# DESIGN AND ANALYSIS OF A REACTOR PRESSURE VESSEL FOR THE SUPER PRESSURIZED WATER REACTOR

**Álvaro Juan Fernández[1], Thomas Schulenberg[2]**
[1] Instituto Tecnológico de Buenos Aires
[2] Karlsruhe Institute of Technology

## Abstract

A reactor pressure vessel has been designed for a Super Pressurized Water Reactor (SPWR) to be operated at 25 MPa core inlet pressure and 370°C core outlet temperature. Dimensions of the reactor internals have been taken from the European Pressurized Water Reactor (EPR), including its four inlet and outlet nozzles for the cold and hot legs of the primary system and its control rod positions. The pressure vessel is designed for a pressure of 27.8 MPa, at which the pressure relief valves are supposed to open, and for a design temperature of 400°C, the hottest temperature to be achieved during normal operation. Wall thicknesses of cylindrical and spherical parts, reinforcements around the nozzles as well as nuts and bolts of the closure head have been dimensioned according to the German safety standard KTA 3201.2. The advanced material SA508 Gr.4N has been selected for the pressure vessel and alloy 26 NiCrMo 14 6 for the bolts. Both materials still need to be qualified for temperatures up to 400°C. A double O-ring gasket has been chosen from present offers of Technetics Group.

The total arrangement has been designed with the CAD system Autodesk Inventor and the geometry has been exported to ANSYS Workbench for Finite Element Analyses of stresses and deformations. The pressure vessel is assumed to rest upon support pads underneath the nozzles. Bolts have been pre-stressed to keep the pressure vessel leak tight even under cold pressure tests. Steady-state analyses at design pressure and temperature confirm that membrane and bending stresses of the reactor pressure vessel comply with the maximum allowable stress according to load case "Level 0" of the KTA standard.

Moreover, a transient temperature and stress analysis has been performed to check the flange deformations at the gaskets and to check local peak stresses for fatigue according to load case "Level A" of the KTA standard. The deformation analysis confirms that the flange stays leak tight during the entire cycle. The fatigue analysis confirms that more than 1000 of such cycles can be allowed.

**Keywords:** Reactor Pressure Vessel Design, Super Pressurized Water Reactor, Structural Analysis.

## 1. Introduction

In the present article, design and analysis of a Reactor Pressure Vessel (RPV) are carried out, with the purpose of being used for a high temperature pressurized water nuclear reactor. The process is done according to the German standard KTA 3201.2 [1]. The framework, in which it is inscribed, is a broader design concept for a Supercritical Pressurized Water Reactor (SPWR) [2].

The purpose of the aforementioned concept is to increase the net power output to 1700 $MW_e$ and the thermal efficiency of the cycle by around 9% when compared to a conventional third generation Pressurized Water Reactor (PWR), like the European Pressurized Reactor (EPR) or the AP1000, by increasing the core outlet pressure and therefore the primary cycle's temperature. This in turn allows to generate steam of a higher temperature in the secondary loop, hereby increasing the cycle efficiency

[2]. The main intention behind this concept is to achieve an increase of power and efficiency without making a major re-design of the primary loop, but instead being constructively very close to the PWR models existing nowadays. The main argument behind this design premise is an evolutionary development based on as many already tested and used components as possible.

## 2. Design requirements and conditions

The inlet temperature of the water that enters the core shall be 345°C and the average outlet temperature 370°C, as explained in [2]. The design pressure of the vessel, at which the pressure relief valves open, is considered to be 27.8 MPa and the operational pressure shall be 25 MPa. This allows producing superheated steam of 350°C in the steam generator [2]. The design temperature, i.e. the maximum achievable temperature during normal operation, is assumed to be 400°C.

The proposed design should be able to withstand at least 1000 warm-up and cool-down cycles without fatigue damage. Such loads are caused by loading and unloading the RPV, e.g. for shut down and refuelling the reactor, whereas the pressure test during commissioning, as a single event, does not cause fatigue.

The concept of the RPV presented in this work keeps a close resemblance with the EPR design. The decision has been made to keep all internal dimensions the same, including the inner diameter of the vessel and the nozzles, the number and positioning of nozzles, the location and number of the standpipe holes for the control rod mechanisms, and the overall inside RPV height, among other relevant parameters. These design constraints are summarized in Table 1.

Table 1. Design constraints that limit the model.

| Dimension | Value | Unit |
|---|---|---|
| Vessel inner diameter | 4,885 | mm |
| Vessel inner height | 13,385 | mm |
| Height of the core barrel | 8,790 | mm |
| Diameter of the core barrel | 4,305 | mm |
| Distance from nozzles to flange | 1,666 | mm |
| Inner diameter of nozzles | 780 | mm |
| Pitch of openings in closure head | 305 | mm |

### 2.1 Loading levels

Relevant for the present analysis are the following loading levels, detailed in the KTA standard [1]:

Level 0 covers all static loadings which are due to the effect of design pressure and additional design mechanical loadings, such as weight, bolt forces and forces from attached components. The limits of level 0 are fixed, "such that the loadings generate equilibrium with external mechanical loadings, in a way that neither deformation nor fast fracture occurs" [1]. The dimensioning of the components is conducted taking this premise into account.

Level A takes into consideration static loadings due to pressure, temperature, dead weight, mechanical loads, reaction forces, restriction to thermal expansions, and transient loads from various sources, such as pressure and temperature. This loading level is relevant to assess load cycles and shall be used for the transient analysis to evaluate fatigue damage.

Lastly, Level P takes into account test loads. The pressure tests under cold conditions exceed the design pressure by 30%. It is relevant for this work, because it determines the load of the bolts.

## 3. Material properties

The material SA508 Gr.4N was selected for the shell of the reactor pressure vessel because of its better mechanical properties (Kim, 2015 [3]) when compared to the material 16 MND 5 (KTA 3201.1 [4]), which is a typical material found in existing reactors, such as the EPR. This material, however, would still need to be qualified according to [4] if used for this application.

In accordance with section 7.7.3.4 of the standard KTA 3201.2 [1], the design stress intensity $S_m$ is obtained based on the temperature $T$ of the respective component and the room temperature $RT$. For ferritic materials except for bolting materials, the design stress intensity is calculated according to equation (7.7-3) of [1]:

$$S_m = \mathrm{MIN}\left\{\frac{R_{p0.2T}}{1.5}, \frac{R_{mT}}{2.7}, \frac{R_{mRT}}{3}\right\} \tag{1}$$

Where $R_{p0.2T}$ is the yield stress of the material at the design temperature (400°C), $R_{mT}$ is the tensile strength at the design temperature and $R_{mRT}$ is the tensile strength at room temperature. The tensile properties for SA508 Gr.4N alloy had been determined experimentally up to 360°C [3]. Since the design temperature for the reactor pressure vessel is 400°C, it was necessary to extrapolate these data using linear regression and a minus two-sigma criterion to estimate the minimum strength. The obtained value for the design stress intensity is $S_m = 200$ MPa [5].

## 4. Dimensioning of the RPV

### 4.1 Dimensioning of the cylindrical shell

The first dimension to be determined for the design of the reactor pressure vessel is the wall thickness of the cylindrical portion of the vessel's body. According to section A 2.2.2 of the standard KTA 3201.2 [1], the expression used to obtain the wall thickness $s_0$ for a cylindrical shell under internal pressure is:

$$s_0 = \frac{d_i . p}{2. S_m - p} \tag{2}$$

For an inner diameter $d_i$ of 4885 mm and a design pressure $p$ of 27.8 MPa, the wall thickness for the cylindrical shell of the reactor pressure vessel shall be equal to 370 mm.

### 4.2 Dimensioning of the spherical heads

The second step in the dimensioning process is to calculate the wall thickness of both spherical heads. For the calculation of the required wall thickness of thin-walled spherical shells, with a ratio $s_0/d_i$ not exceeding 0.05, the following applies (Eq. A 2.3-5 in [1]):

$$s_0 = \frac{d_i \, . p}{4 \, . S_m - 2. p} \tag{3}$$

The calculated value for the unpierced spherical head (bottom header) is $s_0 = 184$ mm. On the other hand, in order to calculate the wall thickness of the closure head (top header), it is necessary to take into account the stress concentration due to the holes for the standpipes for the control rod drives. Even though the design of the standpipes exceeds the scope of this work, it is necessary to estimate their diameter and wall thickness to properly dimension the closure head. Their inner diameter is chosen to be 100 mm in order to provide enough space for the control rods, even if the standpipes are violently vibrating during an earthquake. As a reference for this, the design of the control rod of the AP1000 reactor was considered [6], because such information was missing for the EPR design. In order to calculate the standpipe wall thickness, Eq. (2) was applied and a value of 10 mm was obtained. This resulted in an overall hole diameter of 120 mm to accommodate for the standpipes.

Given the geometrical characteristics of the pierced header, a stress concentration factor of 1.5 is taken from [7]. Therefore, the wall thickness for the pierced closing head should be at least 50 % greater than the one for the unpierced bottom spherical shell. The shell thickness should be in this case equal to 276 mm.

### 4.3 Calculation of nozzle reinforcements

Following the line of the EPR design [8], the RPV counts with 8 nozzles: 4 inlet nozzles that guide the coolant that comes from the coolant pump into the reactor core, and 4 outlet nozzles, that conduct the hot coolant water coming from the core, towards the piping that leads to the steam generator. These nozzles constitute a weakening of the basic shell of the vessel. Therefore, this weakening must be compensated by adequately adding material at specific places. Openings in the shell were reinforced by increasing the wall thickness both in the basic shell and in the nozzle.

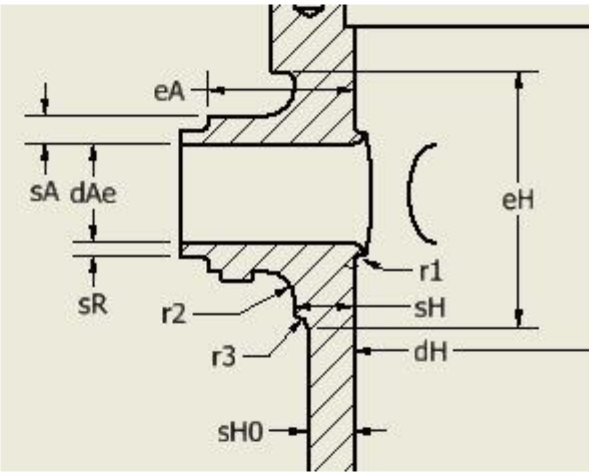



Figure 1. Nozzle geometry designed for the RPV and the relevant nomenclature for the design parameters.

The nozzle wall thickness must be at least twice the one calculated for the connecting piping. For the connecting piping, an internal diameter of 780 mm is needed in order to supply the necessary coolant to the core. All the relevant nomenclature for the nozzle reinforcement design can be seen in Figure 1. The total cross-sectional area $A$ of the required reinforcement of any opening in cylindrical, spherical, and conical shells as well as dished heads under internal pressure shall satisfy the following condition (Eq. A 2.7-1 in [1]):

$$A \geq d_{Ae} \, . \, s_{H0} \, . \, F \tag{4}$$

Here, $d_{Ae}$ shall be equivalent to the internal diameter of the nozzle. The correction factor F equals 0.5 in this case. The dimensions obtained for the nozzle and basic shell reinforcements can be seen in Table 2.

The expression used to obtain the shell reinforcement length $e_H$ is (Eq. A 2.7-4 [1]):

$$e_H = 0.5\, d_{Ae} + s_H + s_A \tag{5}$$

Table 2. Main geometrical dimensions of the inlet and outlet nozzles.

| Nomenclature | Dimension | Value [mm] |
|---|---|---|
| $s_R$ | Unreinforced nozzle wall thickness | 112 |
| $s_H$ | Reinforced shell wall thickness | 472 |
| $s_{H0}$ | Unreinforced shell wall thickness | 370 |
| $s_A$ | Reinforced nozzle wall thickness | 224 |
| $e_H$ | Shell reinforcement length | 1090 |
| $e_A$ | Nozzle reinforcement length | 285 |
| $d_{Ae}$ | Nozzle inner diameter | 780 |

According to Eq. A 2.7-7 of the KTA 3201.2 standard [1], the equation that shall be applied in order to calculate the nozzle reinforcement length $e_A$, measured from the cylindrical shell's outer surface is:

$$e_A = 0.5\left(\sqrt{0.5\, s_A\,(d_{Ae} + s_A)} + r_2\right) \tag{6}$$

The unreinforced nozzle wall thickness $s_R$ was chosen to be equal to 112 mm in order to keep the same $s_R/s_{H0}$ ratio as in the EPR reactor [8]. The radius $r_2$ is shown in Fig. 1. The wall thickness of the reinforced nozzle portion $s_A$ is equal to 224 mm since it must be twice the value of the unreinforced nozzle wall thickness $s_R$ [1]. Therefore, the calculated reinforcement area is $A =$ 183792 mm$^2$.

## 4.4 Design of the closure head bolts

The flange selected for the connection with the closure head is designed with metal-to-metal contact. In this type of flange design, the closure head makes contact with the flange on the body of the vessel, such that the gasket is completely enclosed by metal. The flange is fastened with bolts, since the joint has to be disassembled periodically in order to allow for refuelling, inspection and repair. Therefore, the bolts had to be dimensioned first. For this design, a total number of bolts $n = 44$ was chosen. One key factor, that had to be taken into account for this decision, was being able to locate the bolts on a circumference, leaving enough space for the cylindrical nuts, and at the same time having space for the bolting machine around the nuts to tighten up the bolts. These bolts are evenly spaced along a circumference of diameter 5600 mm.

Under design conditions, the allowable stress is equal to 498 MPa for the chosen bolt material 26 NiCrMo 14 6 (KTA 3201.1, 2017 [4]). Since the properties of this material have only been tested up to a temperature of 350°C, an extrapolation was needed. For later application, this material will have to be qualified up to 400°C according to [4]. As a result, size M220x6 nuts and bolts were selected. The calculated thread length is equal to 180 mm. Details of this analysis can be found in [5].

## 4.5 Calculation of the closure head flange

Finally, in order to correctly dimension the flange of the RPV, a minimum required flange section modulus $W_{erf}$ must be calculated (Eq. A2.9-3 in [1]):

$$W_{erf} = \frac{F_{SBU}\, a_D}{S_m} \tag{7}$$

Where $a_D$ represents the distance between the bolt hole center and the point of application of the compression load on the gasket $F_D$, as indicated in Figure 2, and $F_{SBU}$ is the bolt load required to keep the flange leak-tight.

With this in mind, the required effective flange thickness $h_F$ can be obtained as (Eq. A2.9-34 in [1]):

$$h_F = \sqrt{\frac{4\,W_{erf}}{\pi\,(d_F - d_i - 2\,d_L')}} \tag{8}$$

Where $d_F$ is the flange outside diameter and is equal to 6100 mm and $d_i$ is the inside diameter of the flange.

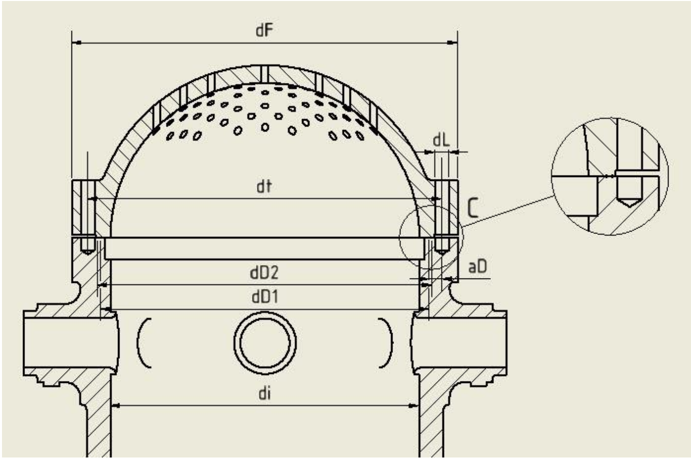



Figure 2. Metal to metal contact flange type used in the RPV closure head with detail view of the double gasket.

The calculation of the bolt circle design diameter $d_L'$ is done according to the expression (Eq. A2.9-26 [1]):

$$d_L' = \frac{n\,d_L^2}{4\,d_t} \tag{9}$$

Where $d_L$ is the bolt hole diameter, which is equal to 220 mm, and $d_t$ is the bolt circle diameter, equal to 5600 mm. The value obtained for the required flange thickness is $h_F = 772$ mm.

## 5. Design of the reactor pressure vessel

The RPV was modelled in 3D, using Autodesk Inventor, taking into consideration the calculated dimensions obtained in Section 4. The domain, that was considered for the development of the model, starts in the inlet nozzles and finishes in the outlet nozzles. Therefore, the connecting piping was not calculated or modelled. Moreover, the system that contains the control-rod drives was not designed either and is out of the scope of this work.

The general geometry and design of the reactor pressure vessel can be seen in Figure 3. Cross sections are shown in Figures 1 and 2. That figure shows an assembly of the RPV shell, the closure head, the 44 bolts and the 8 nozzles. Some of the relevant geometric features of the RPV can be seen in Table 3.

It is relevant to mention that a double gasket design was used for the closure head. The O-rings selected from the Technetics catalogue [9] allow for a total spring-back of 0.16 mm, which means that they can guarantee that the RPV stays leak tight as long as the gap between the body of the vessel and the closure head stays below that value. In the Finite Element Analysis, it needs to be checked that the gap between both flange sides does not exceed this spring-back value.

Table 3. Main geometrical dimensions and features of the RPV.

| Dimension | Value | Unit |
|---|---|---|
| Overall height | 13.847 | m |
| Shell outer diameter | 5.625 | m |
| Flange outer diameter | 6.100 | m |
| Number of standpipe holes | 105 | - |
| Number of nozzles | 8 | - |

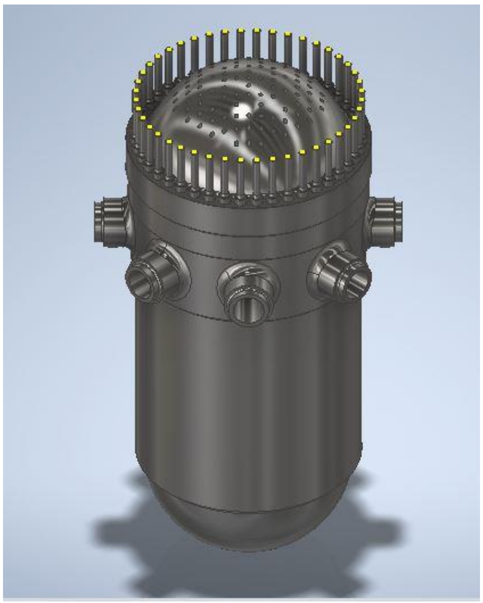


Figure 3. CAD render of the Reactor Pressure Vessel design created using Autodesk Inventor.

## 6. Structural analysis

### 6.1 Static structural analysis

The first analysis that was performed to the RPV design was a static structural simulation. As it has been previously mentioned, the standard KTA 3201.2 [1] describes several loading cases that should be considered and analyzed. Level 0 takes into account the loadings that are due to the effect of design pressure and additional mechanical loads to yield the maximum primary stresses from normal operation. This load case includes the design pressure, which is 27.8 MPa, and the design temperature, 400°C.

For the purpose of this analysis, the static structural module of the ANSYS Workbench was used. The CAD Model was imported from Autodesk Inventor and the ANSYS meshing tool was used to generate the mesh. The bolts and nuts where not included in the model, because meshing such small parts would add too much complexity and computational time. Instead, they were modelled as two forces with equal magnitude and opposite sign, acting on the flange and on the closure head, closing the pressure vessel. In order to save computational time and simplify the model, a quarter symmetry was used. This was made possible thanks to the symmetry of geometry and loads as well.

Forces were added at the nozzles in order to compensate for the internal pressure acting on the nozzles. The same was done to compensate the pressure acting on the cross-section of the standpipe holes. Finally, the bolt load, which was applied, was the one needed to resist the test pressure. However, the bolts were dimensioned for (hot) operation conditions since they are more demanding, as discussed in [5]. Even though the simulation was run for operational conditions, the bolts are fastened for the pressure test conditions. Therefore, the forces that were used to simulate the bolt loads acting on the flange and header are equivalent to the bolt load under test conditions. A zero-displacement condition in the vertical direction was imposed at the pads underneath the nozzles. In order to avoid rotations, no additional restriction was required, since the symmetry conditions already impose a restriction to displacements in the direction normal to the symmetry planes. The loads and constraints applied to the model can be seen in Table 4.

Table 4. Loads and constrains applied on the reactor pressure vessel for the static structural simulation.

| **Denomination** | **Value** | **Unit** |
|---|---|---|
| Design internal pressure | 27.8 | MPa |
| Force nozzle 1 | 13.28 | MN |
| Force half-nozzle 2 | 6.64 | MN |
| Force half-nozzle 3 | 6.64 | MN |
| Bolt force header | 225.62 | MN |
| Bolt force shell | 225.62 | MN |
| Force standpipe openings | 6.93 | MN |

According to the standard KTA 3201.2 [1], membrane stress $P_m$ shall not be greater than $S_m$, the localized stress $P_l$ may not exceed 1.5 $S_m$ and the sum of membrane plus bending stresses $P_m + P_b$ and localized plus bending stresses $P_l + P_b$ shall be smaller than 1.5 $S_m$ (see Table 5). In order to compare the results with the acceptable limits, first, the most compromised area of the RPV had to be identified. Then, a path was plotted across the wall thickness in order to obtain the stress profile and divide the stresses into membrane and bending components. It is important to note that the peak stresses at the nozzles, reaching more than 500 MPa, can relax and they are not relevant for this static structural analysis. Instead, they shall be assessed in the following chapter, that deals with transient loads.

Table 5. Stress limitations for Level 0 loading.

| **Stress category** | **Limit** | **Value** |
|---|---|---|
| $P_m$ | $S_m$ | 200 MPa |
| $P_l$ | 1.5 $S_m$ | 300 MPa |
| $P_m + P_l$ | 1.5 $S_m$ | 300 MPa |
| $P_b + P_l$ | 1.5 $S_m$ | 300 MPa |

As it can be seen in Figure 4, the critical area for the Von Mises equivalent stress is the nozzle reinforcement ring area. In order to perform a numerical assessment, a path was created along the wall thickness in this position in order to evaluate the Von Mises stresses. The plot of the Von Mises stresses vs the position along the wall thickness can be seen in Figure 5a. The location of the path, along which the stresses are plotted, can be seen in Figure 5b.

The calculated membrane stress $P_m$ is equal to 182 MPa, which is below the 200 MPa limit, and the bending stress $P_b$ is equal to 66 MPa. When added to the membrane stress, $P_m + P_b = 248$ MPa, which is clearly under the 300 MPa limit.

A verification for the displacements in the flange area was performed in order to check that the gap does not exceed the spring-back of the O-rings, thus securing that the joint remains leak tight. Let us remember that the spring-back value of the selected O-ring is of 0.16 mm [9]. For determining this gap, the contact tool available in the ANSYS Mechanical was used. A rough contact was assigned to the interface, that means that it allows for slip with a given friction coefficient, but it imposes no impediment to the separation of the two surfaces.

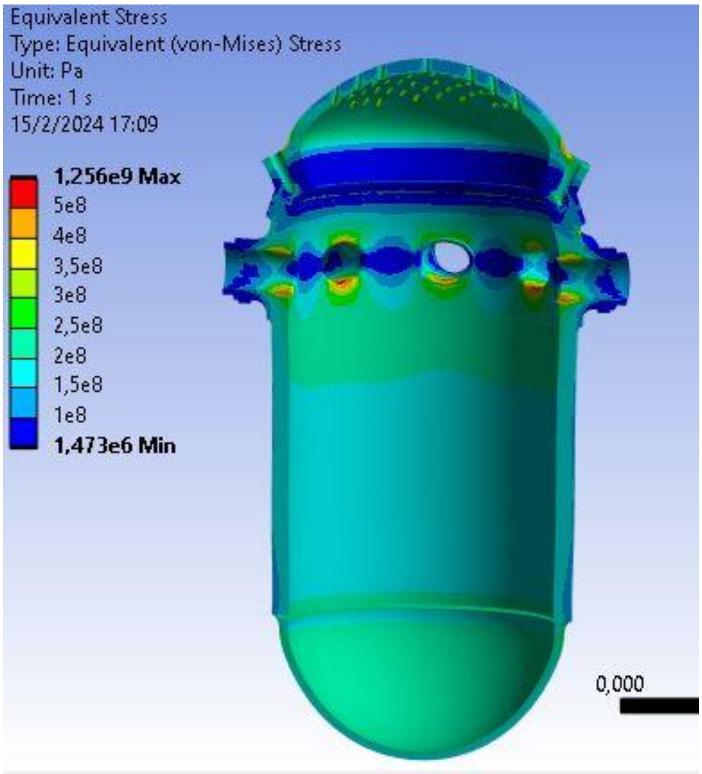
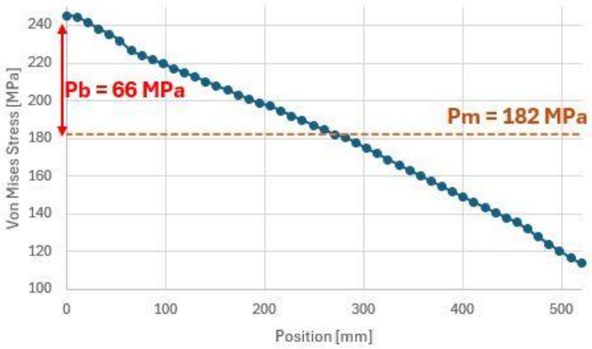
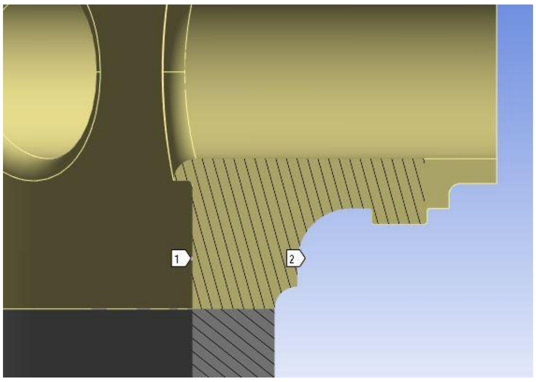



Figure 4. Von Mises stress distributions of the entire vessel (half-section view), deformed structure 130 times magnified.




(a) Von Mises stress distribution




(b) Location of the path.

Figure 5. Stress distribution along a path through the wall thickness of the vessel located at the nozzle reinforcement ring.

As it is shown in Figure 6, the colours red and orange are present in the area surrounding the gaskets. This represents a gap value of around 0.028 to 0.056 mm, which is an order of magnitude below the maximum allowable value. This means that the O-ring seals are able to fulfil their task of keeping the joint leak tight.

### 6.2 Transient thermal analysis

According to the KTA 3201.2 standard [1], loading level A takes into account all transient loadings due to pressure, temperature and mechanical loading.

For this purpose, a transient thermal analysis was performed for the warm-up and the cool-down processes to determine the temperature field of the RPV structure. Once the temperature field was determined, it was possible to establish the stress field due to all the loadings.

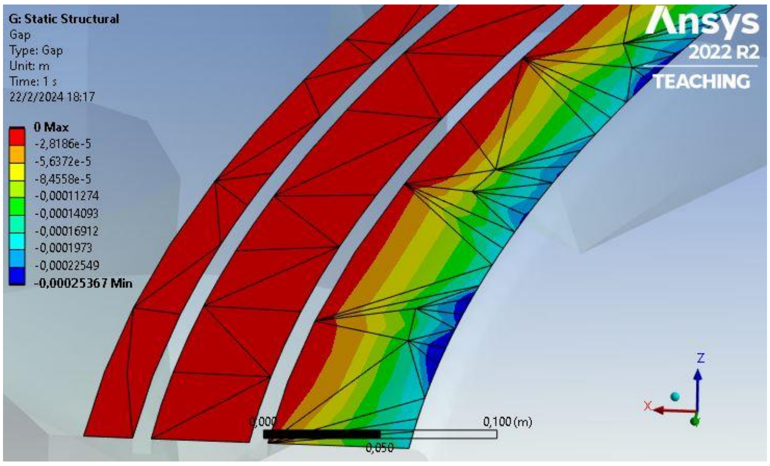



Figure 6. Gap between the closure head and the RPV body.

As a typical example, a warm-up cycle of 6 hours followed by a waiting time of another 6 hours is considered here. The same period was used for the cool-down cycle. The warm-up period includes an increase of the pressure from ambient pressure to 27.8 MPa, along with an increase of water temperature from 22°C to 350°C. The ramp-up of the pressure is linear and lasts for the first 6 hours. The subsequent 6 hours of simulation consist of maintaining the design pressure and temperature constant. After this, the cool-down process starts, ramping down pressure and temperature for 6 hours back to ambient conditions, which are maintained for another 6 hours of simulation. The simulation was conducted with time steps of 100 seconds. The inlet and outlet core temperature have the same value for this simulation, since water is considered to be heated by the pumps, not by the core.

The heat transfer coefficient on all surfaces in contact with water was determined by the Dittus-Bölter correlation [10] as:

$$\mathbf{Nu}_{\mathbf{D_h}} = 0.023\, \mathbf{Re}_{\mathbf{D_h}}{}^{0.8}\, \mathbf{Pr}^{0.4} \tag{10}$$

Where $\mathbf{Nu}_{\mathbf{D_h}}$ is the Nusselt number with reference to the hydraulic diameter, $\mathbf{Re}_{\mathbf{D_h}}$ is the Reynolds number with reference to the hydraulic diameter, and $\mathbf{Pr}$ is the Prandtl number:

$$\mathbf{Nu}_{\mathbf{D_h}} = \frac{h\, D_h}{k} \tag{11}$$

$$\mathbf{Re}_{\mathbf{D_h}} = \frac{\rho v D_h}{\mu} \tag{12}$$

$$\mathbf{Pr} = \frac{c_p \mu}{k} \tag{13}$$

Where $h$ is the heat transfer coefficient, $k$ is the thermal conductivity, $\rho$ is the density, $v$ is the velocity, $\mu$ is the dynamic viscosity and $c_p$ is the specific heat at constant pressure.

A total coolant mass flow of 21,208 kg/s has been proposed in [2] for this reactor type, or 5,302 kg/s per inlet or outlet nozzle. This causes a velocity of 17.7 m/s inside the nozzles at 25 MPa and 350°C. The heat transfer coefficient is dependent on pressure and temperature, but a single value was averaged out for the entire cool-down and warm-up cycle in order to simplify calculations. As a result, an average heat transfer coefficient of 20,000 $W/m^2\ K$ was obtained for the inlet and outlet nozzles, which was taken also for the inside of the downcomer and the lower plenum. A heat transfer coefficient had to be estimated for the upper plenum as well. In this area, heat transfer conditions are poorer, since there is less water circulation, and therefore less convection. For simplification, it is estimated that the heat transfer coefficient in the upper plenum is around 10% of the one at the other inner walls of the vessel. Therefore, a value of 2,000 $W/m^2\ K$ was used for the simulation at the inner wall of the upper plenum. All outside surfaces were assumed to be thermally insulated.

The result of the transient thermal simulation served as input for a structural simulation, which took into account not only mechanical loads, but also the thermal ones (Level A). The support conditions for said structural simulation are the same that were used for the static structural simulation, discussed in the previous section. After running the structural analysis, the results for the Von Mises stresses were exported to a text file for further processing.

The following step was to find, for each node, the maximum and minimum stress value. This allows to calculate the stress range that each one of the nodes undergoes throughout the total warm-up and cool-down cycle. The stress amplitude $S_a$ is calculated as half of this stress range. After some plastic deformations with cyclic hardening during the first load cycles, the stress-strain-cycle remains linear elastic if the stress range, determined by a linear elastic analysis, does not exceed 3 $S_m$. In these cases, the KTA standard [1] is allowing therefore in its section 7.8.3 to predict fatigue simply by a linear elastic analysis. Consequently, if an area of the RPV were subject to a stress amplitude above $1.5\,S_m = 300$ MPa, an elasto-plastic fatigue analysis would be necessary. Nevertheless, after analysing and cleaning up all the data, no points where found, where the stress amplitude exceeded this limit. The Von Mises stress curves for the 10 most compromised points can be seen in Figure 7. The maximum stress amplitude there is 270 MPa. The linear elastic analysis is sufficient there to predict the number of cycles to fatigue failure since the 300 MPa limit is not surpassed. This results in 10,000 cycles to failure according to the fatigue curve, Fig. 7.8-1 in [1]. The design target of 1,000 cycles is clearly exceeded there.

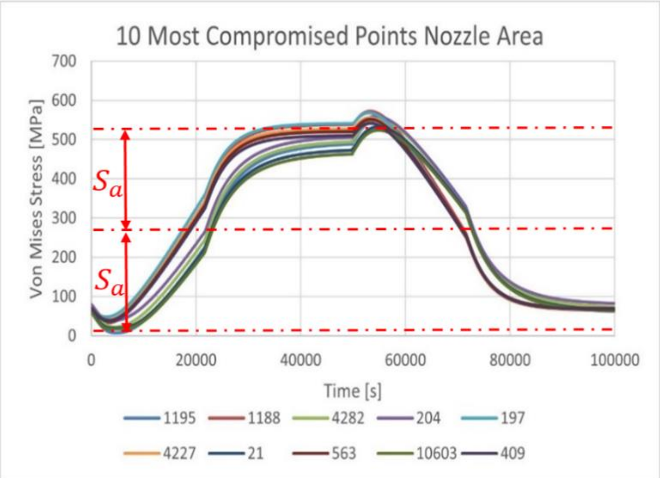



Figure 7. Von Mises stress plotted for the 10 most compromised nodes due to fatigue throughout the whole cycle. It is shown that 2 Sa = 540 MPa < 600 MPa.

## 7. Conclusions

These results show that a reactor pressure vessel can be designed indeed for a supercritical pressure of 27.8 MPa using existing nuclear standards. Despite the thicker walls, the pressure vessel can be heated up or cooled down within 6 hours each with an expected fatigue life of more than 1000 cycles.

Some points should be highlighted though. The first one is in connection with the material used to build the RPV body. Although the alloy selected is promising and in principle it has better properties than the materials already found in existing reactors, that can also be found in the KTA Standard 3201.1 [4], extensive material testing must be carried out in order to qualify the material up to 400°C.

Another key aspect is that the whole primary system should be designed as well to have more accurate data of the loadings that the RPV would receive from the piping. This, in turn, would allow to perform a more realistic simulation, with stresses closer to those ones that could be encountered in reality. This task could be adressed in a follow-up work.

One point that should also be made clear in this conclusion is that this study is just an exercise demonstrating how such a reactor pressure vessel can be designed in principle. The design shown here is not qualified for production.

Finally, it is also worth mentioning that, in order to have a better characterization of the thermal loads, it would be worthwhile to perform a CFD analysis of the model to determine the heat transfer coefficient. This, however, would require that the internals of the reactor pressure vessel are modelled as well.

## 8. References

[1] KTA, Components of the Reactor Coolant Pressure Boundary of Light Water Reactors Part 2: Design and Analysis, en, KTA, Salzgitter, Germany, Standard KTA 3201.2:2017-11, 2017.

[2] Schulenberg, T., Czifrus, S., Design concept of a super pressurized water reactor, 11th International Symposium on SCWRs (ISSCWR-11), P005, Pisa, Italy, February 3-5, 2025.

[3] Kim, M.C., Park, S.G., Lee, K.H., and Lee, B.S., Comparison of fracture properties in SA508 Gr.3 and Gr.4N high strength low alloy steels for advanced pressure vessel materials, International Journal of Pressure Vessels and Piping, vol. 131, pp. 60–66, 2015.

[4] KTA, Components of the Reactor Coolant Pressure Boundary of Light Water Reactors Part 1: Materials and Product Forms, en, KTA, Salzgitter, Germany, Standard KTA 3201.1:2017-11, 2017.

[5] Fernández, A.J., Design and analysis of a reactor pressure vessel for high temperature application, Master Thesis, Institute for Thermal Energy Technology and Safety, Karlsruhe Institute of Technology, 2024.

[6] Westinghouse Electric Company, AP1000 European Design Control Document EPS-GW-GL-700, Revision 1, 2009.

[7] Pilkey W.D., and Pilkey, D.F., Peterson's stress concentration factors. Wiley, 2007. Doi: 10.1002/9780470211106.

[8] AREVA NP, UK-EPR Fundamental Safety Overview, Vol. 1: Head Document, Chapter A: EPR Design Description, 2007.

[9] Technetics Group, Metal Seals Technical Catalogue, https://technetics.com/resource_library/metal-seals-master-catalog/, [Online; accessed 19-November-2023], 2023.

[10] VDI-Gesellschaft, VDI Heat Atlas. Springer, 2010.