Flow-Induced Vibration Modeling

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Abstract*: The current work pertains to a re-closeable annular flow valve oil completion tool (X-AFV) with high gas flow passing through the valve, causing pressure oscillation due to flow separation and high turbulence energy. The objective is to assess the vibration level to protect the tool from damage. The first step was to perform CFD simulation by using Fluent on The valve port region. The power spectral density (PSD) of pressure fluctuations at the high turbulence region was obtained. The next step of the work was to extract natural frequencies of the tool by using Abaqus. The last step was to utilize random vibration analysis (RVA) to obtain the PSD and root mean square of stresses to assess the random vibration effect by using CFD predicted PSD of excitation pressure.*

Keywords*: X-AFV Tool, Flow- Induced Vibration, CFD, Power Spectral Density (PSD), FEA, Random Vibration Analysis (RVA)*

1. Introduction

The modeling of flow-induced vibration is important to understand the structure-fluid interactions, maintain the safety of the equipment and promote productivity.

Figure 1 shows a re-closeable annular flow valve (X-AFV), which is an oil completion tool. Concern about flow-induced vibration was raised when high gas flow passing through the valve caused pressure oscillation because of flow separation and high turbulence energy.

The random nature of the X-AFV tool vibration makes it almost impossible to perform the direct transient-nonlinear dynamic FEA analysis. One effective method for analyzing random vibration effect is random response analysis in frequency domain, which is statistically analyzed. (Izuchi, 2013), (Khan, 2011) and (Guo, 2011) are examples in this area.

Instead of using experimental data, the current authors use computational fluid dynamics (CFD) to predict excitation pressure. The first step was performing CFD simulation using Fluent. The valve port region was identified as the primary vibration source, therefore, CFD was only conducted on this part of the tool. Turbulent transient flow was modelled with the detached eddy simulation (DES) turbulence model. The fast Fourier transform (FFT) was performed to yield the power spectral density (PSD) of pressure fluctuations at the high turbulence region.

The next step of the work was to extract the natural frequencies of the tool using Abaqus. The frequency range extracted was on the high value range of the PSD of the pressure fluctuation. High PSD value represents high excitation force.

The last step was to utilize RVA to obtain the PSD and root mean square of stresses to assess the random vibration effect by using PSD of excitation pressure obtained from CFD.

All simulations were done on a Dell Precision 7600 work station, which has two Intel Xeon CPU E5-2687 @ 3.10GHz with eight cores.

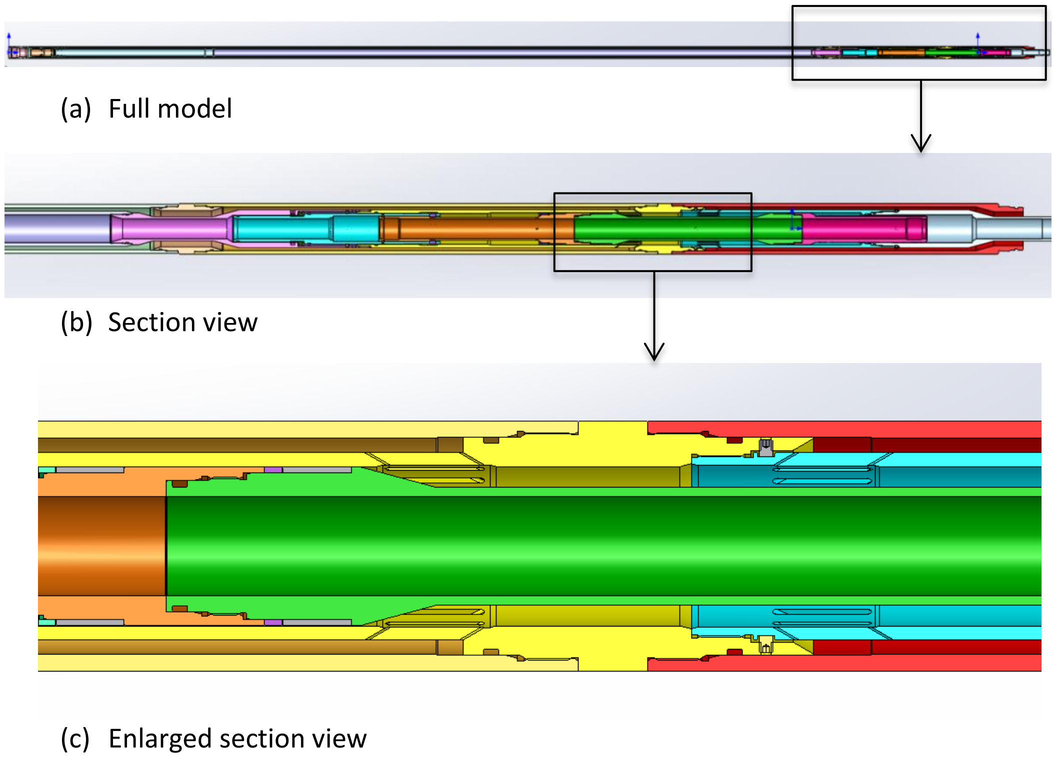


Figure 1. X-AFV.

2. CFD turbulence flow modeling

A gas flow rate of 160 MMscf/day flowed through the X-AFV tool, causing pressure oscillation from flow separation and high turbulence energy. The objective was to assess the stresses caused to the tool by flow-induced vibration of the X-AFV. Theoretically, this is a fluid-structure, two-way interaction modeling case. However, the computational cost makes impractical to obtain the instantaneous turbulent pressure field using CFD for the whole tool, even without consideration of the deflection of the structure. Thus, it is virtually impossible to conduct FSI two-way modeling. A practical approach is to adopt RVA to assess the random vibration effect. The valve port region, which generates the highest level of turbulence, was identified as the primary vibration source. CFD was only conducted on this part of the tool using Fluent. Turbulent transient flow was modelled with the detached eddy simulation (DES) turbulence model, which gives pressure fluctuation at the high turbulence region. A fast Fourier transform (FFT) was performed to yield the power spectral density (PSD) of this pressure fluctuation.

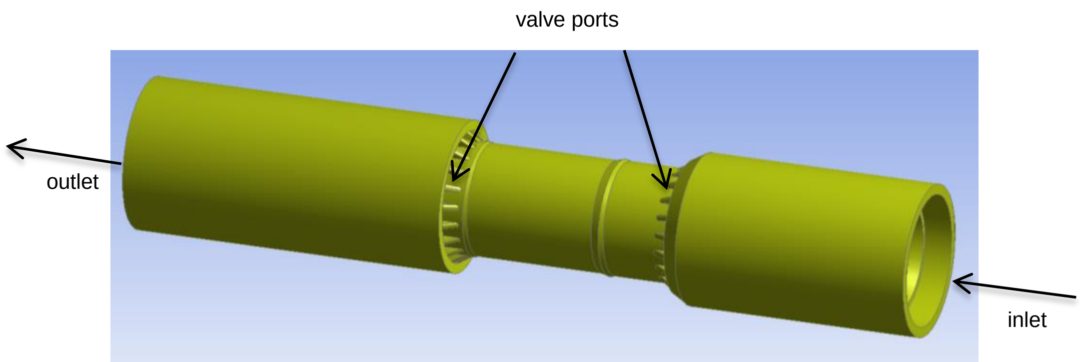


Figure 2. CFD model.

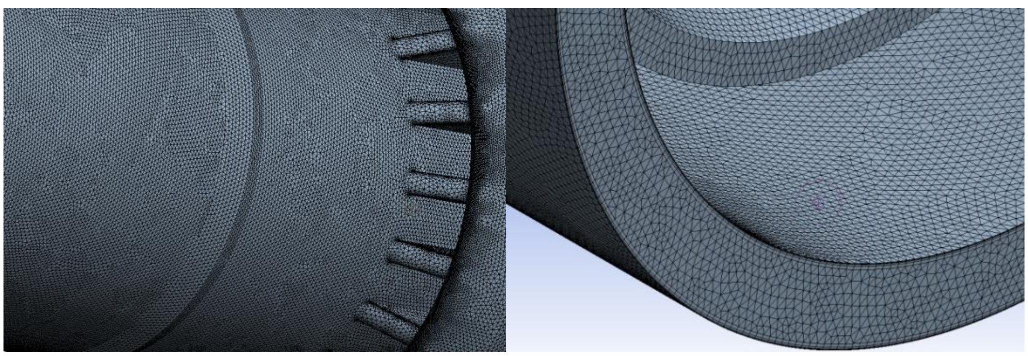


Figure 3. CFD mesh.

Figure 2 shows the CFD model. The valve modeled has 20 ports. A pressure inlet boundary condition was used at the inlet, with a total pressure of 4,400 psi specified. A fluctuating velocity algorithm, the spectral synthesizer, was adopted as perturbation of turbulent transient flow. A pressure outlet boundary condition was used at the outlet, with targeted flow rate of 160 MMscf/day. The fluid was assumed to be idea gas, with molecular weight of 28.293 kg/kgmol and viscosity of 0.031 cP.

Tetrahedral mesh, shown in **Figure 3**, was first generated in ANSYS Mesh for CFD application. There are 7,576,810 nodes and 23,231,802 elements, with refined mesh in the boundary layer region to meet the turbulence model requirement. To reduce memory usage, the mesh was then converted into polyhedral type after being read into Fluent solver. The computing time for DES model on the workstation is about five days (16 cores used).

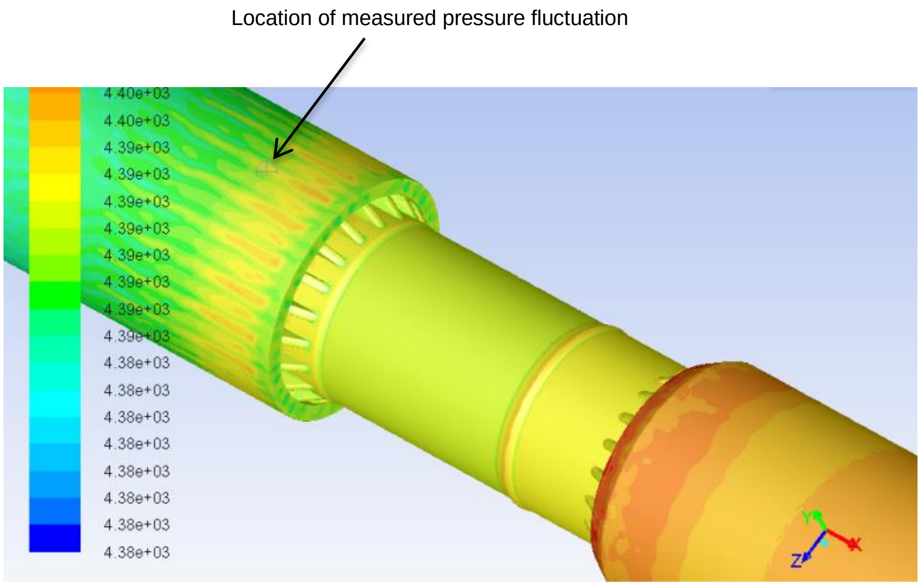


Figure 4. Pressure contours.

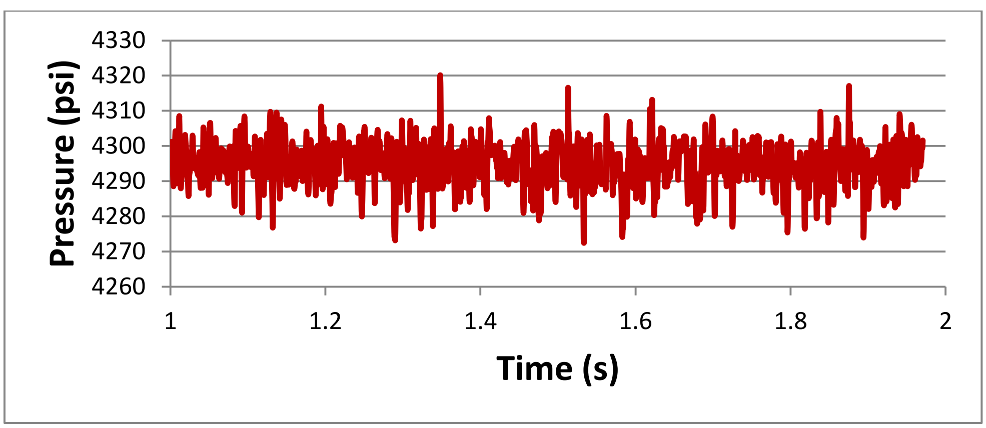


Figure 5. Time history of pressure fluctuation.

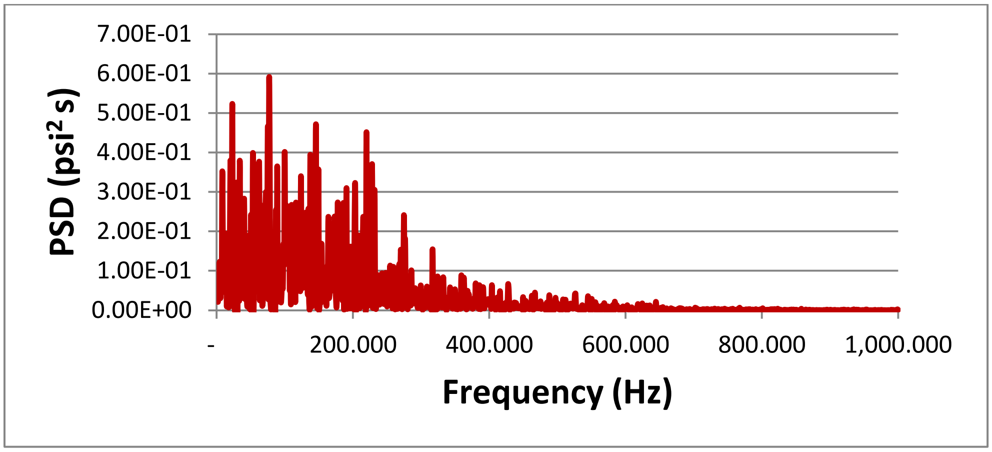


Figure 6. PSD of pressure fluctuation.

The high fluctuation of pressure occurs at the area of high turbulence. The location of measured pressure is determined by maximum turbulent kinetic energy, which is predicted using the RANS turbulence model (**Figure 4**). The pressure fluctuation was measured at the location using the DES model for a gas flow rate of 160 MMscf/d (**Figure 5**). **Figure 5** shows the random nature of the excitation pressure.

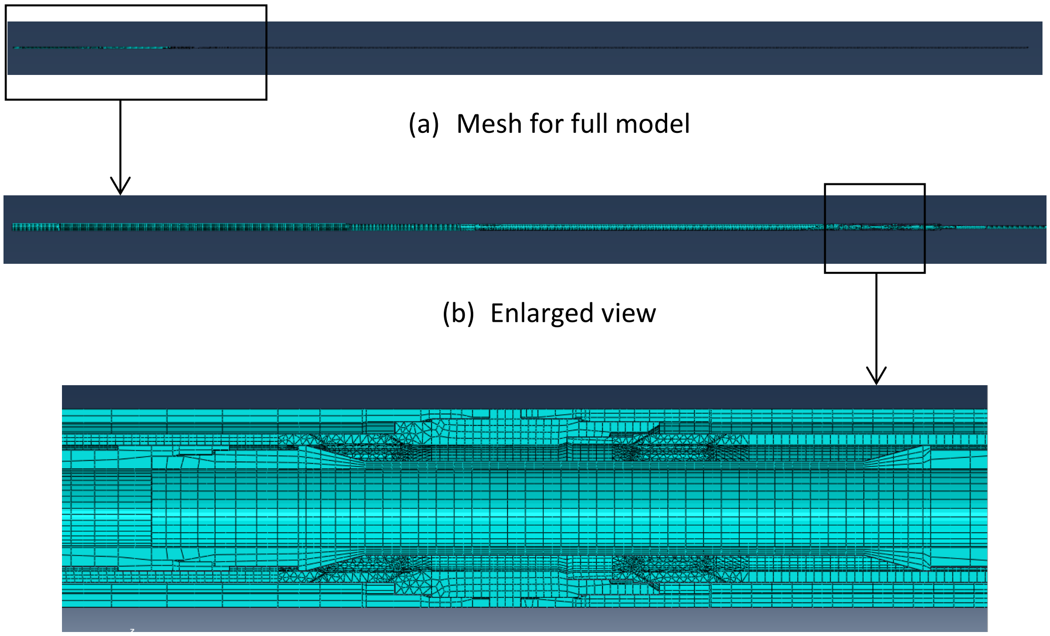
Figure 6 shows the PSD of the calculated pressure fluctuation, which can be obtained by FFT. The power spectral density represents the excitation power of pressures at various frequencies.

3. FEA random vibration analysis

Abaqus 6.14-1 was used to perform the random vibration analysis. Gas in the annulus flowed from right to left. The turbulent excitation force exists in the valve port area after flow exits the port, which excites vibration of the parts to the left. The left-most tubing in the model was used to represent the packer element. The objective is to assess if the vibration can damage the packer.

The full model was modeled by FEA. **Figure 7** shows 186,486 elements (including 124,535 C3D20R hex mesh and 61,951 C3D10 tet mesh) created for the full FEA model. The mesh size effect was studied and found that mesh resolution was not critical because of the linear feature of the FEA model.

The tool is fixed at the packer (left end) and fixed radially but not axially at the right end.



(a) Mesh for full model

(b) Enlarged view

(c) Enlarged section view

Figure 7. FEA mesh for full model.

The first step of the random vibration analysis is to extract natural frequencies. The first 400 natural frequencies for the full model were extracted, and were in the range of 0.02447 to 255.09Hz, which comprises the frequency range of high excitation power (**Figure 6**). More natural frequencies extracted significantly increase RVA computing time. The current RVA modeling on the workstation took about 10 days to complete.

RVA was performed using pressure PSD data obtained from CFD. Response was obtained up to 500 Hz to reduce computing cost. Response over higher frequency is significantly small.

Element Set 6 with 8 nodes on the modeled packer, shown in **Figure 8**, was picked for measuring the response PSD and the root mean square (RMS) of stress components.

FEA results are shown in **Figure 9** and **10**. **Figure 9** is the PSDs of stresses, showing how the structure is responds to excitation force at various frequencies. High peaks of the PSDs represent resonance at certain frequencies. However, the effect is small as the excitation power is low. **Figure 10** is the RMS stresses integration over frequency. The total RMS stresses are the integration over frequency from 0 Hz to ∞ Hz (0 Hz to 500 Hz is used in this simulation); the vibrational response is small.

Figure 11 shows the RMS value of the equivalent stress (1 σ stress). The value of the stress represents the dynamic vibrational magnitude of the stress excited by the excitation force. The full stress is the combination of the static stress under averaged pressure (not performed in this simulation) and the dynamic vibrational stress. The maximum vibrational stress is 0.5875 psi. Though no fatigue analysis was attempted, the risk of fatigue failure of the packer is negligible because the vibrational response is very small.

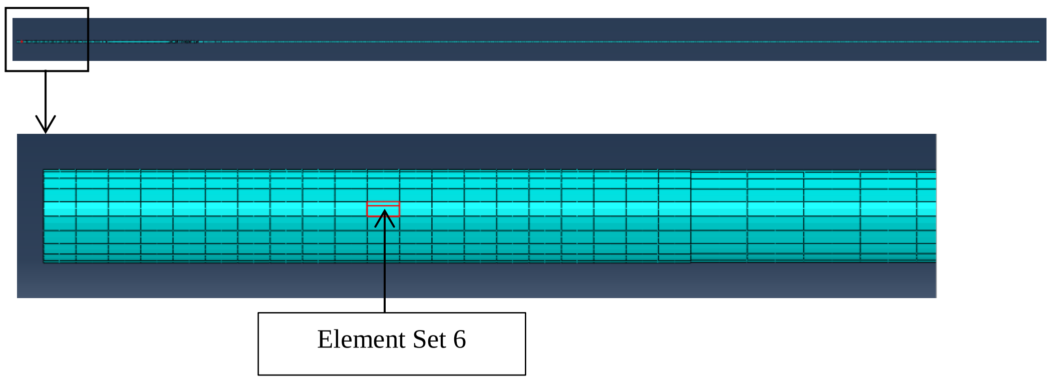


Figure 8. Element Set 6.

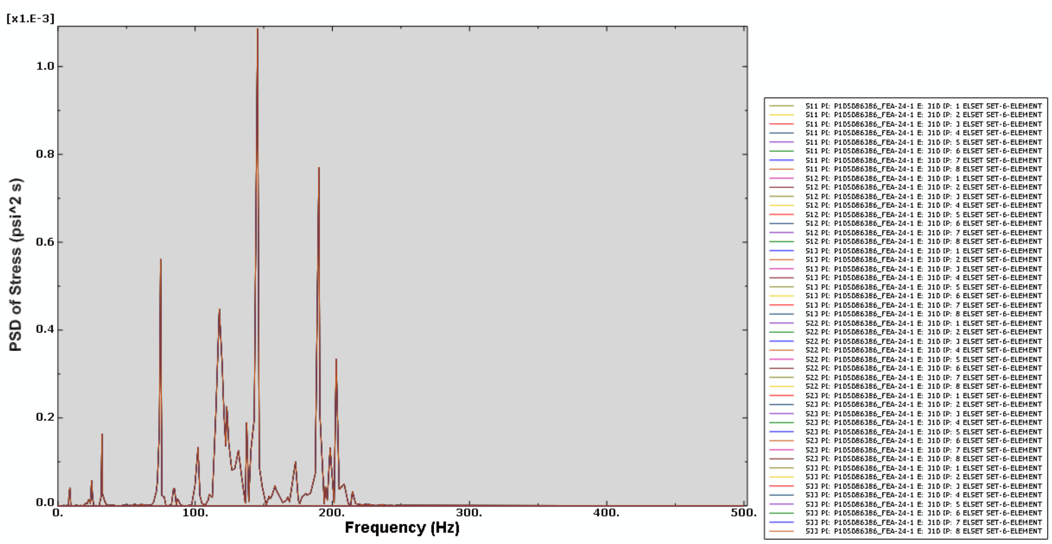


Figure 9. PSD of stresses at Element Set 6.

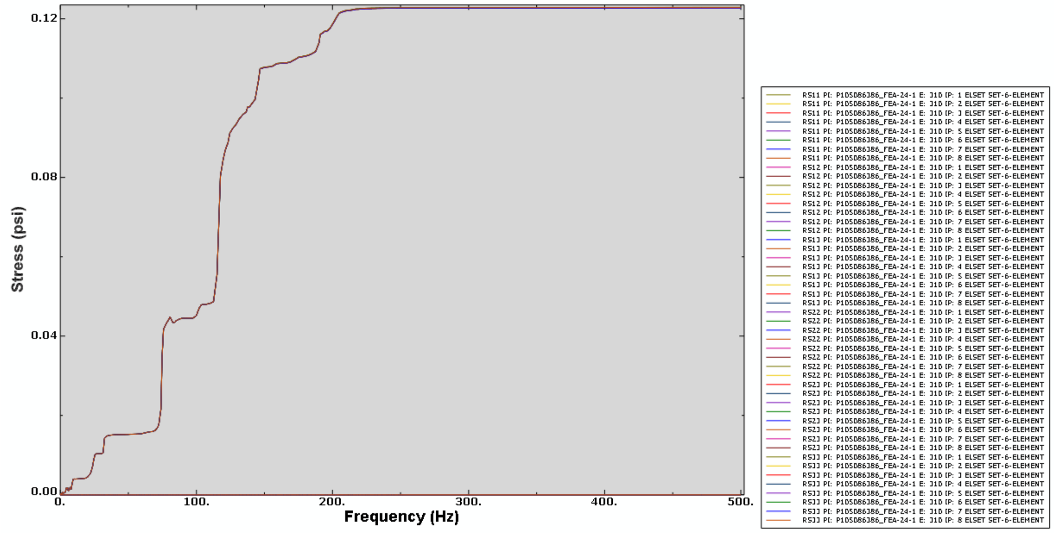


Figure 10. RMS stresses at Element Set 6.

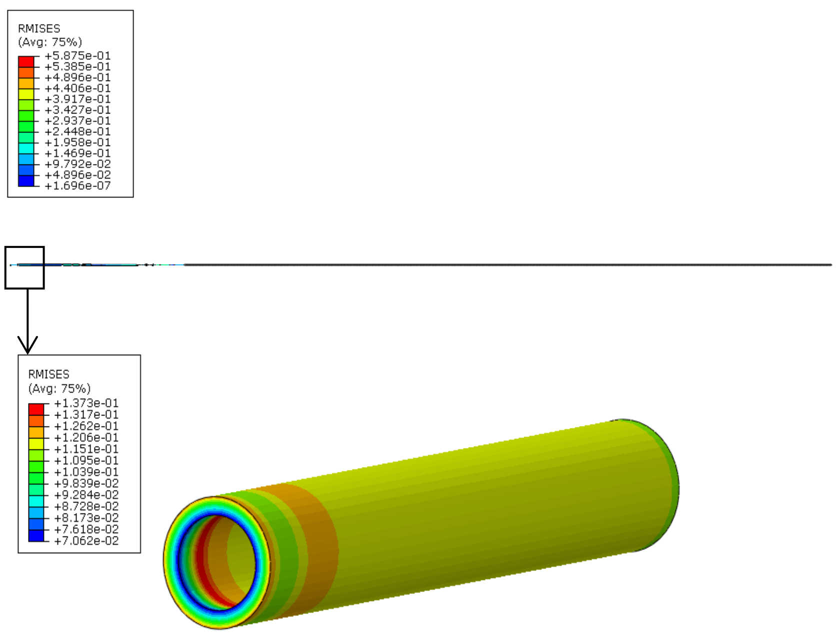


Figure 11. RMS value of von Mises stress.

4. Conclusion

X-AFV turbulent flow-induced vibration was numerically analyzed. The following is a brief summary:

1. DES turbulence modeling was performed on the reduced CFD model for the valve port area, where highest power of excitation force exists.

2. The excitation pressure fluctuation and its PSD at the maximum turbulent kinetic energy location were obtained for gas rate 160 MMscf/d.

3. The first 400 modes of natural frequencies were extracted for the full FEA model. The natural frequencies fall well within the frequency range of high excitation power.

4. The excitation pressure fluctuation has the random and nondeterministic continuous characteristics.

5. Random Vibration Analysis was performed using CFD predicted PSD of excitation pressure as the random vibration load. The PSD and RMS of stresses were obtained. The results show the random vibrational effect is very small, and thus the risk of fatigue failure is negligible.

5. References

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