \frac{65901 + 0.885 \cdot 103856}{6.424 \cdot 10^{-4}} = 246 \frac{\text{N}}{\text{mm}^2} \]  
  \[ (27) \]

- There shall be determined the torque:
  \[ M_t = 9550 \frac{\text{N}}{\text{n}} = 9550 \frac{15}{110} = 1302.3 \text{ N} \cdot \text{m} \]  
  \[ (28) \]

- There shall be determined the torque:
  \[ W_t = \frac{\pi \cdot 0.0286^3}{16} = 4593.3 \text{ mm}^3 \]  
  \[ (29) \]

- There shall be determined the tangential stress:
  \[ \tau_{t} = \frac{1302.3 \cdot 10^3}{4593.3} = 284 \frac{\text{N}}{\text{mm}^2} \]  
  \[ (30) \]

- There shall be determined the equivalent stress in the case of compound demand:
  \[ \sigma_{ech1} = \frac{1}{2} \left( \sqrt{\frac{246^2 + 246^2 + 4 \cdot 284^2}{2}} \right) = 432 \frac{\text{N}}{\text{mm}^2} \]  
  \[ \sigma_{ech2} = 0.35 \cdot 246 + 0.65 \cdot \sqrt{246^2 + 4 \cdot 284^2} = 488 \frac{\text{N}}{\text{mm}^2} \]  
  \[ (31) \]
  \[ (32) \]

The dimensioning is correct because the resistance condition is met.

The test pump data (KUDU 180TP3000) is shown in Table 2.

<table>
<thead>
<tr>
<th>dn [mm]</th>
<th>e [mm]</th>
<th>hsr [mm]</th>
<th>Q [m³/h]</th>
<th>n [rot/min]</th>
<th>Δp [bar]</th>
<th>N [kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>18</td>
<td>3</td>
<td>189</td>
<td>0÷5,6</td>
<td>110</td>
<td>0÷12</td>
<td>15</td>
</tr>
</tbody>
</table>

where: \( d_{sr} \) – diameter of the PCP rotor; \( e \) – eccentricity; \( h_{sr} \) – height of the PCP stator.
4. Finite element analysis of a sucker-rod pump

Sucker-rod pumps are designed to transmit the rotation motion from the drive-head to the PCP rotor.

The loads acting on the sucker-rod pumps, in the case of helical pumping, are given by the weight of the submerged sucker-rod gasket, the weight of the fluid column acting on the cross section of the PCP pump, the torque expected to be transmitted to the pump and the bending momentum (after loss of stability).

It follows that, in the case of helical PCP, the sucker-rod pumps are subjected to compression, torsion and bending, thus to a compound load. The compression stiffens the sucker-rod gasket increasing the speed at which the loss of stability occurs, while torsion has a contrary effect.

The normal stress occuring from the axial load has a maximum value at the top of the sucker-rod gasket and is given by the relation (11), in which the \( P \) - weight of the column of liquid in the extraction pipes (8) is defined; \( b \) - buoyancy factor (9); \( P \) - the weight of the sucker-rod pump, in the air (10); \( a \) - the inner cross-sectional area of the sucker-rod pumps; \( d \) - sucker-rod pump diameter.

The transmission of the torque required to rotate the PCP rotor leads to the development of tangential stresses along the entire length of the sucker-rod pump. The mean torque is determined by the relation (12) and the tangential voltage is determined by the relation (14). The two stresses, axial compression and torsion, give rise to a compound stress. In order to determine the equivalent stress specific to the compound stress \( \sigma_{\text{ech}} \), one of the resistance theories (15) and (16) is adopted and the condition of checking the resistance of the sucker-rod gasket is imposed, where \( \sigma_{\text{ech}} \leq \sigma \),

where \( \sigma \) is the allowable stress \( \sigma \leq \sigma \), with \( c \) - safety coefficient (\( c = 1.5 \)).

Finite element analysis method for the solid model of the sucker-rod pump is performed using the finite element analysis software, Ansys Academic R15.0 [16], compared to the finite element analysis software, Cosmos / M [17]. The material of the workpiece is a 1530M-steel, low in carbon, easy-welding, resistance against intergranular corrosion, low temperature resistance.

a. Modeling the landmark using the ANSYS Academic Software R15.0

In the first stage, there shall be determined the units of measurement, then geometric characteristics of the solid landmark Sucker-rod pump shall be introduced, Tables 3 ÷ 4.

<table>
<thead>
<tr>
<th>Bounding Box</th>
<th>Properties</th>
<th>Statistics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length X</td>
<td>22.2 mm</td>
<td>Volume</td>
</tr>
<tr>
<td>Length Y</td>
<td>22.2 mm</td>
<td>Mass</td>
</tr>
<tr>
<td>Length Z</td>
<td>7620. mm</td>
<td>Scale Factor Value</td>
</tr>
</tbody>
</table>

Table 3: Geometry elements of a sucker-rod pump

<table>
<thead>
<tr>
<th>Properties</th>
<th>Statistics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Centroid X</td>
<td>1.6837e-016 mm</td>
</tr>
<tr>
<td>Centroid Y</td>
<td>-2.1648e-016 mm</td>
</tr>
<tr>
<td>Centroid Z</td>
<td>3810. mm</td>
</tr>
</tbody>
</table>

Table 4: Geometry elements of the solid

For the sucker-rod pump, the following are known: \( d = 22.2 \) mm; \( l = 7620 \) mm; \( G = 24.384 \) kg. The PCP rotor was loaded with torque and, in turn, with three compression forces. In the first stage, the axial force \( F_1 = -171675 \) [N] and a torque \( M = 610 \) [Nmm] shall be applied on the Z-axis, Figure 4 and Figure 5.
The procedure shall be repeated, applying other two axial forces to the Z-axis, \( F_2 = -190314 \) [N] and \( F_3 = -208953 \) [N], but the torque \( M = 610 \) [Nmm] remains unchanged. The modeling factor determines the multiplication factor, as defined in Table 5. With this multiplication factor, the critical load can be determined, by which the sucker-rod pump can be loaded so that the landmark can withstand.

Table 5: Values of the \( K \) multiplication factor - Ansys Academic R15.0

<table>
<thead>
<tr>
<th></th>
<th>( F_1 ) [N]</th>
<th>( F_2 ) [N]</th>
<th>( F_3 ) [N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load Multiplier</td>
<td>5.8932E-4</td>
<td>5.3159E-4</td>
<td>4.8418E-4</td>
</tr>
</tbody>
</table>

b. Modeling using the COSMOS/M software

For finite element analysis using the COSMOS / M software package, the landmark was created by defining elements used for discretisation BEAM3D (three-dimensional), material properties. The cross section in Figure 6 was defined, starting from the definition of points and curves, Figure 7.
Fig. 7. Creating key points by defining coordinates (Geometry>Points>Define)

The model was loaded with axial force and torque (LoadBC>Structural>Force>Define by Points (by Node), Figure 8, and the model was deprived of degrees of freedom LoadBC>Structural>Displacement>Define by Points> All6DOF.

Following the finite elements analysis, the multiplication factor was determined with this software, Table 6.

In Table 7 and Figures 9 and 10, the comparative values obtained under the same loading conditions for the $K$ multiplication factor are presented using the two finite element analysis software packages Ansys Academic R15.0 and COSMOS / M.

### Table 6: Values of the multiplication $K$ factor - COSMOS/M

<table>
<thead>
<tr>
<th>F1 [N]</th>
<th>F2 [N]</th>
<th>F3 [N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load Multiplier</td>
<td>5.90248E-4</td>
<td>Load Multiplier</td>
</tr>
</tbody>
</table>

### Table 7: Comparing the results from the two softwares

<table>
<thead>
<tr>
<th>Compression force [N]</th>
<th>$K$ multiplication factor (COSMOS)</th>
<th>$K$ multiplication factor (ANSYS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>171675</td>
<td>5.90248E-4</td>
<td>5.8932E-4</td>
</tr>
<tr>
<td>190314</td>
<td>5.3244E-4</td>
<td>5.3159E-4</td>
</tr>
<tr>
<td>208953</td>
<td>4.84945E-4</td>
<td>4.8418E-4</td>
</tr>
</tbody>
</table>

Fig. 9. Variation of $K$ vs. F1, F2, F3

Fig. 10. Variation of $K$ vs. Ansys and COSMOS
Conclusions

Progressive helical PCPs are widely used in the industrial field, especially in crude oil extraction, as their constructive and functional characteristics correspond to the severe operating conditions specific to this field, where the aggressiveness of the chemicals involved is a problem. Their construction allows the transfer of high viscosity fluids and the achievement of relative speeds with required precision. They are easy to install and characterised by a high reliability and maintainability, much more convenient than a beam pumping units or hydraulic jet pumps. For work surfaces that come into contact with the fluid to be delivered, stainless steel or anticorrosive material shall be used, while the non-metallic parts are made of rubber, plastics, the essential advantage being the normal displacement of the fluid through the conjugated PCP rotor, with continuous opposite displacement-opposing force inside the pumping chamber. Helical pumps use energy only to lift the fluid, not the sucker-rod pumps, thus eliminating ruptures of the sucker-rod pumps, caused by the weight of the fluid. However, given the long length of the sucker-rod pumps and the operating conditions, their loading must be taken into account, in terms of the forces and moments applied.

In this paper, the correct choice of the helical pumps was demonstrated by a comparative resistance calculation of the sucker-rod pumps, starting from the physical characteristics of the extraction fluid, respectively from the specific operating conditions.

Following modeling with finite elements, a multiplication K-factor has been obtained, similar to modeling with Ansys Academic R15.0 and COSMOS / M softwares, an essential factor in determining critical loads for loss of stability.

References