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Fatigue Strength of Screws in Submersible Centrifugal Pumps under a Tipping Torque

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Abstract—A method is proposed for calculating the fatigue strength of screws in submersible centrifugal pumps under a tipping torque, with allowance for the contact pliability of the thread turns and the flanges. Pump reliability may be increased by improving the finish of the contacting flange surfaces and by repeated tightening of the screws.

Keywords: submersible pump, threaded joint, screw, contact deformation, fatigue strength, tipping torque
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Submersible centrifugal electric pumps used in oil fields are compact (housing diameter 93, 103, or 114 mm) multistage units (length around 5 m), driven by an electric motor. The sections of the pump are connected by screwed flanges. Pump failure on account of problems with the threaded joints are associated with flange disassembly and the loss of sections in the borehole; 30% of pump faults are of this type. The screws fracture on account of fatigue failure after the tension in the threaded joint is reduced. This is associated with flattening of the surface microprojections on first loading of the joint.

Fatigue failure of the screws occurs in two cases.

(1) The accumulation of damage in the screws on account of the initial pump descent in the borehole to a depth of 1500–2500 m. As a rule, the pump is lowered in stages and is stopped after each 12.5 or 25 m, so that the pipe supplying oil from the borehole to the surface can be extended. The threaded joints in the pump are loaded at each stage of the descent by a pulsating opening force, which is determined by the inertia of retardation. The fatigue strength of the screws in this case was discussed in [1].

(2) The accumulation of damage in the screws during pump operation. This is associated with loading of the threaded joint by a static opening force F_G created by the weight of the petroleum column above the pump and also by a tipping torque M that rotates relative to the threaded joint and appears on account of radial oscillation of the pump in the borehole (Fig. 1). Torque M is created by force F_G acting at a distance r equal to the radial gap between the pump and the lining pipe in the borehole (Fig. 2).

We will now consider the second case. We assume that the flanges are held together by z screws.

If the joint is subject only to tipping torque M relative to the y axis, the section of the parts coupled by the first screw is loaded by an external tensile force F, while the section coupled by screw z is loaded by an external compressive force F. These sections are analogous to single-turn threaded joints subject, respectively, to tipping and compressive forces F. The loading diagrams in Fig. 3 are plotted on the assumption that there is no plastic strain in the joint. In a joint that employs one screw tightened by force F_{ti} and loaded by external opening force F (Fig. 3a), the screw is subject to force F_s, according to [2–4]. When a single-screw joint is subject to compressive external force F, we obtain Fig. 3b. The notation in Fig. 3 is as follows: $\varphi_s = \arctan(1/\lambda_s)$; $\varphi_p = \arctan(1/\lambda_p)$; λ_s is the pliability of the screw; λ_p is the pliability of the parts being joined; $\delta_s = \lambda_s F_{ti}$ is the elongation of the screw under the action of force F_{ti}; $\delta_p = \lambda_p F_{ti}$ is the shortening of the part under the action of force F_{ti}; F is the external force; F_s and F_p are the external forces on the screws.

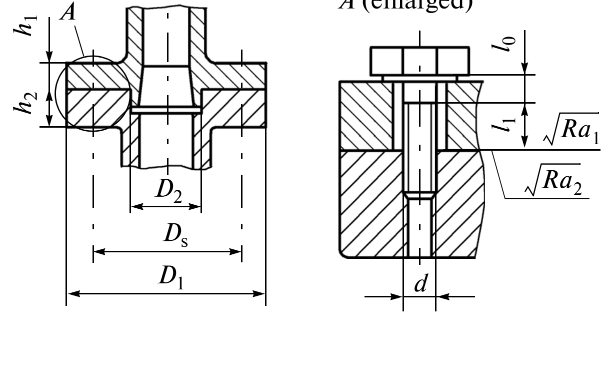

Fig. 1. Threaded joint in submersible pump.

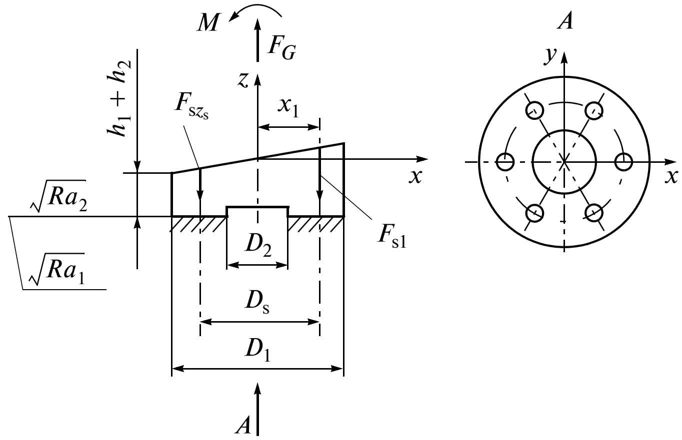

Fig. 2. Joint in submersible pump.

In Fig. 4a, we show the experimental dependence of the decrease δ in the distance between plane contact surfaces on the force F for 40X steel samples from [5]; the contact surfaces of the samples are turned on a lathe. In the second and third load applications (curves *2* and *3*), δ is much less than in the first (curve *1*). This may be attributed to plastic strain of the surface microprojections in the first load application. After the second and third load applications, the microprojections become elastic. In Fig. 4b, we plot δ as a function of the pressure p for samples of unquenched steel 45 in the first and subsequent load applications [5]. We see that δ is larger by a factor of 3–4 in the first load application than in the subsequent load applications.

In Fig. 5, we show the loading of a threaded joint by a tipping torque when the redistribution of the external load among the screws and the coupled parts under the influence of the contact pliability is taken into account. Such redistribution is due to the flattening of the surface microprojections on first loading of the joint as a result of plastic deformation (shown by curves in Fig. 5, where the elastic deformation is linearized). After tightening of the screw by force F_{ti1} and application of the tipping torque, the screw is also subject to the unscrewing force. The section of the joint tightened by screw z_s is subject to an external compressive force. With reversal of the tipping torque, the section connected by the first screw experiences an external compressive force, while screw z_s is subject to additional tensile force. Under these loads, the contact surfaces of the thread and the flanges are subject to plastic deformation, and the tightening force declines to F_{ti}. A cyclic external force F_s acts on each screw. Together with F_{ti}, it forms the static F_m and dynamic F_a components of the force loading each screw.

In Fig. 5 $\varphi_s = \arctan(1/\lambda_s)$, $\varphi_p = \arctan(1/\lambda_p)$, where λ_p is the pliability of section A (Fig. 1) of the parts tightened by the screw; $\delta_s = \lambda_s F_{ti}$; $\delta_p = \lambda_p F_{ti}$; $\delta_{s.pl}$ and $\delta_{p.pl}$ are the displacements due to the plastic deformation of the screw thread and the coupled surfaces of the part; F is the external force on section A (Fig. 1); F_a is the amplitude of the force on the screw; F_m is the static force on the screw; t is the time.

According to Fig. 5, the reduction $\Delta F_{ti} = F_{ti1} - F_{ti}$ in the tightening forces on the screws in repeated loading is

$$\Delta F_{ti} \geq F_{ti}(\delta_{s.pl}/\delta_s + \delta_{p.pl}/\delta_p). \tag{1}$$

Here

$$\delta_{s.pl} = 3Ra_s c_0 \left(\frac{d}{50}\right)\left(\frac{p_s}{E_s}\right)^{0.5}, \tag{2}$$

$$\delta_s = F_{ti}\lambda_s + Ra_s c_0 \left(\frac{d}{50}\right)\left(\frac{p_s}{E_s}\right)^{0.5}, \tag{3}$$

$$\delta_{p.pl} = 3Rac_0\varepsilon\left(\frac{p}{E}\right)^{0.5}, \tag{4}$$

$$\delta_p = F_{ti}\lambda_p + 3Rac_0\varepsilon\left(\frac{p}{E}\right)^{0.5}. \tag{5}$$

In Eqs. (2)–(5), $Ra_s = (Ra_{s1}^2 + Ra_{s2}^2)^{0.5}$ is the reduced roughness of the thread; c_0 is a dimensionless parameter depending on the type of surface treatment

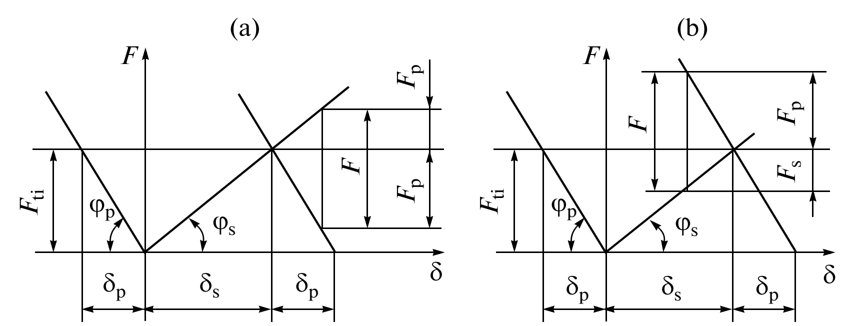

Fig. 3. Loading diagrams of single-screw threaded joints in the case of tension (a) and compression (b) on the assumption that there is no plastic contact strain in the thread and in the joined surfaces.

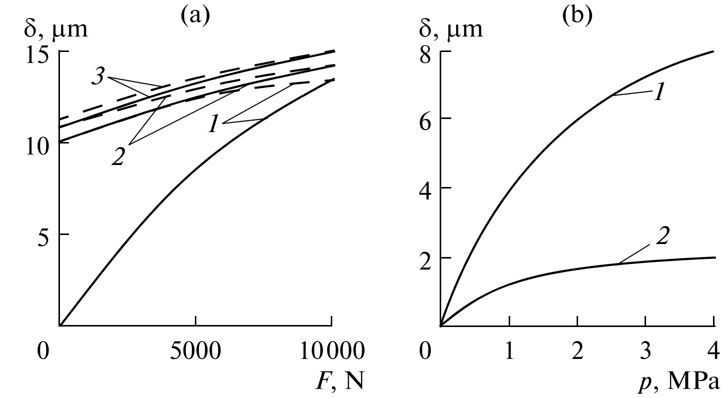

Fig. 4. Dependence of δ on the load F [5] and pressure p in the first (*1*) and subsequent (*2*, *3*) load applications for 40X steel (a) and steel 45 (b) samples turned on a lathe: the continuous curves correspond to loading and the dashed curves to unloading.

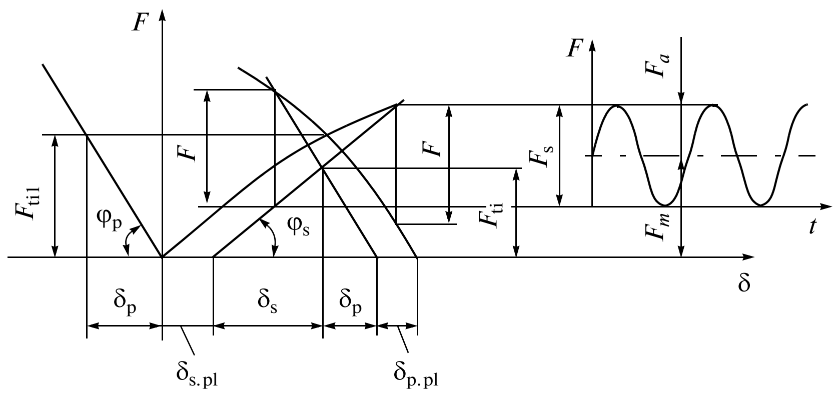

Fig. 5. Loading diagram of section A (Fig. 1) of the threaded joint with a tipping torque in the case of contact plastic deformation of the thread and the flange surfaces.

and the direction of the machining tracks (in repeated loading, $c_0 = 355$); d is the rated thread diameter; $p_s = 0.34F_{ti}/A_{th}$ is the pressure in the first loaded turn of the thread created by the tightening force; $A_{th} = \pi(d^2 - d_3^2)/4$ is the supporting area of the thread turn; $d_3 = d - 1.227P$ is the internal diameter of the thread turn; P is the thread pitch; E_s is the elastic modulus of the screw; λ_s is the pliability of the screw.

The pliability of the screw is found from the formula

$$\lambda_s = 4[(0.5d + l_0)/(\pi d^2) + (0.5d + l_1)/(\pi d_3^2)]/E_s. \quad (6)$$

Here l_1 is the length of the grooved loaded section of the screw (Fig. 1); l_0 is the length of the smooth section of the screw; $Ra = (Ra_1^2 + Ra_2^2)^{0.5}$ is the reduced mean height of the microprojections at the contact surfaces; $\varepsilon = f(\Delta - W_{max})$ is a scale factor depending on the flatness tolerance Δ (determined by the degree of precision according to State Standard GOST 24643–81 and the maximum dimension D of the contact surface) and on the maximum wave height W_{max} of the surface roughness [6]; $p = z_s F_{ti}/A$ is the contact pressure created by the tightening forces; $E = 2E_1E_2/(E_1 + E_2)$ is the reduced elastic modulus of the flanges; λ_p is the pliability of the coupled parts (flanges)

$$\lambda_p = (h_1 + h_2)/(EA). \quad (7)$$

In Eq. (7), h_1 and h_2 are the thicknesses of the flanges.

In Eq. (1), the ≥ sign indicates that the actual $\delta_{s.pl}$ and $\delta_{p.pl}$ values will be greater, since they are determined by the total forces on the screws and the flanges and not by the force F_{ti}. A more precise value of ΔF_{ti} may be found by successive approximation.

In Fig. 5, the external opening force F_G is ignored. If we take account of F_G, the sequence of calculations is as follows [6].

(1) Specify the tightening force on the screw

$$F_{ti1} = \sigma_{ti.s}\pi d_3^2/4, \quad (8)$$

where $\sigma_{ti.s} = (0.6–0.8)\sigma_y$; σ_y is the yield point of the screw.

(2) From Eqs. (1)–(7), estimate the reduction in the tightening force

$$F_{ti} = F_{ti1} - \Delta F. \tag{9}$$

(3) Calculate the basic load coefficients χ_F and χ_{My}, which determine what proportion of the external load on the screws corresponds to the opening force and the tipping torque relative to the y axis, respectively.

The basic tensile load for a group threaded joint with z screws, when the contact surface area of the flanges is $A_{co} = \pi(D_1^2 - D_2^2)/4$, corresponds to the following coefficient, according to [6]

$$\chi_F = \frac{\lambda_{co} + \lambda_p}{\lambda_{co} + \lambda_p + \dfrac{\lambda_{co.s} + \lambda_s}{z_s}}, \tag{10}$$

where λ_{co} is the pliability of the flanges' contact surfaces, which depends on the pliability k and the scale factor ε

$$\lambda_{co} = k\varepsilon/A_{co}, \tag{11}$$

$\lambda_{co.s}$ is the contact pliability of the first turn of the thread under a force $0.34F_{ti}$ [1]

$$\lambda_{co.s} = 0.5Ra_s c_0(d/50)/(0.34F_{ti}E_s A_{th})^{0.5}. \tag{12}$$

We know that

$$k = 0.5Rac_0/(Ep)^{0.5}. \tag{13}$$

According to the results in [6]

$$\chi_{My} = (\lambda_{co} + \lambda_p)/\{\lambda_{co} + \lambda_p + (\lambda_{co.s} + \lambda_s)I_y/\{A_{co}\Sigma[0.5D_2\cos(2\pi i/z)]^2\}\}, \tag{14}$$

where $I_y = \pi D_1^4[1 - (D_2/D_1)^4]/64$ is the moment of inertia of the flange's contact surface relative to the y axis.

(4) Determine the amplitude of the force on the screw

$$F_a = \chi_{My}My\frac{0.5D_s}{\sum_{i=1}^{z_s}[0.5D_s\cos(2\pi i/z_s)]^2}. \tag{15}$$

(5) Determine the static force on the screws

$$F_m = F_{ti} + \chi_F\frac{F_G}{z_s} + \chi_{My}My\frac{0.5D_s}{\sum_{i=1}^{z_s}[0.5D_s\cos(2\pi i/z_s)]^2}. \tag{16}$$

(6) Find the margin of strength of the screw shaft in terms of the fatigue limit, with allowance for the presence of a stress concentrate (the recess close to the screw head) [3]

$$S_{\sigma sh} = \sigma_{-1}/(K_{\sigma D}\sigma_{ash} + \Psi_\sigma\sigma_{msh}), \tag{17}$$

where σ_{-1} is the long-term strength of the material; $K_{\sigma D} \approx 1.42$ is the reduction in the part's fatigue limit [3]; σ_{ash} is the amplitude of the stress in the screw shaft

$$\sigma_{ash} = 0.5F_a/(\pi d^2/4), \tag{18}$$

the factor $\Psi_\sigma \approx 0.1$ characterizes the sensitivity of the material to asymmetry of the cycle; and the mean stress in the cycle is

$$\sigma_{msh} = F_m/(\pi d^2/4). \tag{19}$$

(7) Find the margin of strength of the screw's threaded section in terms of the fatigue limit [2–4]

$$S_\sigma = \sigma_{-1}/(K_{\sigma D}\sigma_a). \tag{20}$$

The effective $K_{\sigma D}$ value is 3.3–3.6 for a carbon-steel screw; 3.6–4.0 for low-alloy steel; and 4.0–4.5 for alloy steel [3].

The amplitude of the stress in the screw is

$$\sigma_a = F_a/(\pi d_3/4). \tag{21}$$

The fatigue strength is assumed acceptable if the margin of strength is no less than the permissible value $[S_\sigma]$, which is 2.5 according to [2–4].

We now assess the fatigue strength of screws in a submersible pump: screws of strength class 10.9 (σ_y = 900 MPa, σ_{-1} = 250 MPa); thread roughness $Ra_{s1} = Ra_{s2} = 2.5$ μm; steel flanges ($E = E_s = 2.1 \times 10^5$ MPa) with D_1 = 103 mm; D_2 = 76 mm; D_s = 84 mm; h_1 = 12 mm; h_2 = 45 mm. After tightening, an opening force $F_G = 8 \times 10^4$ N is applied; it acts at a distance r = 10 mm to create a tipping torque $M_y = 8 \times 10^5$ N mm. We assume that $l_0 = 0$, $l_1 = 12$ mm, $c_0 = 355$, $\Delta = 0.02$ mm, $W_{max} = 0.01$ mm.

We consider two cases: a joint with six M12 screws; and a joint with eight M10 screws. In both cases, $Ra_1 = Ra_2 = 3.2$ μm. The stress in the screw due to the tightening force is (0.6–0.9)σ_y.

From Eqs. (6)–(21), we obtain the results in Fig. 6. Calculations based on Eqs. (1)–(5) show that ΔF_{ti} is comparable with the tightening force.

CONCLUSIONS

(1) The reduction ΔF_{ti} is comparable with the tightening force. That may be attributed to the small length of the screws and flange thickness. Therefore, repeated screw tightening is required for the joints considered. Tightening in two stages is expedient: 1) tightening of all the screws by a force $1.2F_{ti}$; 2) reduction in the force to F_{ti}. Dynamometric keys must be used to preliminarily determine the required tightening torque.

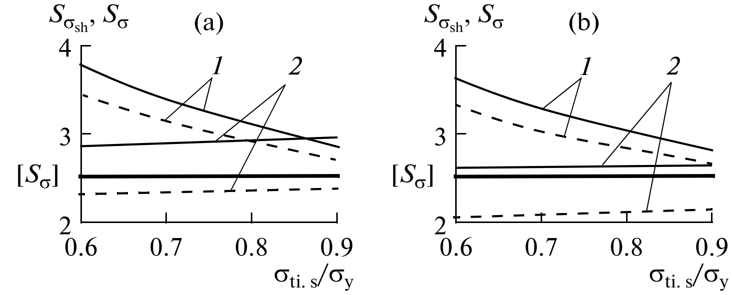

Fig. 6. Margin of strength of the screw shaft (*1*) and its threaded section (*2*) in terms of the fatigue limit for joints with six M12 screws (a) and with eight M10 screws (b) when $Ra_1 = Ra_2 = 3.2$ μm (dashed curves) and 1.25 μm (continuous curves).

(2) The margin of strength $S_{\sigma sh}$ of the screw shaft is satisfactory in joints of different design, with different tightening forces.

(3) In repeated tightening with a stress of (0.6–0.9)σ_y, when the surface roughness of the contacting flanges $Ra_1 = Ra_2 = 3.2$ μm, the margin of strength $S\sigma$ of the threaded section in terms of the fatigue limit is insufficient. If $Ra_1 = Ra_2 = 1.25$ μm, S_σ is acceptable. With increase in the tightening force, S_σ increases and $S_{\sigma sh}$ decreases.

(4) With $Ra_1 = Ra_2 = 1.25$ μm, it is expedient to replace the joint with six M12 screws by the joint with eight M10 screws, since the margin of strength is sufficient in both cases, but the distance from the edge of the threaded hole to the outer edge of the flange is increased by about 30% (from 3.5 to 4.5 mm). That ensures more uniform loading over the screw cross section.

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