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ON THE BEHAVIOR OF A BOLT-NUT JOINT WITH PLASTICALLY DEFORMED THREAD

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The results of investigations connected with influence of tightening bolts above their yield stress on the fatigue life and loosening of the bolt-nut joints are presented. From numerical approach by the FEM, the dependence of the load distribution at the threads and maximal value of stress at the roots of thread on the tightening level of bolts is determined. The obtained results indicate, that tightening of a joint above its yield stress gives more uniform load distribution at the bolt thread, and stress distribution at the roots in part of the bolt screwed with the nut. Experimental investigations of the fatigue life of the bolt-nut joint with the tightening of the bolts below and above their yield stress are presented. The results indicate that tightening above the yield stress considerably increases the fatigue life of the joint. Experimental test of creep in the fatigue process and introductory test on dynamic loosening of the joint have been carried out. It was stated that the phenomenon of creep during the fatigue has no greater influence on the joint, and that the resistance to loosening of the joint increases with tightening of the bolt above its yield stress.

1. Introduction

A threaded joint shown in Fig. 1 is one of the most frequently used joints in machines and installation components. A correct designing of such joints has substantial influence on the life of the whole design. Till now, a lot of attention has been devoted to the influences of the thread geometry, shapes of bolt and nut, the technology of production and, finally, of material properties, on the fatigue life of the bolt-nut joint. As a result, it is obvious that the stress concentration in the thread roots and the rate of the load distribution at the screw threads have a fundamental influence on the behavior of a joint.

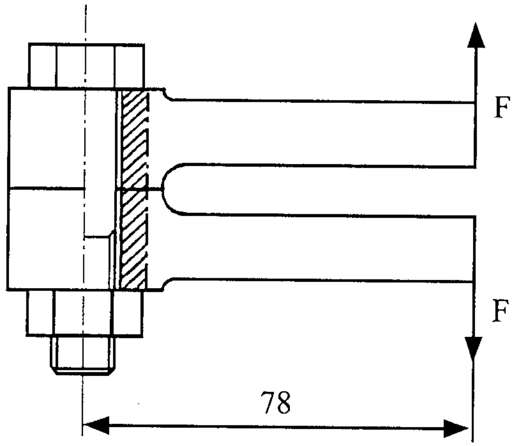

FIG. 1. The threaded joint bolt-nut and two fastened plates.

Amongst the early publications on the rate of load distribution at threads in the joint, let us mention the papers by MADUSCHKA [1], ZHUKOWSKI [2] and SOPOWITH [3]. Goodier carried out experiments with the use of an extensometer, and Hetenyi applied the method of photoelasticity. The corresponding results are presented in the papers [4, 5]. It results from the theoretical analysis and the experimental investigation as well that the greatest load is acting on the first thread at the bearing surface of the nut, and it diminishes with the growth of the distance from this bearing surface. Results obtained from the numerical analysis by the finite element method published in the papers [6, 7, 8] are similar to those inferred from both the theoretical and experimental investigations [1, 2, 3, 4, 5].

Until quite recently in practice, and consequently in the above mentioned papers, tightening of bolts was allowable only within the elastic range. However, in papers [9, 10, 11, 12] some new method of tightening the bolts above the yield stress of materials was proposed. In particular, the results presented in the works of SAKAI [12], CORNELIUS and KWAMI [13], KANESAKA [14] indicate that tightening above the yield stress increases the fatigue life and the resistance to self-loosening of the threaded joint. The other fatigue researches [15, 16] show that preload of the joint to or above the yield stress increases its fatigue strength. Also, as it was ascertained in [17], the economic need to realize high preloads in slip-resistant joints has promoted the tightening of bolts above the yield stress.

However, the above mentioned papers do not bring sufficient explanation why the tightening of bolts over their yield stress increases the fatigue life and the fatigue strength. Experiments performed in this paper confirm the existence of the increase effect of the fatigue life. The obtained results of the numerical approach indicate one of the possible causes of this effect.

2. Numerical computation of stresses

The considered object consists of two plates, fastened by a bolt and a nut, as shown in Fig. 1. Introductory tightening of the bolt and tearing of the plates on one edge by the forces F cause simultaneous tension and bending of the bolt.

The purpose of the numerical approach by the finite element method was the following:

- Evaluation of stresses and strains in the part of the bolt screwed with the nut, and in the nut;
- Description and comparison of the load distribution at the thread for two levels of the bolt tightening below and above the bolt yield stress;
- Evaluation and comparison of stresses at the roots of the bolt thread for two levels of the bolt tightening, below and above the bolt yield stress.

The computations has been performed by FEM with "Adina" code, version 6.1.4. In the computational model we have assumed:

- Elastic-plastic with strain hardening, multilinear model of the material, different for the bolt with the nut and for the plates;
- Both the bolt and the nut are made of steel for which the stress-strain relation $\sigma(\varepsilon)$ for tension with strain rate $\dot{\varepsilon} = 10^{-4} 1/s$ is shown in Fig. 15;
- For the bolt and the nut material, the values of the modulus of elasticity and Poison's ratio are $E = 2.06 \times 10^5$ MPa and $\nu = 0.30$, the yield stress for tension is $\sigma_y = 670$ MPa;
- For the plates material, the quantities E, ν, σ_Y have the values $E = 2.02 \cdot 10^5$ MPa, $\nu = 0.30$, $\sigma_Y = 450$ MPa;
- Dimensions of the bolt and the nut threads correspond to the metric coarse thread M12 – the considered length of the bolt and in the nut contains 6 threads.

In numerical approach, the effects of lead angle of thread and the friction effects on contact surface are not taken into account. The values of the stresses evaluated in the plates are in the elastic range. In discretisation, three-dimensional elements have been applied. Division into elements for the computed problem is shown in Fig. 2 (number of degrees of freedom is 19210). For the part of the bolt screwed with the nut, the discretisation is shown in Fig. 3. In general, the formulated problem was a spatial, nonlinear problem of statics.

In numerical modeling, the level of tightening depends on the magnitude of the bolt tensile force P acting in the section $\alpha - \alpha$ (the plates contact plane) – see Fig. 2. The computations have been performed for two kinds of the load acting on the bolt-nut joint:

- introductory tightening of the bolt in the range of elastic strains –

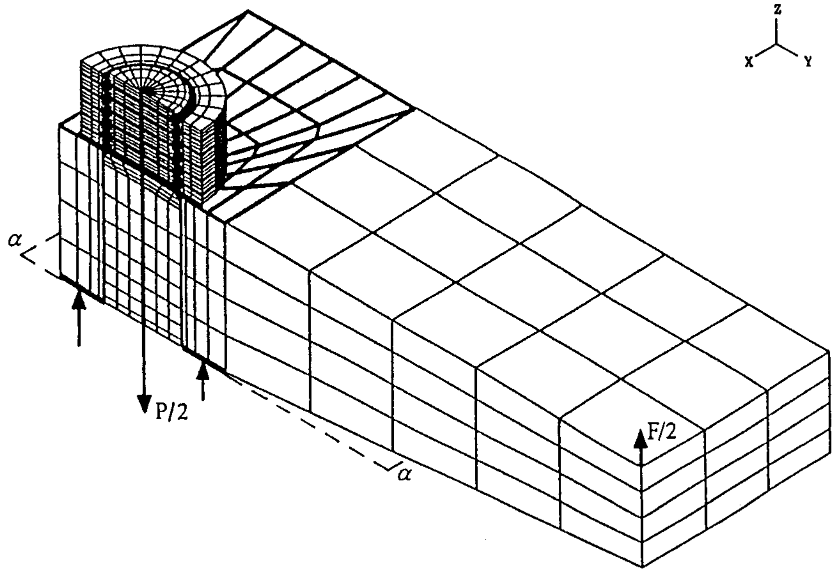

FIG. 2. Elements division for the analysed problem. The contact surfaces are thickened.

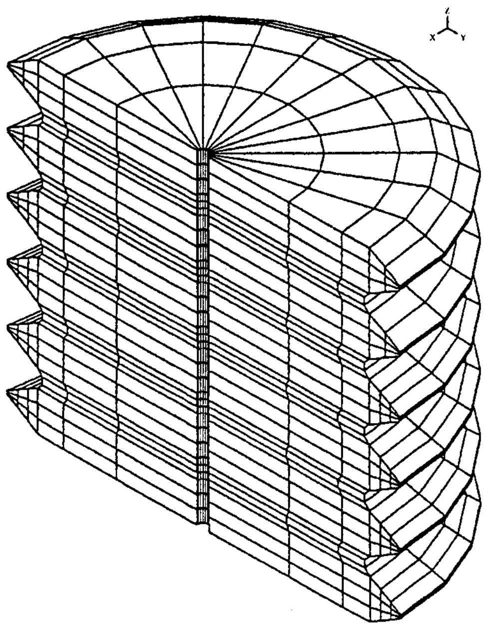

FIG. 3. Elements division for the part of bolt screwed with the nut.

$P = 0.5 S_A \sigma_Y$, (S_A is the stress area of the thread) – and bending forces $F = 3000$ N attached to the plates,

- introductory tightening of the bolt above the range of elastic strains and bending forces $F = 3000$ N attached to the plates.

For the second kind of the load, the force P modeling the tightening causes exceeding of the yield stress of the bolt material in a vicinity of the bottom of the thread roots.

The calculation of the stress state enabled to determine the rate of load distribution P_N/P, at individual threads. P_N is the force loading the individual thread denoted by subscript N and P is the force carried by all the threads. The rate of load distribution P_N/P, at six threads, in the part of the bolt screwed with the nut, for the tightening below the bolt yield stress and with loading the bending force is given in Fig. 4. The figure also shows MADUSCHKA'S [1] and ZHUKOWSKI'S [2] results for elastic problem without loading by the bending force. It is seen that for external loads, which do not lead to a plastic deformation, the rate of load distribution is the greatest on the first thread at the bearing surface of the nut and decreases gradually on the further threads. Similar results were obtained in the papers [1, 2, 3, 4, 5, 6, 7].

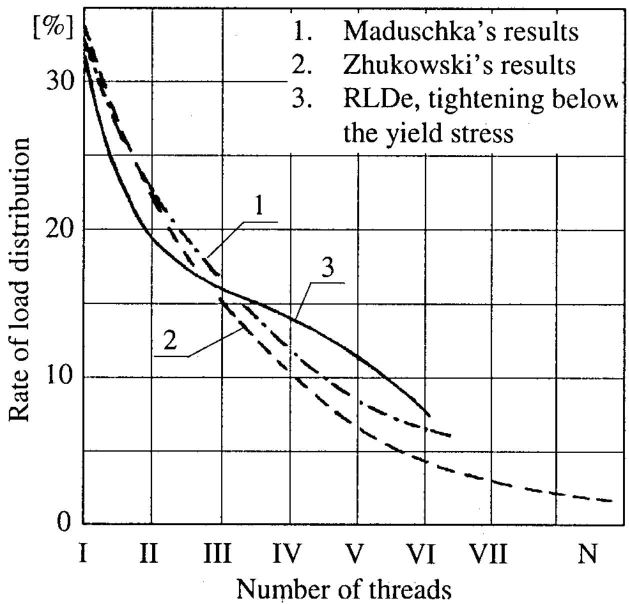

FIG. 4. Rate of load distribution at individual bolt threads for tightening the bolt in elastic range.

From the present computation it follows that for the bolt tightening which produces plastic deformation of the thread, the rate of load distribution becomes more uniform. Fig. 5 shows the rate of load distribution at threads for two levels of tightening the bolt below and above the yield stress. Decreasing and increasing of the rate of load on individual threads, which is caused by loading above the yield point of the bolt thread, is shown in Fig. 6. It is seen that concentration rate of load at the first thread diminishes, and on the second, third, fourth threads it increases.

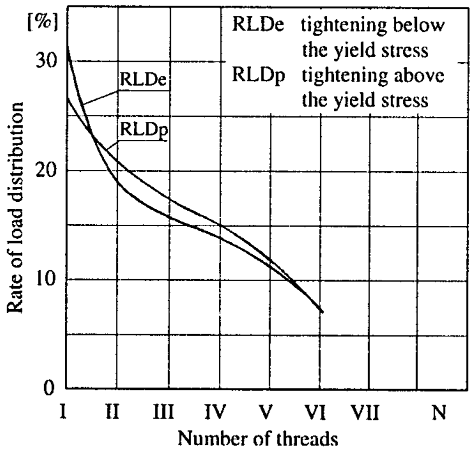

FIG. 5. Rate of load distribution at the bolt thread for tightening the bolt below and above its yields stress

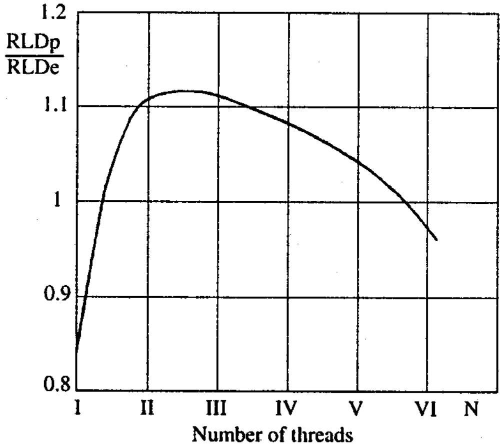

FIG. 6. The relative change of the rate of load distribution caused by the tightening above the bolt yield stress.

The values of stress for the tightening, which causes plastic strains at the bolt thread are presented in Fig. 7 and Fig. 8. Distribution of reduced stress σ_{red}, where $\sigma_{\text{red}} = \sqrt{\frac{1}{2}\left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 3(\tau_{xy}^2 + \tau_{xz}^2 + \tau_{yz}^2)\right]}$, in the part of the bolt screwed with the nut, is shown in Fig. 7 in the form of bands at the same values of stress. The force F is the cause of the non-uniform stress distribution along the thread circumference. The non-uniform stress distributions on the circumference of the roots of thread are presented in Fig. 8.

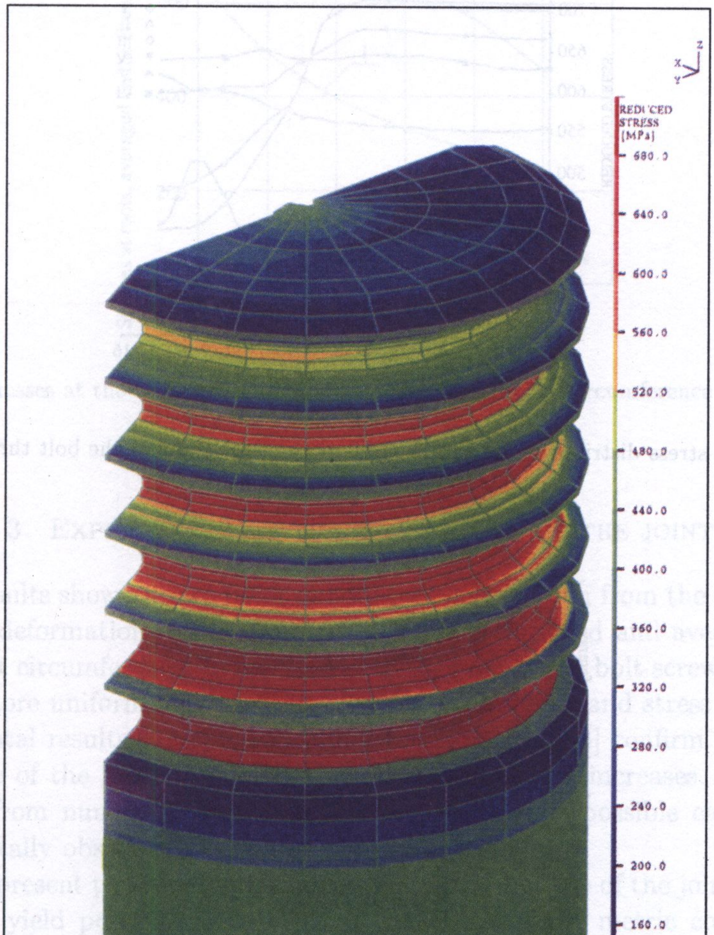

FIG. 7. The bands of constant stresses in the bolt for the tightening above the bolt yield stress.

The stresses at the roots of the thread, presented in Figs. 8, 9 and 10, are computed as the mean value of the stresses in the three elements which form the bottom of the root. The maximal – on the circumference of the roots – values of stresses at the bolt thread for tightening below and above the yield stress are shown in Fig. 9. As it is seen in Fig. 9, the localization of the maximal stresses in the joint, independently of the external load level, was at the root of the first bolt thread. The stresses are also greater for tightening the bolt above its yield stress than below the yield stress.

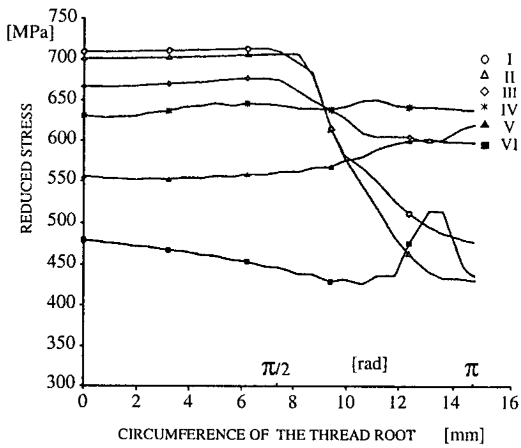

FIG. 8. The stress distribution on the circumference of the roots of the bolt thread for the tightening above the yield stress.

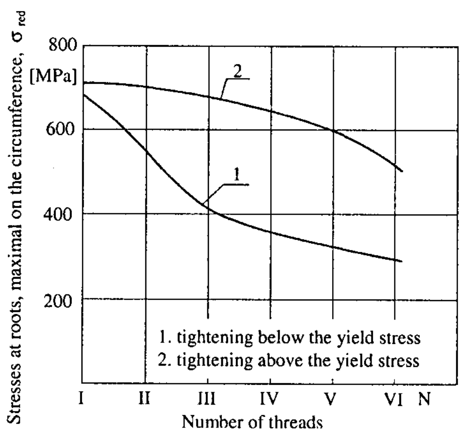

FIG. 9. Maximal stress at the roots of the bolt thread for two levels of the tightening.

If stresses at the roots are presented in the form of not maximal – like in Fig. 9 – but average values of stress on the root circumference, then average value of stress at the first root is smaller for tightening above the yield stress than below. It is seen in Fig. 10. However, the obtained decrease of the average stress on the root circumerence is small and this effect still needs an additional examination.

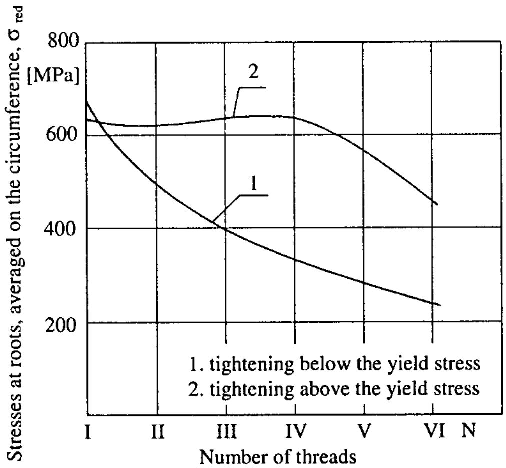

FIG. 10. Stresses at the roots of the bolt thread averaged on the circumference for two levels of the tightening

3. EXPERIMENTS ON THE FATIGUE LIFE OF THE JOINT.

The results shown above point out that after transition from the elastic state to plastic deformations state, the distributions of the load and averaged stress on the root circumference, at the thread in the part of the bolt screwed with the nut, are more uniform and the concentration of the load and stress diminishes. Experimental results presented in publications [12, 13, 14] confirm a view that fatigue life of the bolts tightened above the yield stress increases. The results obtained from numerical approach indicate one of the possible causes of the experimentally observed increase of the bolt fatigue life.

In the present paper the experiments on the fatigue life of the joint tightened above the yield point were performed. The bolts of the metric coarse thread $M12 \times 1.75$ of the class 8.8 were investigated. The bolt-nut joint with the two joined plates and the handle part of the testing machine are presented in Fig. 11.

The plates are torn off on one edge by the force F pulsating from zero (with minimal value $F_{\mathrm{min.}} = 0$). The force causes simultaneous tension and bending of the bolt. The joint was investigated by means of a special testing machine

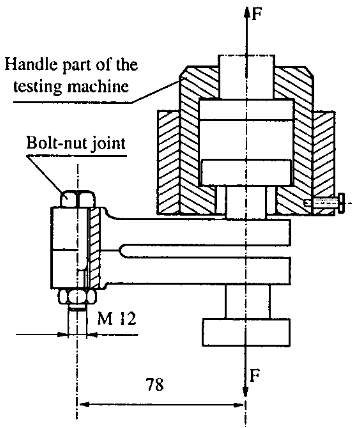

FIG. 11. Specimen with the joint bolt-nut for the fatigue test.

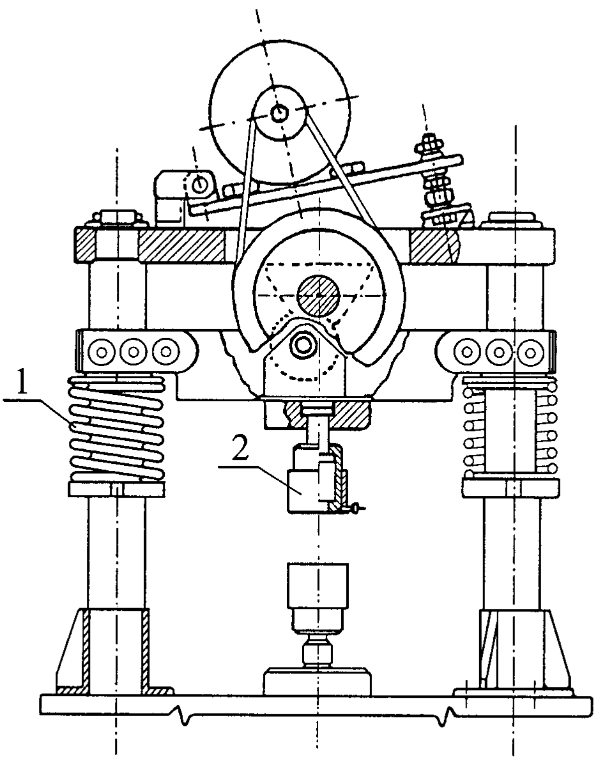

FIG. 12. Scheme of the machine for fatigue testing of thread joints.

presented in Fig. 12, applied for investigation of the fatigue life of the thread joints. The pulsating load was realized by tightening the springs (1) and steering by means of the cam mechanism the cyclic movement of the handle part (2) of the testing machine.

The first group of the investigated bolts was tightened in elastic range to the stress $\sigma = 0.65\sigma_Y$, where $\sigma_Y \equiv R_{eL}$(R_{eL} is lower yield stress). This value of stress σ is defined by the Polish Standard PN-81/M-82056. Experimental results of the investigations for three values of force F equal 2.15; 3; 4.2 kN and frequency 6 Hz are shown in Table 1. The dependence of the tightening torque M and the torque angle gradient $dM/d\varphi$ on the angle φ of the nut rotation has been also determined experimentally.

Table 1.

No	M Nm	F kN	F Hz	n Cycles	n_{av} Cycles
1	84	4.2	6	15859	
2	"	"	"	24700	20376
3	"	"	"	20570	
4	84	3	6	62075	
5	"	"	"	40186	69420
6	"	"	"	106000	
7	84	2.15	6	176200	
8	"	"	"	232320	283600
9	"	"	"	442280	

Table 2.

No	M Nm	$dM/d\varphi$ decrease	F kN	F Hz	n Cycles	$n_{av.}$ Cycles
1	139.2	50%	4.2	6	48988	
2	116.7	"	"	"	62571	61301
3	142.3	"	"	"	72345	
4	149.3	50%	3	6	297368	
5	152.1	"	"	"	320480	299232
6	132.3	"	"	"	279850	
7	148.6	50%	2.15	6	1450000	
8	133.9	"	"	"	2210000	without
9	135.5	"	"	"	2000000	fracture

The bolts investigated in the second group were tightened over the elastic range of strain in the thread. The maximal tightening reached the level of plastic deformation for that the torque angle gradient decreased by 50% in relation to the maximal value of this gradient in elastic state of the thread. The results of investigations for values of the force F equal 2.15; 3; 4.2 kN and frequency 6 Hz are shown in Table 2. Experimental results of the fatigue life of the joint obtained for the bolts tightened in elastic range and with admitting a small plastic strain of the thread, are presented in Fig. 13.

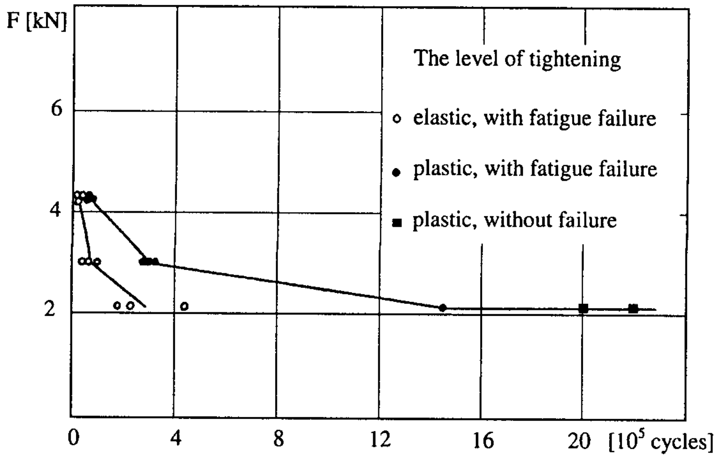

FIG. 13. Experimental results of the fatigue life.

Comparison of the results indicates that the joint with the bolt tightened over the yield stress has fatigue life several times grater than the joint with the bolt tightened in elastic range.

4. CREEP-FATIGUE EXPERIMENTAL TEST OF BOLTS MATERIAL

Due to the fact that within the range of plastic strains, the cyclic loads cause the creep phenomenon during the fatigue process, material of the bolts was tested on dynamic creep. The purpose of the experiment was testing of the initial period of vibrocreep process. Tested specimens of the shape presented in Fig. 14 were made of commercial class 8.8 bolts and were not annealed. The tests were carried out using a two-axis servo-hydraulic tension and torsion machine Instron 1343.

Stress-strain curves for tension with strain rate 10^{-4} and $10^{-1} 1/s$ are presented in Fig. 15. In the creep-fatigue tests the specimens were subjected to one of the following loads:

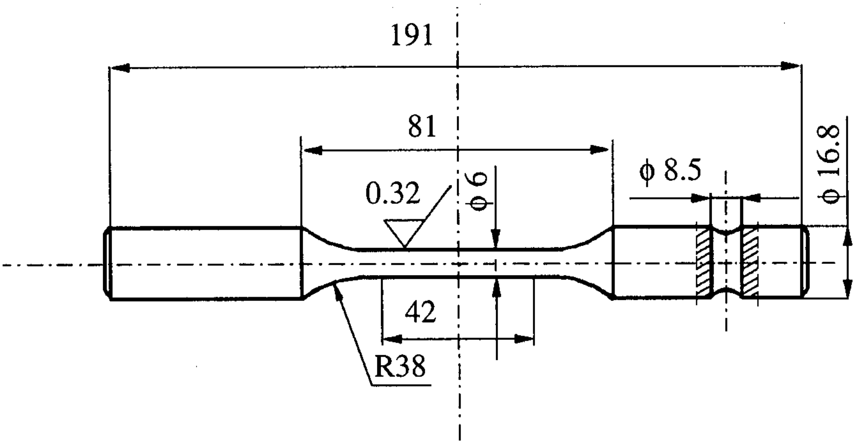

FIG. 14. The specimen for creep-fatigue tests of bolt material.

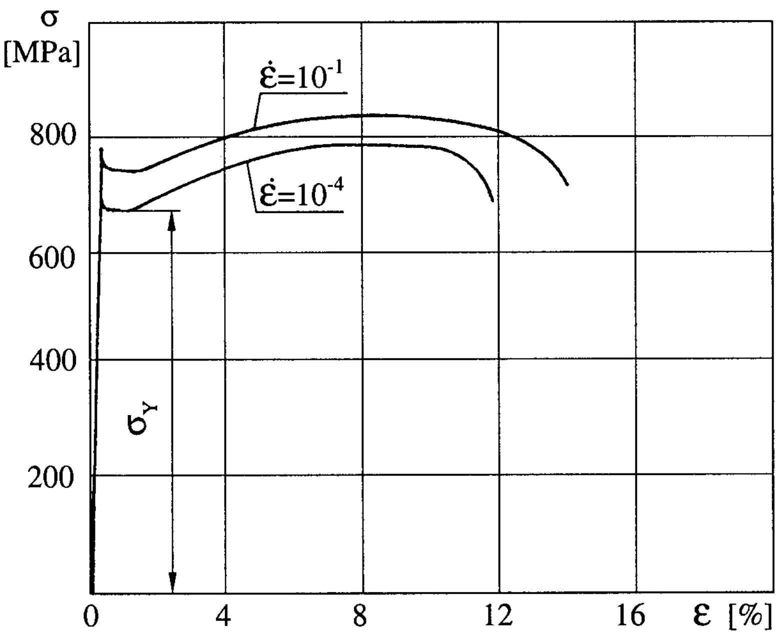

FIG. 15. Stress-strain curve for tension of bolt material.

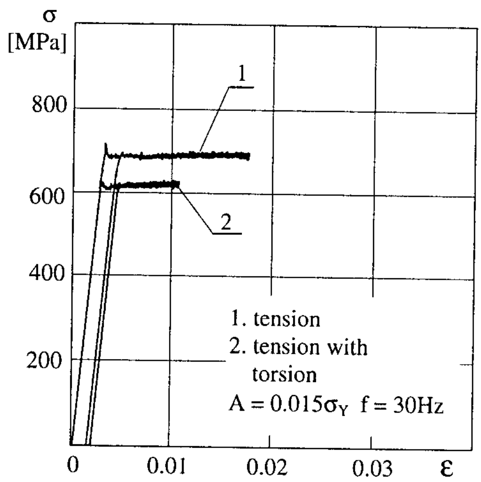

FIG. 16. Stress vs. strain relations for creep-fatigue tests.

1. Quasi-static loading of the specimen by torque with constant value of $M = \frac{(0.15d0.62\sigma_Y \pi d^2)}{4}$, where d is the diameter of the specimen, $\sigma_Y \equiv R_{eL}$. Next, loading by cyclic tension force with stresses $\sigma = 0.62\sigma_Y \pm 0.015\sigma_Y \sin\omega t$, where pulsation $\omega = 2\pi \cdot 30\text{Hz}$.
2. Quasi-static tension of the specimen up to the value of plastic strain $\varepsilon_p = 0.15\%$ and unloading of the specimen. Next, loading by cyclic tension force with stresses $\sigma = \sigma_Y \pm 0.015\sigma_Y \sin\omega t$ where pulsation $\omega = 2\pi \cdot 30\text{Hz}$.
3. Quasi-static loading of the specimen by torque with constant value of $M = \frac{(0.15d\sigma_Y \pi d^2)}{4}$ and tensile force up to the value of plastic strain $\varepsilon_p = 0.15\%$, and then by unloading tension. Next, loading by cyclic tension force with stresses $\sigma = \sigma_Y \pm A\sin\omega t$. Values of A and frequency $f = \omega/2\pi$ were as follows: 0.015 σ_Y, 30Hz; 0.045 σ_Y, 10Hz; 0.1 σ_Y, 1Hz.

The time of duration of the cyclic load for each tested specimen was 1.4h.

The results obtained for the loads, which do not cause exceeding the elastic range, indicate that the creep strain in the fatigue process does not exist. Stress-strain relations under tensile force only and simultaneous tension and torsion are shown in Fig. 16. It can be observed that the strain caused by creep during fatigue is smaller for composite state of stress – probably due to the smaller value of longitudinal stresses in this case. Results obtained for greater values of the amplitude $A = 0.045\sigma_Y$ and $f = 10\text{Hz}$ and for $A = 0.1\sigma_Y$ and $f = 1\text{Hz}$, are presented in Fig. 17.

Relations of the stress σ and the strain ε vs. time for the specimen under tension only and under simultaneous tension and torsion are shown in Fig. 18. As it is seen from relations $\sigma(\varepsilon), \sigma(t), \varepsilon(t)$ in Fig. 17 and Fig. 18, the strains caused by creep during fatigue at the beginning – within the first 10 ÷ 15 cycles of the load – increases rapidly. After a short initial period of a very fast deformation, the strain rate becomes very small and decreases with time.

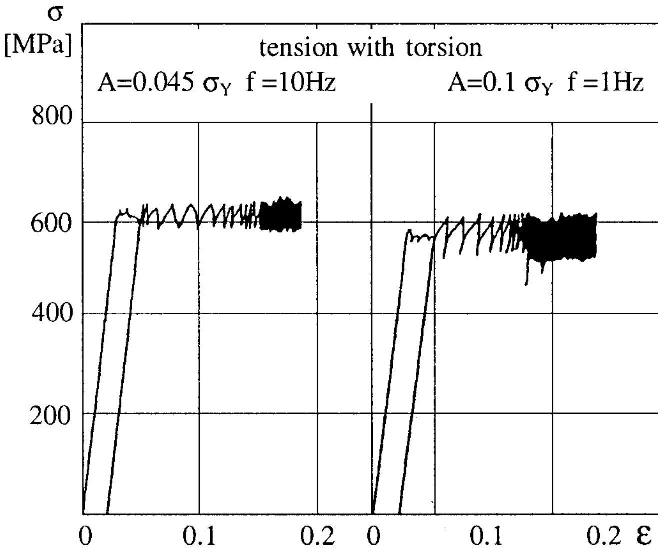

FIG. 17. Stress vs. strain relation for creep-fatigue tests.

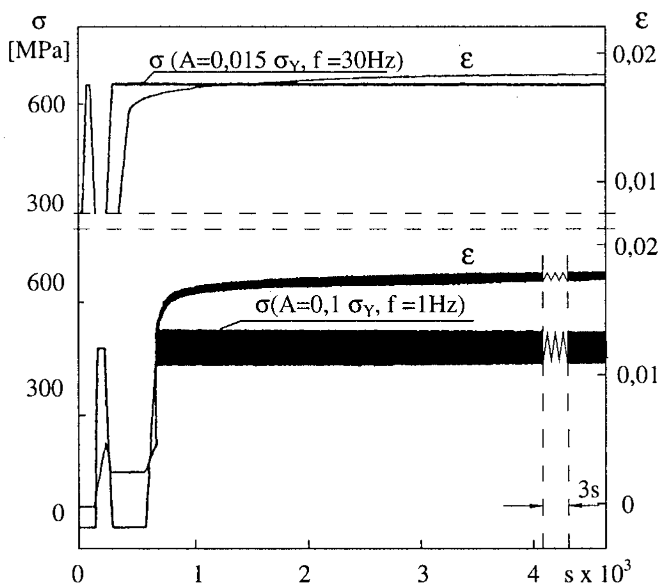

FIG. 18. Stress and strain vs. time relations for creep-fatigue tests.

5. Conclusions

Numerical computations and experimental investigations of the bolt-nut joint carried out in the paper lead to the following conclusions:

1. The tightening of the bolt up to the level of 50% of the torque angle gradient decrease, leads to plastic deformation of the threads and gives a more uniform distribution of the load and stress at the bolt thread and decreasing of the concentration of the load on the first thread (at bearing surface of the nut).
2. The tightening of the bolt with admission of plastic deformation in the threads can increase considerably the fatigue life of the joint in comparison to the tightening within the elastic range.
3. The experimental tests of the material carried out on creep in the fatigue processes and initial test of the joint on loosening, confirm the results obtained in the Sakai paper [12], that tightening of bolt above its yield stress decreases significantly the loosening effect of the joint – the loosening does not exist except for the very short initial interval.

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