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FATIGUE ASSESSMENT OF GROOVES FOR GASKETS IN REACTOR MAIN FLANGE

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ABSTRACT

Reactor pressure vessel (RPV) is a key component of all nuclear power plants. Ensuring its integrity during the whole lifetime is therefore of high importance and main objective of the RPV assessments. One of the main aging mechanisms of the RPV is fatigue. Significantly fatigue-loaded regions in the RPV include the sealing surface of the reactor main flange. During tightening of the flange, gaskets are pressed into the V-shaped grooves in the austenitic cladding on the sealing surface by the reactor closure head. After the tightening, the gaskets are fully yielded, and they are fully squeezed into the grooves. The cladding on the flange and on the reactor closure head, which is in contact with the gaskets and near the gaskets, becomes also fully yielded. One factor that significantly affects the fatigue damage development is the fact that the gaskets are replaced after each campaign. Because of the very large tightening force, which causes very large plastic deformations, and considering very complicated loading history, classical linear-elastic stress analysis and fatigue assessment cannot be used for the area of the grooves and their surroundings. Sequence of elastic-plastic finite element analyses was performed for the repeated tightening and untightening of the RPV main flange connection for the whole anticipated lifetime of the RPV.

INTRODUCTION

There are several places in RPV and reactor internals, where a different computational method than usual is needed, the usual method being linear-elastic analysis. In these places it is necessary to use material plasticity (i.e., to perform elastic-plastic analysis) to obtain correct stress/strain fields. This kind of analysis is very time-consuming. But nowadays, with the massive development of computational tools, this analysis is not a big problem, and we can perform a very demanding analysis in short time periods. For elastic-plastic analysis, it is not possible to use the superposition rule and it is necessary to take into account the entire loading history. One of the main problems is selection of correct material properties for modelling the cyclic plasticity and the choice of computational approach. Another problem is compiling loading history or loading sequence and the last problem is fatigue assessment based on elastic-plastic analyses.

One of the representative places, where it is necessary to perform the elastic-plastic analyses, is the sealing surface of the RPV main flange, more precisely, the grooves for nickel gaskets which ensure sealing. When tightening the flange with screws, the nickel gaskets are fully pressed into the grooves by the reactor closure head and they are fully yielded. The material (austenitic cladding on the flange and on the reactor head) is also fully yielded and strongly loaded when in contact with the gaskets. The gaskets are replaced after each campaign, and this is one of the main reasons that contributes to the significant fatigue of the sealing surface and groves.

In this paper, the procedure for fatigue assessment of the grooves located on sealing surface is presented. For this purpose, a fatigue procedure was developed, which is based on elastic-plastic calculations. This procedure is not directly implemented in the Czech standard for fatigue assessment NTD AME (2020). In this standard, there is only stated that it is possible to carry out this type of analysis.

CONSTRUCTION DESCRIPTION AND LOADING HISTORY

The main flange of the VVER 440 reactor is composed from screws M140, nuts, lower and upper washers, pressure screws M64 including inserts M85, free flange, pressure rings and two pairs of nickel gaskets (in total, four nickel rings). All mentioned parts serve for reactor sealing. The sealing is guaranteed by tightening the screws M140, which press nickel gaskets into the V-grooves via the reactor head. Details of the main flange and position of nickel gasket are shown in Figure 1.

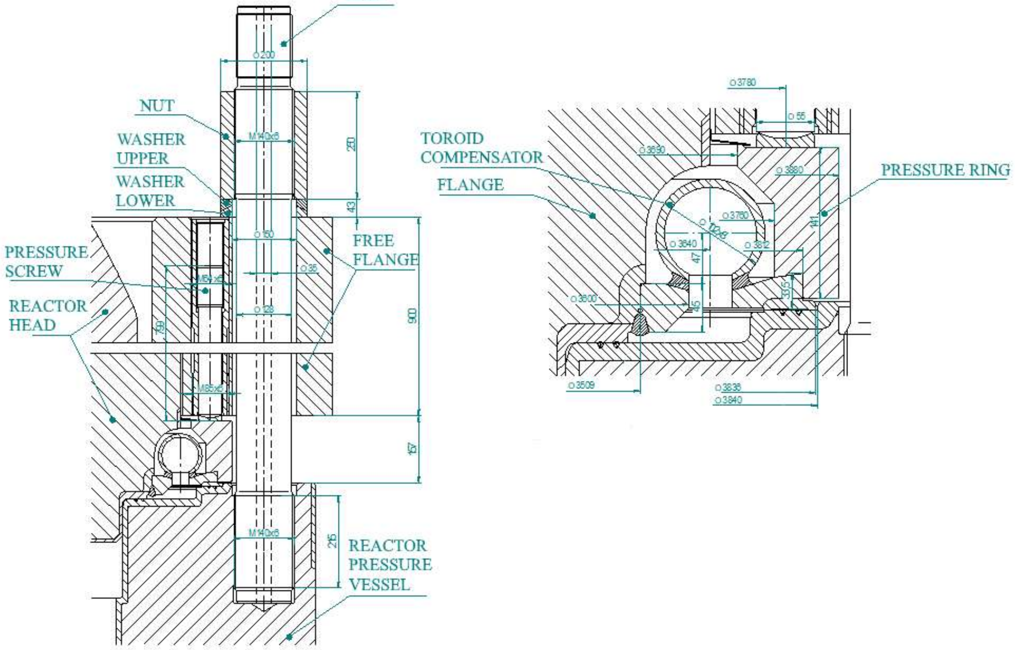

Figure 1. Part of the main flange of reactor VVER 440 and details of sealing and its positions.

Loading history

Most dominant loading for sealing surface, more specifically for V-shaped grooves located in austenitic cladding, is represented by tightening the flange, heating up to the operating temperature and then cooling down and unsealing. After each campaign (unsealing), the nickel gaskets are replaced with new ones. This whole regime is most dominant, since when the flange is tightened (via reactor head and due to tightening with the screws M140), the gaskets are fully squeezed into the grooves. After tightening, the gaskets are fully pressed into the grooves, while the grooves and the austenitic cladding which is in contact with gaskets, as well as near surroundings, are also significantly loaded (deformed). When the reactor is heated to the operating temperature, the grooves are more stressed/loaded due to different expansion coefficients of materials of austenitic cladding and nickel gaskets and due to the different temperature between the flange and reactor head and bolt.

Other loads, such as reactor internal pressure, have insignificant effects, and could lead to the small reduction of the gasket loading and this is the reason, why the pressure load (e.g., at pressure tests) was not taken into account in the computational loading history. Most dominant loading history, consisting in changing the gaskets after each campaign, was used in the simulation.

FATIGUE ASSESSMENT ACCORDING TO THE CZECH NATIONAL STANDARD

National standard for fatigue assessment for VVER reactors in the Czech Republic is NTD AME (2020); it is similar to the Russian standard PNAE G-7 002-86. In what follows, basic steps for low-cycle fatigue calculations are described for the standard fatigue assessment based on linear elastic stress analysis. These main steps are:

- Establishing sequence of stress or strain tensors in all assessed points (stress or strain history),
- Converting the stress tensors to scalars – equivalent stress (or stress intensity),
- Accounting for the effect of plasticity – converting equivalent stress (linear) to "fictitious stress", e.g. using Glinka approach (method of equivalent energy) or Neuber rule,
- Splitting the sequence of stresses into cycles and half-cycles using rain-flow algorithm or maximum range counting algorithm,
- Construction of the material-dependent fatigue design curves
- Calculation of the fatigue usage factor.

All these basic steps are used for fatigue assessment based on linear-elastic analysis, but in the case of elastic-plastic assessment some steps are omitted or modified. The conversion of Hook stress into the "fictitious" stress and finding the corresponding deformation is not necessary, because the true (total) strain is already available from the results of the elastic-plastic calculation.

In addition, in linear elastic fatigue assessment is used the fictitious stress and on fatigue assessment form elastic-plastic analyses is used the equivalent strain. The second case (fatigue assessment from elastic-plastic analyses) is further specified below.

For detailed description of transformation of Hook stress into the "fictitious" stress see Molski (1981). The above-mentioned steps of fatigue analysis, except of the effect of plasticity, are briefly described below and are valid for fatigue assessment for both linear-elastic and non-linear (elastic-plastic) cases.

Converting the stress tensors to scalars

Sequence of equivalent stress values (obtained from FEM calculations) cannot be used directly because we need to properly describe changing the characteristics of stresses (compressive to tension). According to the procedure specified in the standard, the principal stresses are numbered for time, where are the maximal stress achieved $\sigma_i > \sigma_j > \sigma_k$. For rest of the loading history is used the fixing direction, that corresponds to the directions of the maximal stress time. After finding the principal stresses, the sequence of differences of principal stresses is created over all time steps (Tresca approach).

These sequences of scalars are the inputs to the fatigue assessment. The fatigue usage factors are calculated for all three scalars separately and their maximum is then taken as the final result.

Splitting the sequence of stresses into cycles

Calculated sequence of stresses (by FEM or analytically) shall be split into the half-cycles. This is achieved by decomposition of calculated stresses using the "rainflow" method, or using the "maximum range" method. The "rainflow" method was proposed by the authors of the work Matsuiski (1969) for decomposition of a sequence of true deformations. The other approach is not discussed here, because the maximum range method is more conservative and may lead to overestimation of the results. In this case, the rainflow method is applied on the history of fictitious stress or strains.

Calculating the number of allowable cycles

Fatigue assessment of the components according to the standards (NTD AME and PNAE G-7) is theoretically based on the strain-life curves (ε-N curves), which are defined by relations:

$$\frac{\Delta\varepsilon(N)}{2} = \varepsilon_a(N) = \varepsilon_{apl}(N) + \varepsilon_{ael}(N), \tag{1}$$

where $\Delta\varepsilon$ means the strain range, N is the number of cycles to initiation of a defect, ε_a is the amplitude of deformation, ε_{apl} is plastic part and ε_{ael} is elastic part of total amplitude. Based on relation (1), it is clear that it is possible to split the curve into an elastic and a plastic part. The elastic part describes the area of small elastic amplitudes, and the plastic part describes the distinctive amplitudes of plastic deformations.

Elastic part ε_{ael} from equation (1) can be possibly expressed in several ways, the following expressions is often used:

1. Elastic part is defined only as fatigue limit $\varepsilon_{ael} = \frac{\sigma_c}{E}$ and the equation (1) is then in the form

$$\varepsilon_a(N) = \varepsilon_{apl}(N) + \frac{\sigma_c}{E} \tag{2}$$

2. Elastic part is defined as a Basquin relation (see Basquin (1910))

$$\varepsilon_{ael}(N) = \frac{\sigma_{fr}}{E} N^{-m_e}, \tag{3}$$

where σ_{fr} has meaning of fracture stress and m_e is Basquin exponent.

Plastic part could be defined using the Manson-Coffin relation, as defined by equation (4), which is defined in the work of Manson (1953) and Coffin (1954).

$$\varepsilon_{apl}(N) = \varepsilon_c N^{-m_p}, \tag{4}$$

where $-m_p$is the exponent of plasticity and ε_c is coefficient of plasticity which is defined according to NTD AME by the following equation (5).

$$\varepsilon_c = \frac{1}{2}\left\{\varepsilon_{fr} - max_t\left[\varepsilon_{pl}(t)\right]\right\}. \tag{5}$$

In relation (5), ε_{fr} has meaning of fracture deformation during the tensile test and $max_t\left[\varepsilon_{pl}(t)\right]$ is maximum reached plastic deformation during loading. After some substitution of coefficient of plasticity ε_c (see publication Manson (1953)), we can re-define the equation (4) to relation (6).

$$\varepsilon_{apl}(N) = \frac{\varepsilon_c}{(4N)^{m_p}}. \tag{6}$$

In this manner we have defined the elastic and plastic parts of fatigue curves and it is possible to define the equation (1) (and (2)) in more details.

$$\varepsilon_a(N) = \frac{\varepsilon_c}{(4N)^{m_p}} + \frac{\sigma_{fr}}{E(4N)^{m_e}}, \tag{7}$$

or with second relation of elastic part (defined as fatigue limit above)

$$\varepsilon_a(N) = \frac{\varepsilon_c}{(4N)^{m_p}} + \frac{\sigma_c}{E} \tag{8}$$

Relations (7) and (8) may be expressed as dependences of amplitude of fictitious stress σ_{aF} on number of cycles, as is done in the following equations (9) and (10).

$$\sigma_{aF}(N) = \frac{E\varepsilon_c}{(4N)^{m_p}} + \frac{\sigma_{fr}}{(4N)^{m_e}}, \tag{9}$$

$$\sigma_{aF}(N) = \frac{E\varepsilon_c}{(4N)^{m_p}} + \sigma_c. \tag{10}$$

In this manner defined relations (9) and (10) are used for assessment of the fatigue life and the derivation was briefly introduced above.

Relations (7) to (10) are defined for alternating symmetrical cycles. The value of mean stress has significant influence on elastic part of fatigue curves and is included in the fatigue curves by using Goodman approach or by Morrow approach, see publications Goodman (1899) and Morrow (1965).

The fatigue curves are finally reduced using the safety factors. In standards (NTD AME or PNAE G-7), there are introduced the safety factors on stress n_σ or on number of cycles n_n. These coefficients can have different values for different construction elements (parts). This aspect of fatigue curves construction is not discussed here, because it is not the main objective of this paper.

On the basis of the fatigue curves definition, it is possible for each cycle (or each half-cycle) to determine the number of allowable number of cycles [*N*], and then partial fatigue damage can be expressed as

$$D_i = \frac{n_i}{[N_i]}, \tag{11}$$

where D_i is partial damage caused by a cycle, n_i is number of cycles of i-th type and $[N_i]$ is allowable number of these cycles. For summation of all values of partial damage, the Palmgren-Miner hypothesis is used in the standards, see Palmgren (1924) and Miner (1945); the fatigue damage is then defined by the following relation (12).

$$D = \sum_{i=1}^{k} \frac{n_i}{[N_i]} \tag{12}$$

In relation (12), *D* has the meaning of cumulative fatigue usage factor and *k* is the total number of cycle types in the loading history. To fulfil the conditions of fatigue assessment needs to be the cumulative usage factor $D < 1$.

In the standards NTD AME and PNAE G-7, there are presented other "simplified" approaches, e.g., there are given the S-N (the dependence of allowable amplitudes of fictitious stress on number of cycles) curves in the form of their graphical representations (graphs) for different types of materials. This approach is recommended for simple evaluation only. The basic evaluation of fatigue is based on the relations (9) and (10). Construction of fatigue curves, specifically on relations (9) and (10), is dependent on the material properties obtained from tensile tests ($R_{p0,2}$, R_m, Z, E).

FATIGUE ASSESSMENT ACCORDING TO THE CZECH NATIONAL STANDARD FOR NON-LINEAR PROBLEMS

Standards for fatigue calculations, such as NTD AME (in Czech) allow for the fatigue assessment of elastic-plastic problems. Unfortunately, the detailed instructions how to perform this type of fatigue assessments are not presented there. An attempt is currently being made to add such a procedure to the Czech national standard NTD AME and the appropriate preparatory work is underway. Below in the text, the first proposal of such procedure is briefly described, which is conceptually based on the "usual" linear-elastic analysis.

In the previous chapter, the procedure of usual evaluation of fatigue has been briefly described (usual means fatigue assessment based on linear-elastic analysis). This procedure cannot be directly used for fatigue assessment of elastic-plastic problems. The basic assumption is that there is no need for transformation from Hook stress to fictitious stress, because from the results of elastic-plastic FEM calculation the strains (tensors) are already known and will be used in the fatigue assessment. It is clear from the description above that this fatigue procedure is based on Hook stress (expressed in general form), defined by the following relation (13), which represents the state of 1-D tensile idealization.

$$\sigma = \varepsilon \cdot E \tag{13}$$

Such idealization is no longer valid for cases, where the multiaxial stress and strain state occurs as well as in the case where the material is in plastic state (then the effect of Poisson constant ν applies). But still a question remains how to compose the equivalent strain (scalar quantity). In literature, there are several methods how to compose the equivalent strain for fatigue assessment, see e.g. Métais (2015). In the first proposal, the Tresca approach was used, based on total strain tensor which is defined by relation (14) below.

$$\varepsilon_{eq} = \frac{1}{1+\nu} Max_{i \neq j}\left(\left|\varepsilon_i - \varepsilon_j\right|\right), \tag{14}$$

where ν means the Poisson's constant and ε_i and ε_j are principal strain components.

Some questions still remain, for example, Poisson's constant value, whether to use the value of 0,5 (adequate to the plastic state) or to determine this number in dependence on deformation. During the transition from the elastic stress state to the fully developed elastic-plastic state, values of ν ranging in interval $\langle 0{,}3; 0{,}5 \rangle$ may be considered.

Another approach could be to apply the additive rule and compose the equivalent deformation from elastic and plastic part of the calculated deformation, defined e.g. by the following relation (15).

$$\varepsilon_{eq} = \frac{1}{1+\nu} Max(|\varepsilon_1^e - \varepsilon_2^e|; |\varepsilon_2^e - \varepsilon_3^e|; |\varepsilon_1^e - \varepsilon_3^e|) + \frac{1}{1.5} Max\left(\left|\varepsilon_1^P - \varepsilon_2^P\right|; \left|\varepsilon_2^P - \varepsilon_3^P\right|; \left|\varepsilon_1^P - \varepsilon_3^P\right|\right), \tag{15}$$

where constant 1,5 means Poisson's number for plastic state (defined as 1+ ν_{pl}), ε_{1-3}^emeans elastic principal strains and ε_{1-3}^P means plastic principal strains.

The problems associated with fatigue assessment based on elastic-plastic analysis are currently being discussed, and in near future it is expected that the procedure will precisely defined and implemented into the Czech National standard NTD AME.

COMPUTATIONAL MODEL FOR FATIGUE ASSESSMENT OF THE SEALING SURFACE

The sealing surface of reactor VVER 440 is approximately axisymmetric, and the computational model was created also as axisymmetric. All important components of the sealing surface are included in the model and were described above. This approach allows for detailed analysis (stress/strain field) of sealing grooves and surrounding areas and requires shorter computational time than 3-D model.

The FEM model was created with some simplifications. Some parts were not exactly axisymmetric, and therefore advanced techniques of modelling were needed. The free flange, bolt holes and other parts were modelled with using artificially specified anisotropic material properties. The appropriate procedure is described in the ASME code, Section III (2004). This approach was used in articles by Estrada (2015) and Kasahara (2008). The effective material properties are calculated using a reduction factor. This factor is equal to one minus the ratio of the volume e.g. of the bolt holes to the volume swept by the bolt diameter along the entire circumference of the flange along the bolt circle diameter, see relation 16. The original material properties are multiplied by the reduction factor and are used in the computational model. The Young modulus in the hoop direction should be very small ($E_{hoop} \approx 0$), and for Poisson number applies the same ($\nu \approx 0$). For these components the material properties are set as anisotropic.

$$V_r = 1 - \frac{V_0}{V}, \tag{16}$$

where V_r means the reduction factor, V_0 is for this case the volume of all bolt holes and V is the modelled volume of bolt hole.

Reactor head was modelled without influence of nozzles.

Finite element computational model was created in pre- and post-processor of Abaqus version 2019 FEM code. The model was meshed by axisymmetric quadratic elements marked CAX8HT. This type of elements is used for coupled temperature - stress analysis and has hybrid formulation (see Abaqus help).

This type of elements is convenient for large plasticity analysis. Preview of computational model is shown in Figs. 2 and 3.

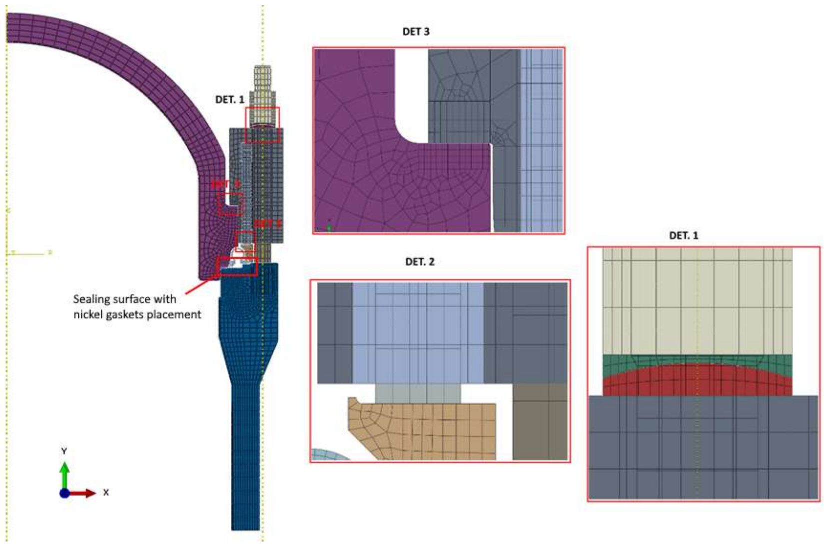

Figure 2. Computational model with details. Det. 1 – details of Bolt M140, nuts and free flange. Det. 2 – Details of internal pressure ring, pressure screw and free flange. Det. 3 – contact between reactor head and free flange.

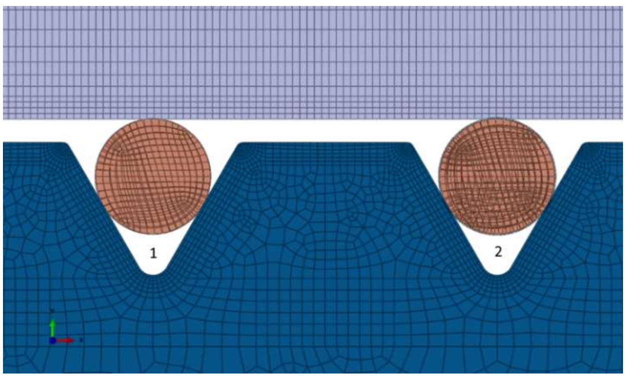

Figure 3. Detail of sealing grooves and nickel gaskets.

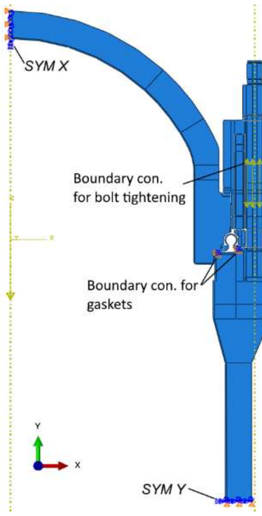

Figure 4. Boundary conditions for computational model

FEM analysis was performed as non-linear, including all types of non-linearities. This means material non-linearity, contacts, and large deformations.

Material properties

For this analysis, a combined material approach was used, i.e., some parts were modelled with linear material behaviour, and the parts of interest were modelled with elastic-plastic material behaviour, this means the austenitic cladding on the flange and on the reactor head. For elastic-plastic parts, the Chaboche material model was used, with three kinematic parts defined by relation (17). This relation is used for determination of parameters of cyclic hardening curve (in this form the isotropic hardening is neglected).

$$\sigma_a = \sigma_Y + \frac{C_1}{\gamma_1}\tanh(\gamma_1 \varepsilon_{ap}) + \frac{C_2}{\gamma_2}\tanh(\gamma_2 \varepsilon_{ap}) + \frac{C_3}{\gamma_3}\tanh(\gamma_3 \varepsilon_{ap}), \qquad (17)$$

where C_{1-3} and γ_{1-3} are material constants, σ_Y is yield stress, ε_{ap} is amplitude of plastic deformation and σ_a is the amplitude of true stress. For determination of Chaboche material properties, the least square method was used. Parameters of Chaboche material model are shown in Table 1.

Table 1: Parameters for Chaboche material model of austenitic clading.

C_1 [MPa]	γ_1 [-]	C_2 [MPa]	γ_2 [-]	C_3 [MPa]	γ_3 [-]
10180	231,96	1210,48	3,26	507,6	0,1

Since one of the main cyclic loading regimes is heating up and cooling down, the thermal-mechanical parameters such as specific heat c_p, thermal conductivity λ and density ρ were used in thermal part of the analysis, and the parameters such as thermal expansion coefficient α and Young modulus E (depending on temperature) were used in the mechanical part of the analysis. All the preceding material parameters were considered as temperature dependent.

For gaskets was used simply isotropic hardening and was used the tensile curve (stress-strain curve).

Boundary conditions

All boundary conditions are shown in the Figure 4. In all nodes lying in the plane of the reactor pressure vessel horizontal cross-section, zero displacements were specified in the normal direction to the section, i.e. in axial direction (lowest part of the pressure vessel model, marked as sym y, see Figure 4). In the plane of the reactor head section (at its pole), all nodes were given zero displacements in the normal direction to the section plane (denoted as sym x, see Figure 4). Since the model is axisymmetric, the boundary condition does not have to be defined on the axis of revolution. Since the calculation takes into account the gasket replacement after each campaign, each gasket has its own defined boundary condition for the stability of the calculation. This means that when the seals are not subjected to a load cycle, they are fixed in space. Conversely, when a seal is subjected to a load cycle, the boundary condition is deactivated, and the seal is included in the calculation.

One of the important boundary conditions is the tightening of the screw (Only one screw is modelled because of the axial symmetry of the model.). For this purpose a special tool in Abaqus was used. On the middle of the bolt, the FEM mesh is divided by a cut (section) perpendicular to the bolt axis. The mutual displacement of the nodes on both sides of the cut (section) is prescribed in such a manner that the both parts of the bolt are elongated. The values of mutual displacement are prescribed in such a way that the calculated elongation of the bolt (determined as sum of elongations of both parts of the bolt determined as the difference of displacements of the end nodes of these parts in the vertical direction) reaches the specified value of the M140 bolt elongation (measured during tightening).

Several contact pairs were considered in the model. The formulation of all contacts was “Surface to surface contact” with friction coefficient of 0,1.

RESULTS OF THE CALCULATION

As mentioned above, the entire history was calculated for the selected loading regimes. After each campaign, the gasket replacement was considered. In Figure 5, there is shown the contact status for first pair of gaskets after bolt tightening. From the FEM results (see Fig. 5), it is obvious that the whole gaskets are pressed into the grooves. Another finding is that reactor closure head is in contact with flange (in particular, cladding on both parts).

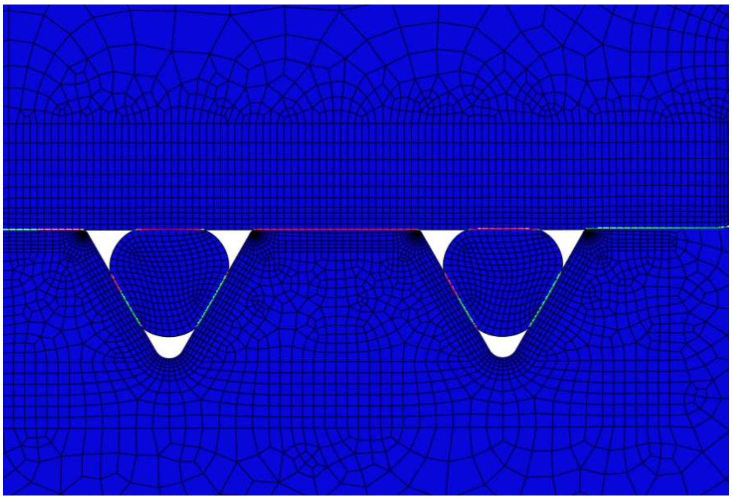

Figure 5. Contact status on first pair of gaskets after RPV main flange bolt tightening

Figure 6 shows the distribution of equivalent plastic strain (PEEQ) on first pair of grooves at the end of the expected operation lifetime (this means after 60 cycles). It is seen from this figure that the corners of grooves are the most loaded region. This is caused by the direct contact between the closure head and the flange, and the consequence is flattening of the corners (their original shape is circular with small radius). On the sides of grooves, there are seen strong “corrugations” from the gaskets and places with plastic strain concentration under the lower end of the contact with gaskets (under the surface). This is caused by embedding the cladding material lower and lower by gaskets, which further leads to a significant load of the groove bottom.

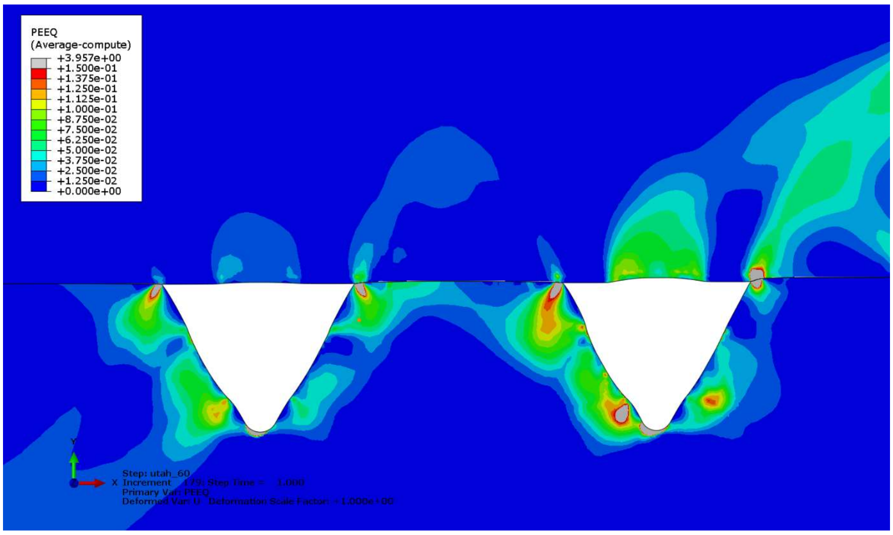

Figure 6. Distribution of equivalent plastic strain on the first pair of grooves at the end of the expected operation lifetime (without scale factor).

Based on the FEM computational results, the areas of the grooves were selected, in which the fatigue assessment was performed. These areas include the groove corners. In the groove corners the maximal fatigue usage factor was evaluated.

CONCLUSION

In the paper, fatigue assessment of the sealing surfaces, mainly of the flange grooves, was described. In particular, the design of the RPV main flange connection and the corresponding computational model were described. Various types of non-linearities were included in the model, specifically large deformations, material non-linearities and contacts. Moreover, the basic approach for fatigue assessment based on linear-elastic calculations according to the Czech national standard NTD AME and Russian standard PNAE G-7 was briefly described. Since NTD AME does not provide detailed instructions for performing the elastic-plastic fatigue assessment (although it is allowed for such type of analysis), the paper discuses a fatigue assessment procedure based on elastic-plastic calculations. This elastic-plastic fatigue assessment procedure represents certain modification of the usual fatigue assessment procedure that is based on linear-elastic calculations.

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