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Improving Threaded-Joint Reliability in Submersible Centrifugal Pumps

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Abstract—Means of improving threaded-joint reliability in submersible centrifugal pumps are considered.

Keywords: submersible pump, reliability, threaded joint, fatigue strength
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Submersible electrical centrifugal pumps are used in oilfields. Threaded joints are used to connect the flanges of adjacent sections of the pumps (Figs. 1 and 2). Damage to the joints results in separation of the sections and their descent deeper into the borehole. Their extraction costs exceed the cost of the pump itself by an order of magnitude.

Repeated tightening of the screws (bolts, pins) was shown to be expedient in [1, 2]. In addition, recommendations were made regarding the roughness of the contact surfaces of the flanges connected by the screws and the tightening forces required to increase their fatigue strength. In the present work, we analyze the influence of other factors—aspects of the design and assembly—on the reliability of such threaded joints.

We begin by considering the influence of the flange thickness on the screw joint. With small thickness, flexurally pliable flanges bend the screws under the action of the opening load on the joint. The flexural pliability of the flanges may be reduced by increasing their thickness or by adding ribs.

In the submersible centrifugal pumps generally employed, the flange thickness is around $1.2d$, although the flange thicknesses h_1 and h_2 should be no less than their width, which is $0.5(D_1 - D_2)$ in order to prevent flexural deformation (Fig. 1). This value is $2d$ for steel flanges and $2.6d$ for cast-iron flanges [3].

We now consider the influence of the basing of the connected parts on the reliability of the threaded joint. The surface removing three or four degrees of freedom from the connected parts is regarded as the primary base, while a surface removing two degrees of freedom is a centering base, according to [4]. The dimensions of the primary base may expediently be increased, while those of the centering base may be reduced in order to remove the indeterminacy in basing [5]. In the threaded joint s of a submersible centrifugal pump, the primary base is the contacting surface of the flange, and the centering base is a collar, whose height $h \approx 1.1d$ (Fig. 1). It is expedient to reduce h by a factor of 2.5, so that it is no more than 5 mm; otherwise, the centering base may be the primary base for some of the joints. That would result in relative tipping of the contacting

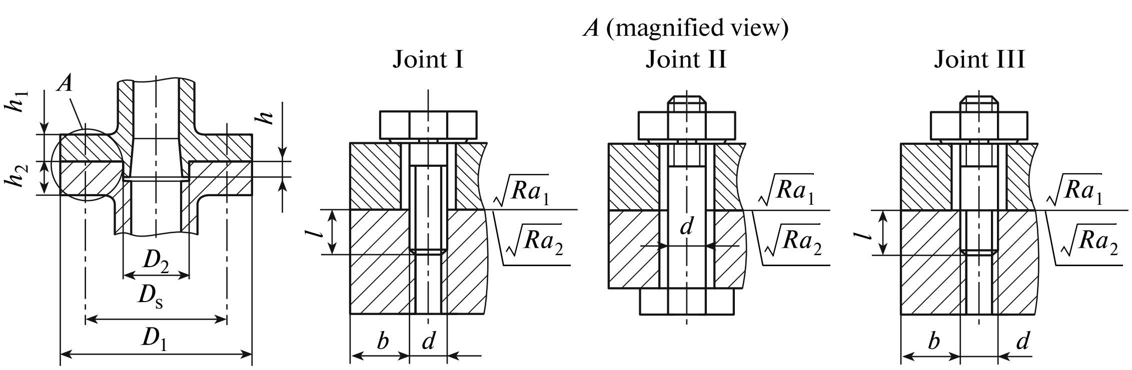

Fig. 1. Threaded joints for the flanges of submersible centrifugal pumps: (I) screws; (II) bolts; (III) pins.

Fig. 2. Threaded joint for submersible centrifugal pumps, with pins.

flange surfaces and nonuniform loading of the screws under the action of the external opening force.

We now turn to the tightening depth l of the screws, which is assumed to be around $2d$ for the threaded joints of the submersible centrifugal pump, although a depth $l \approx 1.2d$ is regarded as sufficient for tightening in steel, since greater values do not increase the strength of the joint because the force on the first loaded turn is not reduced [6]. Reducing l lowers the cost of the joint, since the screw length and depth of the threaded hole are reduced.

The next recommendation is based on the results in [2]: calculations of the fatigue strength for the screws in the threaded joints of submersible centrifugal pumps indicate that the six M12 screws in the joint may expediently be replaced by eight M10 screws (with contact surface roughness of the flanges $Ra_1 = Ra_2 = 1.25$ μm), since the strength is sufficient in both cases, and the distance b from the edge of the threaded hole to the outer edge of the flange (Fig. 1) is increased by about 30% (from 3.5 to 4.5 mm). That ensures more uniform screw loading over the cross section.

In our view, it also makes sense to consider the possibility of replacing screws with an external hexahedral contour at the head by screws with an internal hexahedron. That ensures more reliable tightening of the threaded joints, because a more convenient tightening key may be employed. In addition, that substitution permits reduction in the size of the flange's supporting surface, since the required hole diameter under the head of the screw with an internal hexahedron is smaller.

We now consider how to prevent spontaneous loosening of the joints in the submersible centrifugal pumps. At present, split spring washers (Grover washers) are used for that purpose (Fig. 1). However, they are effective only with a static load, in the absence of vibration. Therefore, the use of spring washers in automobiles was discontinued almost twenty years ago in the United States and Western Europe [7, 8]. With variable load or vibration, as a rule, spontaneous loosening of the joints is prevented by plastic deformation of a washer or wire or by the application of a fixing liquid or gel to the screw thread.

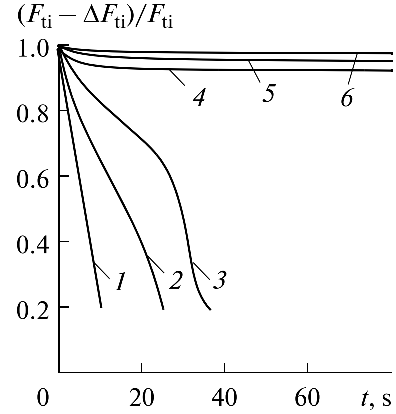

Fig. 3. Dependence of the relative decrease in the tightening force $(F_{ti} - \Delta F_{ti})/F_{ti}$ on the time t according to dynamic tests of the spontaneous loosening of threaded joints: (*1*) without any means of preventing spontaneous loosening; (*2*) with a spring washer; (*3*) with a plastic insert in the nut; (*4*) with a toothed washer; (*5*, *6*) with the application of dry and wet anaerobic glue to the thread, respectively.

To assess the resistance to spontaneous loosening under the action of vibration, the Junker test was developed in Germany (DIN 65151–2002 standard).

In Fig. 3, we show the dependence of the relative decrease in the tightening force $(F_{ti} - \Delta F_{ti})/F_{ti}$ on the time t according to dynamic tests of the spontaneous loosening of threaded joints (Junker tests) [9]. The preliminary tightening force is monitored by tensoresistors. Each method is tested until 80% of the preliminary tightening force has been lost. If that does not occur within 80 s, the tests end. The duration of the tests is 10 s in the absence of any means of preventing spontaneous loosening (curve *1*); 25 s with a spring washer (curve *2*); 35 s with a plastic insert in the nut (curve *3*); and 80 s with a tooth disk (curve *4*) and with the application of dry (curve *5*) and wet (curve *6*) anaerobic glue to the thread. Thus, the application of sealants is the most reliable method.

In Russia, Anaterm, Unigerm, and other glues and sealants are applied to the thread. Globally, one of the major producers of such products is Henkel, which manufactures Loctite glues and sealants. Anaterm-114 and Loctite 243 are used for joints subject to vibration and threaded joints that may need to be disassembled, while Unigerm-9 and Loctite 2701 are used for threaded joints that do not require regular disassembly.

The working temperature range for anaerobic fixing fluids runs from –60 to +150°C. We know that, in an oil borehole, the mean bed temperature increases by 3°C with each 100-m increase in depth and is 125–145°C at 2500–4500 m. Therefore, with increased thermal requirements, special-purpose materials must be used, such as Anaterm-117, which retains its properties in the solid state at temperatures up to 200–250°C.

The use of anaerobic sealants not only eliminates the need for any preventive measures but also increases

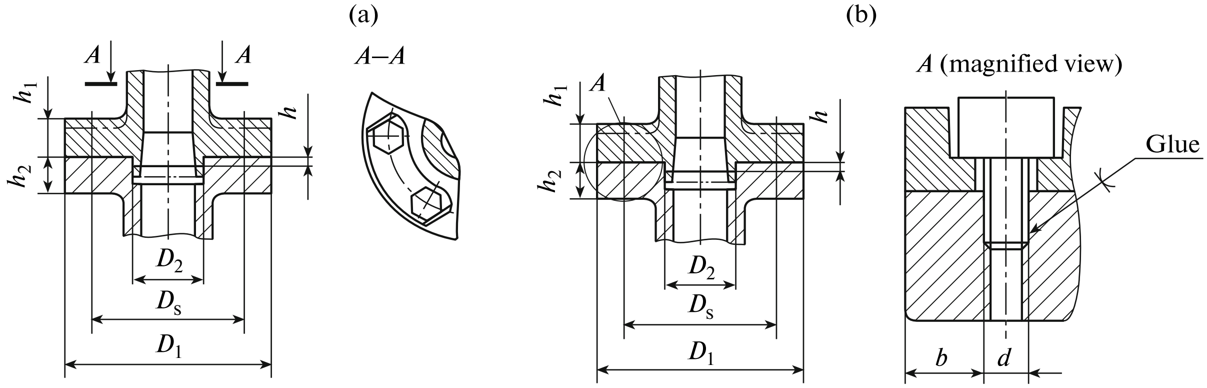

Fig. 4. Means of increasing the reliability of threaded joints in submersible centrifugal pumps: (a) increase in h_1, decrease in h, use of a washer for two holes; (b) increase in b by using more screws of smaller diameter or screws with an internal hexahedron. The dash–dot lines indicate the structure before modernization.

the static and cyclic strength of the joint as a whole, since the hardened polymer filling the space between the thread turns evens out the load distribution over the thread [10].

Thus, for threaded joints with an external hexahedron, we may recommend the use of a disk permitting plastic deformation; for threaded joints with an internal hexahedron, we recommend the use of a thread sealant or glue.

Analysis of the proposed measures for increasing the reliability of threaded joints in submersible centrifugal pumps (Fig. 4) permits the following recommendations.

(1) To increase the flexural rigidity of the flange, its thickness h_1 may be increased to $2d$.

(2) For screws and nuts, spring washers must be replaced by washers with two holes for two screws (Fig. 4a, *A*–*A*) permitting plastic deformation or the use of fixative sealant or glue (Fig. 4b, view *A*).

(3) Since the forces in the screws due to external loads decline with increase in the tightening force and decrease in contact surface roughness, we must improve the machining of those surfaces so that Ra = 1.25 μm.

(4) The reliability of the joint will be improved by repeated tightening of the screws or nuts in a specified sequence, with the use of a dynamometric key.

(5) To remove the indeterminacy in basing, decrease in height h of the collar's centering surface is recommended.

(6) Reducing the diameter of the screws with increase in their number and using screws with an internal hexahedron permits increase in the flange wall dimension b (Fig. 4b).

(7) To increase the fatigue strength of the screws when lowering the pump in the borehole, its rate of descent must be monitored, and the rate of braking at stops must be reduced.

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