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DESIGN OF STUD ENDS AND THEIR INFLUENCE ON LOAD OF MACHINE HOUSINGS

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Abstract: *The article discusses a calculation procedure for determining the load and stress conditions in stud joints. There are considered two based types of stud fixing: with a circular rim and with an abutting end. There are given the equations for calculation of tightening torques for both types of studs. It is shown the load distribution in joined parts and by help of experimental research is shown that studs with the circular rim are a bit stronger that the studs with the abutting end.*

Keywords: CONDITION OF COMMON DEFORMATIONS, LOAD DISTRIBUTION, STUD, CIRCULAR RIM, ABUTTING END

1. Introduction

Studs are widely used for joint of machine housings and frames made of light alloys and cast iron and sometimes made of steel and titanium alloys. Components with studs can for example be some joints, covers, frames, or cylinders. In some designs, when it is necessary to dismount the parts very often, is not recommended to use the screws, which must be screwed into the frame because the internal thread can be broken. In these cases, the studs are usually used. To eliminate the rotation during tightening, the stud must be fixed into the frame enough tightly. There are various methods of stud fixing into the frame [1], which can be evaluated, based on the efficiency of stopping, stability of the tightening and the reliability of joints.

The breakage of the joints in the nut zone occurs more often than in the stud end [2]. However, in many designs, the stud end is at more intense condition than the nut because of its screwing up in the frame with the preliminary tightening. Stress values in critical points depend on load distribution on the threads, bending and shearing stresses into threads, coefficients of stress concentration, mechanical properties of part materials, and method of stud fixing.

In this work is considered changing of the maximal load on the thread by some methods of stud fixing in the frame. The frame is presented as a nut with the external diameter $d_e = 4d$ since the stress on the external surface of the nut accounts only for 1/16 of the maximal stresses [3].

2. Condition of common deformations

Studs usually are loaded by more complicated loading schemes than classical bolt-nut joints. Therefore, it is necessary to analyze load distribution in common case of joint loading before of considering of load distribution into studs.

Condition of common deformations has a usual form

$$\Delta_1 + \Delta_2 = [\delta_1(z) + \delta_2(z)] - [\delta_1(0) + \delta_2(0)], \quad (1)$$

where $\Delta_1 = \int_0^z \frac{\sigma_1(z)}{E_1} dz$ and $\Delta_2 = \int_0^z \frac{\sigma_2(z)}{E_2} dz$ are stud elongation and frame contraction, accordingly. Hereinafter index 1 and index 2 will be used for the stud and for the housing accordingly. $\delta_1(0) + \delta_2(0)$ and $\delta_1(z) + \delta_2(z)$ are axial displacements of the threads along the pitch diameter d_2 in cross sections $z = 0$ and z. $\sigma_1(z)$ and $\sigma_2(z)$ are stresses in the cross section z. E_1 and E_2 are modulus of elasticity of the materials.

Commonly, four forces may act on a threaded joint. These forces are (Fig. 1)

$$F_1 + F_2 = F_3 + F_4 . \quad (2)$$

Let us consider a case when both threaded parts are stretched. From the equilibrium condition of the forces acting on the screw, can be obtained

$$\int_0^H q(z) dz = F_1 - F_4 \text{ and for the nut } \int_0^H q(z) dz = F_3 - F_2 . \quad (3)$$

The forces $F_B(z)$ and $F_N(z)$ acting on any cross-sections z of the screw and the nut will be

$$F_B(z) = F_4 + \int_0^z q(z) dz \text{ and } F_N(z) = F_3 - \int_0^z q(z) dz . \quad (4)$$

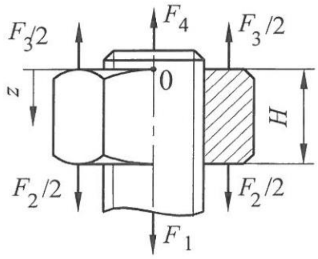

Fig. 1. *Common case of loading on the threaded joint.*

The deformations of the screw shank and the nut body can be determined as follows

$$\Delta_1 = \frac{1}{E_1 A_1} \int_0^z F_B(z) dz = \frac{1}{E_1 A_1} \int_0^z \left(F_4 + \int_0^z q(z) dz \right) dz , \quad (5)$$

$$\Delta_2 = \frac{1}{E_2 A_2} \int_0^z F_N(z) dz = \frac{1}{E_2 A_2} \int_0^z \left(F_3 - \int_0^z q(z) dz \right) dz , \quad (6)$$

where A – area of cross-section, q – intensity of load distribution.

Let us substitute these equations into the Eq. (1). The value of Δ_2 is accepted with (–) since the nut is stretched. Differentiating Eq. (1) with respect to z can be obtained

$$\beta \cdot \int_0^z q(z) dz + \frac{F_4}{E_1 A_1} - \frac{F_3}{E_2 A_2} = \gamma \cdot \dot{q}(z) + \xi \cdot q(z), \quad (7)$$

where

$$\delta_1(z) = \delta_1'(z) + \delta_1''(z), \quad \delta_2(z) = \delta_2'(z) + \delta_2''(z),$$

$$\delta_1'(z) = \frac{p(z) \cdot P}{E_1} \lambda_1, \quad \delta_2'(z) = \frac{p(z) \cdot P}{E_2} \lambda_2 ,$$

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$$\delta_1''(z) = \frac{\sigma_1(z)}{E_1}\mu_1 \frac{d_2}{2}\tan\frac{\alpha}{2},$$

$$\delta_2''(z) = \frac{\sigma_2(z)}{E_2}\mu_2 \frac{d_2}{2}\tan\frac{\alpha}{2},$$

$$\beta = \frac{1}{E_1A_1} + \frac{1}{E_2A_2},$$

$$\gamma = \left(\frac{\lambda_1}{E_1} + \frac{\lambda_2}{E_2}\right)\cdot\frac{P^2}{\pi\cdot d_2 t},$$

$$\xi = \left(\frac{\mu_1}{E_1A_1} + \frac{\mu_2}{E_2A_2}\right)\cdot\frac{d_2}{2}\tan\frac{\alpha}{2},$$

P – thread pitch, t – thread overlap, α – angle of thread profile, μ – Poisson's ratio of material, λ – coefficient depending on mechanical properties of material and geometrical parameters of thread and joint as a whole.

Let us repeat the operation of differentiation to obtain

$$\ddot{q}(z) + \frac{\xi}{\gamma}\dot{q}(z) - mq(z) = 0.$$

The common integral of this equation will be

$$q(z) = C_1 \cdot e^{(b-a)z} + C_2 \cdot e^{-(b+a)z}.$$

Then

$$\dot{q}(z) = (b-a)\cdot C_1 \cdot e^{(b-a)z} - (b+a)\cdot C_2 \cdot e^{-(b+a)z}.$$

The equation (7) can be rewritten as

$$\frac{\beta}{\gamma}\cdot\int_0^z q(z)dz + \frac{F_4}{\gamma\cdot E_1A_1} - \frac{F_3}{\gamma\cdot E_2A_2} = \dot{q}(z) + 2a\cdot q(z), \quad (8)$$

where

$a = \dfrac{\xi}{2\gamma}$, $\quad b = \sqrt{\left(\dfrac{\xi}{2\gamma}\right)^2 + m}$, and constants C_1 and C_2 can be determined from the initial conditions on $z = 0$ and $z = H$.

After transformations, the constants C_1 and C_2 will be

$$C_1 = \frac{1}{\gamma(b+a)}\left[\frac{F_4}{E_1A_1} - \frac{F_3}{E_2A_2}\right] + $$
$$+ \frac{e^{aH}}{2\gamma(b+a)\sinh bH}\left[\frac{F_1}{E_1A_1} - \frac{F_2}{E_2A_2} + \frac{F_3 e^{(b-a)H}}{E_2A_2} - \frac{F_4 e^{(b-a)H}}{E_1A_1}\right]$$

$$C_1 = \frac{e^{aH}}{2\gamma(b-a)\sinh bH}\left[\begin{array}{l}\dfrac{F_1}{E_1A_1} - \dfrac{F_2}{E_2A_2} + \dfrac{F_3 e^{(b-a)H}}{E_2A_2} - \\ - \dfrac{F_4 e^{(b-a)H}}{E_1A_1}\end{array}\right].$$

Then the intensity of load distribution on the working threads can be presented as follows

$$q(z) = \frac{\pm e^{(b\mp a)z}}{\gamma(b\pm a)}\cdot\left[\frac{F_4}{E_1A_1} - \frac{F_3}{E_2A_2}\right] \pm$$
$$\pm \frac{e^{(\pm H \mp z)a}}{\beta\cdot\sinh bH}\left[\frac{F_1 - F_4 e^{(b\mp a)H}}{E_1A_1} - \frac{F_2 - F_3 e^{(b\mp a)H}}{E_2A_2}\right]\times \quad (9)$$
$$\times\left(b\cdot\cosh bz \mp a\cdot\sinh bz\right)$$

The lower signs correspond to the scheme of loading when the compressing force act on the screw.

3. Analysis of load and torque

It is a usual practice to induce force of preliminary tightening F_0 initially upon the assembly of the parts before the external load F_e on the stud is applied. After screwing of the stud into the frame, the axial force F_t arises. The resultant force that is carried by the stud screwed into the frame may be written as $F = F_t + F_0 + \chi F_e$, where χ is a coefficient of the external load, $\chi = 0.2 \ldots 0.4$ [2].

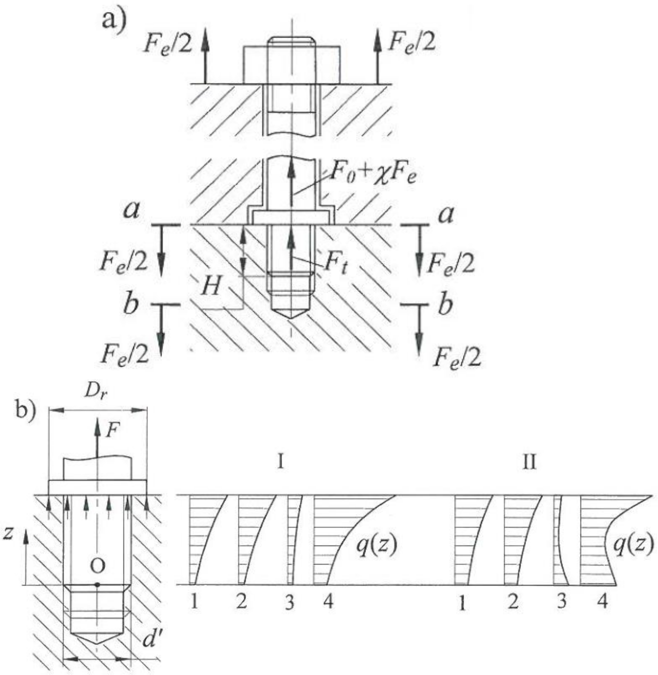

Fig. 2. *Diagram of the stud fixing with the circular rim.*
b) The load distribution on the threads:
I – force F_e in the section a – a is applied,
II – force F_e in the section b – b is applied;
1 – from F_t, 2 – from F_0, 3 – from χF_e, 4 – from total load.

The initial stress is established according to the conditions of joint tightness. It can be decreased due to the squeeze of thread roughness, stress relaxation etc. and increased due to hydraulic impact, works on resonant mode etc. Therefore, the initial stress can be changed by η times. Then the axial tightening force can be expressed as $F_0 = \eta F_e (1 - \chi)$, where the safety factor $\eta = 1.25 \ldots 4.0$ [2]. After assembling of the stud joint is necessary to apply the torque M_0 to create a sufficient axial force F_0 to provide joint tightness. The tightening torque must be

$$M_0 = \eta\cdot F_e(1-\chi)\frac{d_2}{2}\left[\frac{P}{\pi\cdot d_2} + f_1'\right], \quad (10)$$

where $f_1' = f_1 \cos\frac{\alpha}{2}$, f_1 is the friction factor in the frame and stud zone.

At the same time, the torque M_0 is equal to the sum of the friction moment in the threads M_s and the friction moment M_r between the stud and frame supporting surfaces, i.e.

$$M_0 = M_s + M_r\,. \qquad (11)$$

Let us consider stud fixing into the frame with a circular rim (Fig. 2). The fixing of the stud causes the axial force F_t in the stud and frame contact zone. Then

$$M_s = F_t \frac{d_2'}{2}\left(\frac{P'}{\pi \cdot d_2'} + f_2'\right) \quad \text{and}$$

$$M_r = \frac{F_t f_3}{3} \cdot \frac{D_r^3 - (d')^3}{D_r^2 - (d')^2}\,, \qquad (12)$$

where $f_2' = f_2 \cos\frac{\alpha}{2}$, f_2 is the friction factor in the stud and frame zone, f_3 is the friction factor on the supported surface of the rim.

From Eqs. (10) – (12) the necessary axial force F_t can be obtained as follows

$$F_t = \eta \cdot F_e(1-\chi)\frac{d_2\left(\frac{P}{\pi \cdot d_2} + f_1'\right)}{d_2'\left(\frac{P'}{\pi \cdot d_2'} + f_2'\right) + \frac{2 f_3\left[D_r^3 + (d')^3\right]}{3\left[D_r^2 + (d')^2\right]}}\,. \qquad (13)$$

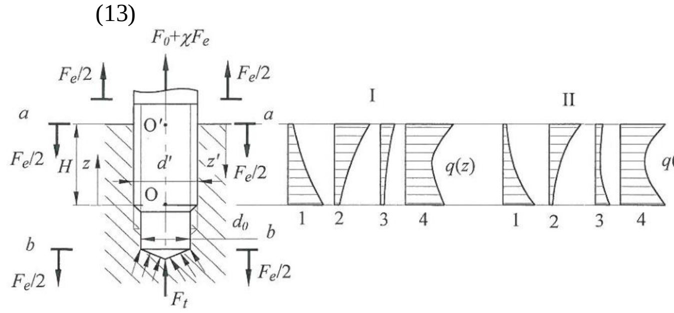

Fig. 3. *Diagram of the stud fixing with the abutting end.*
b) The load distribution on the threads:
I – force F_e in the section a – a is applied,
II – force F_e in the section b – b is applied;
1 – from F_t, 2 – from F_0, 3 – from χF_e, 4 – from total load

If the stud is fixed in the frame with an abutting end (Fig. 3), then

$$M_r = \frac{F_t f_3 d_a}{3} \quad \text{and}$$

$$F_t = \eta \cdot F_e(1-\chi)\frac{d_2\left(\frac{P}{\pi \cdot d_2} + f_1'\right)}{d_2'\left(\frac{P'}{\pi \cdot d_2'} + f_2'\right) + \frac{2 f_3 d_a}{3}} \qquad (14)$$

From Eqs. (13) and (14) it follows that the force F_t is identical in both methods of stud fixing if $d_a = \frac{D_r^3 + (d')^3}{D_r^2 + (d')^2}$. In this case $d_a > d'$, however, it is structurally impossible. Therefore, the force F_t will be always higher at stud fixing with the abutting end than at fixing with the circular rim (with other things being equal).

The forces F_0 and F_e have an identical influence on load distribution in both methods of stud fixing. The force F_0 stretches the stud and compresses the frame. The force F_e compresses the frame and stretches the stud if it is applied to section *a – a* and stretches both the stud and the frame if it is applied to section *b – b*.

The force F_t stretches the stud and compresses the housing at fixing with the circular rim. At fixing with the abutting end this force compresses the stud and stretches the housing. It means that the difference of the maximal load value on the thread takes place only due to the action of the force F_t.

Qualitative diagrams of the $q(z)$ are shown in Figs. 2 and 3. More frequently are used studs M8 and M10 [1]. In Table 1 are shown the results of load calculation in critical sections for such joints.

Table 1. *The load on stud and frame threads, % of F, (the stud made of steel, f = 0.15, n = 10)*

Case material	Thread	Type of fixing			
		circular rim		abutting end	
		cross- section			
		$z = H$	$z = 0$	$z = H$	$z = 0$
Steel	M8x1.25	34.60	1.36	0.48	41.79
	M10x1.5	33.58	1.53	0.55	40.69
Cast iron	M8x1.25	28.98	2.44	1.31	33.75
	M10x1.5	28.08	2.67	1.46	32.77
Aluminum alloy	M8x1.25	27.53	2.82	1.60	31.96
	M10x1.5	26.66	3.06	1.76	31.01

Table 1 shows that most of load F_t is applied to section $z = H$ on fixing with the circular rim and to section $z = 0$ on fixing with the abutting end. Then the complete loading $F(i) = F_t(i) + F_0(i) + \chi \cdot F_e(i)$ in section $z = H$ (critical section of the stud) will be higher on fixing with the circular rim than with the abutting end. It means that the reliability of the stud with the abutting end should be higher than the stud with the circular rim for an ideally made and fixed joint. In practice, because of inaccurate manufacturing and fixing of threaded parts, in the thread a bending moment arises, decreasing the stud's reliability. The circular rim (Fig. 3) allows a decrease in the influence of the bending moment and an increase in the joint reliability [2].

The section $z = 0$ is the critical section for the frame. Total load $F(i)$ in this section will be greater for stud fixing with the abutting end than with the circular rim. It should be remembered that the value of force F_t is also greater for such type of fixing. It means that on joining the frame and parts by the studs with a small number of working threads, fixing with the circular rim should be preferred.

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4. Experimental research

Tests of stud specimens were made on a tensile machine. Two methods of stud fixing in the frame were investigated: with circular rims and with abutting ends. Studs had the thread M10x1.5 and M8 with pitches P = 1.25, 1.0 and 0.75 mm. Studs were made of medium-carbon steel. Nuts were made of the same steel and of cast iron. The number of working threads was equal for both types of studs. The frame was present by nuts with the external diameter d_e = 4d. The torque was equal to 14 Nm for the thread M10 and 8 Nm for the thread M8.

Experimental results have shown that studs with the circular rim are about 6% … 8% stronger than studs with the abutting end. It agree with the results of fatigue tests [2]. All studs are broken on section a – a (Figs. 2 and 3). Studs with the circular rim are stronger owing to a decrease in bending in the critical section of the stud.

Frame material (medium-carbon steel and cast iron), does not influence stud reliability, difference is less than 1%. Studs reliability increases with a decrease in thread pitch explained by an increase in the stud cross-section not by the leveling of load distribution. Circular rims allow a decrease in the influence of the bending moment and an increase in joints reliability. The reliability of the frame also increases when the stud is fixed with the circular rim.

Conclusions

The force of preliminary tightening and external load on the joint have an identical influence on load distribution in stud joints at stud fixing with a circular rim and an abutting end. The difference in load distribution on the working threads takes place only due to the action of axial load arisen after stud fixing in the frame. The total load in the stud critical section will be more on fixing of the stud having a circular rim but owing to manufacturing inaccuracy and fixing of threaded parts, the bending moments lowering the stud reliability occurs. The circular rim allows to decrease the influence of the bending moment and to increase the reliability of the joint. The reliability of the housing also increases at stud fixing by help of the circular rim.

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