Preliminary Design and Stress Analysis of the Transformational Challenge Reactor Vessel

S. Mohanty[a], A. Bergeron[a], A. Talamo [a], S.N.P. Vegendla[a], F. Heidet[a]
B. Ade[b], B.R. Betzler[b]

[a)]*Argonne National Laboratory, 9700 South Cass Avenue, Argonne, IL 60439, USA*
[b)]*Oak Ridge National Laboratory, P.O. Box 2008, Oak Ridge, TN 37831-6170, USA*
[1]*smohanty@anl.gov*

INTRODUCTION

The Transformational Challenge Reactor (TCR) program aims at designing, licensing, building and operating a small reactor by leveraging recent scientific achievements in advanced manufacturing, nuclear materials, machine learning, and computational modeling and simulation. These scientific and technological advances enable a paradigm shift in reactor design and deployment [1, 2].

The preliminary design of the TCR vessel and other pressure boundary components will follow the ASME code [3]. Both the subsection NB and NH of section III, Division -1 will be followed. The vessel will be made from austenitic stainless steel. Note that for austenitic stainless steel, the subsection NB rules are applicable for design temperature lower than or equal to 427 ºC. However, if the vessel wall temperature is beyond 427 ºC the subsection NH rules are applicable. The NB rule requires limits on load-controlled stresses (using time-independent allowable limits such as design stress intensity values) to be satisfied. Whereas, NH rules mandate to satisfy both load-controlled stresses (using time-independent/dependent allowable limits such as design stress intensity values) and limits on deformation-controlled quantities (e.g., allowable strains). In this paper, we present preliminary results related to vessel design by following NB design by rules and followed by stress analysis results aiming at improving the geometry of vessel and its support. The details are discussed below.

VESSEL THICKNESS ESTIMATION

The preliminary vessel thickness was estimated based on the ASME-NB-code design-by-rules [3]. For this study, austenitic stainless steel 316 (UNS No S31600, nominal composition: 16Cr–12Ni–2Mo) is considered as the vessel material. Figure 1 shows the allowable/design stress intensity of the assumed vessel material. A maximum design pressure of 5 MPa is considered. Minimum thickness of the vessel was estimated for different combination of vessel wall temperature and vessel inner radius. Figure 2 shows the design envelope of the minimum vessel shell thickness for different inner radius and temperature combinations. Figure 2 shows that the minimum vessel thickness required increases substantially when the vessel wall temperature is higher than 600 ºC. This is due to the sharp decrease of allowable stress beyond 600 ºC (see Figure 1). For the preliminary design, the TCR maximum operating pressure is 5 MPa and the vessel inner radius is 36.65 cm. In addition, the maximum vessel wall inlet and outlet temperature is assumed to be 300 ºC and 550 ºC, respectively. Considering a vessel wall temperature of 300 ºC, the minimum vessel thickness was estimated to be 1.573 cm. Considering a vessel wall (at outlet) temperature of 550 ºC, the minimum vessel thickness was estimated to be 1.7878 cm (refer Figure 2). However, an initial vessel thickness of 2.54 cm (1 inch) was considered for further analysis [4].

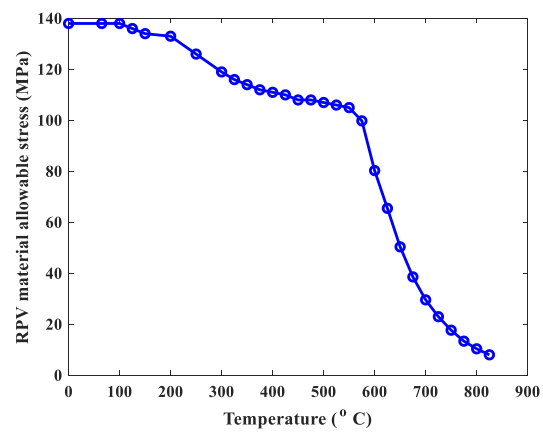

Figure 1. ASME code Section II, Part-D allowable/design stress intensity of 316-grade austenitic stainless steel [3]

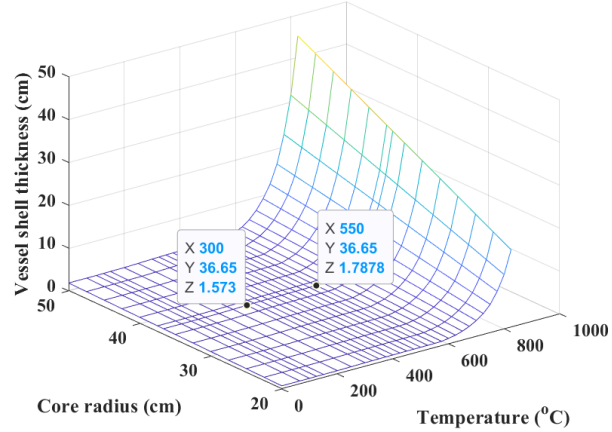

Figure 2. Design envelope of minimum vessel shell thickness for different core radius and temperature.

FINITE ELEMENT STRESS ANALYSIS

Using the 3D finite element (FE) software ABAQUS, a numerical model of the TCR vessel was created. This model was used to analyze the impact on stresses of different support conditions (e.g., with and without skirt support) and temperature and pressure transients. The analysis was also conducted with and without temperature transient to assess the effect of temperature on top of the pressure transient. 3D view of the model can be seen in Figure 3, which shows the FE mesh (OD and ID side) of the vessel with and without the skirt support.

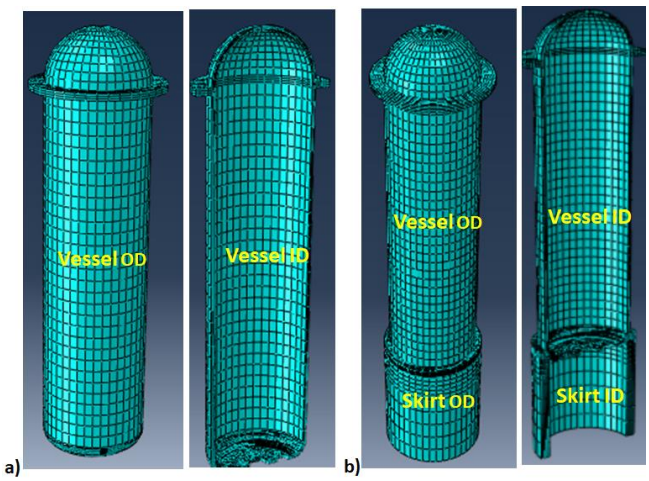

Figure 3. a) FE mesh (OD and ID side) of vessel without skirt support, b) FE mesh (OD and ID side) of vessel with skirt support.

FE Models

Different FE models were subsequently created to estimate the stress-strain in the vessel. Table-1 shows the six FE model cases considered. Case A-1 and A-2 were simulated assuming the vessel was supported at its bottom with all the displacements and rotations restricted/fixed. Whereas case B-1 through C-2 were simulated assuming the vessel was supported at the bottom of a skirt (refer Figure 3b). For models B1 to C2, all the displacements and rotations at the bottom of the skirt were restricted/fixed. As explained later in this paper, a skirt support was considered to push the high stress region from the bottom head of the vessel to the bottom of the skirt.

The six cases described in Table 1 were simulated with/without temperature transient in addition of a pressure transient. For these preliminary simulations, the temperature at a given time was assumed to be everywhere the same throughout the vessel and skirt. Figure 4 shows the applied pressure transient (at the ID surface of vessel only) with a maximum pressure of 5 MPa. Whereas, Figure 5 shows the applied temperature transient (at each node of vessel and skirt) with a maximum temperature of 300 °C. These transient scenarios are conceptual in nature.

Table 1. Different cases of FE model

Case	Loading conditions	Support conditions
A-1	Pressure only	Vessel wall and bottom head thickness = 2.54 cm (1inch), Fixed joint at the vessel bottom head.
A-2	Pressure & Temperature	
B-1	Pressure only	Vessel wall & bottom head thickness =2.54 cm (1inch), skirt thickness = 7.62 cm (3 inch), bottom of skirt is fixed.
B-2	Pressure & Temperature	
C-1	Pressure only	Vessel wall and bottom head thickness =2.54 cm (1inch) and 5.08 cm (2 inch), respectively, skirt thickness = 10.16 cm (4 inch), bottom of skirt is fixed.
C-2	Pressure & Temperature	

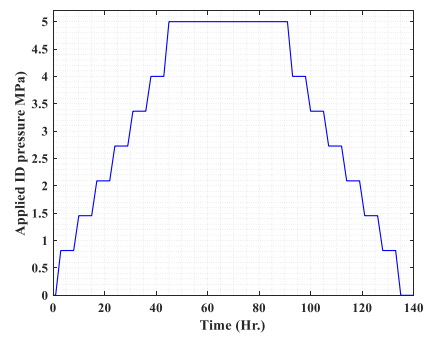

Figure 4. Applied pressure transient (at the ID surface of vessel only) with a maximum pressure of 5 MPa.

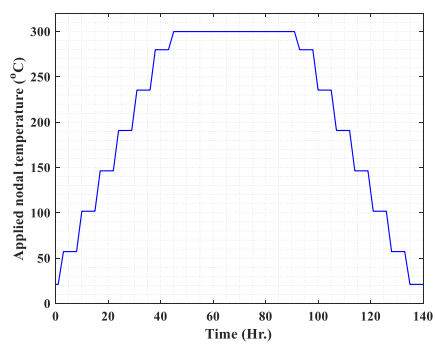

Figure 5. Applied temperature transient (at each node of vessel and skirt) with a maximum temperature of 300 °C.

A-1 & A-2 Model Results

To check the accuracy of the FE models, a thin-wall analytical model was developed and compared against the numerical one (FE model A-1, pressure loading only). The Von Mises and other stresses results are presented in Table 2. As can be seen in this table, the FE model agrees very well with the analytical one. This shows the adequacy of the FE models, particularly regarding the density of the mesh.

Figure 6 illustrates the Von Mises stresses at different vessel locations. Figure 7 shows the principal strain at the vessel bottom head. From Figure 6, under combined pressure and temperature, the vessel bottom-head support experiences a maximum Von Mises stress of 1648 MPa and a maximum mechanical strain of 0.51%. Although the mechanical strain is not high, the maximum stress is found to be substantially higher than the yield stress of stainless steel (i.e., 0.2% offset yield stress of 316SS at 300 ºC is roughly 156 MPa as reported in earlier work [5]).

These high stresses suggest that another type of vessel support is needed. A vessel support based on a skirt (see Figure 3) has been considered. The analytical results are discussed below.

Table 2. A-1 FE model versus analytical model results

Stress (MPa)	FE model (Case: A-1)	Analytical model
Hoop stress	72.0	72.1
Radial stress	-2.3	-2.5
Axial stress	34.7	34.9
Von Mises stress	64.5	64.6

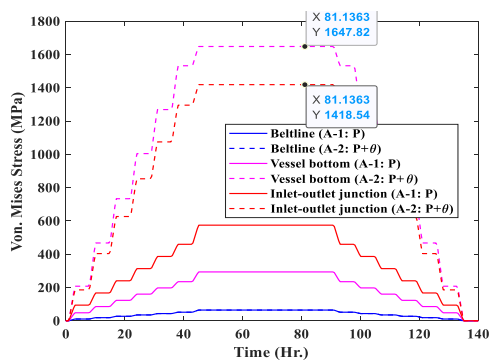

Figure 6. Comparison of Von Mises stress at different locations from A-1 and A-2 FE models.

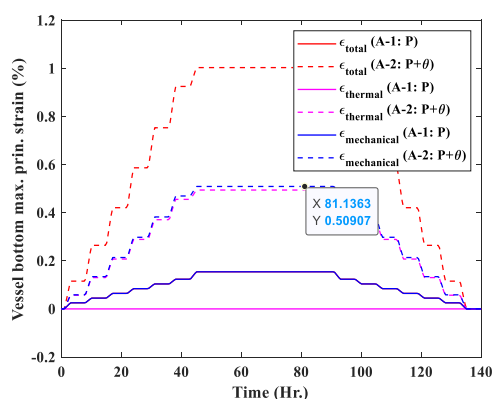

Figure 7. Comparison of various principal strain at vessel bottom head from A-1 and A-2 FE models.

B-1 & B-2 Model Results

The B-1 and B-2 FE models were simulated considering a skirt support, which was attached to the vessel bottom head (refer Figure 3b). The purpose of the skirt support is to push away the stress hotspot regions from the vessel to the support structure. From Figure 8, under combined pressure and temperature, the vessel bottom-head and inlet-outlet junction (i.e., the region of the vessel that connect to the outlet pipe) experiences a maximum Von Mises stress of 292.5 MPa and 580 MPa, respectively. The inlet-outlet junction still exhibits significant stresses, which is likely due to the presence of sharp edges. Geometric optimization (i.e., appropriate curvature and nozzle design) can certainly reduce substantially the stresses in this area. The vessel bottom head shows a maximum Von Mises stress of 292.5 MPa, which is considerably less compared than the 1648 MPa obtained in the case of bottom head supported model (A-2). This result illustrates the efficiency of the skirt support. Figures 9 and 10 show the comparison of the principal strain at the vessel bottom-head and at the bottom of the skirt, respectively. From these figures, it can be seen that the maximum mechanical strains are 0.17% and 0.35%, respectively. These strains are considerably lower than required by the ASME code (maximum cumulative strain requirement of 1% for base metal and 0.5% for weld metal). However, it is important to note that all the simulation discussed in this paper are based on elastic FE models. More realistic estimation of strain would require elastic-plastic and creep (if any for higher temperature) stress analysis.

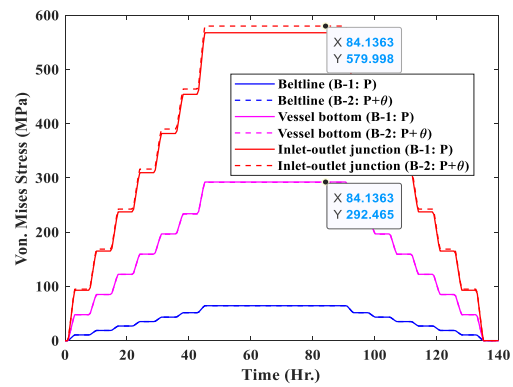

Figure 8. Comparison of Von Mises stress at different locations from B-1 and B-2 FE models.

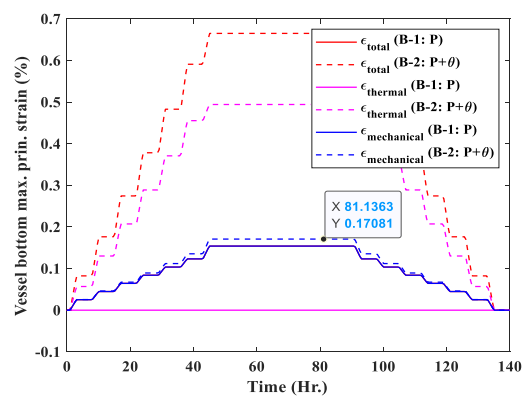

Figure 9. Comparison of various principal strain at vessel bottom head from B1-1 and B2-2 FE models.

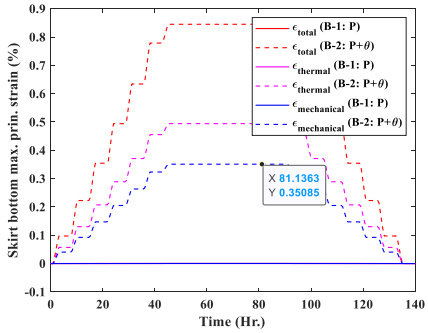

Figure 10. Comparison of various principal strain at bottom of skirt head from B1-1 and B2-2 FE models.

C-1 & C-2 Model Results

To reduce the stress further at the vessel bottom head, the thickness of the bottom head was increased from 2.54 cm (model B-1 and B-2) to 5.08 cm (model C-1 and C-2). Whereas, the thickness of the skirt was increased from 7.62 (model B-1 and B-2) to 10.16 cm (model C-1 and C-2). Figures 11 to 13 show the corresponding model results. Figure 11 shows the comparison of Von Mises stresses at different locations. From the figure, it can be seen that the vessel bottom stresses are reduced to 109 MPa (from 292.5 MPa for the case B-2 and from 1648 MPa for the case A-2). Whereas, in the inlet-outlet junction region, the stresses are substantially reduced to 192 MPa (from 580 MPa for case B-2 and from 1419 MPa for case A-2). Figures 12 and 13 shows the comparison of various principal strain at the vessel bottom-head and at the bottom of the skirt respectively. From these figures, it can be seen that the respective maximum mechanical strains are 0.06% and 0.43%, well below the limit (maximum cumulative strain requirement of 1% for base metal and 0.5% for weld metal).

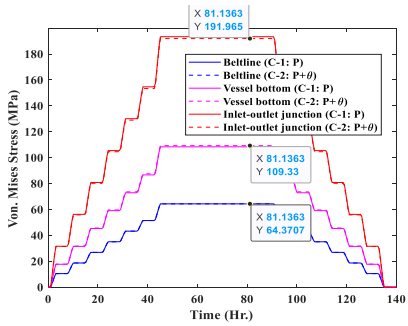

Figure 11. Comparison of Mises stress at different locations from C-1 and C-2 FE models.

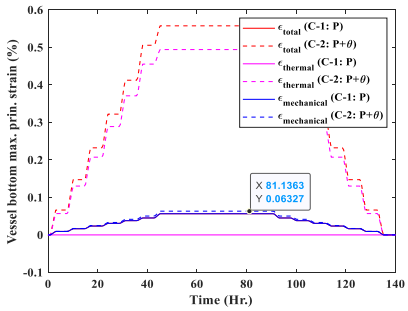

Figure 12. Comparison of various principal strain at vessel bottom head from C1-1 and C2-2 FE models.

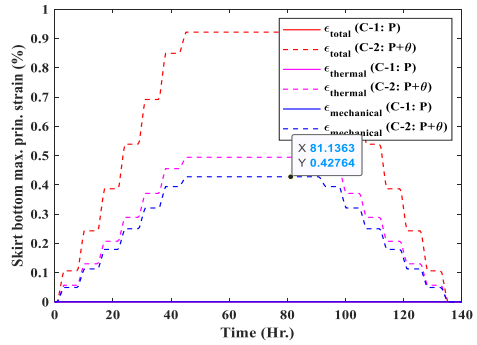

Figure 13. Comparison of various principal strain at bottom of skirt head from C1-1 and C2-2 FE models.

CONCLUSIONS

In this paper, analytical and FE model results in support of the preliminary design of the TCR vessel are discussed. The results show that an extended skirt support (from the bottom head of vessel) can substantially reduce the stress of the main vessel structure. The skirt-based design helps restricting the thickness of the vessel wall to 2.54 cm (one inch).

ACKNOWLEDGMENTS

The Transformational Challenge Reactor program is supported by the U.S. Department of Energy Office of Nuclear Energy (DOE-NE).

The submitted manuscript has been created by UChicago Argonne, LLC, Operator of Argonne National Laboratory ("Argonne"). Argonne National Laboratory's work was supported by the U.S. Department of Energy, Office of Nuclear Energy under contract DE-AC02-06CH11357.

REFERENCES

1. J. BUSBY, M. HACKETT, L. LOVE, S. BABU, K. TOBIN, R. CAO, D. POINTER, A. WHARTON, L. QUALLS, B. BETZLER, "Technologies to Reactors: Enabling Accelerated Deployment of Nuclear Energy Systems," ORNL/SPR-2018/1025, Oak Ridge National Laboratory (2018).

2. K.A. TERRANI, "Transformational Challenge Reactor Demonstration Program," (2019). Retrieved from http://tcr.ornl.gov.

3. ASME Pressure Vessel and Piping Code: Section-II, and III, (July 2017).

4. J. R. BURNS, B. R BETZLER, et al., "Control Element Design for the Transformational Challenge Reactor," *Trans. Am. Nucl. Soc.*, **122** (2020).

5. MOHANTY, S., LISTWAN, J., and PARK, J.P., "A System-Level Framework for Fatigue Life Prediction of a PWR Pressurizer-Surge-Line Nozzle under Design-basis Loading Cycles," Report No. ANL/LWRS-19/01. Argonne National Laboratory, September 2019. (https://www.osti.gov/biblio/1571258)