Interactions between GTMAX Plugs and Host Pipes

FEA Analysis

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Date: July 15, 2024

1. Scope

This report documents an analysis using Autodesk Inventor Stress Analysis software. The purpose of this analysis is to explore the stress behavior in pipes tested with GTMAX® Test Plugs with a focus on how stress response in the gripped area compares to that in the wetted area. A linear static analysis was performed using the finite element model shown in the figure below.

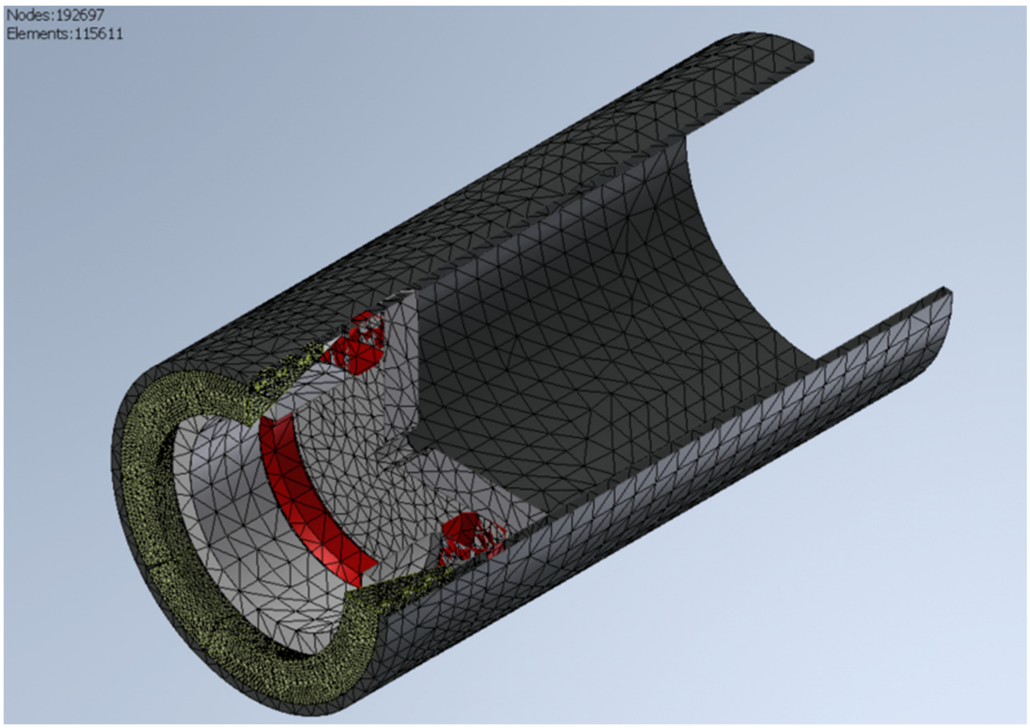

Figure 1: 14” (DN350 mm) Sch. 30/STD Pipe Tested with (simplified) GTMAX14PSTD Plug (Meshed)

2. Assumptions

- Displacements are small
- Follower forces are ignored

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3. Model Definitions

The simplified GTMAX plug is composed of its standard steel components and 80 durometer urethane seal. The Pipe is ASTM A106 Gr. B Steel with a yield strength of 35ksi (240 MPa). Due to limitations of the stress analysis software, the pipe ID was reduced to the plug OD of 13.07” (332.0 mm) to keep displacements small and in the elastic region. The stress analysis was run using a mesh of approximately 200,000 nodes and 115,000 elements.

4. Environment

4.1. Structural Loading

A pressure typical for 300# class hydrostatic applications of 1125psi (78 Bar) was applied to the interior of the pipe and plug bottom washer to simulate a pressure test. This test pressure is well within the 80% maximum yield pressure for this pipe as seen in Appendix A.

The pressure test load was applied only to the region of pipe to the right of the seal in the figure below to simulate a proper seal being formed during testing.

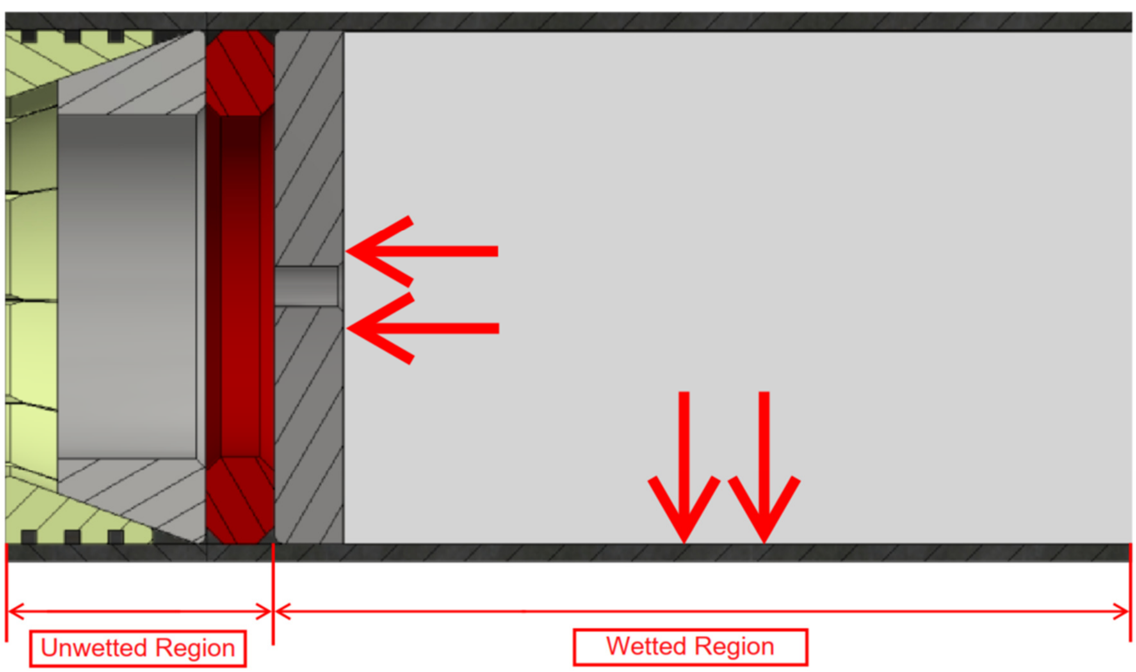

Figure 2: Internal Loading on Pipe Tested with GTMAX-14PSTD

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4.2. Structural Support

The grippers were free to expand radially by sliding on the tapered cone surface to replicate gripper expansion, but were restricted from moving longitudinally (left or right in the figure above) to simulate effectively gripping the pipe.

5. Results

Results show that the wetted region on the interior of the pipe experiences approximately 17ksi (117 MPa) of stress and the exterior experiences approximately 15ksi (103 MPa) of stress with the defined setup. This agrees with theoretical calculations shown in Appendix B.

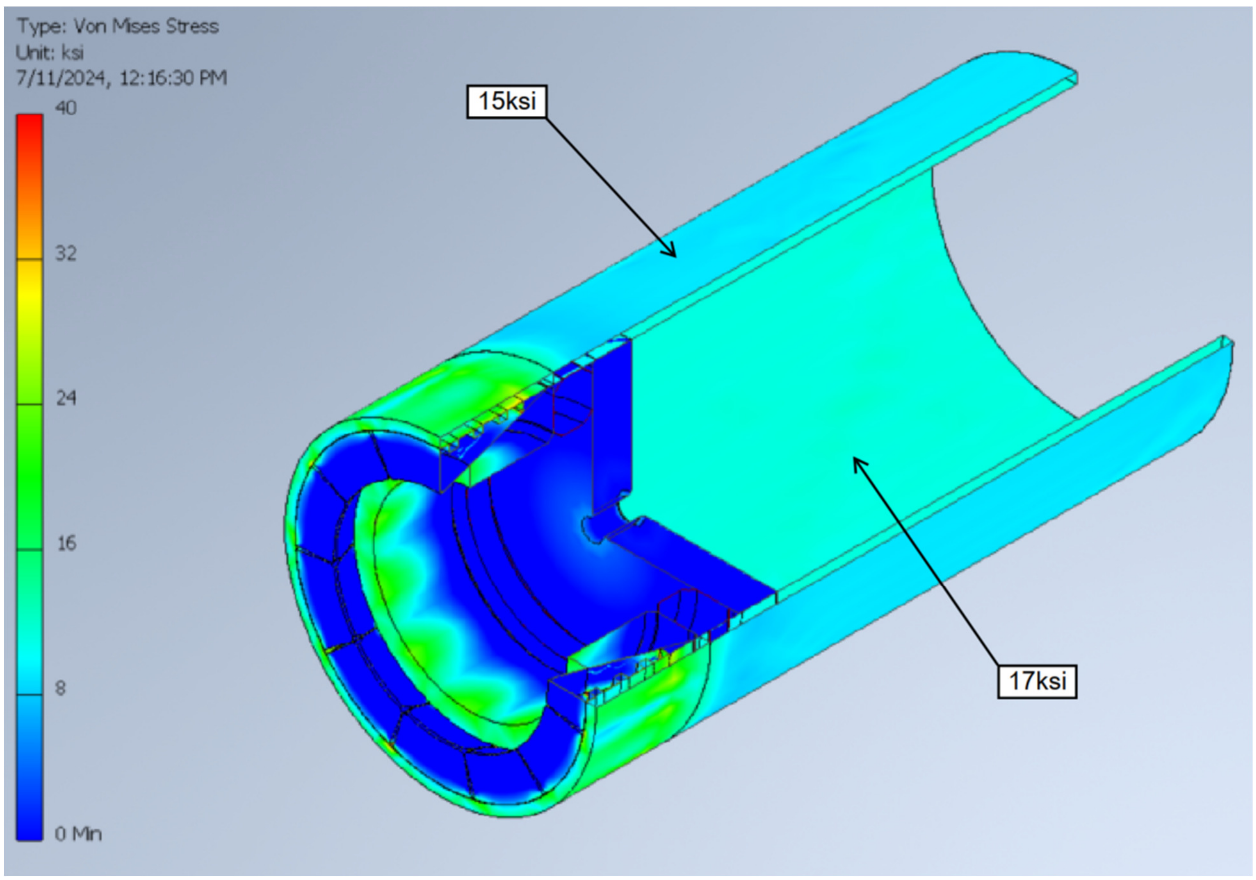

Figure 3: Shaded View of FEA Results. Pipe Interior Stress is Approximately 17ksi (117 MPa).

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In the gripped region, stress is concentrated near the bottom and edges of gripper segments. The pipe experiences 15ksi (103 MPa) at the center of the segments and 40ksi (276 MPa) at the segment edge contact area. These high contact stresses are representative of the grippers biting into the pipe and are critical for proper plug function. This behavior is due to the slight difference in radius between the gripper assembly and pipe internal diameter, and can be seen in the marks left behind after testing which are more pronounced in these areas.

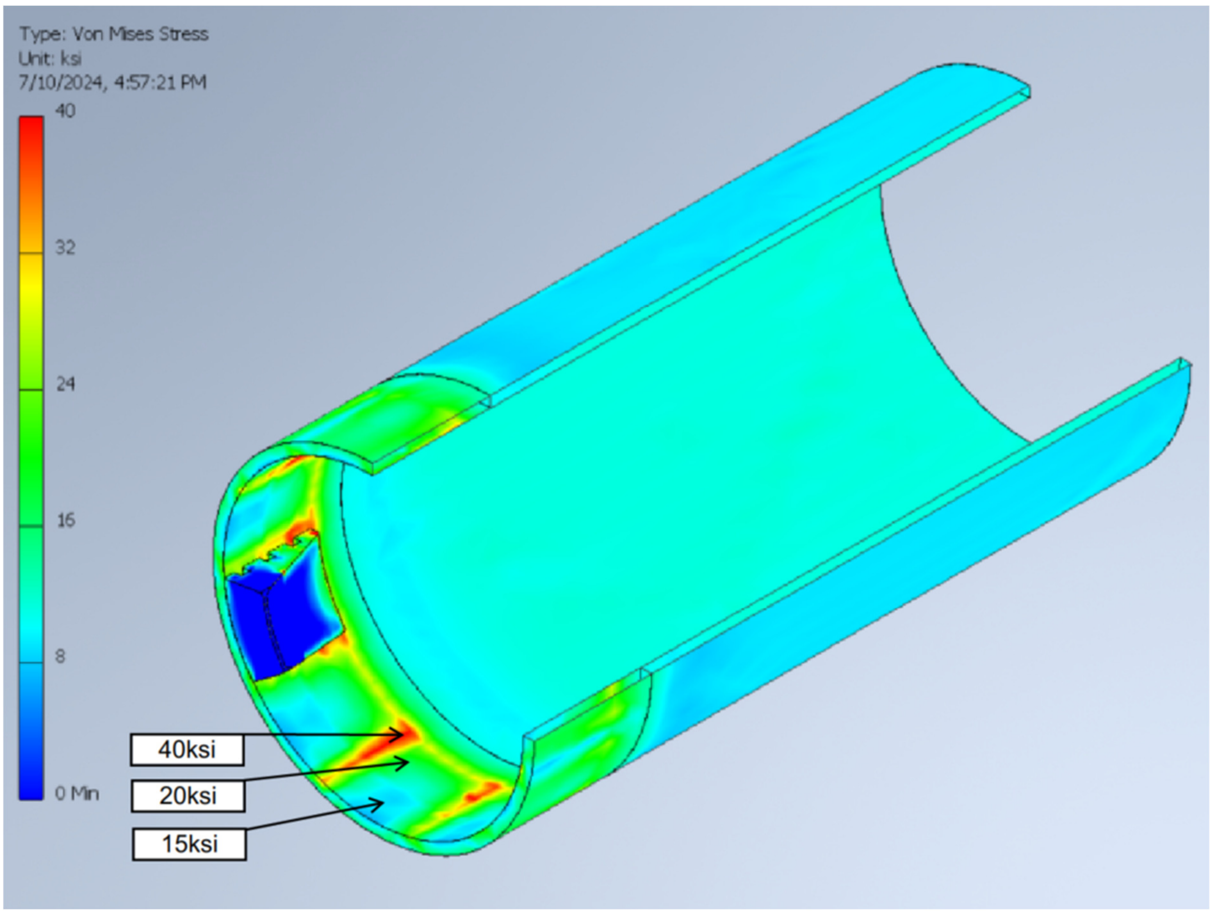

Figure 4: Pipe Interior Stress Concentrated at Gripper Edges

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As stated, the calculated stress on the pipe interior exceeds the pipe yield strength in these (red) sections. However, analysis shows the stresses return to below yield on the pipe exterior meaning these stresses are highly localized where the gripper segments make contact, as seen in Figure 5.

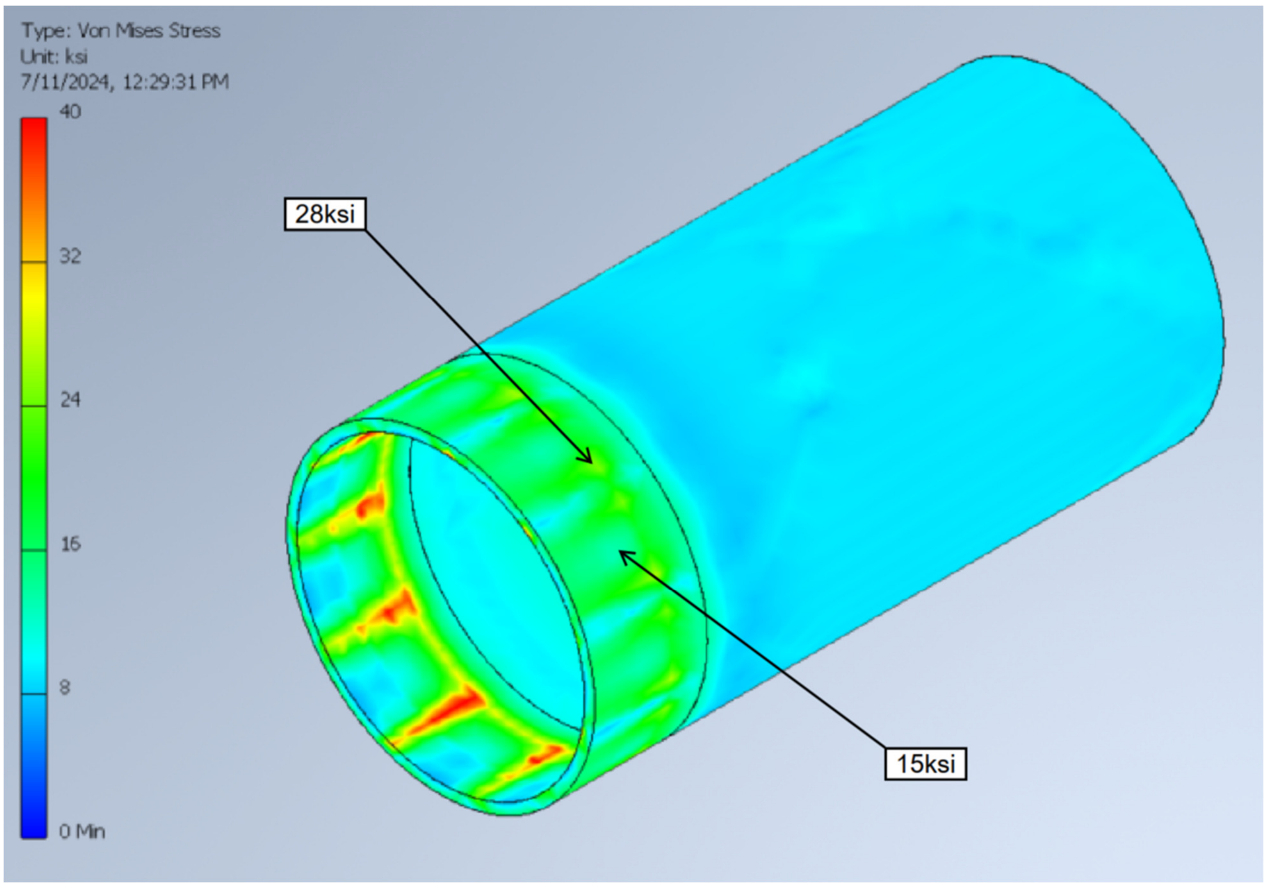

Figure 5: Pipe Exterior with Stress due to Gripper Segments Concentrated near the Bottom of the Segments

Note: Due to the simulation relying on small deformations, the stress due to the seal cannot be fully analyzed with this model. However, the stress on the pipe due to seal expansion can be shown to be small compared to stress from pressure testing (see Appendix C).

Additionally, a ring is visible in the figures above between the gripped region and the pressurized region of the pipe. This is an artifact from the simulation showing a sharp discontinuity between loading environments.

6. Summary

The behavior observed in the FEA study is in line with theoretical calculations to a reasonable degree of accuracy and agrees with practical experience. Some discrepancy is to be expected due to the different analysis approach.

As shown, high stress is felt at the gripper edges and low stress is felt at the center of gripper segments. The areas experiencing high stress are highly localized to small elements and do not propagate to the pipe exterior.

In conclusion, the unwetted region of the pipe is exposed to equal or greater stress than that felt in the wetted region.

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Appendices

Appendix A: Pipe Burst Pressure Calculations

Pipe/Tubing material :	Carbon Steel	
Pipe/Tubing spec :	ASTM A106 Grade B	
Tensile strength :	60,000 (415)	psi (MPa)
Yield strength :	35,000 (240)	psi (MPa)
Pipe/Tubing Od :	14.00 (355.6)	in (mm)
Pipe/Tubing Wall :	.465 (11.81)	in (mm)
Pipe/Tubing Id :	13.07 (332.0)	in (mm)
% of w/t vs radius :	7%	
Pipe/Tubing type (thin or thick) :	thin	
Tensile psi :	4,269 (294)	psi (bar)
Yield psi :	2,490 (172)	psi (bar)
80% of yield psi :	**1,992 (137)**	**psi (bar)**

Appendix B: Theoretical Pipe Stress Calculations

Thin-wall approximations can be used if the pipe wall thickness is less than 5% of the diameter.

In this case: w/t = 0.465”, 0.05*13.07 = 0.654”, so thin wall approximations are applicable.

Hoop Stress can be calculated using the following equation: $\sigma_h = \frac{Pd}{2t} = \frac{(1125psi)*(13.07")}{2*(0.465")} = 15810psi\ (109.0\ MPa)$

Longitudinal Stress can similarly be calculated as follows: $\sigma_l = \frac{Pd}{4t} = \frac{(1125psi)*(13.07")}{4*(0.465")} = 7905psi\ (54.5\ MPa)$

Radial stress is equal in magnitude to the applied pressure: $\sigma_r = -\mathrm{P} = -1125psi\ (7.8\ MPa)$

No shear stress is applied to the pipe during pressure testing ($\tau = 0$).

Finally, Von Mises Stress can be calculated using the equation: $\sigma_{VM} = \sqrt{\frac{(\sigma_l-\sigma_r)^2+(\sigma_r-\sigma_h)^2+(\sigma_h-\sigma_l)^2}{2} + 3\tau^2} =$
$\sqrt{\frac{(7905psi+1125psi)^2+(-1125psi-15810psi)^2+(15810psi-7905psi)^2}{2} + 3(0)^2} = 14677psi\ (101.2\ MPa)$

Appendix C: Calculations for Stress Imposed on Pipe by Seal under Compression

Pipe Parameters			
Pipe Material Yield Strength	σ_y	35,000 (240)	psi (MPa)
Nominal Pipe OD	OD	14.00 (355.6)	in (mm)
Nominal Pipe ID	ID	13.07 (332.0)	in (mm)
Outside Radius (calc)	R	7.00 (177.8)	in (mm)
Inside Radius (calc)	r	6.54 (166.1)	in (mm)
Wall Thickness	t	0.465 (11.81)	in (mm)

Seal Parameters			
Seal OD (uncompressed)	D_o	13.07 (332.0)	in (mm)
Seal ID (uncompressed)	D_i	8.63 (219.2)	in (mm)
Seal Width	W	1.75 (44.5)	in (mm)
Seal Material Durometer	H	80	shore A
Seal Material Young's Modulus	Y	1,120 (7.7)	psi (MPa)
Seal Thickness	t	2.22 (56.4)	in (mm)

Test Parameters			
Test Pressure (Max: 80% Material Yield)	P	1,125 (77.6)	psi (bar)

Calculation			
Uncompressed Volume of Seal	V_{su}	132.43 (2170)	in^3 (cm^3)
Seal Thickness After Compression	W_{sc}	1.75 (44.5)	in (mm)
Force Applied on Seal at Full Pressure	F_s	150,936 (671.4)	lbf (kN)
Axial Compressive Stress on Seal	S_a	1,995 (13.8)	psi (MPa)
Circumference Compressed Seal Area	A_s	71.86 (464.0)	in^2 (cm^2)
Stress to Expand Seal to Pipe ID		0.00	psi (MPa)
Distance Seal would Expand		1.11 (28.2)	in (mm)
Pressure From Compressed Seal	P_s	558 (38.5)	psi (bar)

Stress Analysis			
Type of Calculation	Thin-Wall Vessel Calculation		
Hoop Stress Applied by Seal Under Pressure	**σ_e**	**7,849 (54.1)**	**psi (MPa)**

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