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FINITE-ELEMENT ANALYSIS OF THE TUBING THREAD

During the work in the hole under the frequent cycles of gas storage filling and unfilling the external axial loading are applied to the coupling (muff screw-thread part) of tubing as well as an internal pressure together with the dynamic loading and vibrations. The torque for providing of his monolith state and impermeability tightens tubing coupling thread connection [1]. The action of vibrations can result in reduction of contact stresses in thread and violation of monolith and even impermeability of coupling. For the study of nature of stress distribution in the tubing screw-thread there has been developed a computer finite-element axisymmetrical model of the standard tubing couplings of pipes with the conditional diameter 114 mm GOST 633-80 (Fig. 1) with the use of the ANSYS program.

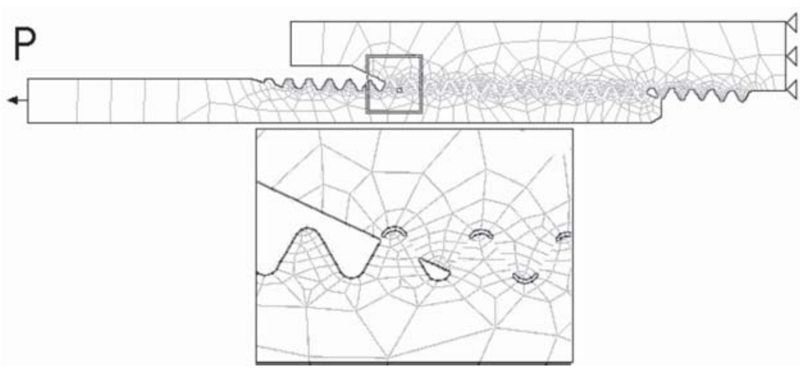

Fig. 1. The finite-element model of tubing coupling

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A material of details for pipe and coupling is a steel after normalization ($E = 2.1 \cdot 10^{11}$ Pa, $\nu = 0.28$). The computation has been conducted taking into account the friction between the contact surfaces (f = 0,1 – friction coefficient). Consequently, it has been resolved the nonlinear task. For the computer simulation of effort of screwing up the coupling together the possibilities of ANSYS for determination of initial penetration between the contact surfaces have been used.

The stress-deformation state of coupling with shoulder axial stretch $A = 0$ mm, $A = 3.175$ mm and $A = 6.35$ mm under the absence of the external loading (a) and in case of action of the external loading, which creates stress of 100 MPa in pipe body (b), is shown on the Figure 2 (on the interleaf).

The deformation is conditionally enlarged on the figure to 10 times. Local stresses of such big value (about 1 GPa) may be explained by application of elastic finite-element model, where the plastic deformations does not take into account.

The contact between all threads of coupling screwed together opens up in case of action of the external axial load.

The biggest stresses exist in first working pitch of the pin. In coupling screwed together the high values of stresses exist near the shoulder end of nipple and in the first threads of coupling. In the coupling ($A = 6.35$ mm) screwed together on the machine tool, these stresses exceed a yield strength. This picture of tensions coincides quit well with the results obtained by method of strain gauges application and by freezing of models made of the optically active material [2]. Distribution of loading between the threads is similar for coupling of such type [3]: the biggest – in the first thread of coupling loaded up, less – in the last and least – in the middle (Fig. 3 on the interleaf).

However the distribution of contact pressure on the thread surface is nonuniform and the most this unevenness exists exactly in the middle threads. The biggest value of contact pressure exists near the top of nipple thread. In case of screwing the coupling together this phenomenon spreads on all threads. In case of action of the external axial loading the contact pressure is almost equal to the zero on the right side of the 3^{rd}, 4^{th}, 5^{th} nipple thread screwed together.

Proper frequencies of coupling screwed together without stretch are found by method of finite elements (Fig. 4 on the interleaf): first – 10 627 Hz, second – 14 226 Hz, third – 15 156 Hz (11 248 Hz, 14 275 Hz, 15 258 Hz – for screwing together with stretch 6.35 mm).

The harmonic analysis of coupling in case of loading by the variable external force, which creates axial stress in the pipe body equal 100 MPa has been conducted in range of the frequencies 0–20 000 Hz. The amplitude-frequency descriptions (AFD) of coupling screwed together by hand and coupling screwed together with stretch ($A = 3.175$ mm) in area of the left end of model pipe are shown in the Figure 5. Notedly, that the tightening of coupling does not almost affect these characteristics.

However the amplitude of contact pressure between the threads changes substantially (Fig. 6), the difference between the AFD in the first, middle and last pitch of threads diminishes in case of screwing coupling together. In case of screwed together coupling ($A = 3.175$ mm) the amplitude of contact pressure substantially increases on left nipple threads in the area of third proper charge frequency.

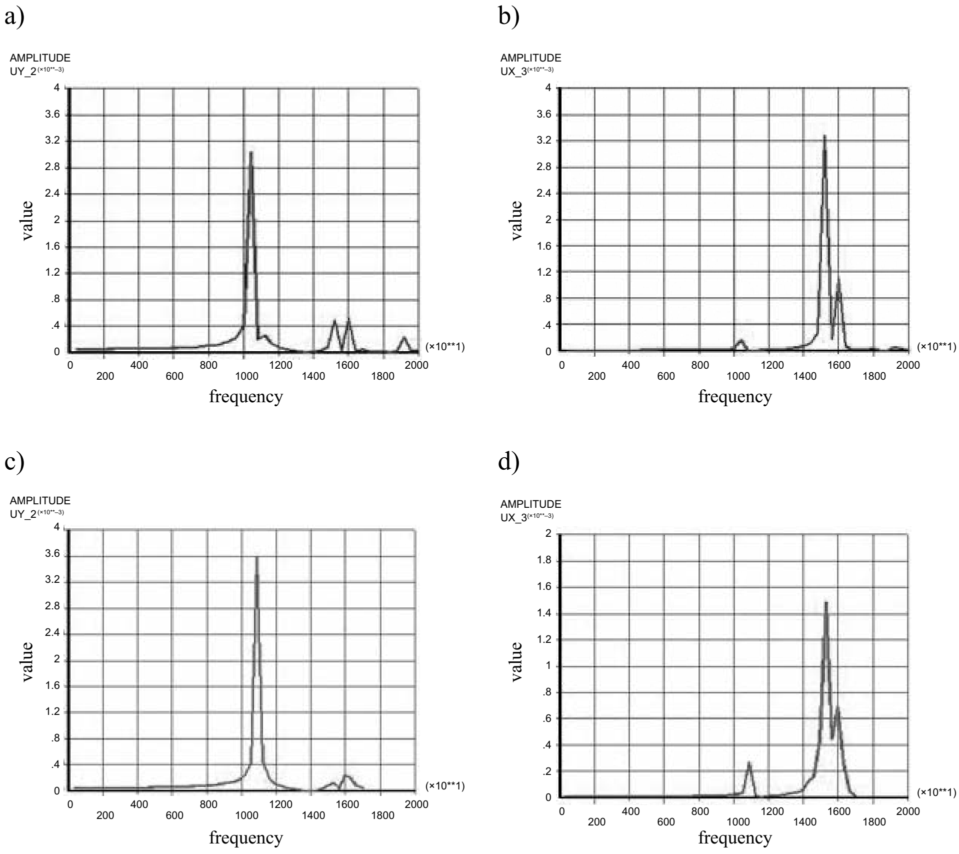

Fig. 5. The amplitude-frequency descriptions of coupling: a) displacement in the axial direction, $A = 0$ mm; b) displacement in the radial direction, $A = 0$ mm; c) displacement in the axial direction, $A = 3.175$ mm; d) displacement in the radial direction, $A = 3.175$ mm

The amplitude of contact pressure of middle (in area of the first proper charge frequency) and last (in area of third proper charge frequency) threads increases.

In comparison with the tool-joint-thread [4] the most dangerous frequency from point of view fatigue destruction of coupling in the dangerous plane of coupling (the first working pitch thread of pin and the last working pitch thread of box) there is only the first proper frequency (Fig. 7).

So, it is possible to make the conclusion, that the most dangerous frequencies of such coupling there is the first and third proper charge frequency. Screwing together coupling results in smoothing amplitudes of contact pressure on the working sides of threads. The first proper charge frequency is the most dangerous from point of view fatigue destruction of coupling in the dangerous planes.

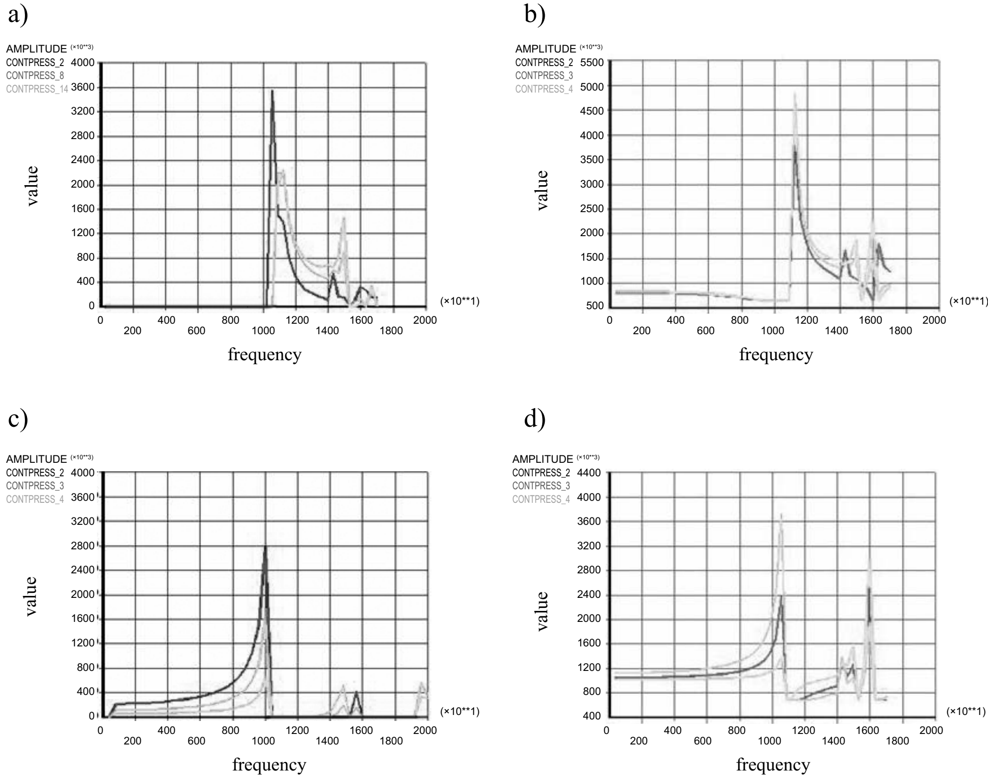

Fig. 6. The amplitude-frequency descriptions of coupling – an amplitude of values of contact pressure between the first (CONTPRES 2), middle (CONTPRES 3) and last (CONTPRES 4) pitch threads: a, b) right (working) side of nipple threads; c, d) left; a, c) $A = 0$; b, d) $A = 3.175$ mm

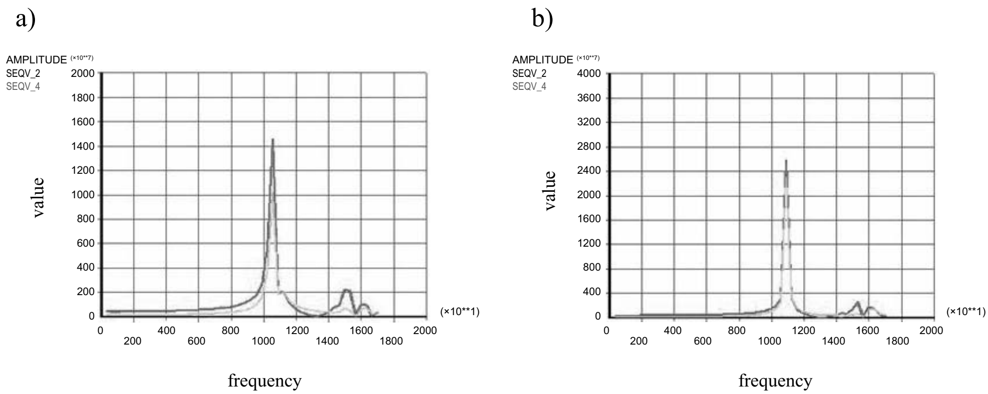

Fig. 7. Amplitude of stresses after the H – M – H (Pa) criterion in the dangerous planes of coupling: a) $A = 0$ mm; b) $A = 3.175$ mm

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