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Fatigue life prediction for cyclically bent threaded connections

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1. Introduction

Threaded connections applied for high pressure vessel covering must have adequate mechanical strength and good leak tightness. Often in gasketed flanges the variations of internal pressure, temperature and deformation of joined-up elements cause cyclic bending of these elements and threaded connections also [1-7]. Due to the high stress concentrations at thread roots and cyclic loading these connections are prone to fatigue failure.

Low fatigue strength of important threaded connections is defined in accordance with the ASME code [6] or Russian Federation standard [7]. These methods require the maximum stress concentration in thread root to be evaluated and this can be done without using a knowledge of the load distribution along the thread helix.

More exact analitical calculation of the stress at the thread roots can be achieved by direct using of load distribution data.

In the field of high cycle fatigue a several methods have been suggested to evaluate the fatigue limit of a bolt by employing appropriate theories or empirical laws to calculate the load along the thread helix followed by the maximum stress concentration and then the fatigue limit. The analysis of these methods is performed in [8].

In the field of low cycle fatigue i.e. in [9], the analytical method [10] of calculation of load distribution between turns (which secondly is used for the calculation of maximum stresses at the thread roots) was incorporated into standard method [7] which in [9] was used for the predicting of threaded connection's fatigue life.

Until now analytical methods for evaluating fatigue limit or fatigue lifetime of threaded connections using load distribution data, were realizable only in the case of theirs axial cyclic loading. Recently the analytical method to describe the load distribution in thread of bent threaded connection was developed [11], and now the above mentioned evaluating can be performed for the cyclically bent threaded connection also.

The object of this paper is to assess potential of the analytical method [11] in predicting fatigue life of cyclically bent threaded connections by combining this method with standard [7]. For this purpose calculation results have been assessed against fatigue test data which are expressed in terms of numbers of load cycles at fatigue crack initiation in studs/bolts of threaded connections.

2. Positions of the segments and thread helix in the threaded connection

In this paper a threaded connection Fig. 1, a, b which after tightening by force F_t is loaded by the bending moment M_f is considered. Due to action of these external loads the appropriate loads per unit length $q_t(z_i)$ and $q_b(z_i)$ arise in the stud and nut threads. This causes the proportional turns pair deflections $\delta_t(z_i)$ and $\delta_b(z_i)$

$$\delta_t(z_i) = \gamma(z_i)q_t(z_i), \qquad \delta_b(z_i) = \gamma(z_i)q_b(z_i) \tag{1}$$

where $\gamma(z_i)$ is turns pair pliability and z_i is cross-section location coordinate.

Threaded connection is divided in three segments: $I = 1, 2, 3$. In the middle segment the length of which is H_2, the turns are engaged over the full profile and depth of turns' engagement is constant. Therefor here pliability of the turns' pair $\gamma(z_2) = \gamma = const$ does not change along all length. In the boundary segments of the connection, i.e. on runouts, where their length is $H_1 = H_3 = P$ (P is thread pitch), due to the changing depth of turns' engagement their pliability is varying being $\gamma(z_1) \neq const$ and $\gamma(z_3) \neq const$. The origin and end of any segment are found at z_{0i} and z_{Hi} respectivelly.

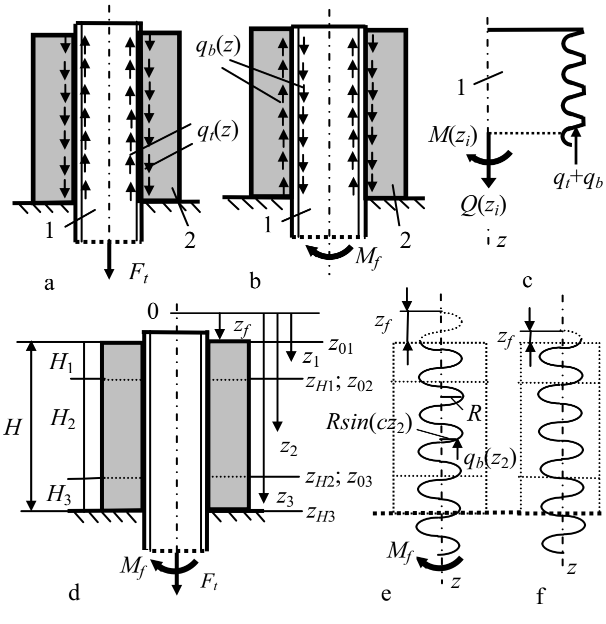

Fig. 1 Loading of the threaded connection: *1* - stud, *2* - nut, a – tightening, b – bending, c – load factors cause the stresses in studs' turn root, d – segments of threaded connection, e, f – two different positions of the thread pitch diameter helix

It can be seen in Fig. 1, d that the origin of coordinate z_i of any cross-section location is receded from the free end of the nut on a phase length z_f, which is designed to set a position of the threaded connection with respect to longitudinal axis thus with respect to bending plane also. Position for any thread helix point now can be expressed by turning angle $\alpha(z_i) = (2\pi/P)z_i = cz_i$. After installation of the threaded connection in to construction and after its tightening the helix can occur in any position with respect to bending plane. Herewith in every new thread helix position the considered helix point on the thread pitch diameter $2R$ where acts the appropriate turn load $q_b(z_i)$ will be differently remote from the neutral line (suppose M_f acts alone). Here there is no trouble to notice that turn load $q_b(z_i)$ then creates different local moments $m(z_i) = q_b(z_i)R\,sin(cz_i)$ in the same cross-section of connection. Two examples of this distance variation (i.e. $R\,sin(cz_i)$) which can be obtained by using two different values z_f are shown in Fig. 1, e, f.

In Fig. 1, c internal axial force $Q(z_i)$, internal bending moment $M(z_i)$ and turn loads $q_t(z_i) + q_b(z_i)$ which act in the studs' cross section at coordinate z_i and produce local stresses in the thread root are shown. They all must be calculated primarily when fatigue life of the threaded connection is predicted.

3. Equations for thread loads and stresses due to tightening of threaded connection

The turn load $q_t(z_i)$ and internal axial force $Q(z_i)$ caused by tightening have been calculated by using the method given in [10]. These loads for every segment are expressed through turns' deflection in the following way

$$q_t(z_i) = \frac{\delta_t(z_i)}{\gamma(z_i)}, \quad Q(z_i) = \frac{\delta_t'(z_i)}{\beta} \tag{2}$$

where $\beta = 1/(E_sA_s) + 1/(E_nA_n)$; E_s, E_n are Young's moduli of the stud and nut; A_s, A_n are cross-sectional areas of the stud core and nut wall.

For the second segment where turns' pliabilities are $\gamma(z_2) = \gamma = const$ the turns' deflection and its derivative are expressed by using Birgers' theory

$$\delta_t(z_2) = A_{t2} sinh(m_t z_2) + B_{t2} cosh(m_t z_2) \tag{3}$$

$$\delta_t'(z_2) = A_{t2} m_t cosh(m_t z_2) + B_{t2} m_t sinh(m_t z_2) \tag{4}$$

where $m_t = \sqrt{\beta/\gamma}$.

Expression for the turns' pliabilities in runouts (where $i = 1$ or $i = 3$) has the following form [10]

$$\gamma(z_i) = \frac{\beta}{C_i^2 e^{2n_i z_i} + C_i n_i e^{n_i z_i}} \tag{5}$$

where factor C_i and power exponent n_i can be defined according to the test results of the turns' pairs, engaged over the incomplete profile for every segment - they have been calculated by using the known turns' pliability in one edge of segment H_1 or H_3 where $\gamma(z_{H1(03)}) = \gamma$, and also the experimental turns' pliability factor in the middle of these segments $\gamma(z_{01}+P/2) = 1.67\gamma$ or $\gamma(z_{03}+P/2) = 1.67\gamma$.

The turns pair deflection and its derivative for the runouts, i.e. for segments $i = 1, 3$ are

$$\delta_t(z_i) = A_{ti} e^{-\frac{\tau_i}{2}} + \frac{B_{ti}}{n_i} e^{-\frac{\tau_i}{2}} \left(ln\tau_i + \frac{\tau_i}{1\cdot 1!} + \frac{\tau_i^2}{2\cdot 2!} + ... \right) \tag{6}$$

$$\delta_t'(z_i) = A_{ti}(-\tau_i n_i e^{-\tau_i/2}) + \\ + B_{ti}\left(-\tau_i e^{-\tau_i/2} \left(ln\tau_i + \frac{\tau_i}{1\cdot 1!} + \frac{\tau_i^2}{2\cdot 2!} + ... \right) + e^{\tau_i/2} \right) \tag{7}$$

where $\tau_i = -2\frac{C_i}{n_i} e^{n_i z_i}$.

To estimate the six unknowns factors A_{ti} and B_{ti} of equations (3), (4), (6) and (7) is used the system of equations, which express boundary conditions of segments

$$Q(z_{01}) = 0, \quad Q(z_{H1}) = Q(z_{02}), \quad Q(z_{H2}) = Q(z_{03}), \\ Q(z_{H3}) = F_t, \quad \delta_t(z_{H1}) = \delta_t(z_{02}), \quad \delta_t(z_{H2}) = \delta_t(z_{03}) \tag{8}$$

In strength calculation norm for nuclear equipments [7] the fatigue durability is estimating according the local alternating elastic stresses σ^*. These stresses for axial loaded stud thread which arise at tightening were calculated by using the following formula [2]

$$\sigma_t^*(z) = \frac{q_t(z)P}{f} K_{m,t} + \frac{Q_t(z)}{A_s} K_{0,t} \tag{9}$$

here $K_{0,t}$, $K_{m,t}$ are concentration factors of stresses due to the axial force $Q_t(z)$ and the stud turn load $q_t(z)$ respectively; A_s is cross-sectional area of the stud core; f is the turns' contact surface projection into the plane, perpendicular to the stud axis; P is the thread pitch. The values of elastic stresses concentration factors, defined in work [2] are: $K_{0,t} = 2$ and $K_{m,t} = 1{,}95$, at the turns' root rounding-up radius being $R = 0.144P$.

4. Equations for thread loads and stresses due to bending of threaded connection

The turn load $q_b(z_i)$ and internal bending moment $M(z_i)$ which arise at the bending of the threaded connection have been calculated by using the method given in [12]. For every segments of connection here are used the relations

$$q_b(z_i) = \frac{\delta_b(z_i)}{\gamma(z_i)}, \quad \delta_b(z_i) = y(z_i)\, sin(cz_i) \tag{10}$$

where $y(z_i)$ is the function of turns' maximum deflections.

For the second segment where turns' pliabilities are $\gamma(z_2) = \gamma = const$ the above mentioned function is the following

$$y(z_2) = A_{b2}\, sinh(m_b z_2) + B_{b2}\, cosh(m_b z_2) \tag{13}$$

where $m_b = \sqrt{-2c^2 + c\sqrt{4c^2 + 2b}}$, $b = R^2\lambda/\gamma$, $\lambda = 1/(E_sI_s) + 1/(E_nI_n)$ are constant factors and I_s, I_n are moments of inertia of the cross-sectional area for the stud core and the nut wall respectively.

The equation obtained in [12] for the calculation of internal bending moment which acts in stud core of the segment H_2 has the following form

$$M(z_2) = A_{b2} \frac{R}{\gamma} \frac{1}{m_b^2 + 4c^2} \left[m_b\, cosh(m_b z_2)\, sin^2(cz_2) - \right. \\ \left. - c\, sinh(m_b z_2)\, sin(2cz_2) + (2c^2/m_b)\, cosh(m_b z_2) \right] + \\ + B_{b2} \frac{R}{\gamma} \frac{1}{m_b^2 + 4c^2} \left[m_b\, sinh(m_b z_2) sin^2(cz_2) - \right.$$

$$-c\ cosh(m_b z_2)\ sin(2cz_2)+(2c^2/m_b)\ sinh(m_b z_2)\big]\quad(14)$$

For mathematical purposes the variation of the turn's pair pliability in length of any runout in the bent threaded connection in [12] is described by other formula than in the case of tension connection. This formula has the following expression

$$\gamma(z_i)=V_i e^{u_i z_i}\quad(15)$$

where V_i and u_i ($i=1$ or $i=3$) are constant factor and power exponent which have been defined in the same way as in the case of tight threaded connection, i.e. by using two experimentall turns' pair pliability values known for these segments.

The analytical expression of the function for the runouts' turn maximum deflection in [12] was obtained in the following form

$$y(z_i)=A_{bi}e^{n_{Ai}z_i}(n_{Ai}z_i+W_{Ai})+B_{bi}e^{n_{Bi}z_i}(-n_{Bi}z_i+W_{Bi})= \\ =A_{bi}f_A(z_i)+B_{bi}f_B(z_i)\quad(16)$$

where n_{Ai}, W_{Ai}, n_{Bi} and W_{Bi} are the factors which need to be found, f_{Ai} and f_{Bi} are designations, $i=1$ or $i=3$.

The equation for the calculation of internal bending moment which acts on the stud core in segments $i=1$ or $i=3$ is

$$M(z_i)=A_{bi}F_A(z_i)+B_{bi}F_B(z_i)\quad(17)$$

where $F_A(z_i)$ and $F_B(z_i)$ are designations which are expressed in the following common form

$$F_\omega(z)=\frac{Re^{tz}}{2V}\left[\pm\frac{n_\omega(tz-1)}{t^2}\mp n_\omega\left(\left(tz-\frac{t^2-4c^2}{p}\right)\frac{cos(2cz)}{p}-\right.\right. \\ \left.-\left(\frac{4ct}{p}-2cz\right)\frac{sin(2cz)}{p}\right)-\frac{W_\omega}{p}\big(t\,cos(2cz)+2c\,sin(2cz)\big)+ \\ \left.+\frac{W_\omega}{t}\right]\quad(18)$$

where index $\omega=Ai$ or $\omega=Bi$, $t=t_\omega=n_\omega-u_i$, $V=V_i$, $p=p_\omega=t^2+4c^2$, $z=z_i$, $i=1$ or $i=3$; where are dual signs the upper sign is valid at $\omega=Ai$ and the under sign is valid in the case of $\omega=Bi$.

The factors n_{Ai}, W_{Ai} and n_{Bi}, W_{Bi} for any runout must be solved from the two equation systems

$$f_A'(z_i^*)/R\lambda=F_A(z_i^*),\quad f_A'(z_i^{**})/R\lambda=F_A(z_i^{**})\quad(19)$$

$$f_B'(z_i^*)/R\lambda=F_B(z_i^*),\quad f_B'(z_i^{**})/R\lambda=F_B(z_i^{**})\quad(20)$$

where $z_i^*=z_{0i}+P/4$ and $z_i^{**}=z_{Hi}-P/4$, by using

$$f_A'(z_i)=n_{Ai}e^{n_{Ai}z_i}(n_{Ai}z_i+W_{Ai}+1)\quad(21)$$

$$f_B'(z_i)=n_{Bi}e^{n_{Bi}z_i}(-n_{Bi}z_i+W_{Bi}-1)\quad(22)$$

The unknowns factors A_{bi} and B_{bi} of Eqs. (13), (14), (16), (17) further must be estimated by using the system of equations, which express segments boundary conditions

$$M(z_{01})=0,\quad M(z_{H1})=M(z_{02}),\ M(z_{H2})=M(z_{03}), \\ M(z_{H3})=M_f,\ \delta_b(z_{H1})=\delta_b(z_{02}),\ \delta_b(z_{H2})=\delta_b(z_{03})\quad(23)$$

The alternating elastic stresses in the stud thread due to bending were calculated on the analogy of Eq. (9)

$$\sigma_b^*(z)=\frac{q_b(z)P}{f}K_{m,b}+\frac{M(z)R\ sin(cz)}{I_s}K_{0,b}\quad(29)$$

where $K_{0,b}\approx k_r K_{0,t}$ and $K_{m,b}\approx K_{m,t}$ are concentration factors of the stresses due to bending moment $M(z)$ and due the stud turn load $q_b(z)$ respectively; k_r is the factor which estimates the difference between local stresses in the stud at bending and at its tension; here was assumed $k_r\approx$ 0.9.

5. Load and stresses distributions along the thread helix

Figs. 2-7 presented in this Section reflect the calculation sequence for the stresses cycle parameters which are used further for the studs fatigue life prediction. Calculations have been performed for the connections M16×2 (height of the nut $H=0.8d=12.8$ mm) – made from steel 25Ch1MF. The same stud-nut connections have been used in the fatigue tests the results of which are presented in the next section. Average indices of mechanical properties of connections steel 25Ch1MF: proof strengths $R_{p0.02}=860$ MPa and $R_{p0.2}=890$ MPa, tensile strength $R_m=1000$ MPa, percentage area of reduction $Z=60.2$ %, module of elasticity E 2 10 GPa. Indices for one turn pair M16×2 were established experimentally by the technique described in [10]: pliability $\gamma=3.78\times10^{-3}$ mm/(kN/mm) and yield turns load i.e. load per unit length at which plastic deformation of the turn pair begins $q_y=12$ kN/mm.

Threaded connection can be cyclically loaded by one-side or two-side bending moment. At one-side cyclic loading the bending moment in the period of a cycle increases and decreases without its direction change. Then the parameters of the alternating elastic stresses cycle in the stud thread (amplitude of stresses, mean stresses, maximum and minimum stresses) are the following

$$\sigma_a^*(z_i)=0.5\sigma_b^*(z_i),\ \sigma_m^*(z_i)=\sigma_t^*(z_i)+0.5\sigma_b^*(z_i), \\ \sigma_{max}^*(z_i)=\sigma_t^*(z_i)+\sigma_b^*(z_i),\ \sigma_{min}^*(z_i)=\sigma_t^*(z_i)\quad(30)$$

At two-side symmetrical cyclic loading the direction of the bending moment changes after every semicycle of the loading. Then the parameters of stress cycle are the following

$$\sigma_a^*(z_i)=\sigma_b^*(z_i),\ \sigma_m^*(z_i)=\sigma_t^*(z_i), \\ \sigma_{max}^*(z_i)=\sigma_t^*(z_i)+\sigma_b^*(z_i), \\ \sigma_{min}^*(z_i)=\sigma_t^*(z_i)\text{-}\sigma_b^*(z_i)\quad(31)$$

Further the figures presented in this section reflect the calculation data (for the connection M16×2) which have been obtained at such external loads F_t and M_f which cause the following studs' nominal stresses:

$\sigma_{t,nom}/R_{p0.02} = 0.57$ at tightening and $\sigma_{b,nom,max}/R_{p0.02} = 0.31$ at bending. It was assumed in the calculation that the helix of threaded connection is set in the position *I*. In this case the studs' thread alternating stresses due to bending $\sigma_b^*(z_i)$ and alternating stresses $\sigma_t^*(z_i)$ due to tightening have maximum values in the same cross-section [12]. This cross-section in the stud is found at the coordinate z_{03} where $R\,sin(cz_{03}) = R$ in this case and where runout at the bearing surface of the nut begins.

In Figs. 2 and 3 are shown distributions of loads $q_t(z_i)$, $Q(z_i)$ and $q_b(z_i)$, $M(z_i)$ which have been calculated according to the methods given in sections 3 and 4. Vertical dashed line here and further marks origin of the runout at z_{03} (at distance z_{03} - z_f = 10.8 mm from the free end of the nut) where dangerous cross-section of the stud at helix position *I* is found.

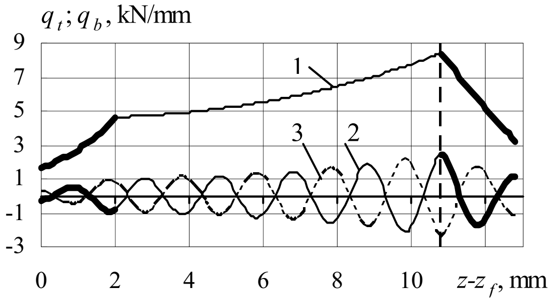

Fig. 2 Turn loads distributions in the thread: *1* – $q_t(z_i)$, *2* – $q_b(z_i)$, *3* – $q_b(z_i)$ at the second semicycle in the case of two-side bending

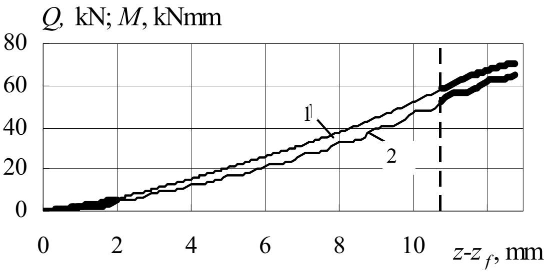

Fig. 3 Internal axial force and internal bending moment of the stud cross-sections: *1* – internal axial force $Q(z_i)$, *2* – internal bending moment $M(z_i)$

Alternating stresses in the thread roots due to tightening $\sigma_t^*(z_i)$ and stresses due to bending $\sigma_b^*(z_i)$ are shown in Figs. 4 and 5. Stresses of the primary importance in the calculation of the fatigue lifetime are $\sigma_b^*(z_i)$, because they give numerical values for the stresses cycle amplitude (Eqs. (30) and (31)). It is well known that fatigue lifetime is mostly sensitive namely to the stresses amplitude. In Fig. 5 are visible three points where maximum amplitudes of the local stresses are practically equal: two black points and one white point on the opposite layers of the stud – all in the bending plane. Therefore in the case of two-side bending almost simultaneous initiations of the fatigue crack can be expected in the stud cross-sections which are marked by these points.

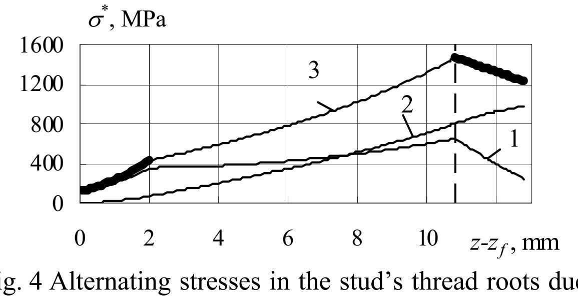

Fig. 4 Alternating stresses in the stud's thread roots due to tightening: *1* – stresses due to $q_t(z_i)$, *2* – stresses due to $Q(z_i)$, *3* – total stresses due to tightening: $\sigma_t^*(z_i)$

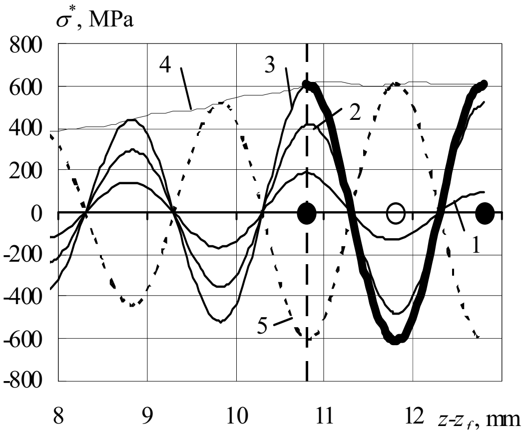

Fig. 5 Alternating stresses in the stud thread roots due to bending: *1* – stresses due to $q_b(z_i)$, *2* – stresses due to $M(z_i)$, *3* – local bending stresses: $\sigma_b^*(z_i)$, *4* – function $y(z_i)/\gamma(z_i)$, *5* – $\sigma_b^*(z_i)$ at the second semicycle in the case of two-side bending

In Figs. 6 and 7 distributions of the total alternaing stresses $\sigma_\Sigma^*(z_i) = \sigma_t^*(z_i) + \sigma_b^*(z_i)$ in the stud thread M16×2 are presented. It is seen in Figs. 6 and 7 that in the position *I* of the threaded connection in both cases of the cyclic bending (in the case of one or two side cyclic bending) the maximum mean and maximum total alternating stresses arise at the origin of runout, i.e. at the distance z_{03} - z_f=10.8 mm from the free end of the nut.

At the high level of tightening ($\sigma_{t,nom}/R_{p0.02} > 0.85$) total load of the turns pair $q_\Sigma(z_i) = q_t(z_i) + q_b(z_i)$ obtained for the thread elastic state at z_{03} exeeds the yield turns load q_y (Fig. 8). Really here due to the turns plastic deformation in the region of runout (at the bearing surface of the nut) the total turns pair load will be less than obtained. In such cases the correction of the amplitude of the turns load

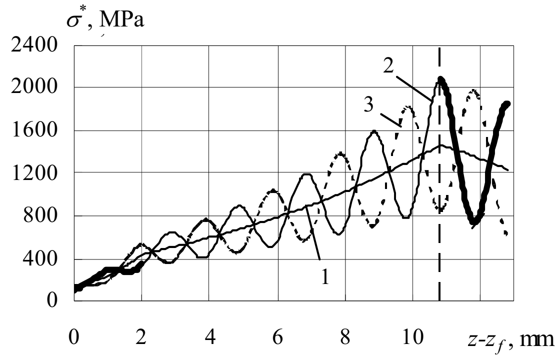

Fig. 6 Alternating stresses of the stud thread roots in the case of two-side cyclic bending: *1* – stresses after tightening $\sigma_t^*(z_i)$, *2* – total stresses, *3* – total stresses at the second bending semi-cycle

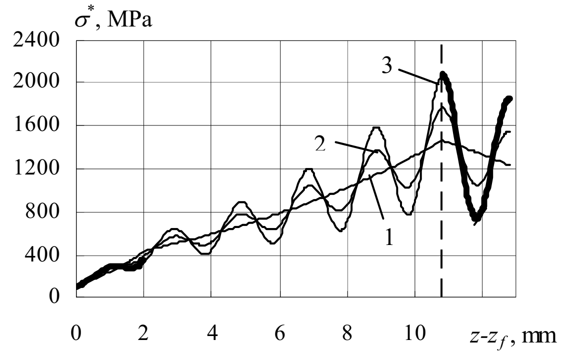

Fig. 7 Alternating stresses of the stud thread roots in the case of one-side cyclic bending: *1* - $\sigma^*_{min}(z_i) = \sigma^*_t(z_i)$, *2* - $\sigma^*_m(z_i)$, *3* - $\sigma^*_{max}(z_i)$

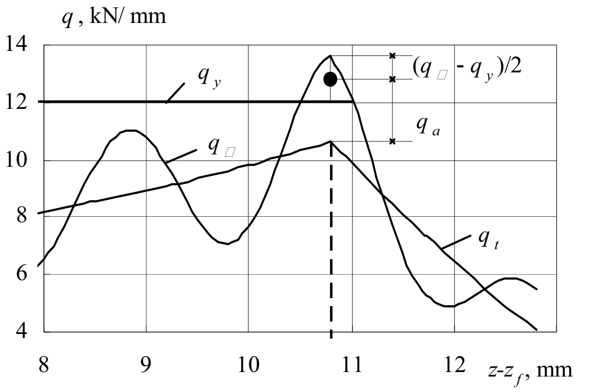

Fig. 8 Scheme of the turn pair load amplitude correction

has been performed in the following way: $q_a(z_{03}) = q_b(z_{03})$-$-[q_\Sigma(z_{03})-q_y]/2$. Then the obtained $q_a(z_{03})$ is beeing used in Eq. (29) (instead $q_b(z_i)$).

Finally in this section it is usefull shortly to analyse the magnitudes of alternating stresses amplitude (in the stud thread) obtained by using elastic model for the cases of helix position different from the position *I*.

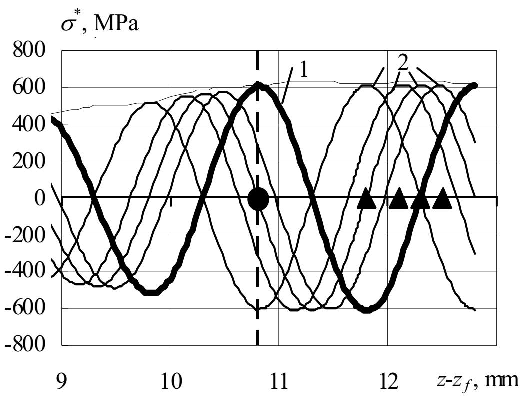

Fig. 9 Amplitudes of alternating stresses in the stud thread at various positions of thread helix: *1* – helix position *I*, *2* – other helix positions

It have been defined that in the cases of other helix position the maximum of local stresses amplitude in the stud thread are within the region of the runout and their values are almost equal (very slight lower) to the value obtained in the case of helix position *I*. Such four cases are shown in Fig. 9, where black triangle points mark the appropriate cross-sections of the stud. Due to the lower mean stresses (tightening stresses $\sigma^*_t(z_i)$) within this region (Fig. 4) the numbers of load cycles up to crack initiation in the stud thread (calculated by the methods presented in the next section) in above mentioned cases are also lower than in the case of helix position *I*, but just very slightly.

6. Fatigue life prediction

In order to estimate fatigue life of the cyclically bent threaded connections the possibilities to upply standard method [7] and modified standard method [9] are considered. Contrary to the standard method in its modification the data of stresses distribution along thread of the stud are used immediately. In the case of cyclic tension of threaded connections this has given calculation results of the connections lifetime notably close to the experimental data [9].

Two Coffin-Manson-Langer type formulae are presented in standard [7]. The smaller value of the number of load cycles up to crack initiation in the stud thread N_0 must be finally chosen out of two assessed values. These formulae, applied to the stud, no safety factors considered, are as follows

$$\sigma^*_a = \frac{E_{1s}e_c}{(4N_0)^m} + \frac{R_{-1}}{1+(R_{-1}/R_m)\left[(1+r_c)/(1-r_c)\right]} \tag{32}$$

$$\sigma^*_a = \frac{E_{1s}e_c}{(4N_0)^m} + \frac{R_c}{(4N_0)^{m_e} + \left[(1+r_c)/(1-r_c)\right]} \tag{33}$$

where R_{-1} is fatigue limit stress of the material, e_c is material plasticity index, m and m_e are exponents of power, r_c is asymmetry factor of the local stresses cycle, R_c is material strength index reliant on tensile strength R_m and σ^*_a is amplitude of alternating local stresses in thread roots of the stud which is calculated by using only one stress concentration factor K_σ according to the following formulas: $\sigma^*_a = K_\sigma \sigma_{a,nom}$ and $K_\sigma = 1 + 1.57\sqrt{P/R_r}$ where $\sigma_{a,nom}$ is amplitude of maximum nominal stresses of stud and R_r is turns' root rounding radius.

By using distribution of the stress $\sigma^*_a(z_i)$ in roots of the stud thread, defined under the methods presented in sections 3 and 4, it is possible to calculate the number of cycles $N_0(z_i)$ until the crack appears for any turn of the stud, as well as $N_0(z_C) = N_0$ for the stud's dangerous cross-section at z_C thereof. Here expression $\sigma^*_a(z_C = z_{03})$ obtained for studs at the thread helix position *I* was used. To prove this the following formula adjusted in [9] has been used

$$\sigma^*_a(z_C) = \frac{E_{1s}e_c}{[4N_0(z_C)]^m} + \frac{R_c}{[4N_0(z_C)]^{\overline{m}_e} + \dfrac{1+r_c}{1-r_c}} \tag{34}$$

where $\overline{m}_e = 0.78(Z10^{-2}) - 0.26(R_m 10^{-3})$ is the power exponent, adjusted in [9] on the basis of the experimental data obtained at axial cyclic loading (in standard [7], i.e. in formula (33) appropriate power exponent is $m_e = 0.132\ lg[(R_m / R_{-1})(1+1.4\cdot 10^{-2} Z)]$.

The possibilities to apply formulas (32), (33) or (34) for cyclically bent threaded connection have been analysed by using comparison of the calculated cycle life $N_{0,calc}$, with the experimental data $N_{0,exp}$.

Tests parameters of the stud-nut connections M16×2 made from steel 25X1MΦ are given in Table. In every specimen rounding radius of the stud thread root was $R_r = 0.144P$ and the other dimensions of metric thread were as specified by ISO 724.

Table

Specimen No.	$\frac{\sigma_{t,nom}}{R_{p0.02}}$	$\frac{\sigma_{b,nom}}{R_{p0.02}}$	$r = \frac{\sigma_{min,nom}}{\sigma_{max,nom}}$	Bending sides
1	0.74	0.4	0.30	two
2	0.72	0.38	0.30	two
3	0.69	0.64	0.04	two
4	0.68	0.62	0.05	two
5	0.61	0.57	0.52	one
6	0.64	0.59	0.52	one
7	0.43	0.294	0.19	two
8	0.437	0.308	0.19	two
9	0.59	0.175	0.54	two
10	0.543	0.162	0.54	two
11	0.547	0.164	0.54	two
12	0.65	0.31	0.36	two
13	0.57	0.31	0.31	two

Cyclic life tests of the threaded connections have been carried out under bending cyclic loads where the displacement $s(t)$ proportional to the angular displacement of the twisted support of the nut is being monitored (Fig. 10). Two pairs of gauges 6 (Fig. 10) have been used in order to control the maximum nominal bending stresses in outside layers of the stud found in bending plane and nominal tightening stresses in layers at neutral plane.

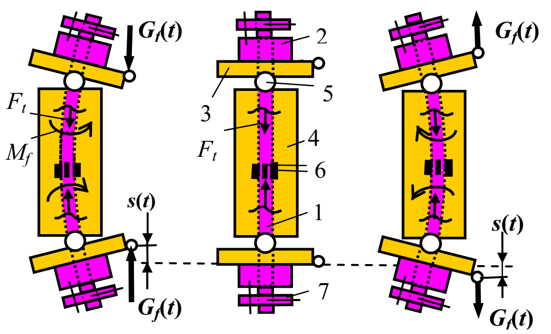

Fig. 10 Sheme of cyclic bending of the stud-nut connection: *1* – stud, *2* – nut, *3* – twisted support, *4* – intermediate detail, *5* – roll, *6* – four strain gauges, *7* – auxiliary nut; $G(t)$ – force varying during loading cycle time t

The nominal tightening stresses of the stud $\sigma_{t,nom}$ at cyclic loading untill crack initiation have decreased about 3-5% but it's nominal bending stresses $\sigma_{b,nom,max}$ practically remained unchanged.

Magnetic luminescent powder method has been employed to define the start point of crack initiation in the stud. Detection of formation of the crack in the stud is being performed on routine basis by knocking down the connection. After each proof inspection, by using auxiliary nuts with pins, no load applied, the joint is assembled in such a way, that in the course of further testing, position of the nut relative to the stud remains unchanged. Now, such a dangerous state of the stud's thread roots, when the length of macrocrack round the periphery thereof reach (3-6) mm, is considered to be the crack initiation.

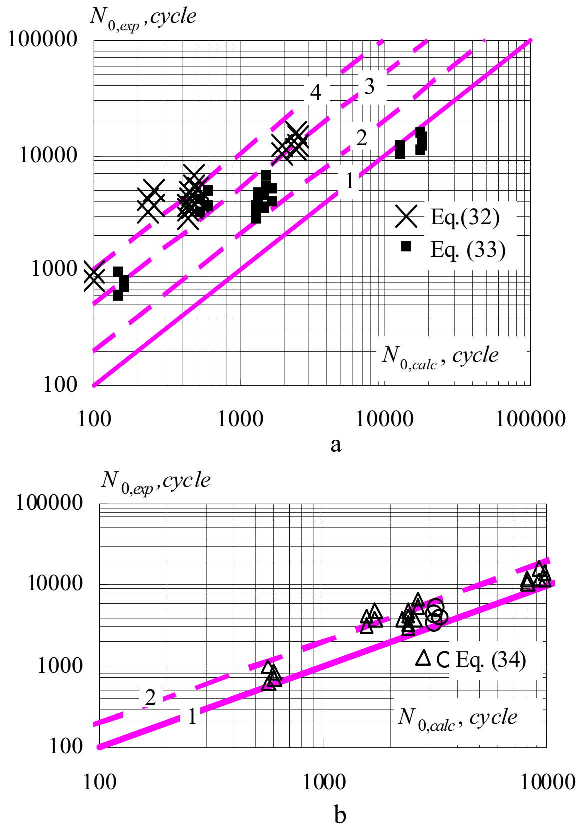

Fig. 11 Comparison of lifetimes for stud-nut connections M16×2: a – calculations as per standard's formulas (32) and (33), b – calculation as per formula (34); Δ and ○ – points for two-side and one-side cyclic bending; *1*, *2*, *3*, *4* – lines at $N_{0,exp}/N_{0,calc}$=1, 2, 5, 10

In Fig. 11 are shown that all points which mark calculated crack initiation lifetimes of the studs lies in the safe part of life time range – over the line *1*. The experimental values $N_{0,exp}$ exceed lifetimes $N_{0,calc}$ obtained by using formulas (34) and (33) up 2 and 5 times respectively and about 10-15 times in the case of using formula (32).

7. Conclusions

1. To incorporate the threads load distribution data into the calculation of fatigue durability of cyclically bent threaded connections the modified formulae (34) could be used.

2. Low cycle durability up to $2\cdot 10^4$ cycles of the cyclically bent threaded connections set according to modified method for cyclic strength is notably higher (about 2-5 times) than the calculated values set according to the Norm of Rusian Federation (Norm RF) [7]. They are close to experimental values, howerer do not exceed them.

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CIKLIŠKAI LENKIAMŲ SRIEGINIŲ JUNGČIŲ ILGAAMŽIŠKUMO PROGNOZAVIMAS

R e z i u m ė

Straipsnyje parodyta, kad modifikuotas norminis metodas, įvertinantis apkrovos pasiskirstymą sriegio vijose, yra tinkamas kartotinai lenkiamų srieginių jungčių ciklinio ilgaamžiškumo prognozavimui. Panaudotas apkrovos ir įtempių pasiskirstymo vijose modelis, kuris atspindi visu ir ne visu profiliu sukibusių vijų porų tampriojo deformavimo savybes. Gauti jungčių M16×2 ciklinio ilgaamžiškumo skaičiavimo rezultatai artimi eksperimentinėms reikšmėms ir jų neviršija.

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FATIGUE LIFE PREDICTION FOR CYCLICALLY BENT THREADED CONNECTIONS

S u m m a r y

In this paper it is shown that the modified normative method improved by incorporating in the calculation the data of load distribution in threads is fitted for the fatigue life prediction of cyclically bent threaded connections. The model for calculation of load and stress distribution along the thread based on elastic deformation properties of the full and partial profile turns pairs is used. Results of fatigue life prediction obtained for the cyclically bent threaded connections M16×2 are close to the experimental values, howerer do not exceed them.

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ЦИКЛИЧЕСКАЯ ДОЛГОВЕЧНОСТЬ ПОВТОРНО ИЗГИБАЕМЫХ РЕЗЬБОВЫХ СОЕДИНЕНИЙ

Р е з ю м е

В настоящей работе показана возможность использования модифицированной нормативной методики, учитывающей распределение нагрузки по виткам резьбы, для расчета долговечности повторно изгибаемых резьбовых соединений. Используется модель распределения нагрузки в витках, отражающая свойства упругого деформирования витков при их полном и частичном сопряжении. Полученные результаты расчета по циклической долговечности изгибаемых резьбовых соединений M16×2 близки к экспериментальным значениям и их не превышают.

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