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Brief Discussion:
Should Bolting Preload be included in a Finite Element Analysis?

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Date: Thursday, July 24, 2008

Discussion

Background:
The ASME VIII-1 code's *general* approach to bolting is stated succinctly in ASME VIII-1, Appendix S-1:

> *In the great majority of designs the practice that has been used in the past should be adequate, viz., to follow the design rules in Appendices 2 and Y and tighten the bolts sufficiently to withstand the test pressure without leakage.*

In other words, when the design complies with the ASME Code requirements, the bolts can be tightened as required. Thus, when a connection leaks due to relaxation of the joint, "*it may suffice merely to retighten the bolts*". (Appendix S-1) This apparently *loose* approach to the assembly and operation procedures stems from protection against over-tightening that is found within the Appendix 2 & Y calculations.

Approach:
A Finite Element Analysis (FEA) is performed when the particular configuration doesn't fall within the standard Code guidelines, such as the flange calculations found in ASME VIII-1 and mentioned above. The component is then approached as a VIII-1 vessel that needs additional rules, which are readily available in VIII-2.

The following study presents two extreme scenarios: a) No preload, & b) Very high preload. The contrast between these two cases is clearly shown by the results:

a) No Preload: unacceptable stress levels.
b) Very High Preload: acceptable stresses with infinite life cycles.

Conclusion:
Should bolting preload be included in a Finite Element Analysis?
The resounding answer is *YES*.

When a Finite Element Analysis is performed, the bolting preload must be considered both to realistically model the assembly and also be in compliance with the applicable pressure vessel code.

Questions remaining for future discussions are: How should the preload be applied in the FEA? How can the required preload be calculated? What financial and practical reasons are there for considering bolting preload when the ASME Code doesn't seem overly concerned with it?

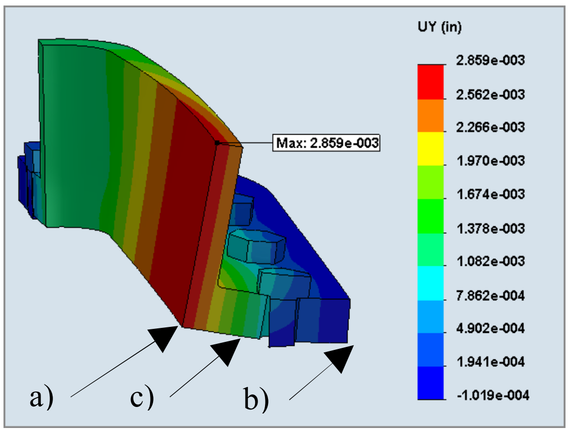
Results

Displacement Plots:

It is necessary to commence by reviewing the deformed displacement plots to make sure that the component's behaviour is reasonable. In the following plots, please note the following reference points, as illustrated in the first image:

a) Maximum displacement at the inner surface
b) Rotation about outer edge
c) Curvature of the bolted plate
d) Bolt bending

No Preload



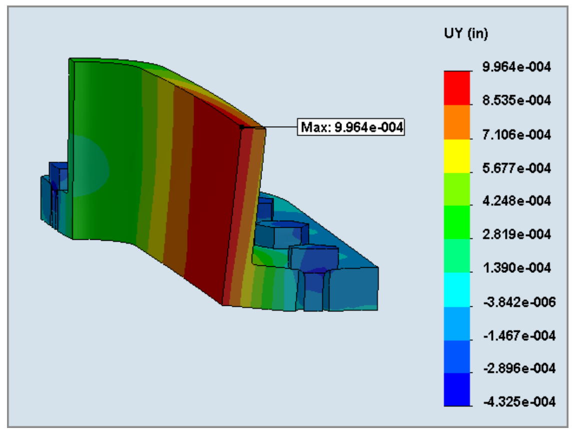
a) The maximum vertical displacement is 0.0028", at the lower surface.

b) The plate rotates about the outer edge.

c) The plate curves up leaving a significant gap and there is no metal-to-metal contact around the bolt hole.

d) The bolting deformation lines (different shades of colour) cross the bolt at an angle, because the bolt is undergoing significant bending.

Preloaded



a) The maximum vertical displacement is 0.0009", at the lower face.

b) There is very little movement about the outer edge.

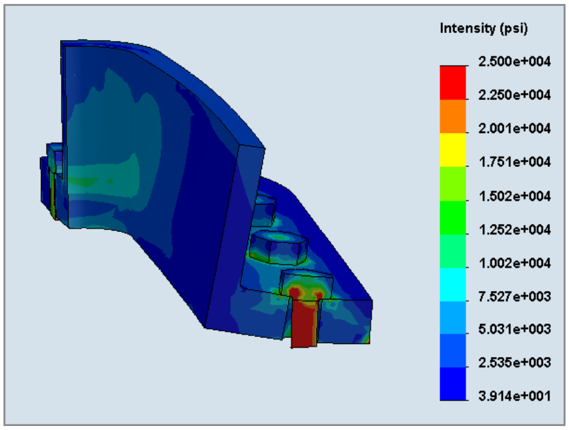
c) The bolting preload is sufficient to keep the lower surface in contact with its mating face, maintaining metal-to-metal contact.

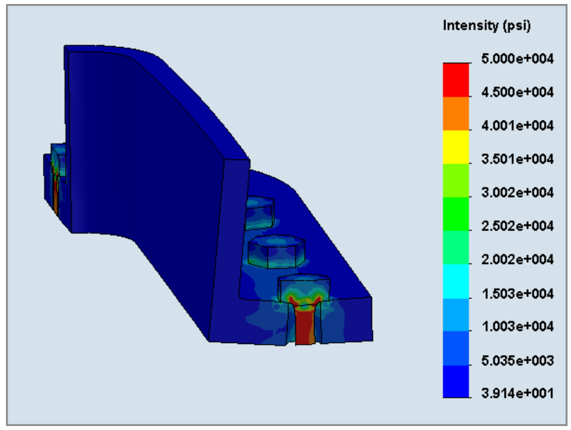
d) The bolting deformation lines run horizontally, indicating that the bolt is primarily in tension and is undergoing very little bending.

Stress Plots:

No Preload

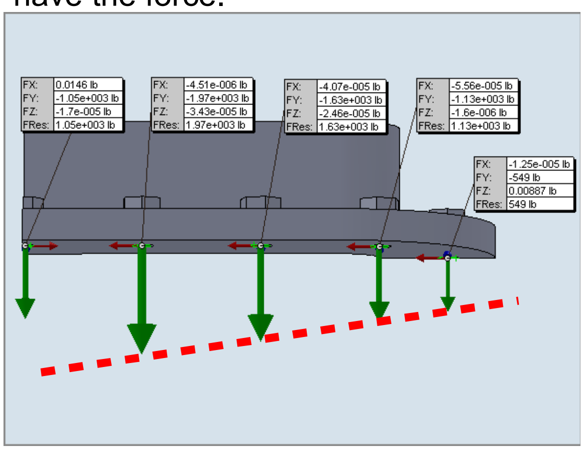
Preloaded



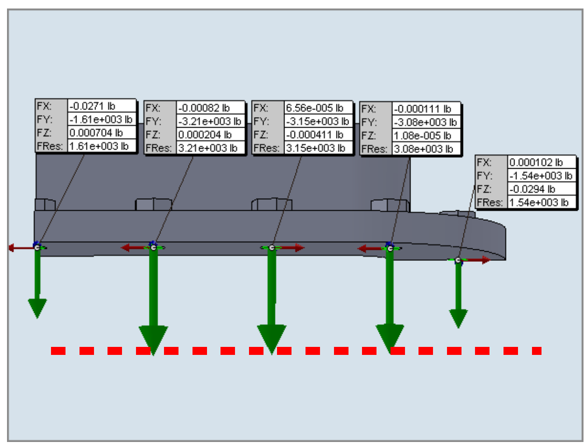


Bolt loading & stress:

1) **Distribution of bolt loads:** When the bolts don't have sufficient preload, the load distribution won't be the same from bolt-to-bolt. This can be seen in the following figures, where the length of each green arrow indicates the relative amount of tensile load in each bolt. In the figure on the left, the force decreases from left to right. In the figure on the right, all the bolts bear approximately the same load. Due to symmetry, the half-bolts will only bear have the force.



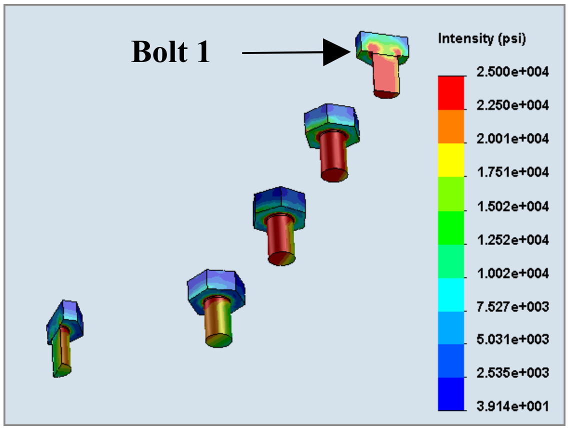
No Preload, Forces in bolts



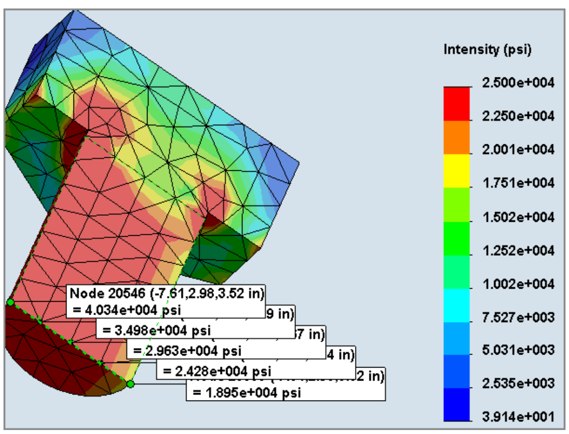
Preloaded, Forces in bolts

2) **Stress in each bolt**: The plate curvature causes the bending in the bolt. For this reason, no significant bending is found in the preloaded bolts.

No Preload



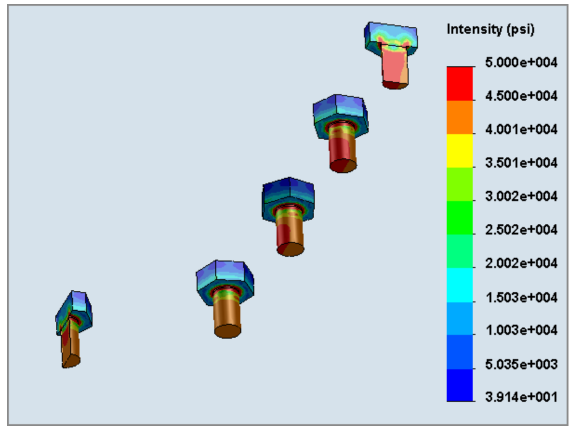
The worst-case scenario for the bolting is Bolt 1, because of highest stress levels. The display is capped at 25,000 PSI, which is tensile limit for the bolt.



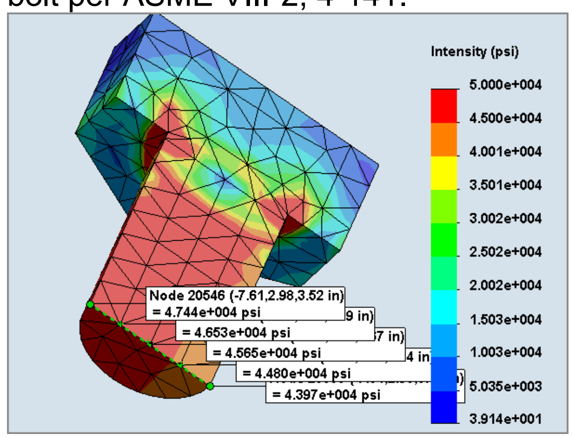
The display is capped at 25,000 PSI, which is the allowable stress in tension for this bolting material.

Bolt stress: 18,950 to 40,340 PSI

Preloaded



As previously, the worst-case scenario for the bolting is Bolt 1. The display is capped at 50,000, which is the allowable average tensile stress in the bolt per ASME VIII-2, 4-141.



The display is capped at 50,000 PSI, which is the allowable average tensile stress in the bolt per ASME VIII-2, 4-141.

Bolt Stress: 43,970 to 47,440 PSI.

The bolt with the highest peak stress is Bolt 1, as expected, and so it will be investigated further. Are the stress levels Acceptable or Unacceptable?

No Preload	Preloaded
Average Tensile Stress: The average tensile stress in Bolt 1 is 29,722 PSI, which is greater than the allowable stress of 25,000 PSI (for a SA-193-B7 bolt). The average tensile stress through the bolt has been calculated by dividing the resultant force by the cross-sectional area. This can also be obtained by the central node's stress along the bolt's cross-section, as shown. Stress is **Unacceptable**	**Average Tensile Stress**: The average tensile stress in Bolt 1 is 45,576 PSI, which is less than the allowable stress of 2X25,000 PSI, per ASME VIII-2, 4-141, when preload is included in the analysis. Stress is **Acceptable**
Peak Stress: The peak stress around the bolt periphery, away from a discontinuity zone, is 40,340 PSI, which is greater than 25,000 PSI. Because preload is not included in the analysis, the additional stipulations of ASME VIII-2, 4-141 cannot be used. Stress is **Unacceptable**	**Peak Stress**: The peak stress around the bolt periphery, due to a combination of tension plus bending is allowed to be 3XAllowable per ASME VIII-2, 4-141, when preload is included in the analysis. Peak stress of 47,440 PSI is less than the allowable 3X25,000 PSI= 75,000 PSI. Stress is **Acceptable.**
Fatigue life analysis: According to ASME VIII-2, Fig.5-120.1, curve MNS<2.7Sm, this peak stress of PSI predicts a lifetime of 12,054 cycles to failure. Acceptability would normaly be determined by the customer's specification.	**Fatigue life Analysis**: According to ASME VIII-2, Fig. 5-120.1, this peak stress of 47,440 PSI includes both the preload stress, which is 43,300 PSI, and the operating (alternating) stress. Actual alternating stress is very low and per ASME VIII-2, Fig.5-120.1, curve MNS<2.7Sm yields greater than 1E6 cycles, or *infinite*.
Result: Average Stress: **Uncceptable** Peak Stress: **Unacceptable** Cycle-Life: **1.2E4 cycles**	**Result:** Average Stress: **Acceptable** Peak Stress: **Acceptable** Cycle-Life: **Infinte, >>1E6 cycles**

3) For additional stress location results and interpretation, such as on the valve body, please contact Key Design Engineering for more information.

Additional information

The setup options used for the FEA model, such as mesh size, mesh error checking, load applications, boundary conditions, and many more details, can be obtained by contacting Key Design Engineering:

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