Preliminary Frequency Analysis of the Transformational Challenge Reactor Vessel

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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]. Figure 1 shows the schematic of the overall TCR layout.

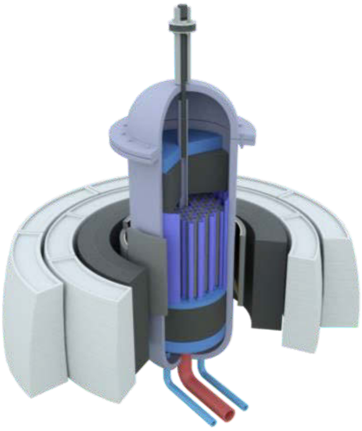

Figure 1. TCR layout

The preliminary design of the TCR vessel and other pressure boundary components follow the ASME code [3]. For the TCR vessel structural integrity evaluation, it is required to perform the stress analysis subjected to temperature-pressure transients under normal-operating-conditions, as well as under severe accident conditions such as loss of coolant accident. In addition, it is also required to assess the structural integrity of TCR pressure-boundary components under earthquake loading. For the stress analysis under earthquake condition it is first required to conduct the frequency analysis. In last ANS winter meeting [4] we presented preliminary stress analysis results of a vessel-skirt assembly under normal-operating-condition temperature-pressure transients. In this paper, we present preliminary frequency or modal analysis results for a vessel-skirt configuration. Modal analysis of the TCR structural system is required to ascertain the dynamic behavior of the structural system under dynamic loading such as earthquake and/or forced vibration loading such as from pumps and fluid-induced vibration sources.

FINITE ELEMENT MODEL

A preliminary Finite element (FE) model was developed to estimate the modal frequency and corresponding mode shapes under different temperature conditions. The FE model was developed by considering a Skirt-Vessel configuration and using the commercially available ABAQUS software. The CAD model and the corresponding FE mesh is shown in Figure 2. The model was meshed such a way to achieve moderately finer mesh across the thickness and at the bottom of the vessel (refer Figure 2). For the discussed results, austenitic stainless steel 316 (UNS No S31600, nominal composition: 16Cr–12Ni–2Mo) is considered as the vessel and skirt material. The related material properties taken from the ASME code. For the discussed FE model, we considered a thickness of vessel wall = 1 in, thickness of bottom-head of vessel = 2 in, Thickness skirt wall = 4 in. The justification of these dimension is based on TCR vessel design related previous works and can be referred from [4, 5]. For all the simulation cases discussed in this paper, the bottom of the skirt was restrained/anchored in all directions (both rotation and translations) to the floor. The Eight-node linear brick element (C3D8) was used for FE meshing the entire skirt-vessel assembly. The reported frequency analyses were performed with/without considering the weight of internals. To be noted that in this preliminary frequency analysis models the details geometry of the core-internals and their position with respect to vessels were not explicitly modeled. Rather the weight of the internals indirectly modeled by artificially increasing the density of the steel by a certain factor. The factor is equal to the ratio of the estimated weight of vessel, skirt, and internals to the weight of vessel and skirt. Frequency analysis were also performed considering zero-stress condition and maximum-stressed condition (e.g. at full-power operation condition). For the zero-stress frequency analysis only a single step method was followed with only performing the linear perturbation frequency analysis. Whereas, for frequency analysis under maximum-stressed condition a two-step procedure was followed. This is by performing first a static stress analysis and then performing the linear perturbation frequency analysis.

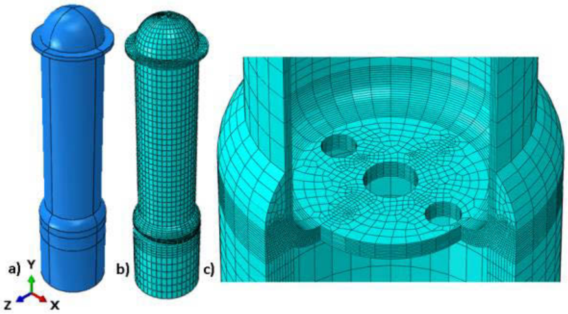

Figure 2. a) CAD model, b) FE mesh of the full model, and c) Magnified (cut-section) mesh at the bottom head.

FREQUENCY ANALYSIS RESULTS

The above-mentioned FE models was simulated for different temperature conditions, pertinent to TCR operating combined heat-up-full-power-operation-cool-down temperature regime. We also included results those consider the effect of core internal load and full-power-operation stressed conditions. We assume that earthquake and other dynamic loading can happen at any temperature during the heat-up-full-power-operation-cool-down temperature regime. The estimated frequency can be used for designing simulated earthquake loading for detail analysis and/or doing necessary design change to push the lower modal frequency away from the earthquake frequency regime. For example, the Tohoku-Oki (in Japan) earthquake, that was happened in 11 March 2011 [6] has a magnitude of 9.0 on the Richter scale. This earthquake generated loading spectra with a maximum highest frequency of 40Hz. Ideally, if the reactor needs to be built in a place like Tohoku-Oki, the lowest modal frequency of the reactor assembly needs to be above the 40Hz earthquake frequency to avoid any frequency-resonance related exponential/unstable dynamic stress. To note, frequency-resonance is the phenomenon of increased amplitude that occurs when the frequency of an earthquake ground acceleration become equal or close to a natural frequency of the structural system on which it acts and on when it acts. For designing against earthquake, one needs to look at location/site specific earthquake data. The proposed modal analysis is to provide a first estimation of the modal frequency of the TCR skirt-vessel assembly under different operating temperature. The frequency analysis results for different conditions are presented below:

Set-1 Frequency analysis results without considering the core internal load and thermal-mechanical stress: For the first set of ABAQUS based frequency analysis the core internal load and thermal-mechanical stress were not considered. However, the frequency analyses were performed considering different isothermal temperature conditions. This is to check how the natural frequency of the skirt-vessel assembly changes with respect to different temperature. Many temperature data points were considered to ascertain the trend. Table 1 shows the frequencies of first 5 modes of vessel-skirt assembly FE model at different temperature. Figures 3 shows the first to fifth mode shapes of vessel-skirt assembly. Note that the depicted mode shapes are normalized displacements of the vessel-skirt assembly if that assembly dynamically subjected due to a forcing condition with frequency equal to the corresponding modal frequency. For example, if the vessel-skirt assembly subjected to an earthquake load with frequency equal to the first mode frequency (at that temperature), the assembly would oscillate in a cantilever mode (refer Figure 3a) with vessel top experiencing the highest horizontal displacement. From the Figures 3a-e, it can be seen that, mode-1 and 2 are two cantilever-bending modes, mode-3 is the rotational mode (around y-axis or the axial-direction of vessel), mode-4 and 5 are the bending-rotation-axial-warping mixed higher modes. Also, from the results shown in Table-1, there is a large gap between mode-2 and mode-3 frequencies. This shows that we should only concern about first two modes since there will be higher stiffness for mode-3 and beyond vibrations. Also, these higher mode frequencies (from mode 3) are irrelevant since it's unlikely to happen an earthquake at a frequency of 50 Hz or higher and at the same time with high amplitude. Also, from the results, 1st and 2nd modes are very similar (with respect to modal frequency and mode shapes). This is due to near symmetry of the two lateral planes (x-y and z-y planes assuming y-direction is along the axial direction of vessel and vessel cross-section is parallel to the x-z plane). Note that at the bottom head of vessel there is a central opening for outlet and two additional inline openings for two inlets (refer Figure 2c) leading to only one-plane of prefect symmetry. Although, there is one-plane of perfect symmetry in the present discussed geometry, rather than considering a half-symmetry-model we have considered the full model for frequency analysis. This is because in future we will add further unsymmetrical features (e.g. piping, cutouts in the skirt, etc.) and it will be easy to include those features in the discussed base model. Nevertheless, new frequency analysis is required to ascertain the change in geometry effect associated with the addition of geometrical features). However, the discussed results are firsthand results and conceptual in nature.

Table 1. First 5 mode frequencies with respect to different temperatures (without considering core-internal load).

T (oC)	Mode-1 (Cycles/ time)	Mode-2 (Cycles/ time)	Mode-3 (Cycles/ time)	Mode-4 (Cycles/ time)	Mode-5 (Cycles/ time)
21.2	56.659	56.662	220.2	229.96	229.97
50	56.338	56.341	218.960	228.650	228.670
100	55.793	55.795	216.840	226.440	226.450
150	55.331	55.333	215.040	224.560	224.580
200	54.774	54.776	212.880	222.300	222.320
250	54.297	54.300	211.020	220.370	220.380
300	53.750	53.752	208.900	218.150	218.160

350	53.243	53.246	206.930	216.090	216.110
400	52.649	52.652	204.620	213.680	213.700
450	52.012	52.014	202.140	211.100	211.110
500	**51.385**	51.387	199.71	208.55	208.56

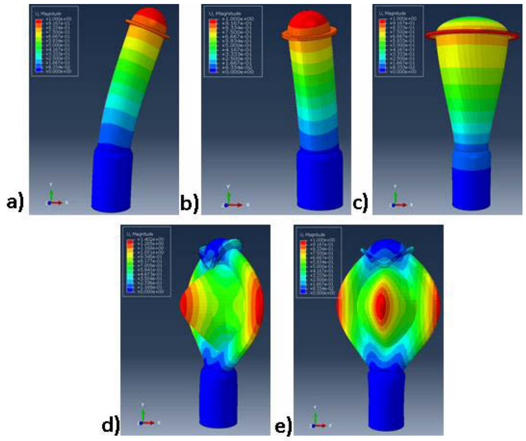

Figure 3. First five mode shape of vessel-skirt assembly with a) 1st, b) 2nd, c) 3rd, d) 4th and e) 5th mode shapes.

Set-2 Frequency analysis results with considering the core internal load but without considering the thermal-mechanical stress: Similar as the first set of frequency analysis cases, we also performed different temperature-dependent frequency analyses but assuming the effect of weight associated with core internals. The geometries of the core-internals are not explicitly included in the discussed preliminary or base model, rather their dynamic/mass effect included by artificially modifying the density of steel (refer FE model section for details). Table 2 shows the corresponding frequencies of first 5 modes of vessel-skirt assembly FE model at different temperatures.

Table 2. First 5 mode frequencies with respect to different temperatures (with considering core-internal load).

T (oC)	Mode-1 (Cycles/ time)	Mode-2 (Cycles/ time)	Mode-3 (Cycles/ time)	Mode-4 (Cycles/ time)	Mode-5 (Cycles/ time)
21.11	47.117	47.119	183.120	191.230	191.240
50	46.850	46.852	182.080	190.140	190.160
100	46.396	46.398	180.320	188.300	188.310
150	46.012	46.014	178.820	186.740	186.760
200	45.549	45.551	177.020	184.860	184.880
250	45.153	45.154	175.480	183.250	183.270
300	44.697	44.699	173.710	181.410	181.420
350	44.276	44.278	172.080	179.700	179.710
400	43.782	43.784	170.160	177.690	177.700
450	43.252	43.254	168.100	175.540	175.550
500	**42.731**	42.733	166.070	173.430	173.440

Figure 4 shows the temperature dependency of 1st mode frequency for both with/without considering the core-internal loads. For both the cases, this Figure shows that modal frequencies are broadly linearly dependent with respect to operating temperatures. The linearity is due to linear-dependency of square root of elastic modules with temperatures (within the discussed temperature range of 20 - 500 °C). Figure 5 shows the corresponding temperature versus square root of elastic modulus of 316 stainless steel. Nevertheless, higher the modal frequency (for mode 3 and beyond), it has minimal effect on the structural integrity since it is rare an earthquake can have a frequency higher than 50 Hz and at the same time with high amplitude of ground vibrations. From Tables 1 and 2, the lowest modal frequency at 500 °C is of 42.731 Hz (this is considering the core-internal loads), which is above the 40 Hz frequency observed at Tohoku-Oki (in Japan). Note that an earthquake of this amplitude is unlikely to happen at ORNL, the likely site of the TCR. However, a detailed ORNL site-specific analysis with more detailed thermal-mechanical boundary conditions are required to ascertain the effect of earthquake on the structural integrity of TCR vessel and other safety critical systems.

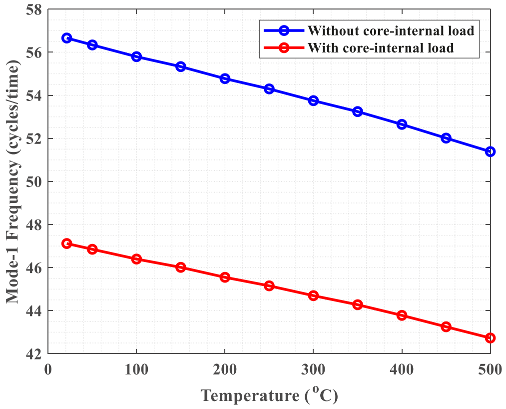

Figure 4. Temperature dependency of first mode frequency for the vessel-skirt configuration

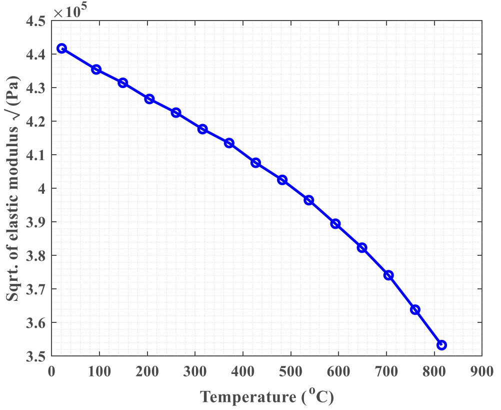

Figure 5. Temperature dependency of square root of elastic modulus of 316 stainless steel.

Set-3 Frequency analysis results considering thermal-mechanical stress: The effect of thermal-mechanical stress on frequency analysis results are also assessed. We have already presented preliminary stress analysis results in the ANS winter 2020 conference [4]. The earlier stress analysis was performed under a full loading cycle comprising of heat-up, full-power-operation and cool-down conditions. The corresponding applied pressure and temperature transients can be found from the earlier publication [4]. However, in the present paper we only considered half-of the loading cycle that includes only the heat-up and part of full-power transients. This is to simulate the maximum stressed condition (at full-power operation). A two-step analysis procedure was followed. First a statis stress analysis was performed to simulate the full-power stress condition in the skirt-vessel assembly. Then a linear-perturbation frequency analysis was performed to estimate the modal frequency at that state of the skirt-vessel assembly. Figure 6 shows the simulated time versus Von Mises stress (at a typical element at the bottom head of the vessel). The two-step simulation procedure includes the effect of core-internal load. Table 3 shows the corresponding estimated frequency analysis results. Comparing the results of Table 3 with Table 2, we can find that the estimated stressed condition frequencies are exactly same as the frequency at 300 ^{o}C (of Table 2). This is obvious, because at the full-power condition the vessel-condition is subjected to a temperature of 300 ^{o}C. Also, we can find the stress has no effect on frequency. This is due to the fact that natural frequency, which is the inherent properties of a component doesn't change due to stress unless there is a significant plastic or permanent deformation. Note that in the discussed model we followed linear elastic stress analysis procedure (accordance with ASME NB code requirements). Under elastic analysis the stiffness matrix doesn't change and hence the frequency associated with the corresponding stressed conditions. Even though, we do not expect much effect associated with stressed conditions, detailed elastic-plastic analysis (following ASME NH code procedure) is required to ascertain the above-mentioned observations. Also detailed time-frequency analysis [7] required to ascertain the detrimental effect of earthquake. Those advanced studies are out of scope of this paper.

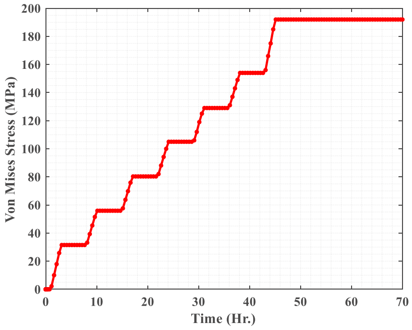

Figure 6. Time versus simulated Von Mises stress (at a typical element at the bottom head of the vessel) with considering core-internal load.

Table 3. First five mode frequencies of vessel-skirt assembly (with considering core-internal load) at full-power-operation stressed condition.

Mode-1 (Cycles/ time)	Mode-2 (Cycles/ time)	Mode-3 (Cycles/ time)	Mode-4 (Cycles/ time)	Mode-5 (Cycles/ time)
44.697	44.699	173.71	181.410	181.420

CONCLUSIONS

In this paper, we present preliminary frequency or modal analysis results in support of TCR program. We estimated the modal frequency of a vessel-skirt configuration at different temperature. From the FE model results, we estimated a lowest modal frequency of 51.385 Hz (without considering load of core-internals) and 42.731 Hz (with considering load of core-internals) in the operating temperature range of 20 - 500 ^{o}C. The discussed results are firsthand results and conceptual in nature. Further detailed ORNL site-specific analysis with more detailed thermal-mechanical boundary conditions is required to ascertain the effect of earthquake on the structural integrity TCR vessel and other safety critical system.

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