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TIGHTNESS OF STEAM GENERATOR BOLT JOINTS IN ACTUAL SERVICE CONDITIONS

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ABSTRACT

Integrity between primary and secondary circuit of nuclear power plants must be ensured in service conditions. Tightness of the bolt joint of the primary circuit collector of the WWER steam generators has influence on their integrity. Cases of tightening of bolts are discussed. Results of experimental measurement of stresses in bolts and a temperature of bolt joint elements of steam generators are discussed and compared with results of theoretical analysis.

1. CONSTRUCTION OF THE BOLT JOINT

The scheme of the bolt joint of the primary circuit collector of the WWER steam generator is on Fig. 1. The primary collector and the primary cover are manufactured from the austenitic steel 08Ch18N10T, twenty bolts M48x5 are manufactured from the alloy ChN35VT-VD, Tab. 1. Two nickel packing rings are put between the primary cover and the flange of the primary collector. Bolts are the most stressed among these elements. The allowed stress of the alloy ChN35VT-VD $[\sigma] = S_y/2 = 441/2 = 220.5$ MPa is in comparison standard |1|.

2. LOADING OF BOLTS

The force in the tightened bolt is $F_u = 272.2$ kN and causes a average stress $\sigma_m = 219.1$ MPa in the bolt section. Deformations of bolt joint elements during tightening of bolts cause a bending stress in them. This bending stress is not desirable, because the stress is higher the influence of a corrosion environments is higher. The cases of stress corrosion cracking of bolts are known. Therefore bending stress has to be very low.

Bolts can be tightened by torque wrench or by the hydraulic prestressing equipment (e.g. NSK 48-4 PGP - made in ORGREZ Brno). This equipment first prestress together all twenty bolts and then temperately turns their nuts. In this time every nut and two pads take mutually suitable position for elimination of deformations of bolt joint elements. Finally pressure is decreased to zero in the system of this equipment.

Bending stress $\sigma_b = 256.3$ MPa in the torque wrench tightened bolt was calculated by theoretical analysis. This value was confirmed by experimental measurement using strain gauges. The stress $\sigma_b = 256.3$ MPa is even higher then σ_m = 219.1 MPa. Bending stress σ_b = 52.4 MPa in a bolt tightened by the equipment NSK 48-4 PGP was calculated by theoretical analysis too. Experimental measurements by strain gauges have confirmed that actual σ_b are lower than theoretical value 52.4 MPa. Bending stress σ_b = 15 MPa was even measured. Comparison of the time trends σ_b for both tightening procedure is shown on Fig. 2.

The tightening procedure has influence on the stress level in the bolt. Theoretical calculated the nominal stress σ_{max} = 552.5 MPa for service conditions and torque wrench tightened bolts decreased to σ_{max} = 420 MPa, when equipment was used. It is a decrease to 76 %. An additional decrease of a bending stress in bolts was caused by increasing of the cover thickness and by lengthening of bolts. Theoretical calculated stress for service conditions decreased to σ_{max} = 376.3 MPa, when the equipment NSK 48-4 PGP was used too. It is a decrease to 68.1 %. Theoretical results was confirmed by experimental measurements at thightening of bolts before hydraulic test, during test and during service conditions on nuclear power plant.

3. TIGHTENING PROCEDURE OF BOLTS

Four cycles of prestressing are used during tightening of bolts by the equipment NSK 48-4 PGP, Fig. 3. The force F_p = 300 kN for every bolt is applied in the first cycle and F_p = 385 kN is usually applied in other three cycles. Nuts are temperately turn, when bolts are prestressed by the force F_p. When a pressure in the system of the equipment NSK 48-4 PGP decreases to zero after fourth cycle, the force 272.2 $\pm$ 20,2 kN has reamain in every bolts. Relation k = F_p/F_u = 385/272.2 = 1.41. This relation decreases in every cycle. The force F_p has not to be higher than the force F_s during service conditions on a nuclear power plant. The force F_p namely loads coils in the opening for the bolt in the flange of the collector. Its material 08Ch18N10T has only S_y = 196 MPa at 20°C.

Characteristic experimental results for tightening of bolts by the equipment NSK 48-4 PGP when thickened cover and lenghtened bolts were used: F_p = 362.6 kN ÷ 372.3 kN, F_u = 260.2 kN ÷ 285.1 kN (average F_u = 274.9 kN), σ_{bmax} = 41.6 MPa [2].

4. SERVICE CONDITIONS

High temperature strain gauges of the type MG 425-01 VH-15-9S produced by ESA Messtechnik have been used for measurement of stress in bolts during service conditions on nuclear power plant. Gauges are fastened in two sections on two bolts. Four strain gauges are fastened in every sections.

Time trends of measured force F_s in bolts during two campaigns are on Fig. 4. The force F_s unsignificantly decreased during nominal service conditions. Two shutdowns of the block are evident on Fig. 4. The forces F_s had decreased about 4.5 % during two campaigns. After shutdown and new start of the block the forces F_s decreased about 1.5 %. Any decrease was observed after transient conditions. Only one non - standard transient condition (no scheduled cooling of the bolt joint of collector during start of the block) caused a decrease of the forces F_s. This situation will be discussed bellow.

An average stresses in the bolt sections σ_m are below S_y during all service conditions. Bolts are so loaded in an elastic state. Coils in connection bolt - collector were loaded in elasto-plastic state during prestressing of bolts by the equipment NSK 48-4 PGP. The shakedown effect has been originated after unloading and the service force F_s was not higher than used force F_p.

Influence of packing ring ductility on the decrease of the force F_u during service condition was investigated too. Relation between force F_t effects on the packing rings and deformation of the ring l_t is evident from Fig. 5. Relation $\Delta l_t = f(\Delta F_t)$ is non-linear for monotonic loading (curve 1) and linear for cyclic loading (curve 2).

When bolts are tightened by key, the maximum force $F_{tmax} = F_u$ and the service force F_{ts} is higher than F_{tmax}. It means that plastic deformation of packing rings originated during service conditions. It causes the decrease of the force F_u after unloading. The packing rings work in an elastic state (curve 2, Fig. 5), when prestressing of bolts by the equipment NSK 48-4 PGP was enough high. It was a case of the investigated bolt joint. The decrease of the force F_u in bolts was most likely caused by a thermal ageing of packing ring material at 270 ^{0}C.

Experimentally measured relation $\Delta l_t = f(\Delta F_t)$ for a monotonic load of packing rings at 20 ^{0}C :

$$\Delta l_t = \left(\frac{F}{K} \right)^{1/n}$$

where n = 1.653 and K_o = 362.9 kN.
For cyclic load at 20 ^{0}C $\Delta l_t = \lambda F = 0.000213\ F$ [mm, kN].

5. NON-STANDART TRANSITION

The theoretical force F_s = 369.6 kN |3| in bolt for service conditions was calculated. The forces ^sF_s = 320.3 kN and 339.4 kN were obtained by means of strain gauges. These forces are less then theoretical ones. This difference of results was investigated.

Analysis of time trends of measured data showed that a non-standard cooling of the top of the collector caused a short time increase of forces in bolts an on packing rings, Fig. 6. Forces in bolts were less than F_p = = 393.3 kN and therefore packing rings worked in elastic state, Fig. 5. But coils which connect nuts with bolts were at the first time more loaded than at their tightening by equipment NSK 48-4 PGP. These coils were loaded in elasto-plastic state and plastic deformed. These plastic deformations of coils cased the decrease of forces in bolts when temperature field began to be homogeneous. The forces F_u = 272.3 kN and 276.8 kN decreased on F_u = = 235.3 kN and 245.9 kN after shutdown block for a removal of a fault. The forces decreased about F_u = 37.0 kN and 30.9 kN. The forces F_B = = 320.3 + 37.0 = 357.3 kN and F_B = 339.4 + 30.9 = 370.3 kN could be without the non-standard cooling of the bolt joint.

This non-standard transition was theoretically analysed. Temperature field in top of collector (cover, bolt, flange) was first calculated for this transition. Theoretical calculated temperatures are compared with experimentally measured temperatures on Fig. 7, 8 and 9 for the cover, flange of collector and bolt. When a good conformity was reached, then stresses in all elements of bolt joint and forces in bolts were calculated. These results are in comparison with an experimental data |4|.

REFERENCES

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Table 1. Mechanical properties of materials

Material	T /°C/	S_y /MPa/	S_u /MPa/	Z /%/	E /MPa/
08Ch18N10T	20 325	196 176	490 353	55 50	201000 184000
ChN35VT-VD	20 325	490 441	834 687	40 40	190000 186000

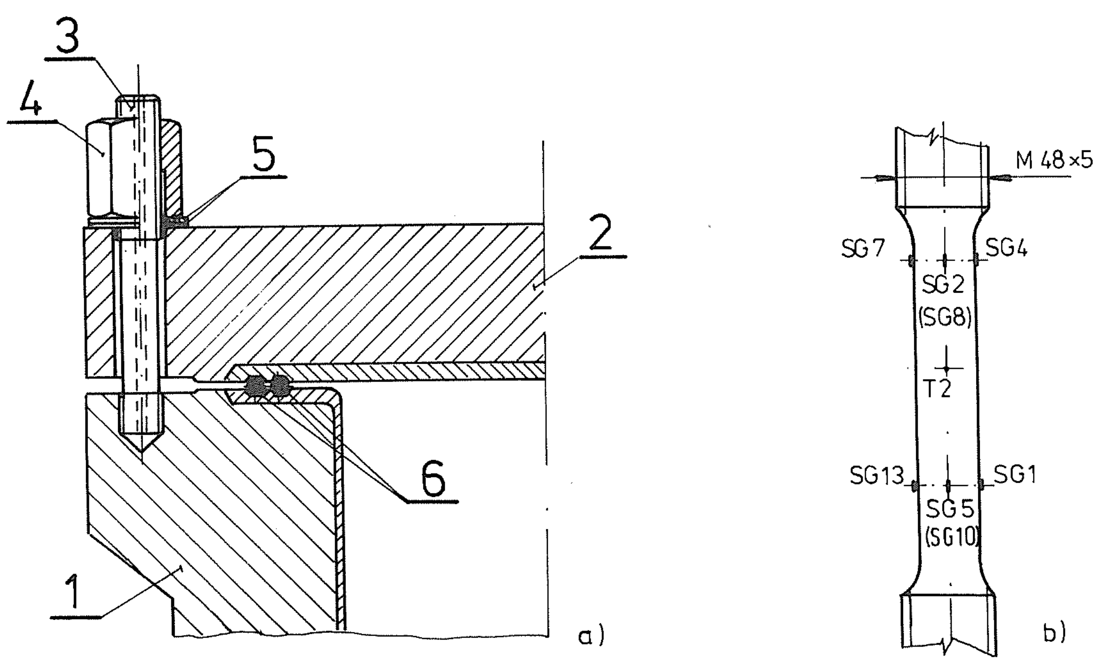


Fig. 1. a) Bolt joint, 1 - flange of primary collector, 2 - cover, 3 - bolt, 4 - nut, 5 - two pads, 6 - two packing rings
b) Location of strain gauges SG and the thermocouple on the bolt

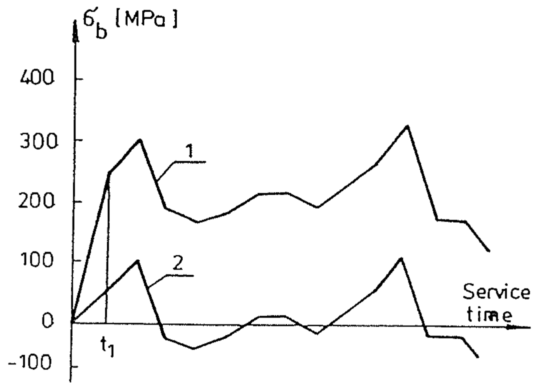


Fig. 2. 1 - tightened by torque wrench
2 - tightened by device
3 - state of tightening

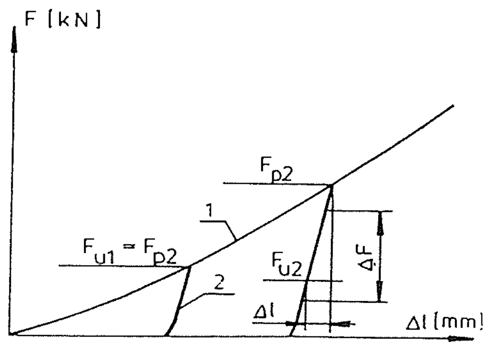


Fig. 5. 1 - monotoning load
2 - cyclic load

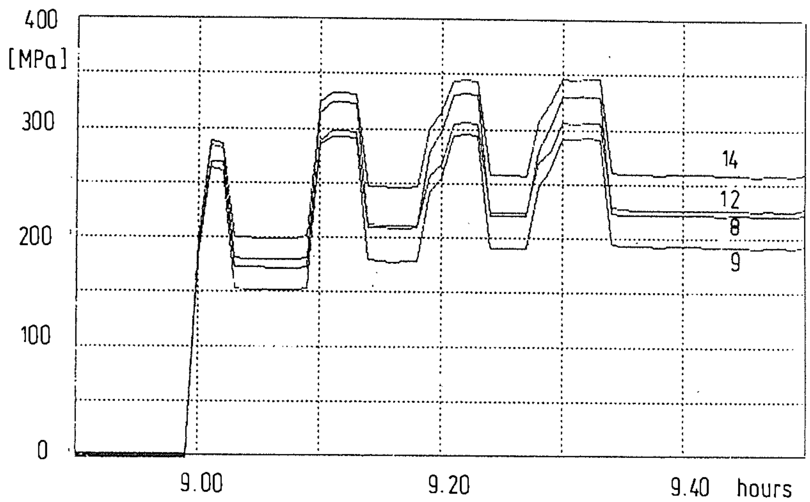
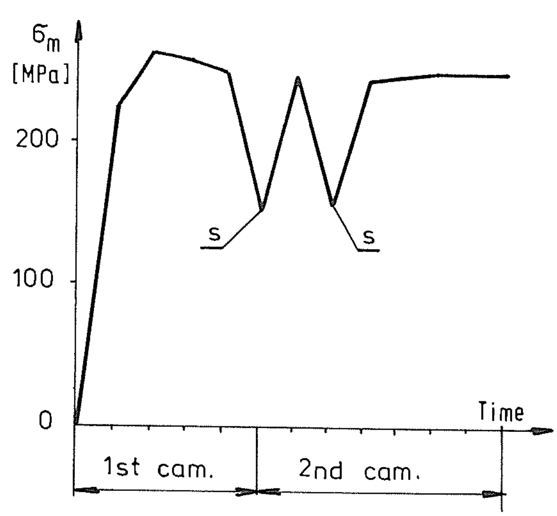


Fig. 3.
Stresses during four cycles of tightening; strain gauges No. 8, 9, 12, 14

Fig. 4.
Two campaigns of the block; s - shutdown; 1st and 2nd campaign



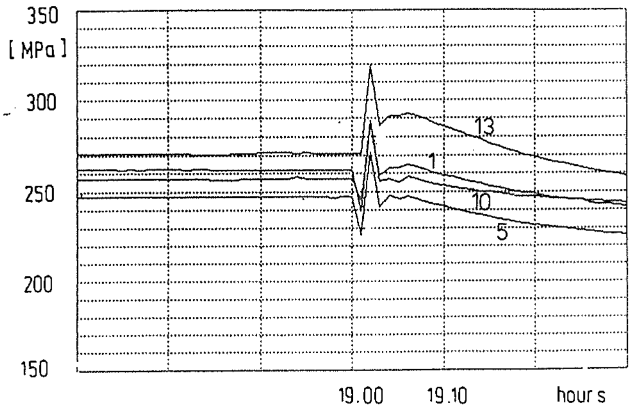


Fig. 6. Increase of stress in the bolt; strain gauges No. 1, 5, 10, 13

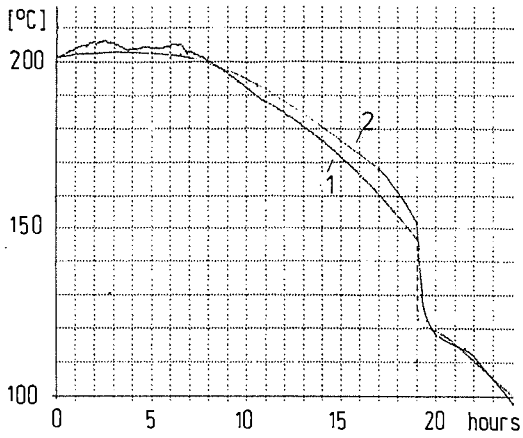


Fig. 7. Temperature of the cover
1 - measured
2 - calculated

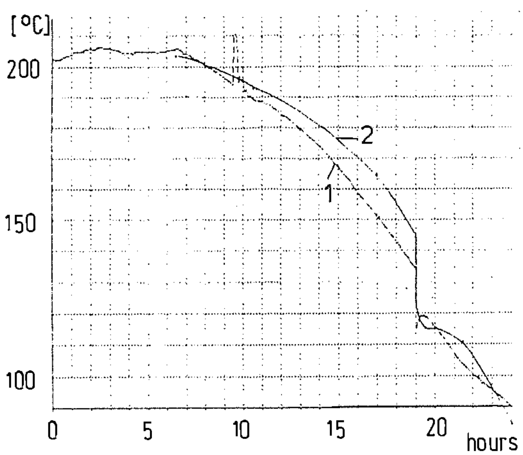


Fig. 8. Temperature of the collector flange.
1 - measured
2 - calculated

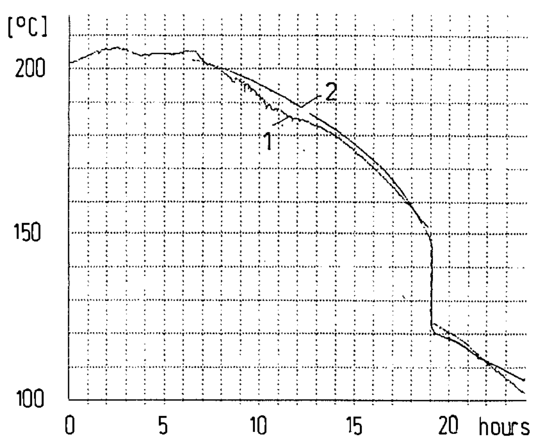


Fig. 9. Temperature of the bolt.
1 - measured
2 - calculated