Liquid Hydrogen Tank Chill and No-Vent Fill Prediction using Computational Fluid Dynamics

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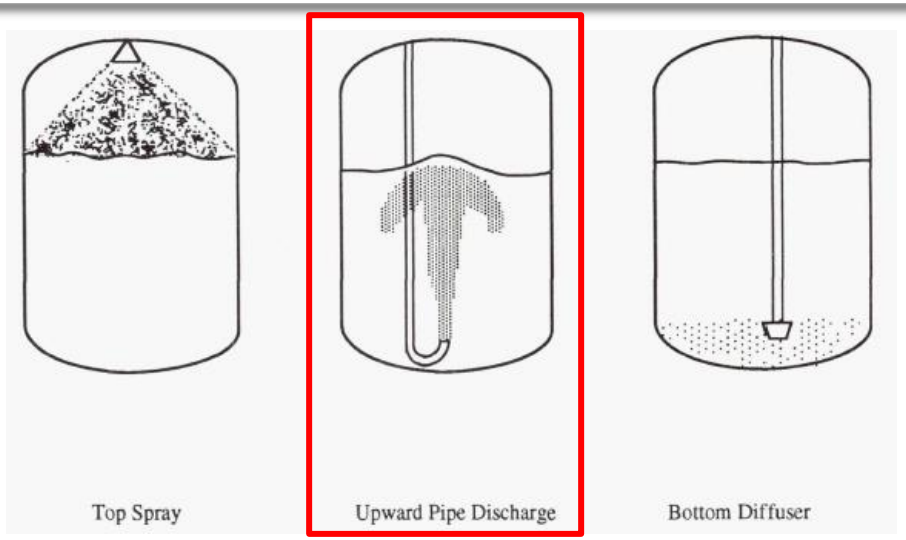
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Motivation

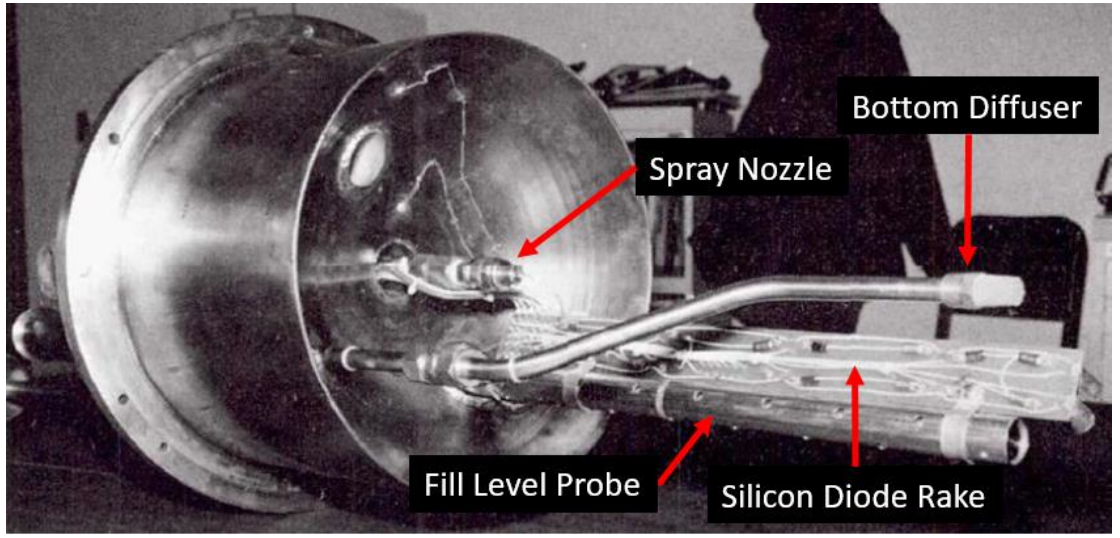
- On-orbit propellant depots for refueling spacecraft can support and enable NASA's long-duration deep space missions
- Cryogenic tank chill and fill is an important cryogenic fluid management (CFM) technology required to realize in-space propellant transfer
- No-Vent Fill (NVF) propellant transfer architectures require the receiver tank pressure to remain below the supply tank pressure during the entire transfer
- In lieu of expensive tests conducted on-orbit, accurate computational models capable of predicting receiver tank pressure may be used to reduce system mass and mission risk
- Computational models must be validated to experimental data to instill confidence

Experimental Overview (1 of 2)

- Liquid Hydrogen NVF ground testing was conducted in 1990 at NASA Lewis Research Center (now Glenn Research Center)
- Cylindrical tank with elliptical domes
  - 20” height
  - 12.5” diameter
  - 1.2 cubic foot volume (34 liters)
- Experiment employed three different liquid injection methods: top spray, upward pipe, bottom diffuser



Liquid injection configurations used for NVF testing (NASA TM-105273, 1991)



Receiver tank lid assembly (NASA TM-105273, 1991)

Experimental Overview (2 of 2)

- Experimental Measurements Taken:
  - Tank Pressure
  - Fluid Temperatures
  - Solid Wall Temperatures
  - Fill Level
  - Mass Flow Rate
  - Inlet Temperature

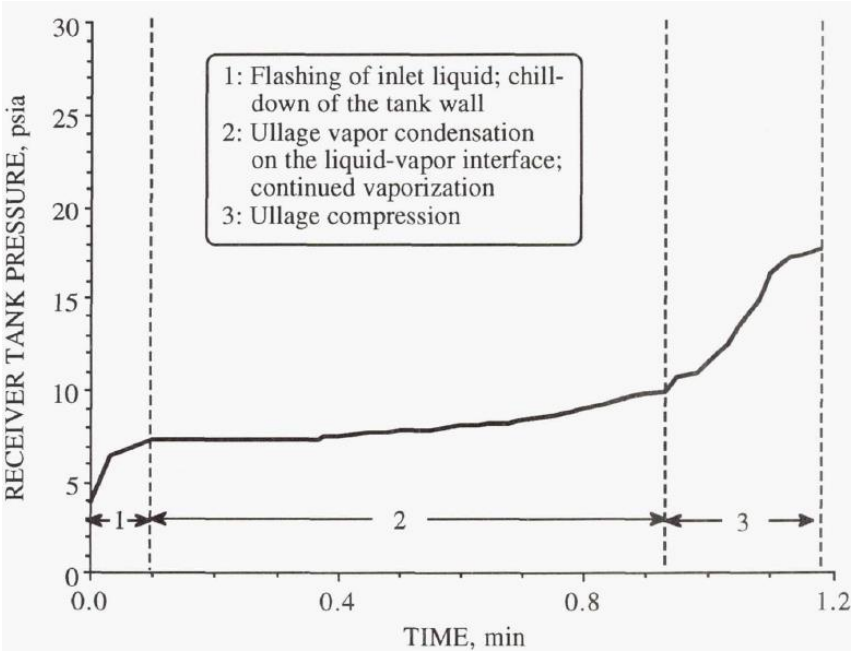
Reported as $f(t)$

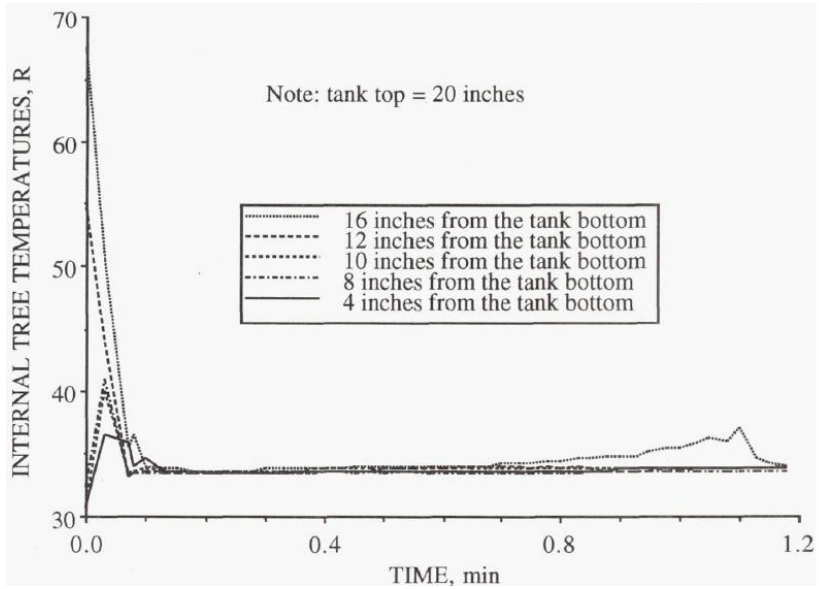
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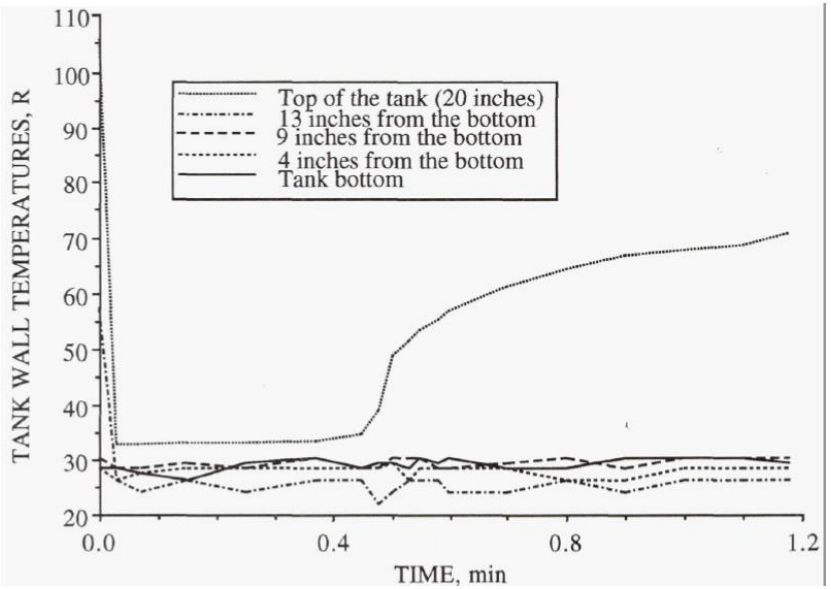
Measured Test Quantities for Test Number 9093G (NASA TM-105273, 1991)

Fill Config	Inlet Venturi Temp	Inlet Venturi Flow Rate	Initial Tank Pressure	Final Fill Level
	K	kg/s	psia	% by volume
Upward Pipe	18.55	0.034	3.8	96

↳ $P_{sat}(T_{inlet}) \approx 8.2$ psia







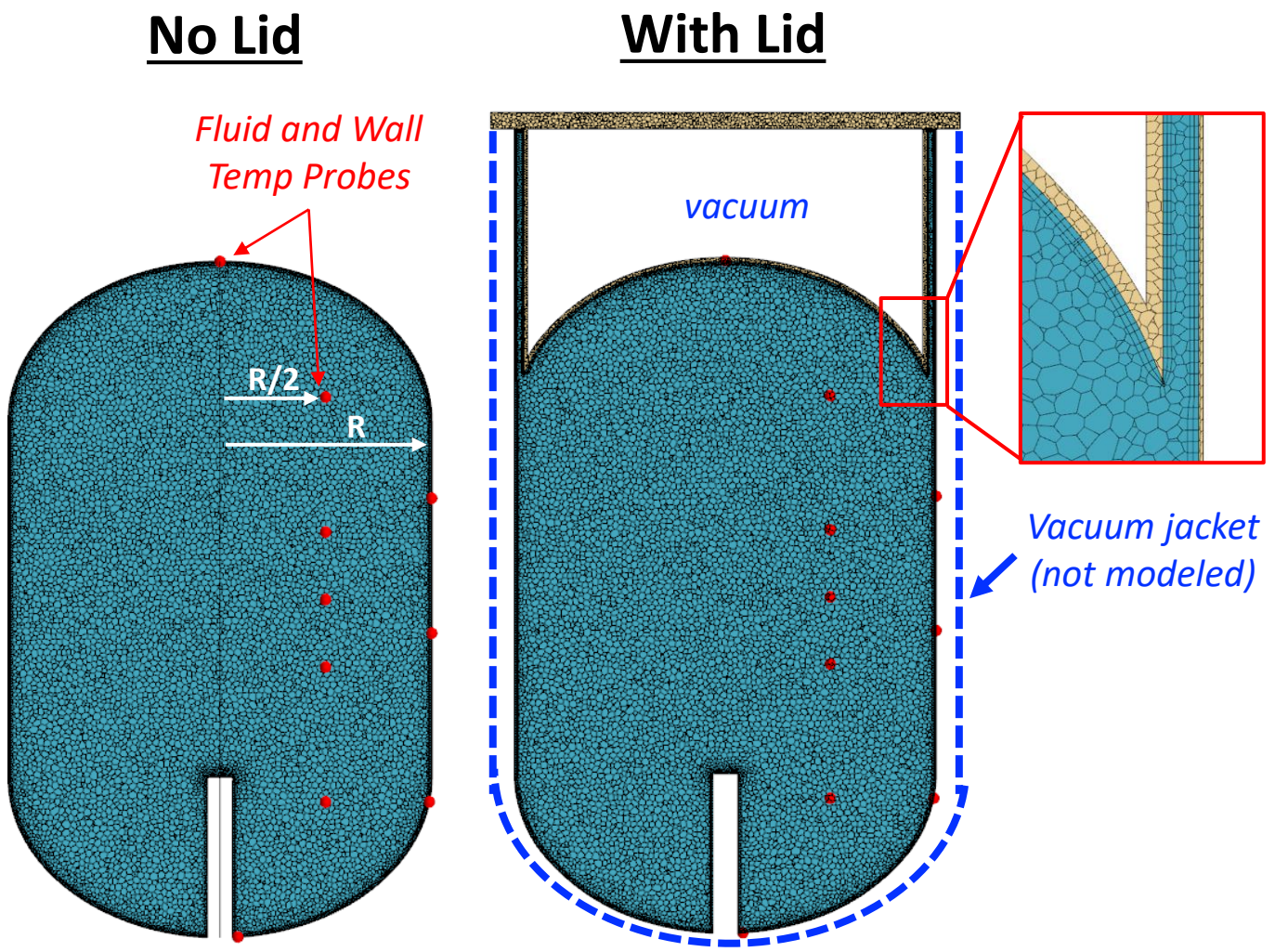
Data reported for upward pipe configuration (NASA TM-105273, 1991)

Notable Sources of Uncertainty in the Experiment

- Geometric dimensions of injection pipe diameter, axial height of pipe exit, tank lid assembly, and tank wall thicknesses
- Mass flow rate and inlet temperature reported as averaged quantities (no time-dependent data)
- Mass flow rate measurement of two-phase flow during initial portion of tank fill
- Initial temperature of tank lid assembly
- Heat leak
- % of helium in tank at start of test (helium gas was used to purge tank between tests)

Computational Model: Geometry and Mesh

- Siemens Simcenter STAR-CCM+ was used to model the experiment
- Two full 3D computational domains were considered for this study:
  1. Internal tank volume with constant wall thickness of 0.81mm
  2. Internal tank volume including tank lid structure with variable wall thickness
- Internal temperature rake assumed to be at R/2 (no specific details given in NASA TM)
- Conjugate Heat Transfer between fluid and solid walls using a conformal polyhedral mesh



Computational mesh for both configurations shown on plane through middle of tank

Computational Model: Physics Models

- Volume-of-Fluid Multiphase Solver
- Surface Tension Force with 0deg contact angle
- Constant liquid properties
- Compressible ideal gas
- Temperature-dependent SS304 properties (C_p and k)
- k-ω Menter SST (Re_d = 1.4e5 → turbulent)
- Schrage Equation for user-defined interfacial mass transfer with Accommodation Coefficient, σ
- Conservation of Enthalpy for liquid flashing across an orifice was used to calculate inlet quality and void fraction every time step
  - Initial tank pressure is well below saturation pressure at inlet temperature
  - Liquid will flash as it enters domain
  - Inlet Void Fraction function of time as it adjusts to current tank pressure predicted by CFD

Schrage Equation

$$\dot{m}_i = \left(\frac{2\sigma}{2-\sigma}\right)\sqrt{\frac{M}{2\pi R T_{cell}}}\left(P_{sat}(T_{cell}) - P_{abs}\right) \quad \left[\frac{kg}{m^2 s}\right]$$

Clausius-Clapeyron Relation

$$P_{sat}(T_{cell}) = P_{ref} exp\left[\frac{L}{R}\left(\frac{1}{T_{ref}} - \frac{1}{T_{cell}}\right)\right]$$

Conservation of Enthalpy

Exit quality: $$X = \frac{Cp_{liq}[T_{inlet} - T_{sat}(P_{tank})]}{H_{vap}}$$

Void Fraction: $$\alpha_{vap} = \left[1 + \frac{1-X}{X}\frac{\rho_{vap}}{\rho_{liq}}S\right]^{-1}$$

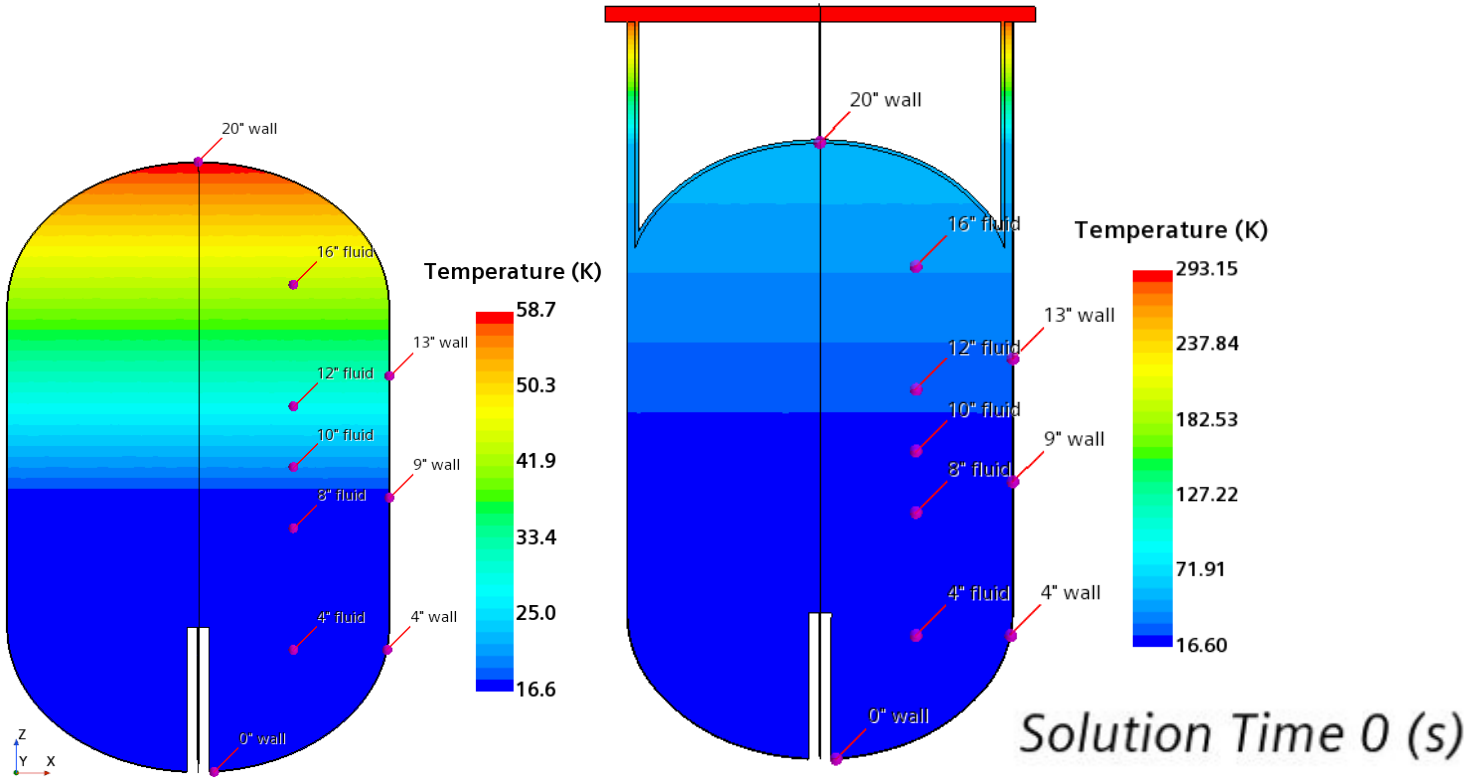
S is the slip ratio (vapor velocity to liquid velocity) and is assumed to be 1.0

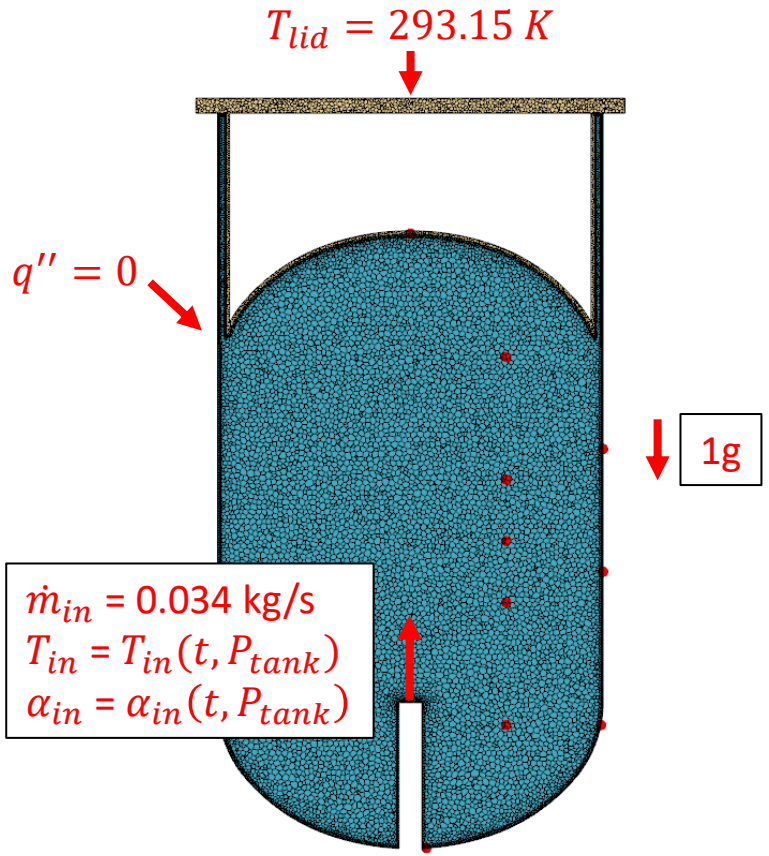
Computational Model: Boundary Conditions and Initial Conditions

- Initial temperature field set to match experimental wall temperatures using linear interpolation between probes
- Initial pressure and constant flow rate specified via experiment
- Since liquid is flashing as it enters tank, inlet temperature is specified as a function of time and tank pressure
- $T_{measured}$ is the averaged measured inlet temperature via experiment

$$T_{inlet} \begin{cases} = T_{sat}(P_{tank}) & P_{tank} < P_{sat}(T_{measured}) \\ = T_{measured} & P_{tank} \geq P_{sat}(T_{measured}) \end{cases}$$

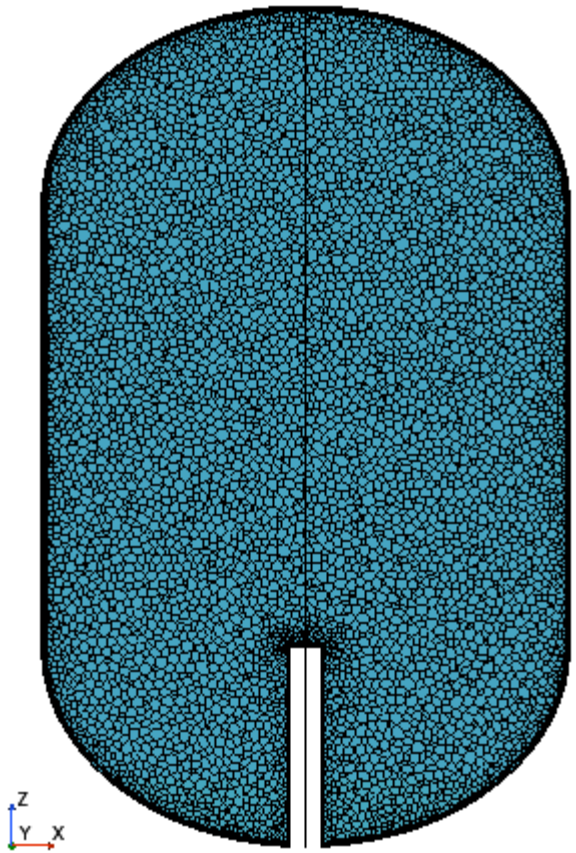
$$T_{measured} = 18.55\ K$$





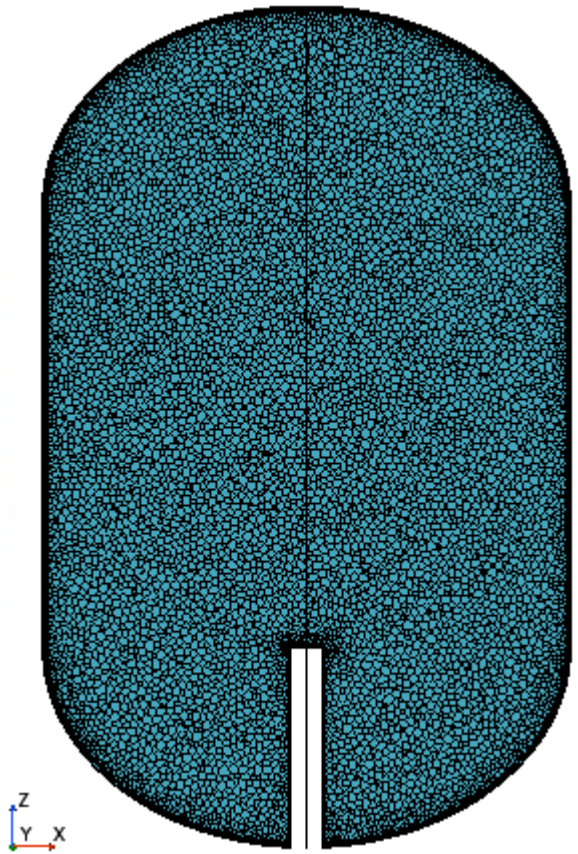
Mesh Independence Study

Coarse



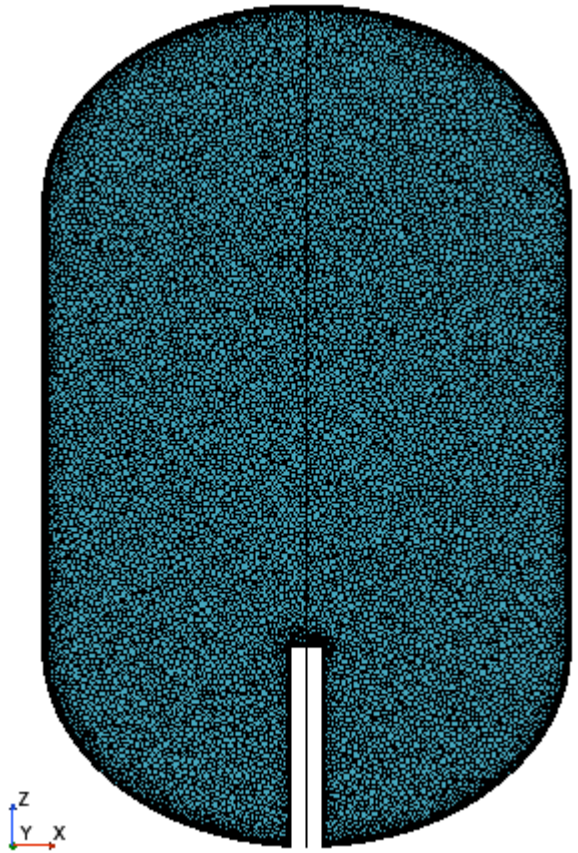
Medium base size * (2)^(1/3)
0.77 million cells

Medium



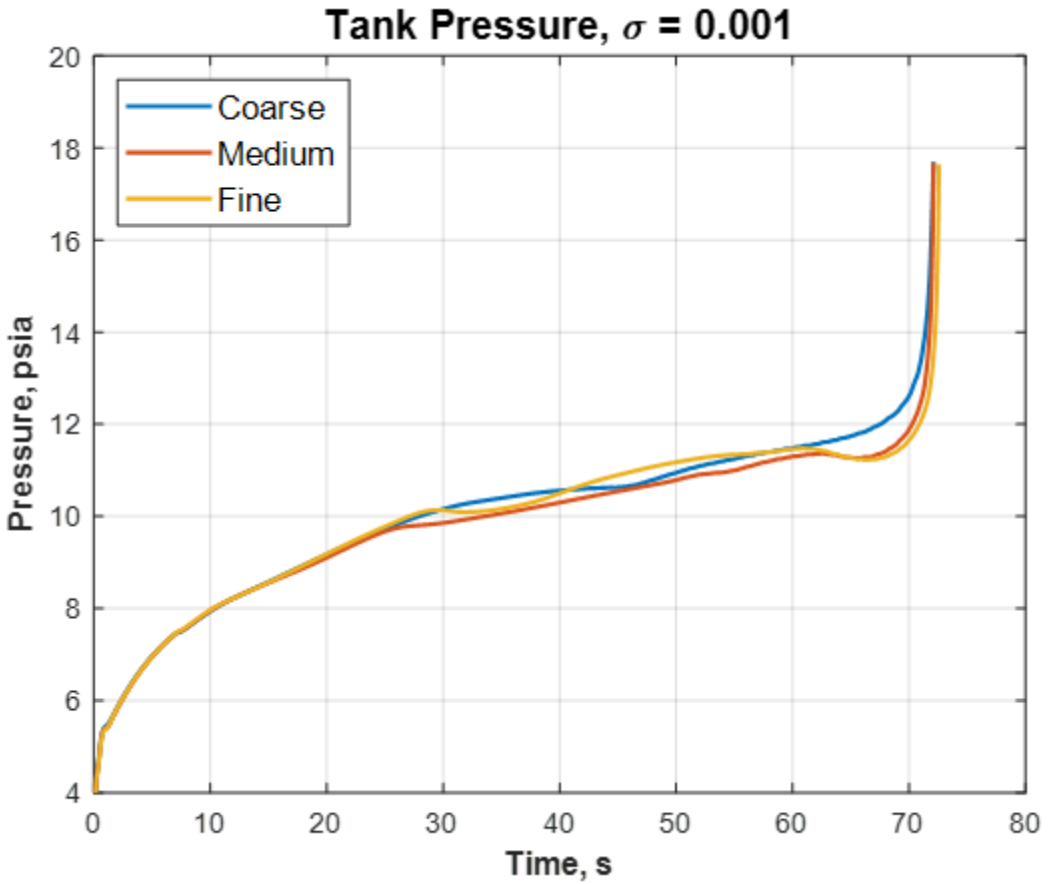
Medium base size = 4 mm
1.47 million cells

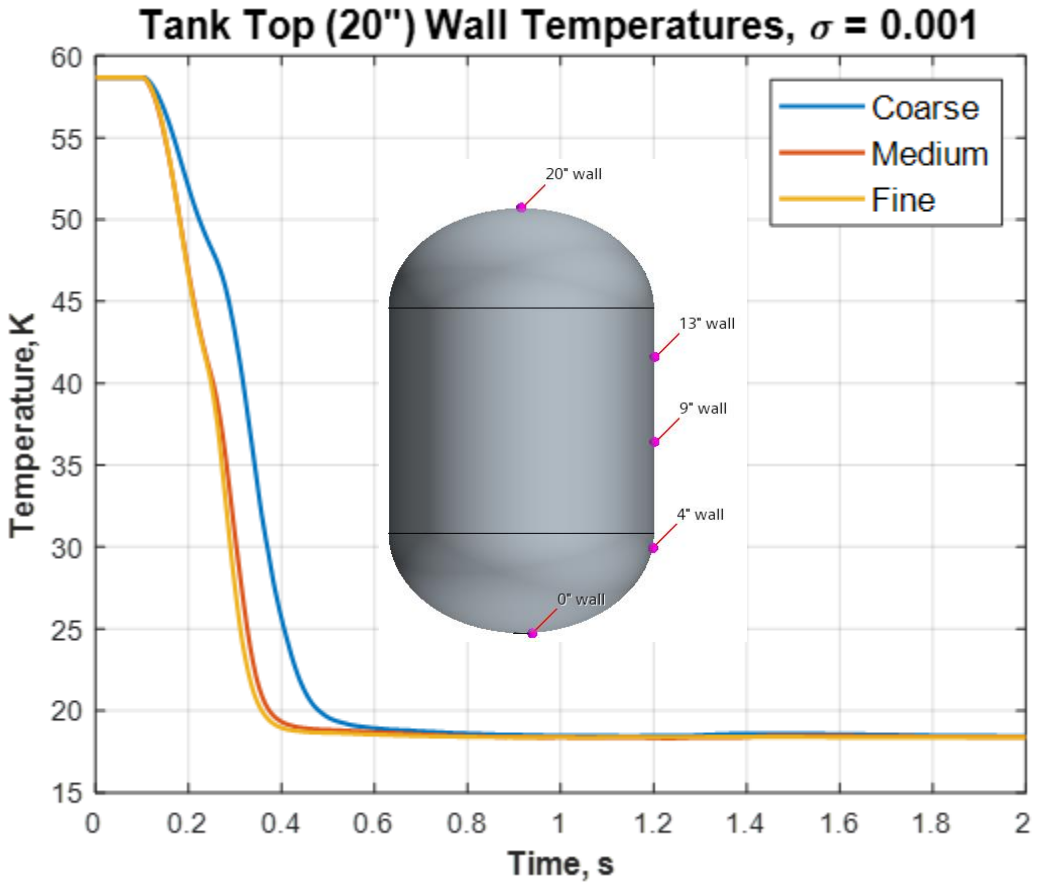
Fine



Medium base size / (2)^(1/3)
2.32 million cells

Mesh Independence Study



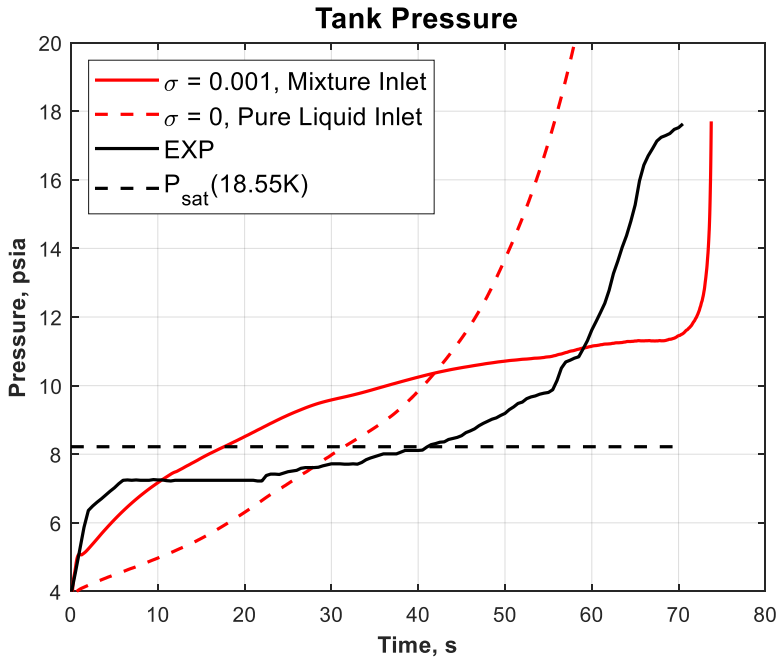


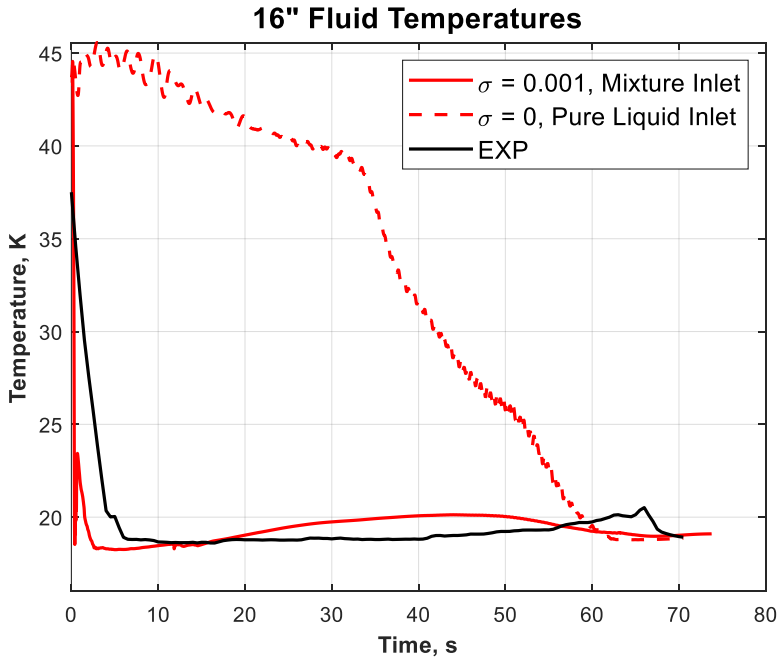
Validation Approach

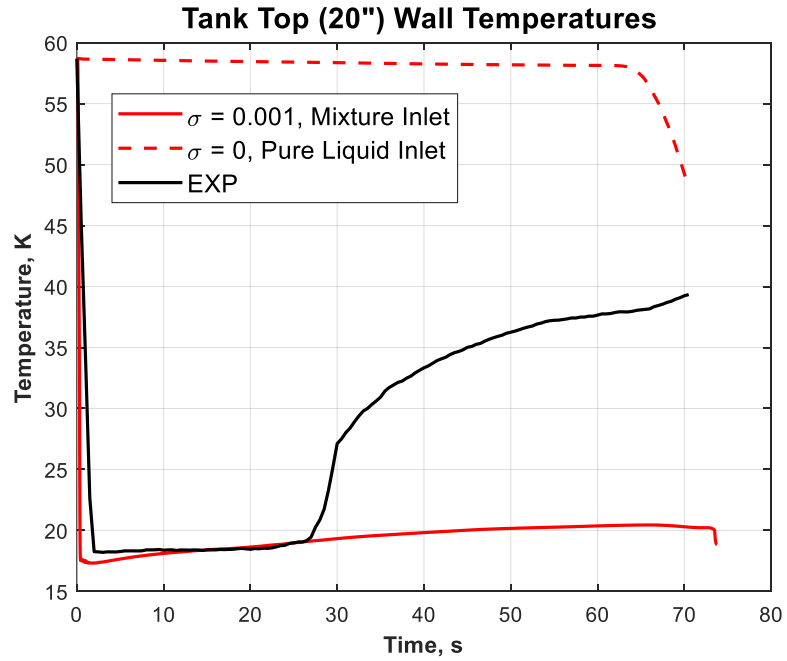
- Due to the amount of uncertainty in the experiment, several sensitivity studies were considered
  - Inlet Condition and Mass Transfer
  - Inlet Temperature
  - Accommodation Coefficient
  - Geometric Configuration and Wall Thickness
  - Presence of Helium
- Note: Each simulation was run until the predicted pressure reached the final pressure in the experiment (as opposed to the fill level)
- This allows us to assess the CFD model's capability of predicting the tank pressure at the corresponding fill level
- In practice, it would be highly desirable for CFD models to accurately predict at what fill level would the transfer fail, reach a MEOP, or for a successful transfer, what the pressure would be at the end of the transfer

Effect of Inlet Condition and Mass Transfer (1 of 2)

- First, a baseline case was compared to a case that implemented a pure liquid inlet (no flashing) and neglected interfacial mass transfer
- The baseline case (mixture inlet) predicts the tank pressure and temperature responses with better accuracy
  - Combination of injecting two-phase flow and evaporation yields a larger initial pressure rise
  - Later in the fill, the Pure Liquid Inlet case predicts large pressure rise due to ullage compression while condensation dominates in baseline case keeping pressure low
  - Temperature response is predicted well with 'mixture' inlet due to higher velocity jet penetrating to top of tank chilling probes and inducing mixing

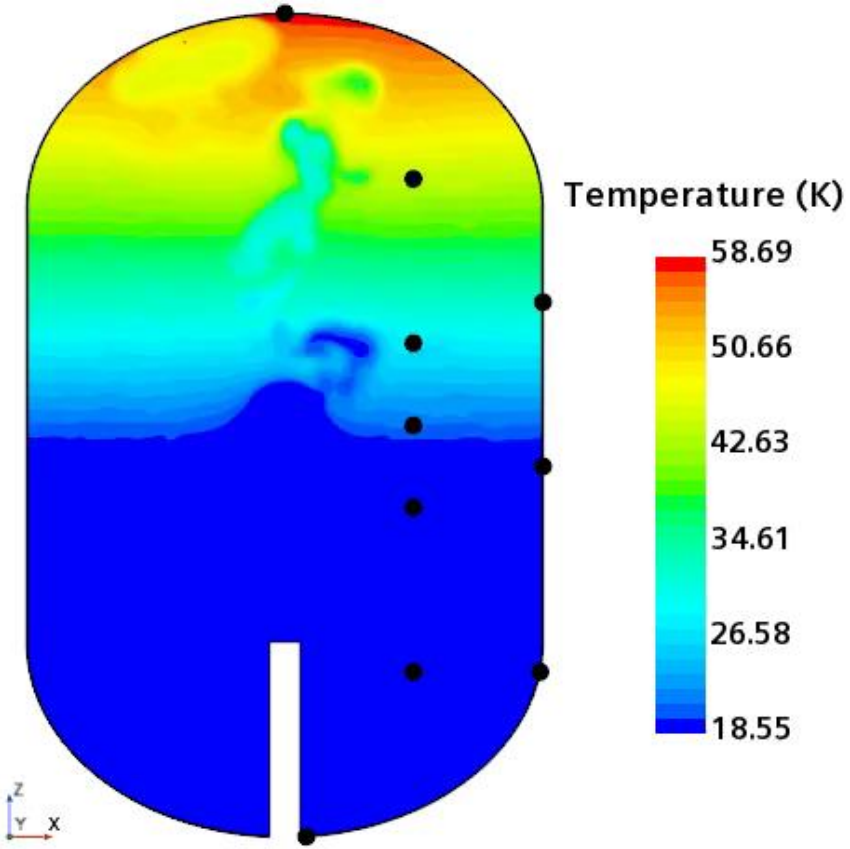






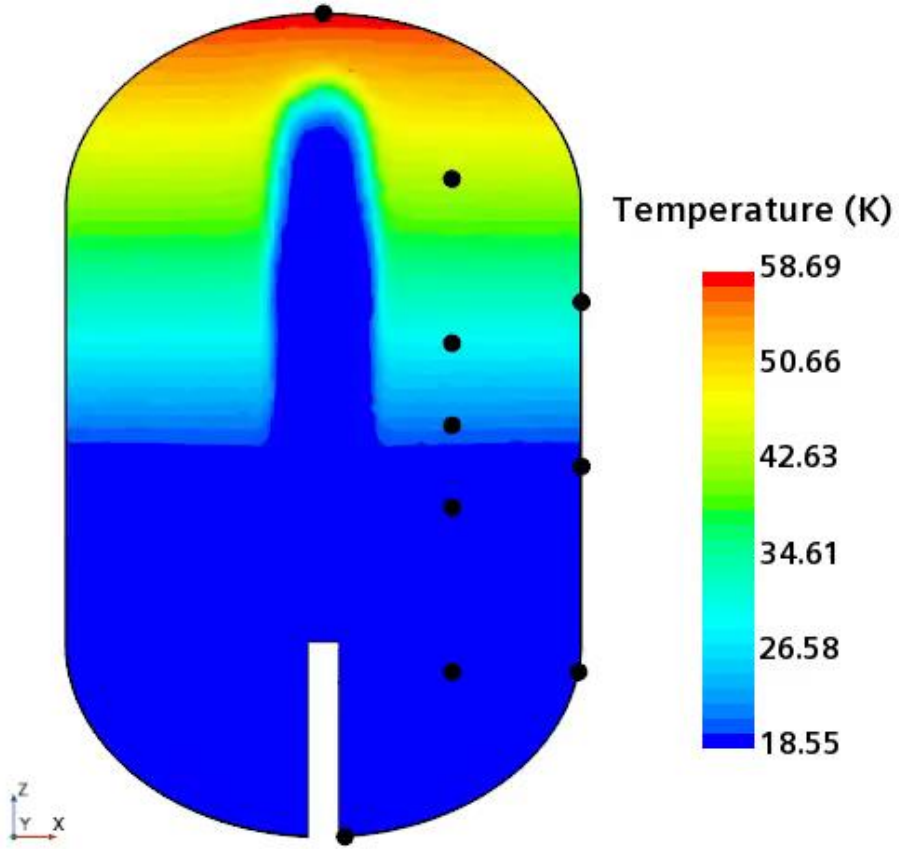
Effect of Inlet Condition and Mass Transfer (2 of 2)

Pure Liquid Inlet



Solution Time 0.2 (s)

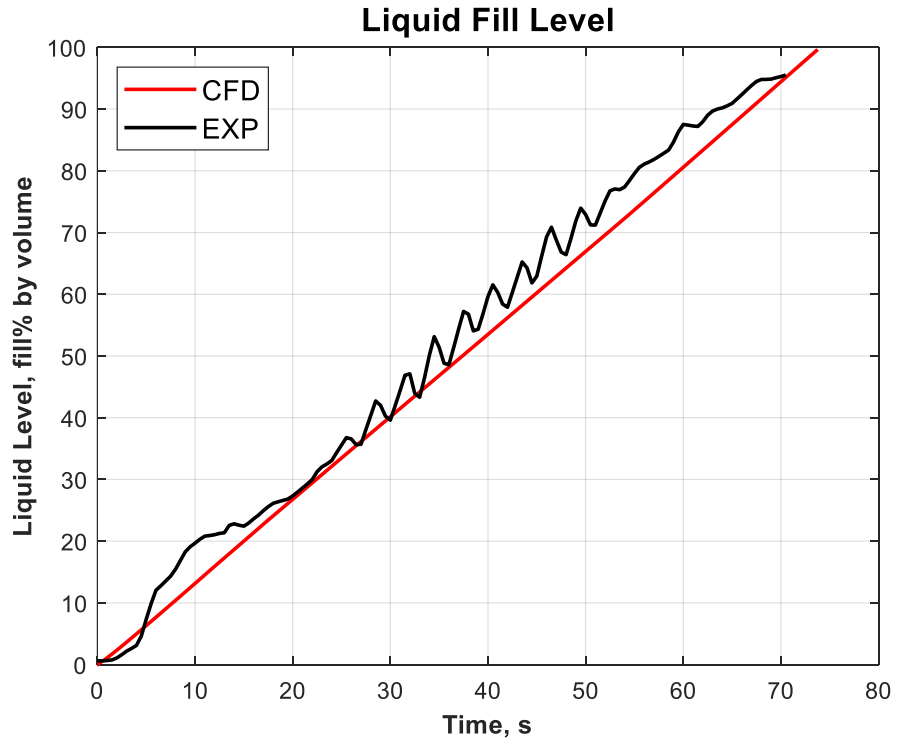
Mixture Inlet

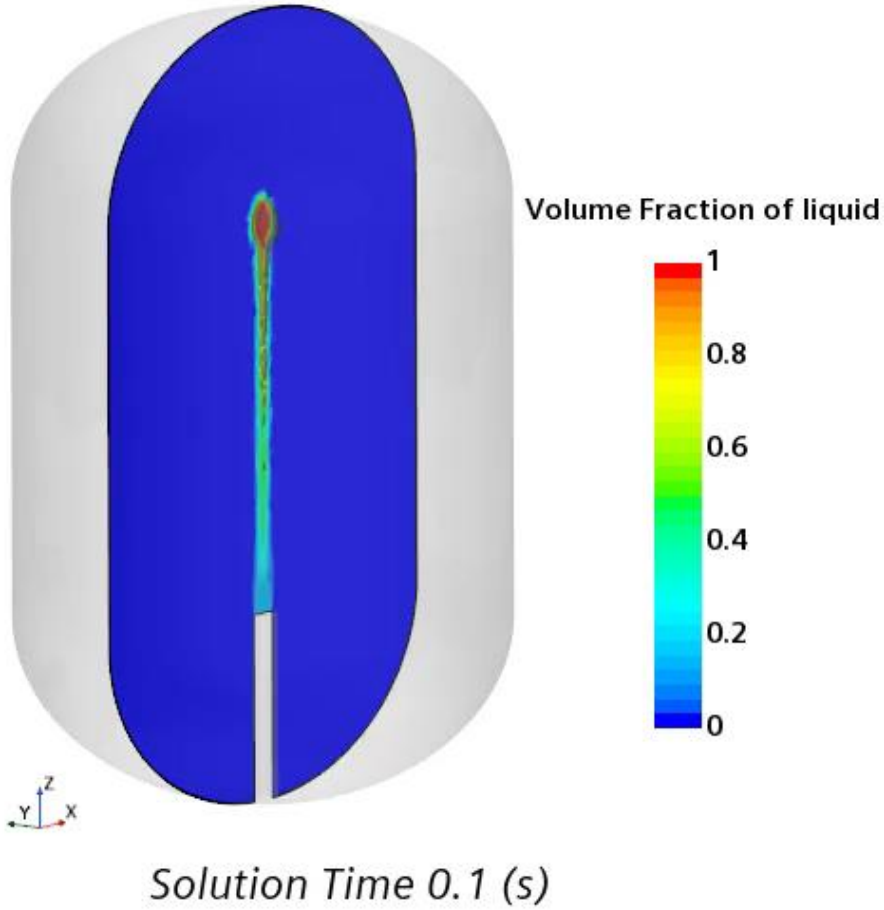


Solution Time 0.1 (s)

Fill Level Evolution

- Oscillations in the liquid fill level are observed during the middle section of the test
- Upward liquid jet acted as a forced perturbation that excited the liquid-vapor interface inducing a slosh mode for that range of fill level
- CFD VOF model captures oscillations during exact time range (25-55 seconds)

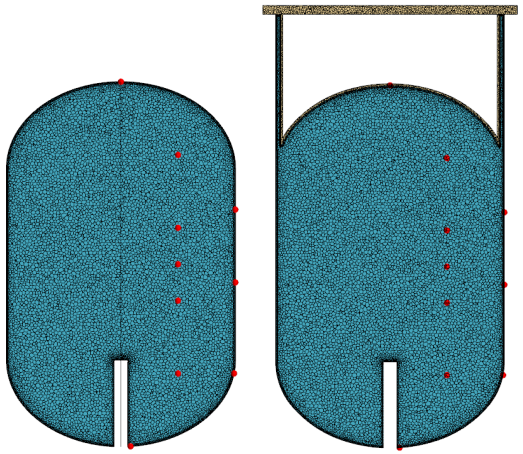


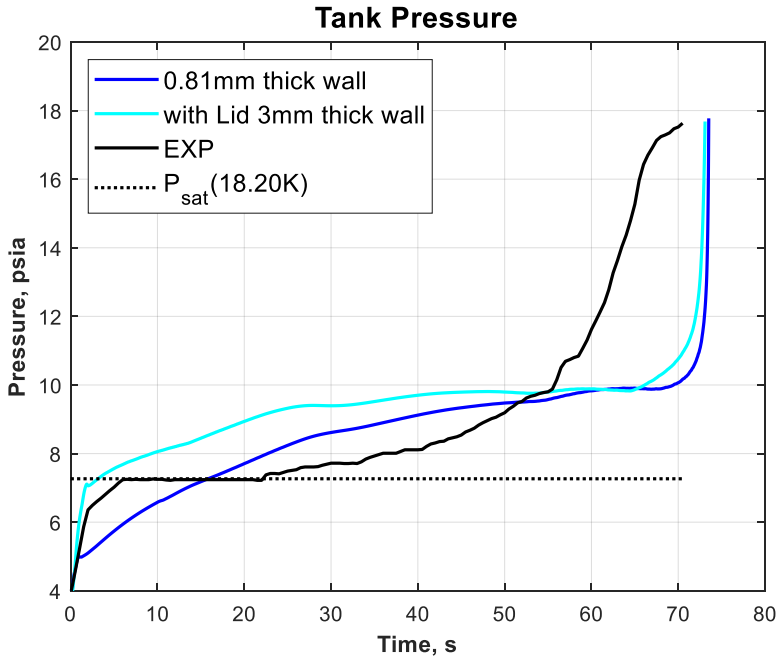


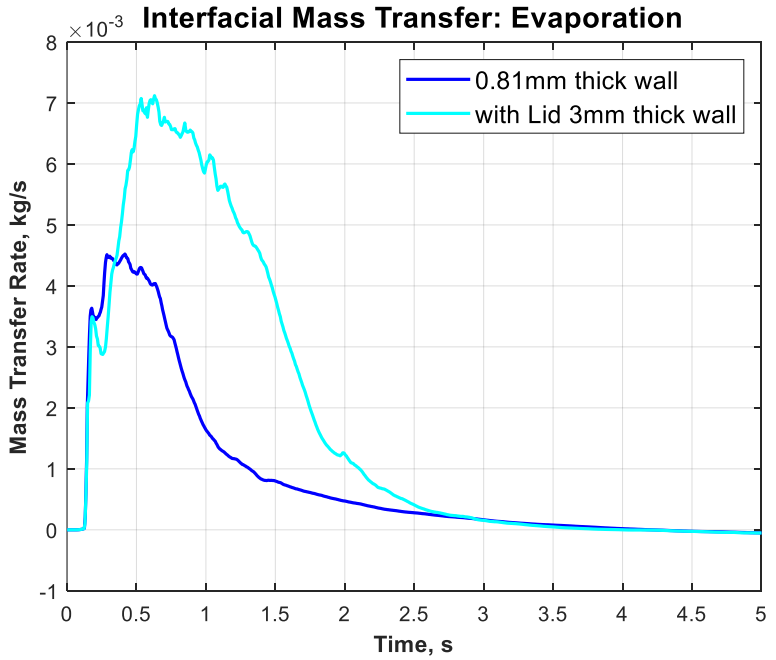
Fill sequence highlighting Volume Fraction on midplane and liquid surface using isosurface

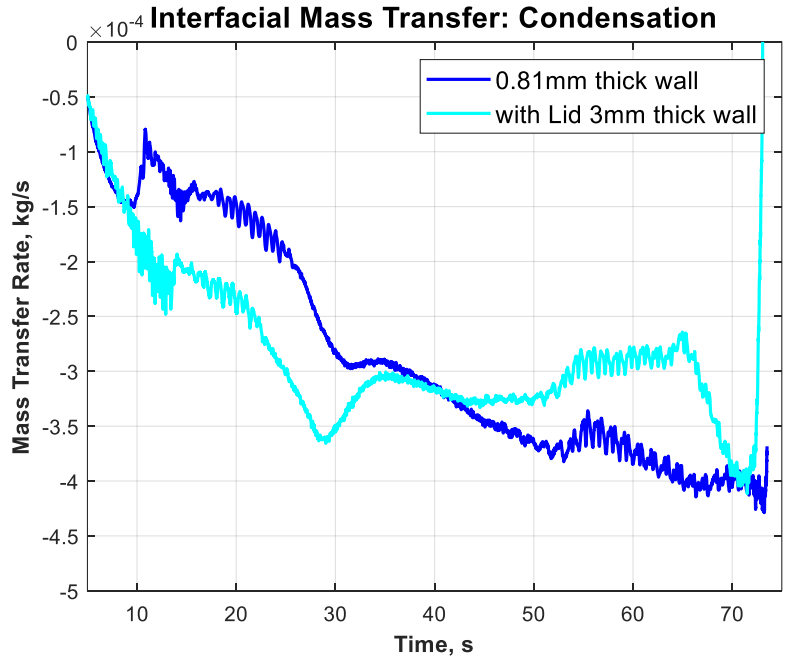
Effect of Geometric Configuration (1 of 4)

- Configuration with lid and thicker walls predicts higher pressure rise due to higher evaporation rate
- Thicker walls increased the thermal mass which extracts more energy from the walls during chilldown resulting in a higher pressure
- Since the two configurations bound the experimental pressure during the initial part of the fill, actual tank wall thickness may be somewhere between



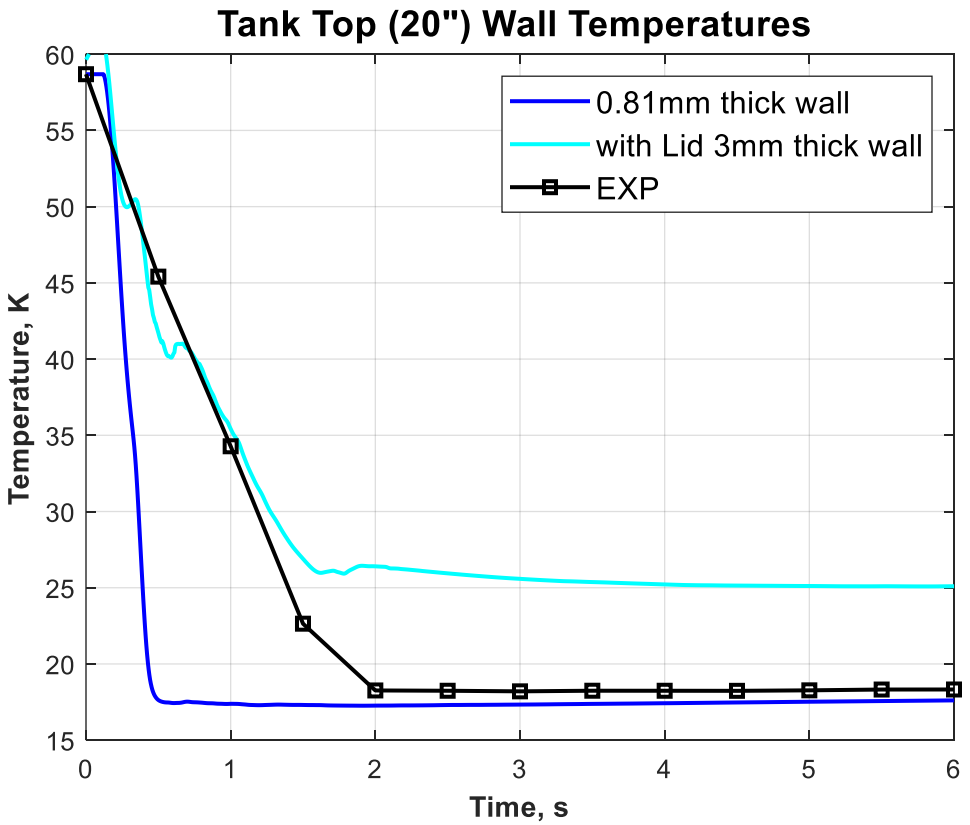


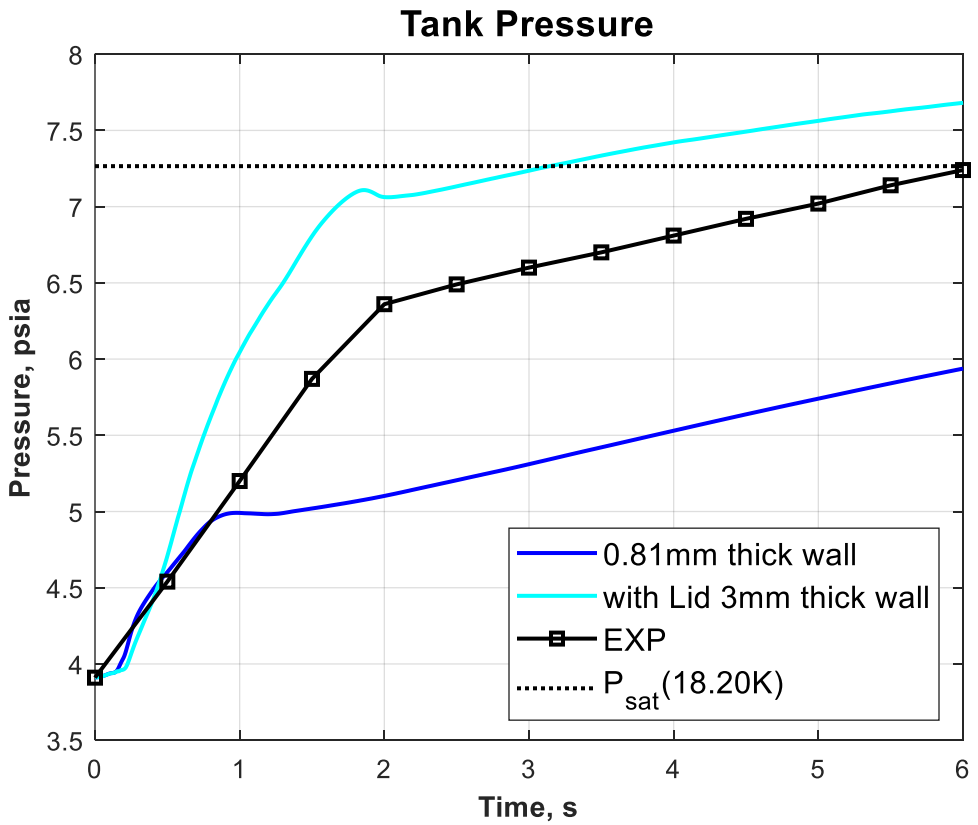




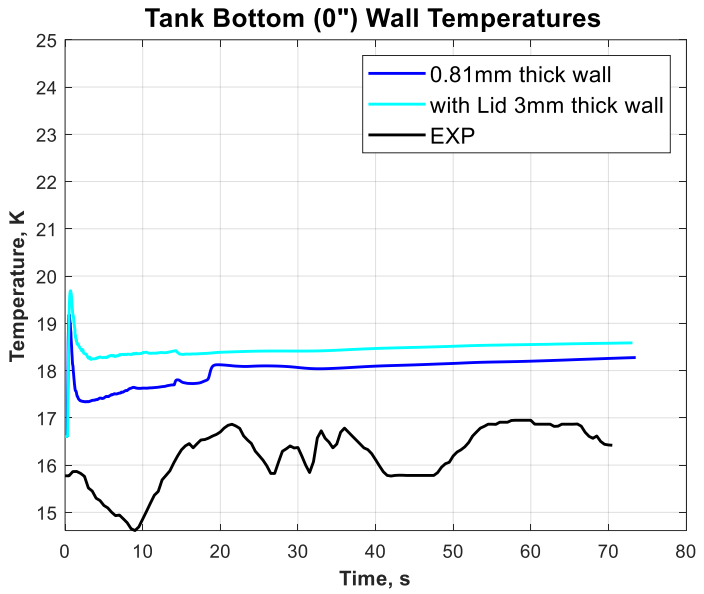
Effect of Geometric Configuration (2 of 4)

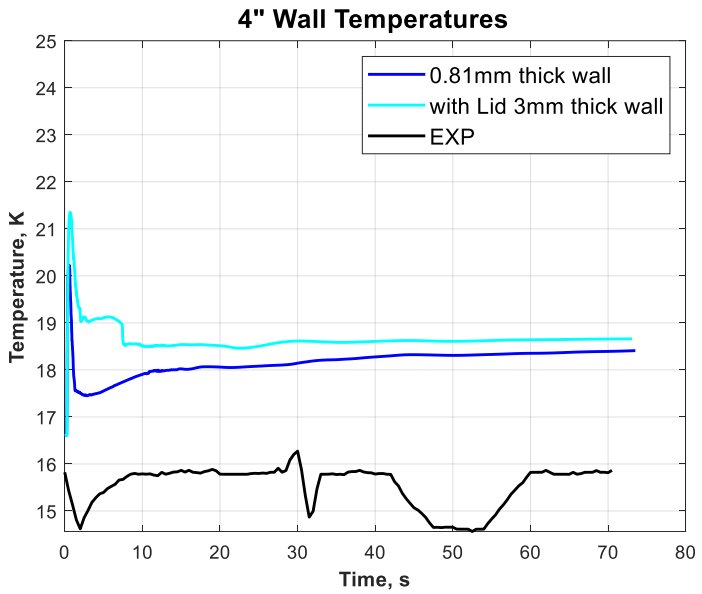
- The chilldown of the top wall temperature probe corresponds to a change in the pressure rise slope
- Once the probe is chilled down to the inlet temperature, the evaporation rate significantly diminishes yielding a slower pressure rise
- The thicker wall configuration doesn't chill all the way down to the inlet temp because it over-predicts the pressure rise so that the liquid jet no longer impinges on the top dome

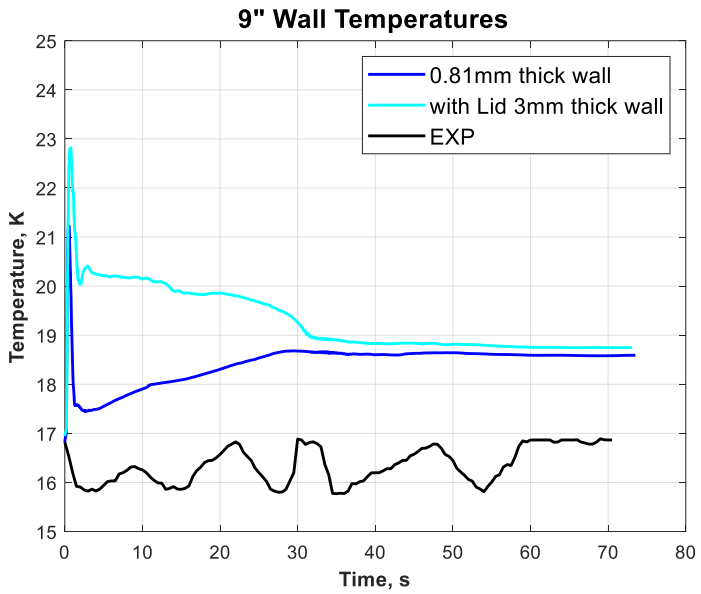


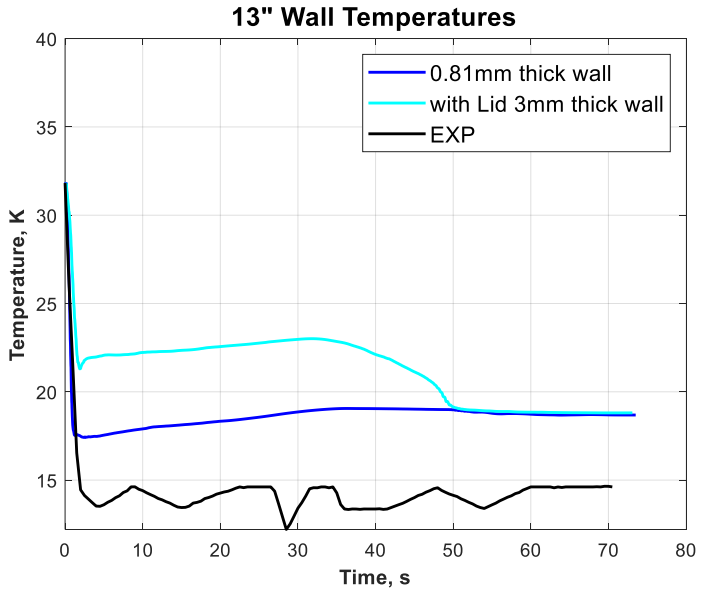


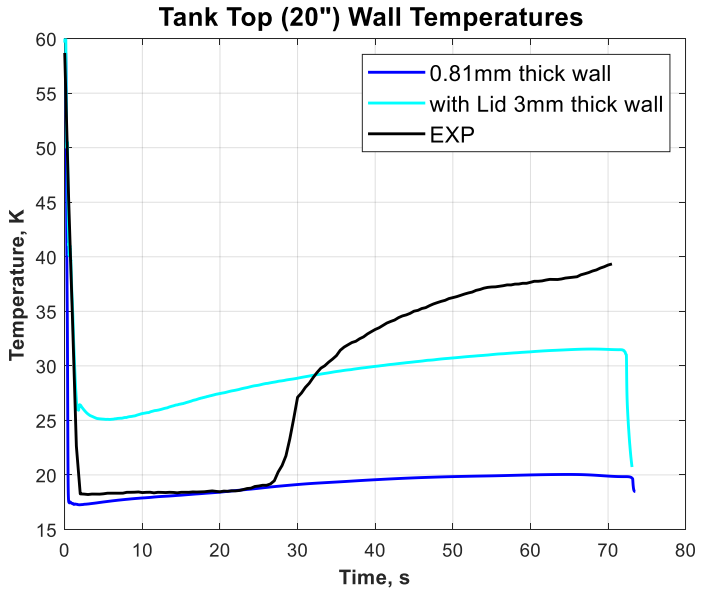
Effect of Geometric Configuration (3 of 4): Wall Temperatures

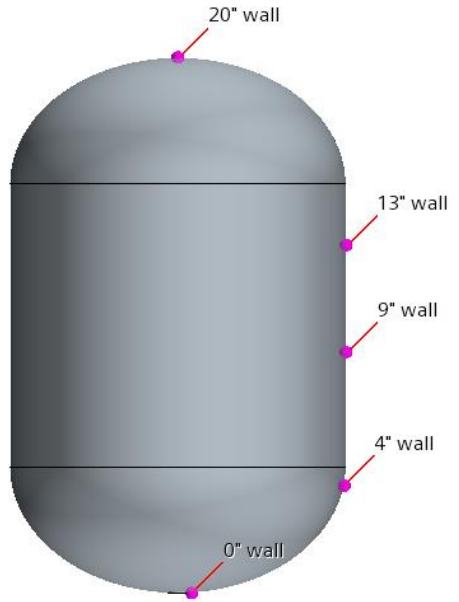




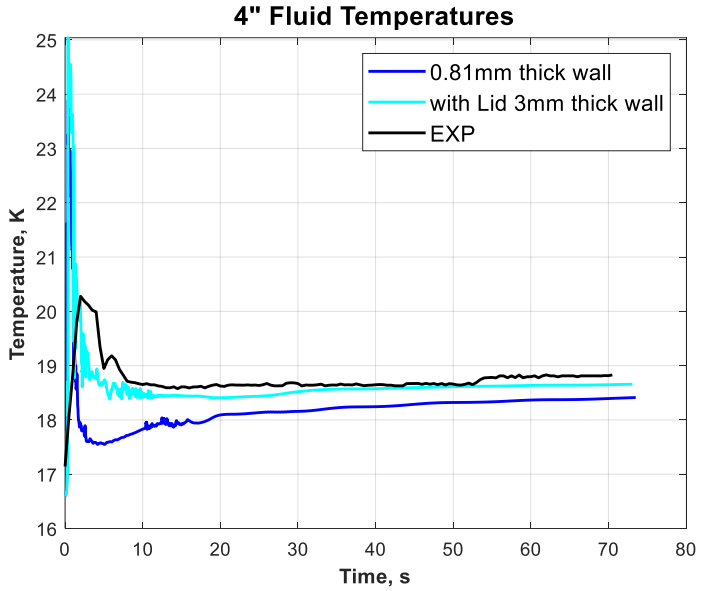


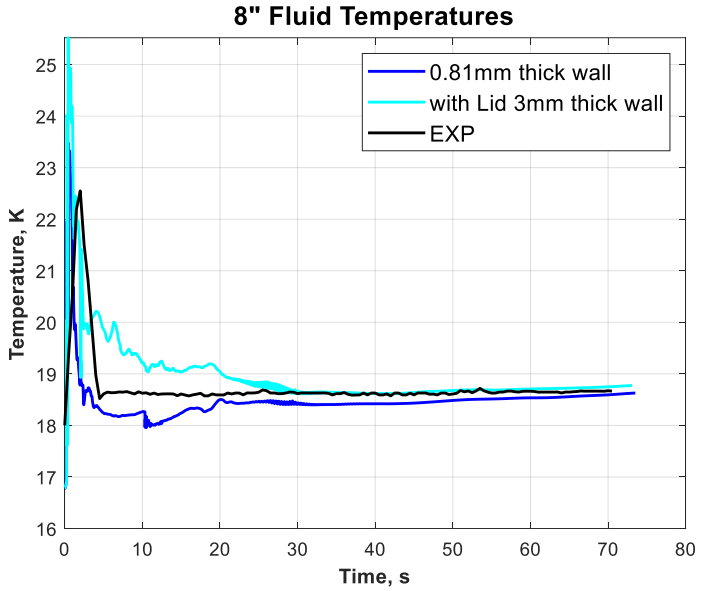


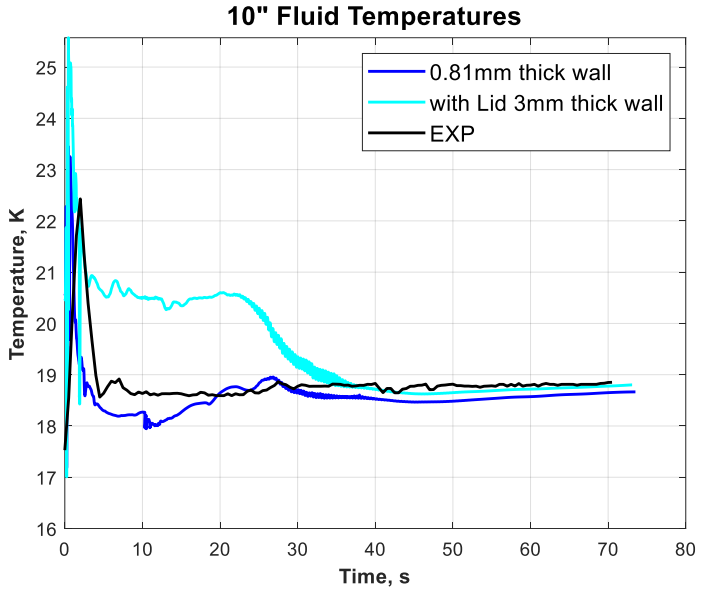


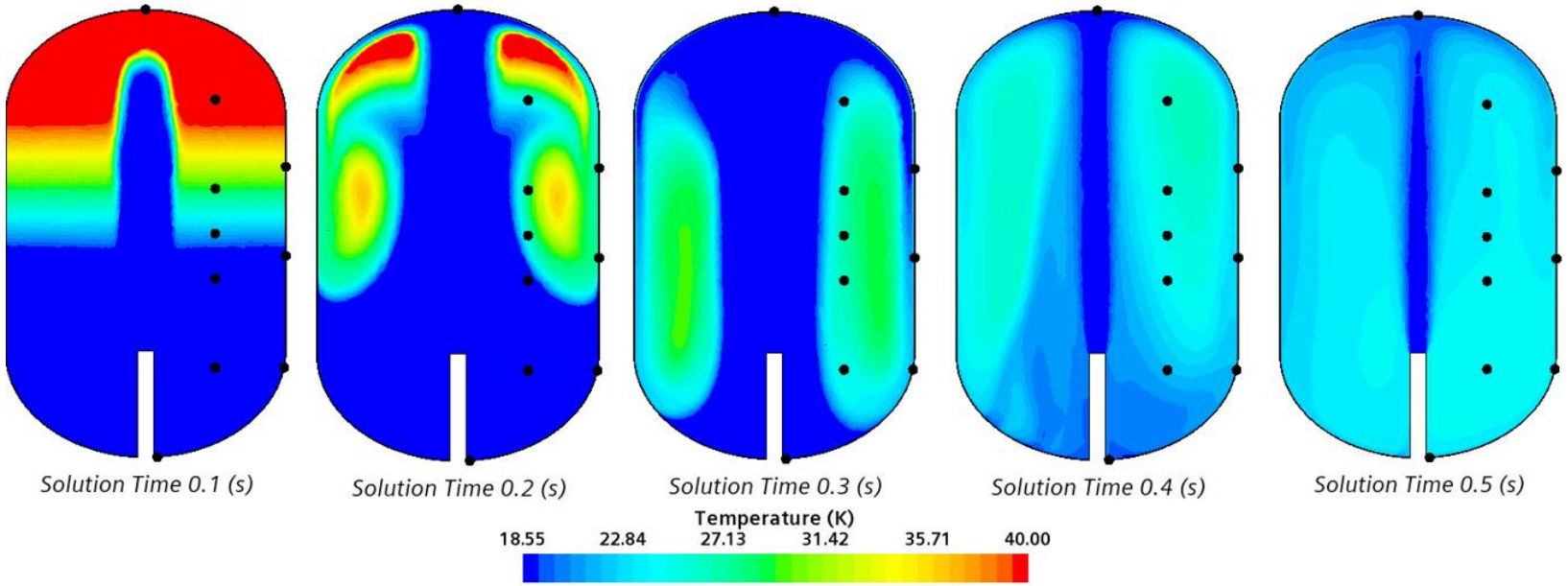


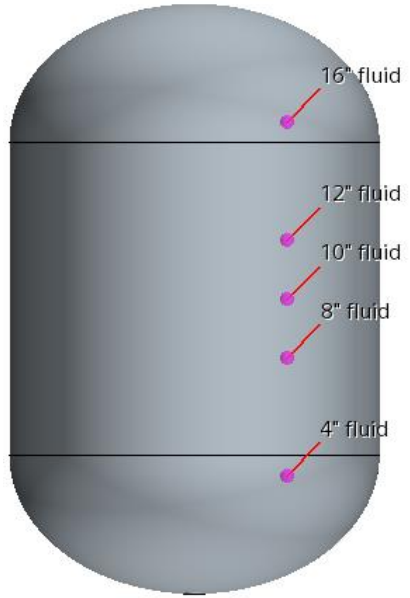
Effect of Geometric Configuration (4 of 4): Fluid Temperatures



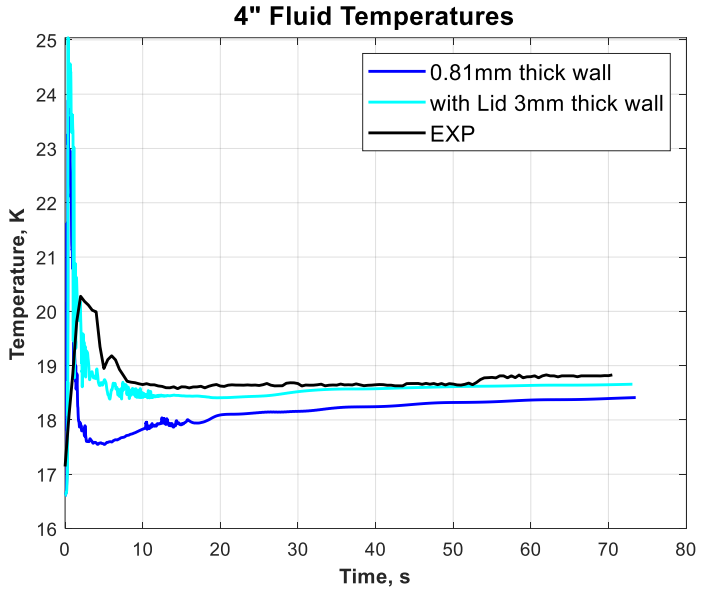


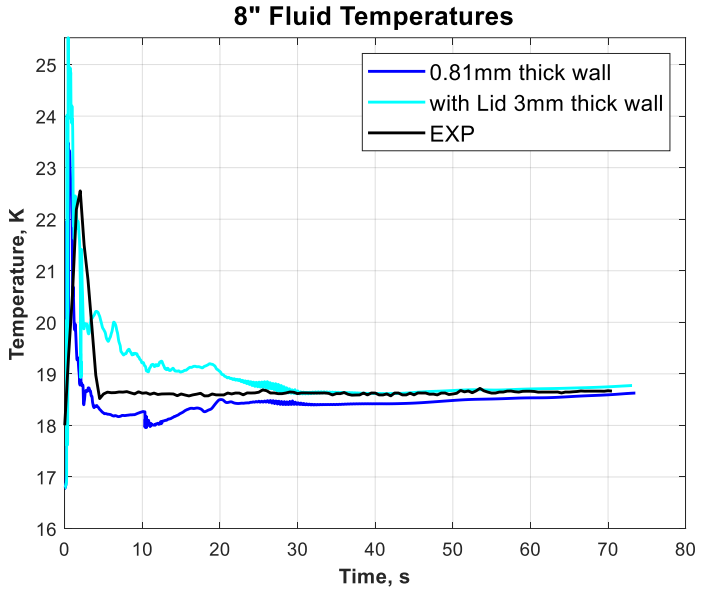


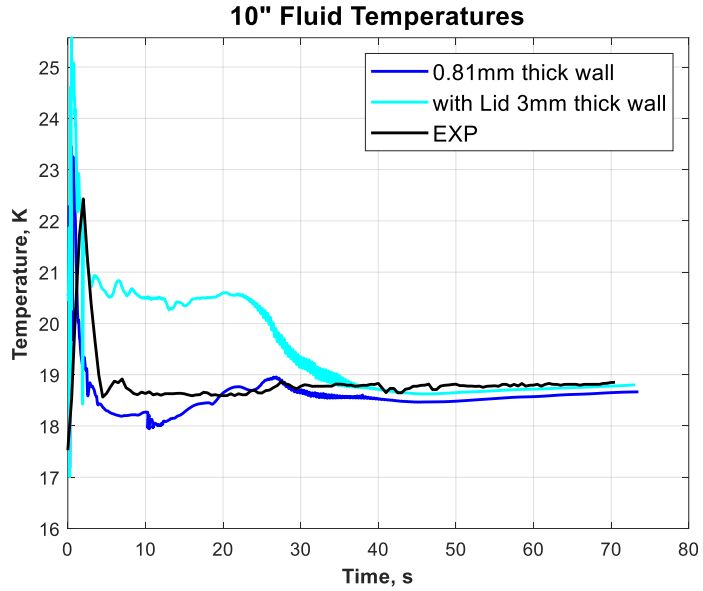


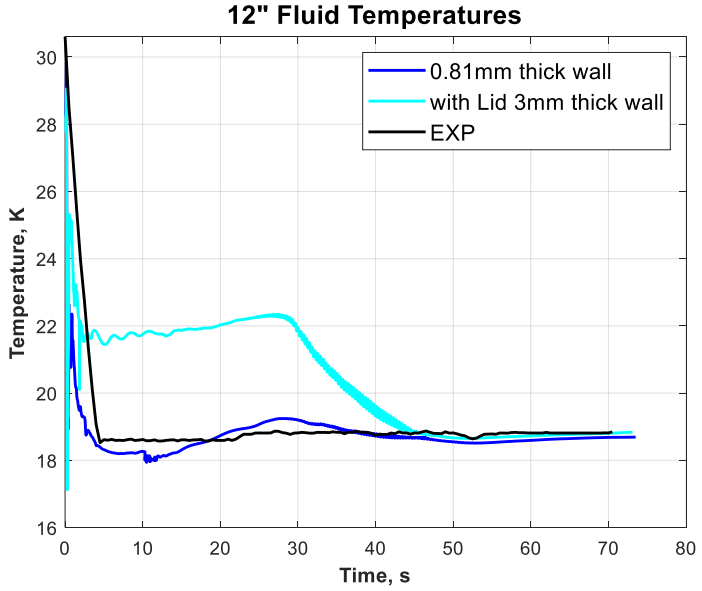


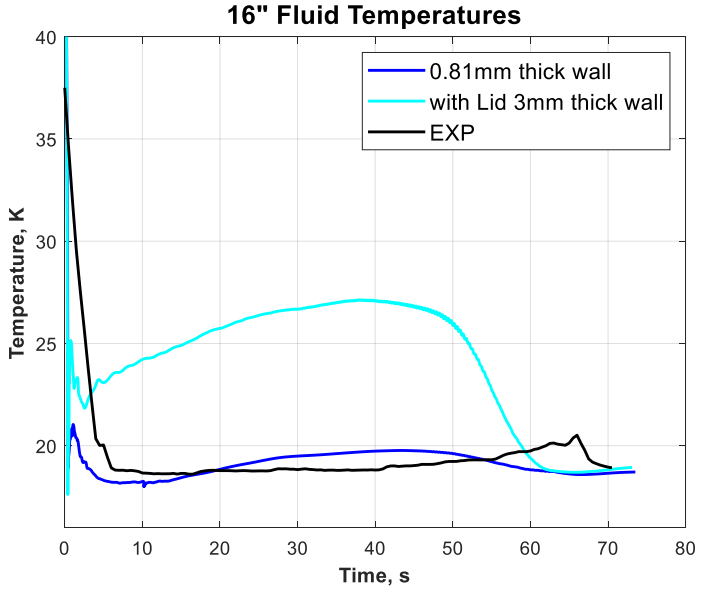
Effect of Geometric Configuration (4 of 4): Fluid Temperatures

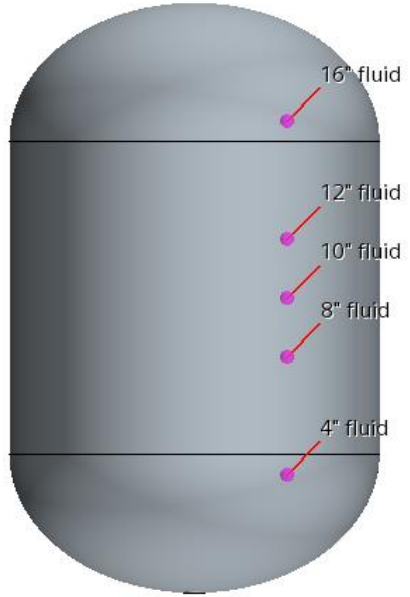






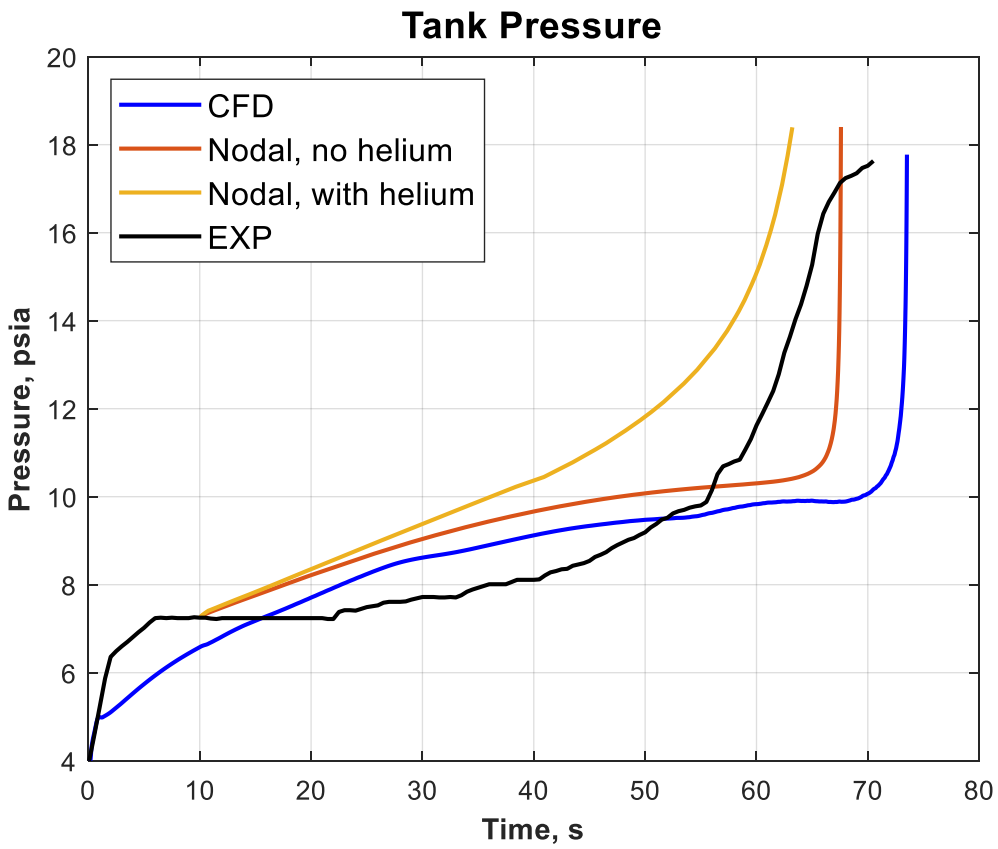






Presence of Helium

- None of the CFD cases considered accurately predicted the ullage compression near the end of the fill
  - Predicted final fill level = 99.6% vs. Experiment final fill level = 96%
- Helium gas used to purge system between tests may have become trapped in the tank and mitigated condensation near the end of the fill
- The effect of the presence of helium on the pressure signature was studied using a SINDA/FLUINT nodal model
- Mass fraction of helium in each of the ullage nodes was set to 10%
- Case without helium resembles CFD while case with helium resembles experiment



Conclusions and Future Work

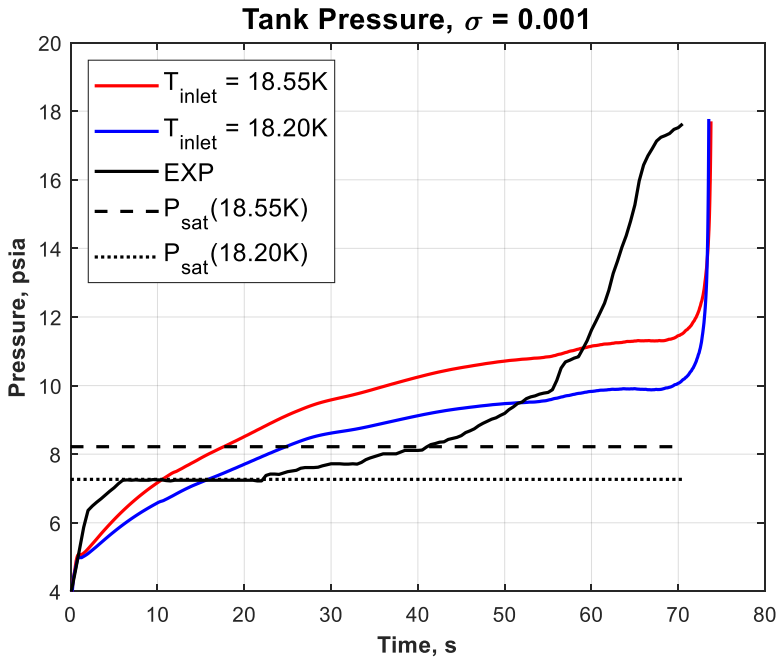
- **Conclusions**
  - The importance of implementing the time-dependent inlet void fraction and interfacial mass transfer on the predicted pressure was shown
  - Accurately measuring the inlet condition for CFM testing is extremely important for modeling considerations
  - Tank wall thickness yielded differences in the pressure and temperature signatures
  - None of the CFD cases considered accurately predicted the ullage compression near the end of the fill and may be due to the presence of helium (consistent between STAR-CCM+, Flow3D, and Fluent)

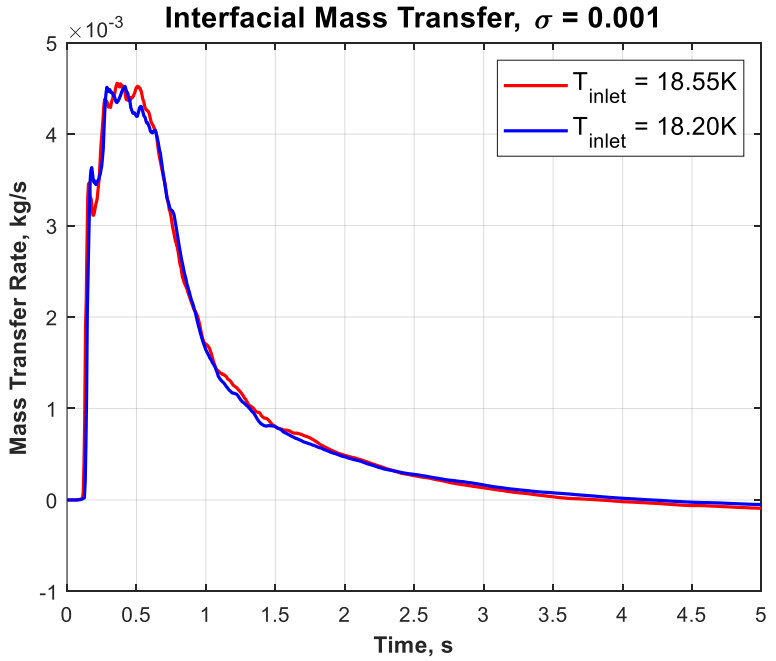
- **Future Work**
  - Amount of uncertainty made it difficult to assess model capability
  - Modeling a well-defined and well-instrumented tank chill and fill experiment

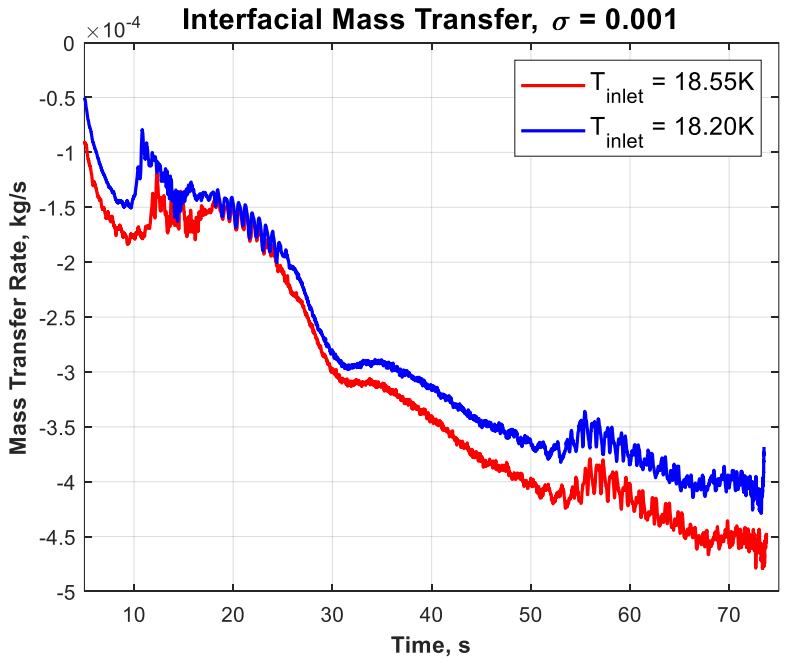
BACKUP

Effect of Inlet Temperature

- Two “final” inlet temperatures of 18.20 and 18.55 K were considered
  - Measurement uncertainty
  - Experimental tank pressure quickly levels off at value very close to saturation pressure at 18.2 K
- Both cases under-predict initial pressure rise, over-predict middle section, and fail to predict ullage compression until extremely high fill levels
  - Condensation is over-predicted near end of fill
  - Predicted final fill level = 99.6% vs. Experiment final fill level = 96%
- Identical evaporation rates, higher temp yields higher condensation rate due to higher pressure

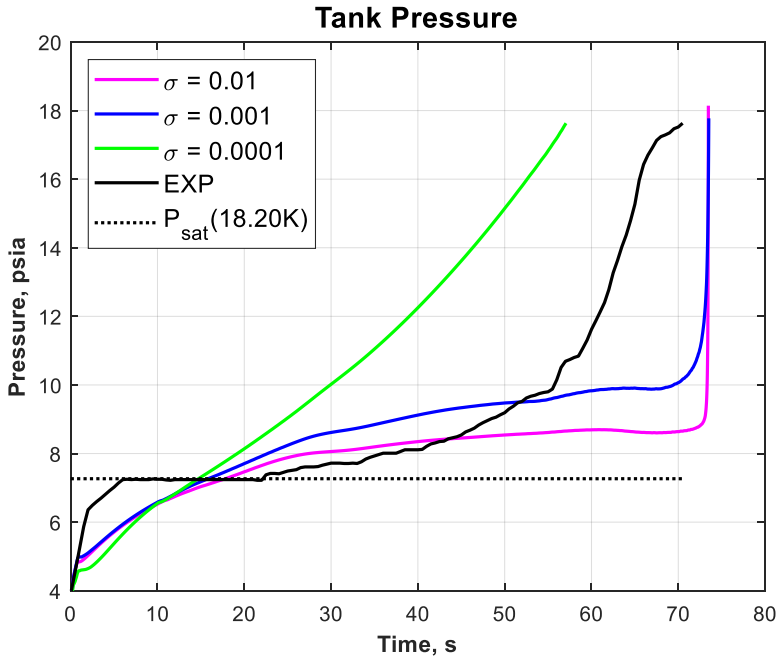


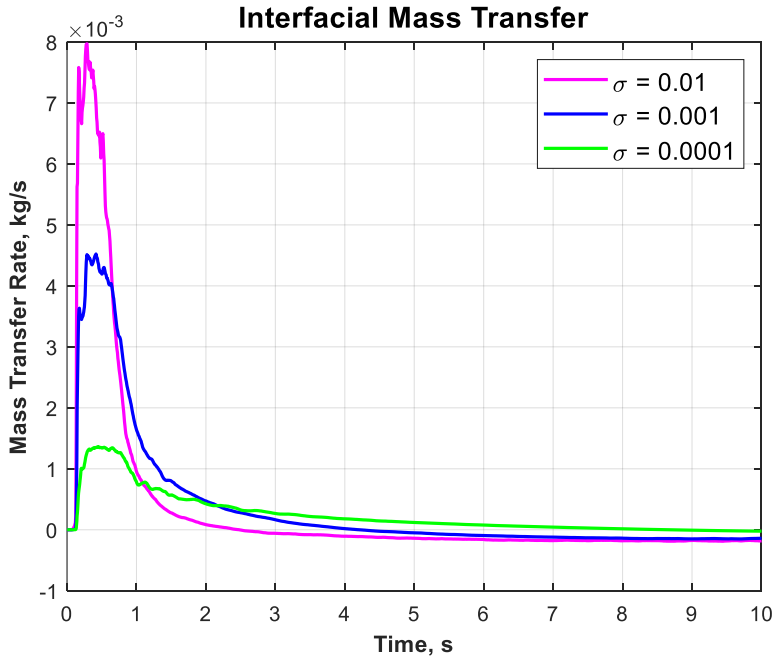


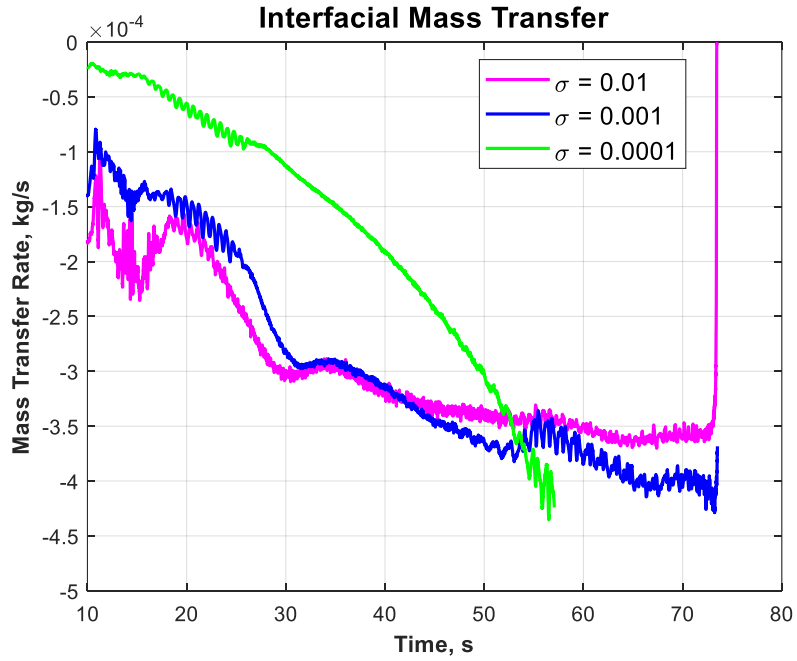


Effect of Accommodation Coefficient

- Larger AC yields higher evaporation rates during beginning of fill and higher condensation rates during majority of fill
- Although σ = 0.01 yields higher evaporation rates, mass transfer transitions to condensation earlier than the lower values
- Lower AC values yield higher condensation rates near end of tank fill because $(P_{sat} - P_{abs})$ is large and negative







Effect of Turbulence Model

