Analytical study of “crosshead – slide rail” wear effect on pump rod stress state

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Abstract. The work presents an analytical study of “crosshead- slide rail” wear effect on stress state of a double piston pump’s rod. The authors proposed an engineering model of pump’s rod stress state and analytically analyzed a pump rod's stress state caused by wear of the “crosshead- slide rail” pair. It was stated, that wear values according to regulations and directive documents can refer to significant values of rod’s residual stresses. The numerical validation of the obtained results was carried out on for drill pump rods, including reinforced functional coatings.

1. Introduction
The modern market of equipment and technique is full of structural parts with protective coating of different types; a number of coating technologies constantly grows both for new parts and restoration of damaged ones [1–3]. At the same time, premature destruction issues limit a scope of coating application for part strengthening. Contrary to not coated parts, sometimes coating can cause decreasing of part performance in conditions of extreme loading or unexpected operating modes. In addition, pump environmental safety depends significantly on sealed moving part surfaces.

The authors faced the specific issues of additional loading when evaluated strength of restored rods of double-acting drilling pumps. This occurs due to wear of reciprocating movement of friction pair “crosshead – slide rail” and rod’s axis displacement relative to its initial position. This leads to changes in rod material deformation patterns and reduces a safety factor (strength coefficient). In particular, additional transverse bending occurs instead of the standard central stretching or compression of the pump rod contrary to design calculations. As a result, there are not only quantitative, but also qualitative changes of rod’s stress state. This way, our study is spawn by the wishes to estimate the impact of additional stresses on the strength of drill pump rods. These issue are especially relevant for operation of pump rods restored by wear-resistant and corrosion-resistant coatings.
Operating conditions, wear rate and causes of failures of parts and assemblies of drilling pumps operated in aggressive and abrasive-containing environments under dynamic loading were analyzed in [4–6]. In order to increase the aforementioned pump’s service life, gears of its driver’s mechanical part are improved by the design and technological ways [7, 8], as well as the hydraulic part is improved by tribological studies of polymeric materials [9, 10], optimal methods of manufacture and assembly using elements of composite materials [11]. Various coatings are used to increase the wear resistance of parts, in particular as follows: galvanic [12], electrospark [13–15], plasma [16–19] and plasma electrolytic [20–22]. An analysis of methods for steel protection from sulfide cracking, inherent for oil and gas equipment, showed the prospects of protective layered coatings for their working surfaces [23]. Layered coatings can improve the mechanical and tribological properties of surfaces and develop effective compositions for a rational combination of different characteristics, such as strength, wear resistance, low coefficient of friction, blocking of unwanted heat fluxes, etc. An overview of fundamentally different approaches to the modeling of layered structures is presented in [24–26]. Some authors considered mechanical effects in layered coatings under temperature [27–29] and local forces [30–32]. Strength and stiffness of layered structures under axisymmetric loads are presented in [33, 34]. The works [35–37] present other samples of design and calculation of shell – filler composite structures. Since the pump rods operate under load in a liquid medium, electrochemical studies are performed to assess the effect of mechanical stresses on corrosion processes [38–40].

To restore the original dimensions of worn rods, it is necessary to provide possibility of coating thickness increasing with the obtaining the “base – coating” sufficient strength. To do this, thin metal layers are placed between the solid top layer and the base [41]. This restoration technology for drill pump rod surface makes it possible to replace a material with another one. Thus, the rod’s part to be restored is transformed into a rod area of layered material. Therefore, the operational strength of the resulting material composition is the issue at stake. This issue is especially relevant in emergency cases, which lead to rod’s strength decreasing due to deformation pattern changes.

The article aims to study the stress-strain state and to assess the strength of the coated section of the double action pump rod.

2. Main content of the paper

2.1. Model of additional pump rod loading

Let’s study the additional stress-strain state of the pump rod associated with the “crosshead – slide rail” friction pair wear (Fig. 1). The pump rod, bended by the crosshead vertical displacement from the horizontal axis, refers to a two-stage, statically indeterminate rod (Fig. 2). The left support of this rod has a forced vertical linear displacement. We assume that vertical component of the external load (caused by the rod connecting through the finger to the crosshead) provides the effect of two-way connection on the left support, so we schematize the left edge of the rod with a rigid fastening. In addition, we assume that the pliability of the rod’s sealing materials is high and practically does not prevent free transverse rotation of the rod contacting with the specified sealing. However, we assume that piston sealing tension do not make possible rotation in the pair of piston-cylinder. It should be noted that longitudinal bending effects should be considered when model long structures. The phenomenon of buckling is not limited to long rod elements. Buckling can occur in many kinds of structures and can have many forms [42, 43], in particular in rod systems of drilling tools [44–46]. As far as, the flexibility assessment of real drill pump rod shows that pump rods are low-flexible rods, we do not consider longitudinal bending [47].

The specified rod model (Fig. 2) pattern for this case will be close to the pattern of the real object and will make possible evaluation and analyzation of the stated problem.

In order to specify rod’s loadings and stresses effected by kinematic loading (Fig. 2) we use the method of initial parameters. The deflection $w(x)$ and angles of rotation $\theta(x)$ caused by the friction pair “crosshead – guide frame” wear, equations for rod’s cross sections are as follows:
\[w(x) = w_0 + \frac{1}{kEJ} \left( \frac{M_1 x^2}{2} - \frac{P_2 x^3}{6} \right) - \frac{1}{EJ} \left( \frac{1}{k} \right) \left[ \frac{P_1 (x-a)^2}{2} \right] + \]
\[\frac{P_2 (x-a)^3}{6} - \frac{M_4 (x-a)^2}{2} \right] H(x-a) - \frac{P_1 (x-\psi)^3}{6} H(x-\psi) \right]; \tag{1}\]
\[\theta(x) = \frac{d\psi}{dx} = \frac{1}{kEJ} \left( M_1 x - \frac{P_2 x^2}{2} \right) - \frac{1}{EJ} \left( \frac{1}{k} \right) \left[ \frac{P_1 (x-a)^2}{2} \right] + \]
\[\frac{P_2 (x-a)^3}{2} - \frac{M_4 (x-a)^2}{2} \right] H(x-a) - \frac{P_1 (x-\psi)^3}{2} H(x-\psi) \right], \tag{2}\]

where \(H(x)\) – Heaviside function, \(E\) – Young’s modulus for the rod’s material, \(J\) – axial moment of inertia of cross section, \(\psi = a + b\). For the sake of compactness in the equations (1) and (2), we have already considered two geometric boundary conditions at the left end of the rod, in particular \(w(0) = w_0\), and \(\theta(0) = 0\).

The boundary conditions at the right end of the rod and at the \textit{rod-sealing} interaction point are as follows:
\[w(l) = 0; \quad \frac{d\psi}{dx}(l) = 0; \quad w(\psi) = 0, \tag{3}\]

where \(l = \psi + c\).

Substituting (3) in (1) and (2), we obtained a system of three equations to determine unknown reactions in the bonds \(M_1, P_2, P_3\). In order to determine two more reactions \(P_4\) and \(M_5\), we used two traditional equilibrium equations for a plane system of forces.

### 2.2. Determination of additional stresses in the rod

In order to make the obtained results more illustrative, we choose the rod of the УНБ-600 drilling double action pump with the following design options (Fig. 1): \(a = 575 \text{ mm}, \ d = 1130 \text{ mm}\); and maximal rod’s running \(\eta = 400 \text{ mm}\); \(D_1 = 120 \text{ mm}, \ D_2 = 70 \text{ mm}, \ E = 2 \cdot 10^6 \text{ MPa}\).

When the pump rod changes its position, the \(b\) and \(c\) options (Fig. 2) change too to the rod’s running distance; for example, \(b = 715 \text{ mm}\) and \(c = 415 \text{ mm}\) for the leftmost position of the rod. The kinematic load is represented by the initial deviation caused by the friction pair “crosshead – guide frame” wear.
Fig. 3 shows reactions in the rod’s bonds refer to rod’s running distance. The zero position of the abscissa axis is the \( \eta \) running distance which corresponds to the leftmost position of the pump rod. It is shown, that reactions \( M_1, P_2, P_3, M_3 \) increase monotonically, and the reaction \( P_1 \) decreases monotonically when the rod moves from the leftmost to the rightmost position. It should be noted, that the direction of reactions does not change.

Fig. 4 shows rod bond reactions refer to initial displacement \( w_0 \). We took a typical case when the rod is in the leftmost position for illustration. In general, analyze of the charts (Fig. 3 and Fig. 4) shows, the rod bond reactions have a quadratic dependence on the rod’s running distance \( \eta \) and depend linearly on the initial deviation \( w_0 \) caused by the friction pair “crosshead – guide frame” operation.

Additional normal stresses in the external fibers of the pump rod were determined by the formula:

\[
\sigma(x) = \begin{cases} 
2k \frac{J}{D_1}(M_1 - P_2x), & (0 \leq x \leq a); \\
2 \frac{J}{D_2} [M_1 - P_2x + P_3(x - \varphi)], & (a \leq x \leq l).
\end{cases}
\] (4)

Fig. 5 shows distribution of the maximum normal stresses along the rod’s length at its various positions. The maximal normal stresses in the rod’s material are observed at the rightmost position, and the minimal ones – at the extreme left. The normal stresses increase monotonically while the rod moves from the leftmost to the rightmost position. Estimation of normal stresses indicates that they can have a significant impact on the strength of the pump rod.
1 – leftmost position of the rod; 2 – intermediate position of the rod; 3 – rightmost position of the rod

**Figure 5.** Distribution of the additional normal stresses along the rod’s length.

2.3. Model of the restored area of the rod

The rod was restored by turning out the worn area and forming a two-layer coating on a steel base: base steel – 40X (State Standard 4543-71) wear-resistant surface layer – alumina, intermediate layer – aluminum. The two-layer coating was formed by applying a layer of aluminum to steel with further microarc oxidation in an electrolyte leaving an unoxidized layer of aluminum. It should be noted that the proposed coating technology is resource saving and environmentally friendly.

For the case of additional loading, there is a problem of bending with compression of the area of a homogeneous cylindrical rod (Fig. 6). Its cross section is a multiply connected region with elasticity and strength variation from layer to layer, i.e. is a piecewise constant functions refer the radial coordinate.

Equations of the stress-strain state for all layers of the materials were developed used the same static and kinematic hypotheses of the linear theory of rods. We neglected the susceptibility of the rod to transverse shear deformations and compression, and we assumed that the hypothesis of a rigid normal is true. The normal $\sigma_x$ and tangential $\tau_{xy}$ stresses in the cross section of the coated area are represented by their integral static equivalents: $N$ – axial force, $Q_y$ – transverse force, $M_z$ – bending moment, which we consider known (defined by the method proposed in paragraph 2.1). The properties of materials in homogeneous regions are determined by Young's moduli $E_i$, Poisson's ratios $\nu_i$, and yield $\sigma_{yi}$ or strength $\sigma_{ui}$ limits. We use the mark $i = 1$ for the internal (base) layer, $i = 2$ – for the intermediate layer, and $i = 3$ for the external layer; we mark $F_i$ the cross-sectional area of the $i$-th layer.
The equilibrium equations are presented as integral relationships referring longitudinal force, bending moment and normal stresses:

\[ N = \sum_{i=1}^{n} N_i = \sum_{i=1}^{n} \int \sigma_x dF_i; \quad M_z = \sum_{i=1}^{n} \int y \sigma_x dF_i. \quad (5) \]

The deformation compatibility expressions of different layers of rod material are:

for stretching \( \sigma_{x,i} = \frac{\delta_{x,i}}{E_i} \), \( i = 1, 2 \); with bending \( \varepsilon_{y} = \kappa y = \text{const} \), \( i = 1, 2, 3 \),

where \( \kappa \) – the curvature of the rod’s deformed axis caused by the additional loading, \( \varepsilon_{x} \) – linear deformations.

Physical relations are represented by Hooke’s law:

\[ \sigma_{x,i} = E_i \varepsilon_{x,i}, \quad i = 1, 2, 3, \quad (7) \]

Integrating (5) and taking into account (6) and (7), we obtained expressions of normal stresses by axial force and bending moment:

\[ \sigma_{x,k}^{(x)} = -\frac{N}{F_k} \sum_{i=1}^{n} E_i F_i, \quad \sigma_{x,k}^{(m)} = \frac{M_{z,k}}{J_{z,k}} \sum_{i=1}^{n} E_i J_{z,i}, \quad k = 1, 2, 3, \quad (8) \]

\( J_z \) – axial moment of inertia of the cross-sectional area of the material’s \( i-th \) layer.

The function of tangential stresses caused by the transverse force was determined by Zhuravsky’s theory:

\[ \tau_{xy}(y) = \frac{Q}{J_z} \sum_{i=1}^{n} k_i b(y), \quad k_i = \frac{E_i}{E_1}, \quad i = 1, 2, 3, \quad (9) \]

where \( J_z = \frac{\pi}{64} \left[ d_1^4 + k_2 d_2^4 (1 - s^4) + k_3 d_3^4 (1 - s^4) \right], \quad \zeta = d_1 / d_2, \quad s = d_2 / d_3, \quad b(y) \) – the width of the cross section.

Huber-Mises theory was used to assess strength. Equivalent stresses were calculated by the formula:

\[ \sigma_{\text{equiv}} = \sqrt{\left( \sigma_{x}^{(x)} + \sigma_{x}^{(m)} \right)^2 + 3 \tau_{xy}^2}. \quad (10) \]

2.4. *Estimation of rod’s restored area strength*

We applied the results obtained above to provide numerical estimation of the strength for a specific engineering problem. Let’s the drilling double action pump UNB-600 operates with the cylinder plug with internal diameter of \( D_c = 160 \text{ mm} \), and provides pressure \( p = 16.5 \text{ MPa} \) at the same time. Normal compressive stresses in the new solid rod of the pump are determined by the formula \( \sigma = p(D_c / d_c)^2 \).

These stresses are equal to 87 MPa for a rod with the diameter of \( d_c = 70 \text{ mm} \).

Due to the technical need, the rod-sealing contact area was restored by the two-layer coating. This coating make the composition of materials in this area as it is shown in Fig. 6. The modulus of elasticity of the base material, intermediate layer, and external layer are \( E_1 = 2.1 \cdot 10^{11} \text{ Pa} \), \( E_2 = 0.71 \cdot 10^{11} \text{ Pa} \), and \( E_3 = 2.9 \cdot 10^{11} \text{ Pa} \) respectively. Characteristic diameters of the rod (fig. 6) are
The following mechanical characteristics of materials were taken as a criterion of static strength: yield strength for the steel 40X – 315 MPa; strength limit for alumina – 350 MPa; yield strength for aluminum – 80 MPa.

Fig. 7, a shows the distribution of equivalent stresses calculated by the energy theory of Huber-Mises strength in the restored area of the rod, taking into account the additional stresses caused by the “crosshead – slide rail” friction pair wear. Along, fig. 7, b shows static safety factor distribution (static strength) $n^*$ for the rod’s restored area operating with additional loading.

![Figure 7](image)

Figure 7. Estimation of static strength of rod restored area with additional loading consideration.

Analyse of the obtained relations (Fig. 7) make possible to conclude that the rod’s restored area meets the condition of static strength for the considered case. It should be noted, that restoration composition of materials with an intermediate layer is quite prominent in terms of strength of the restored object as a whole. Variance of safety factors for layers is relatively small. This means that varying at the design stage geometric, physical-mechanical and technological factors of the restoration makes possible optimization of the stress-strain state in the system "base - layered coating", which will result in uniform compositions.

The calculation of wear and fatigue strength should be the final calculation to determine the suitability of the restored area of the rod for long-term operation. Based on the results of this work, we will be able to determine the nominal normal and tangential stresses for aforementioned calculation. Then, after describing the pattern of cyclic loads and determining the limits of static strength and coefficients of their reduction, we can easily make such calculations.

### 3. Conclusions

The work presents an engineering model of the drilling double action pump rod, which is subjected to additional loading dynamic loading. There were specified additional stresses and loadings because the “crosshead – slide rail” friction pair wear. The change of additional forces and stresses depending on the parts’ wear and operational position of the pump rod is analyzed. It was stated, that additional stresses become significant for the case of wear specified by standards (requirements); and these stresses should be taken into account when assessing the strength of the pump rod.

The work proposes a model of a substantially inhomogeneous pump rod to study the stress-strain state of the pump rod restored area.

Restoration of the pump rod was carried out by forming the two-layer coating on the steel base: base – steel 40X; wear-resistant surface layer – alumina; layer – aluminum. The multiply connected region was used as the rod’s cross section with elasticity and strength variation from layer to layer. The pattern of the model was considered for the non-standard situation caused by “crosshead – slide rail” friction pair wear.
There were obtained the distribution of equivalent stresses in the inhomogeneous material of the restored area. The restored area strength was evaluated taking into account the additional stresses. It should be noted, that in the framework of the model used, one count calculate the stress-strain state of the multilayer coating.

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