TFAWS Fluids Paper Session

Numerical Prediction of Heat Transfer Coefficients during Xenon Tank Fill in Microgravity

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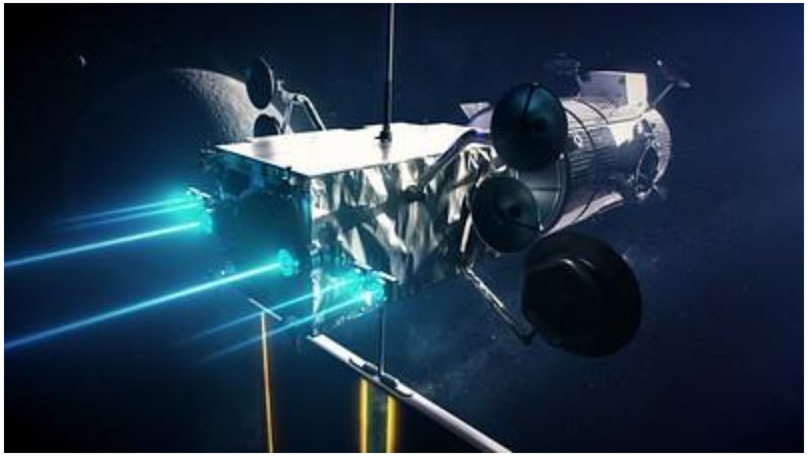
Motivation

- Gateway's Power and Propulsion Element (PPE) will employ solar electric propulsion using Xenon as fuel
- Xenon will be stored in a composite overwrapped pressure vessel (COPV)
- During the propellant refueling process, xenon will compress causing the fluid and tank walls to warm leading to concern:
  - Bond between the tank liner and composite may degrade at elevated temperatures
  - COPV may exceed design pressure
- Numerical models have the potential to predict the PPE tank performance during the refuel operation



Power and Propulsion Element firing its thrusters
nasa.gov/gateway/images

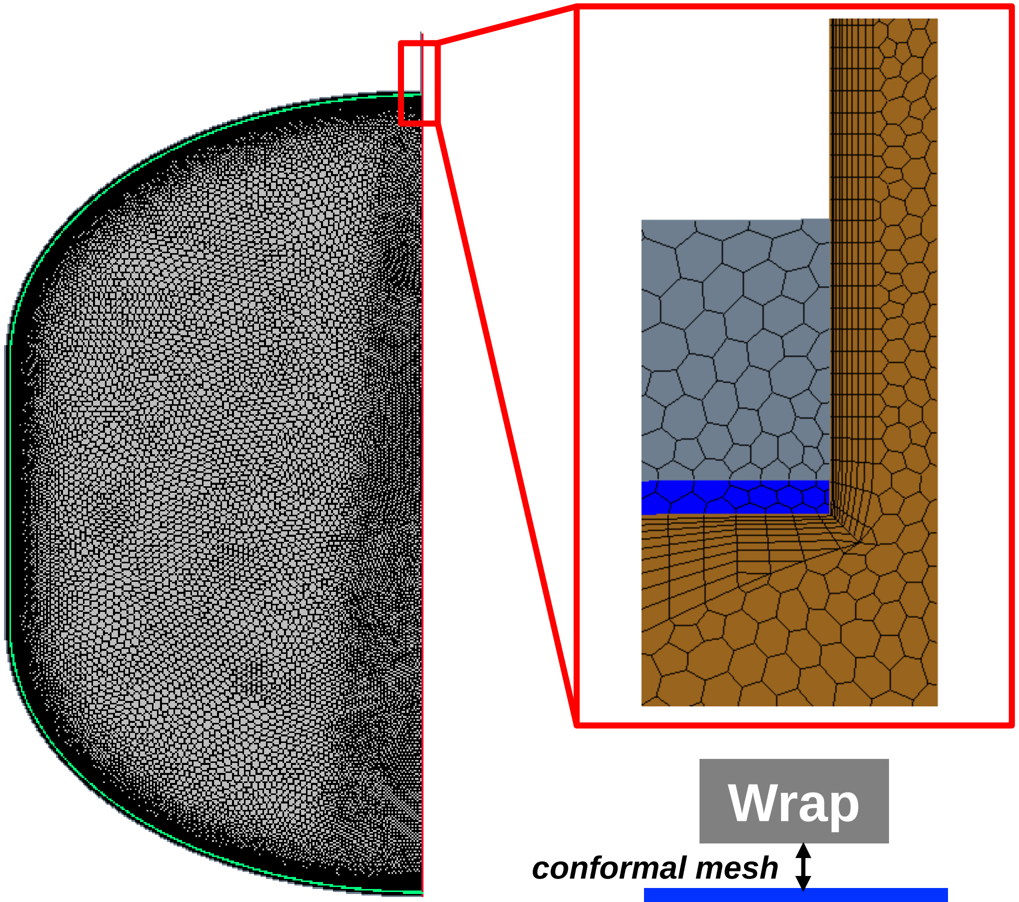
Gateway orbiting the Moon
nasa.gov/gateway/images

Objectives and Challenges

- Objectives:
  - Develop understanding of the heat transfer environment in microgravity during Xenon refueling for the PPE tank through CFD analysis (nat. conv is diminished)
  - Predict the heat transfer coefficients (HTCs) between the Xenon gas and COPV tank walls
  - Implement HTCs predicted by CFD into Thermal Desktop models to increase accuracy
  - Investigate a mixed convection Nusselt Number correlation to be used for predictive TD models

- Challenges:
  - Non-ideal gas behavior (highly nonlinear fluid props)
  - Xenon operates near & above critical point during refuel operation
  - Lack of experimental flight data

Computational Domain and Mesh



Conjugate heat transfer

62.2k cells

Wrap

conformal mesh ↕

Al Liner

conformal mesh ↕

Fluid Region

Assumes constant solid thickness

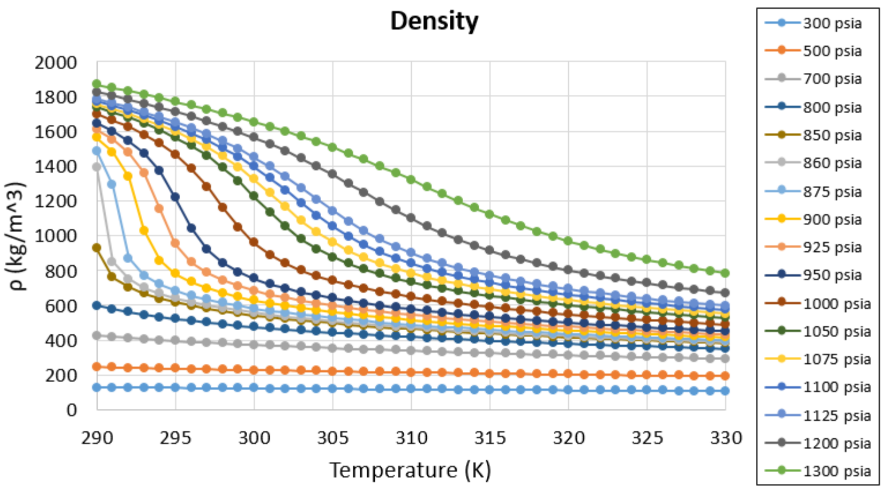
Model Setup

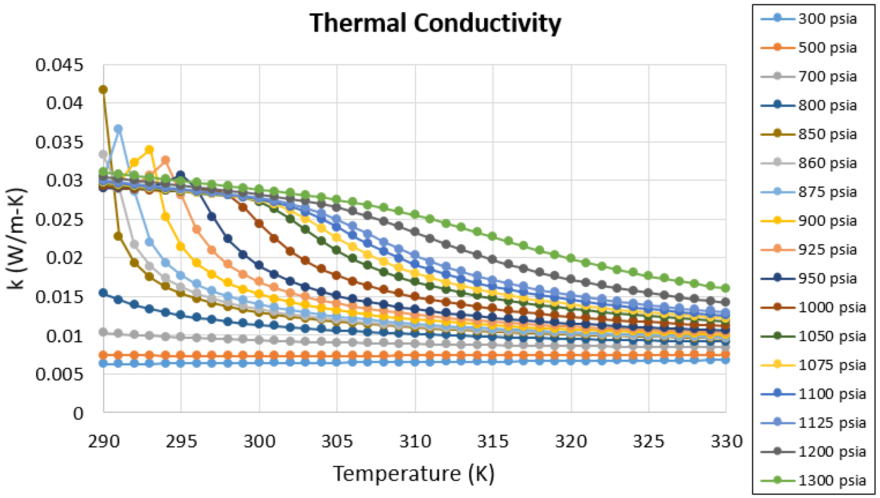
- Siemens Simcenter STAR-CCM+ commercial CFD code
- 2D axisymmetric
- Unsteady, $\Delta t = 0.25$ s
- 2nd order numerical schemes
- Microgravity @ 10^{-6} m/s^2
  - NRHO orbit
- Tank Rayleigh# → Laminar, Transfer Tube Flow → Turbulent
  - k-ω Menter SST
- Conjugate Heat Transfer with Al 6061 Liner and Wrap
  - Assumes perfect contact between liner and wrap
- Constant solid properties
  - Wrap properties are volume-weighted accounting for fiber and resin
- User-defined Equation of State for Xenon gas
  - Compressible gas with Temperature & Pressure-dependent fluid properties

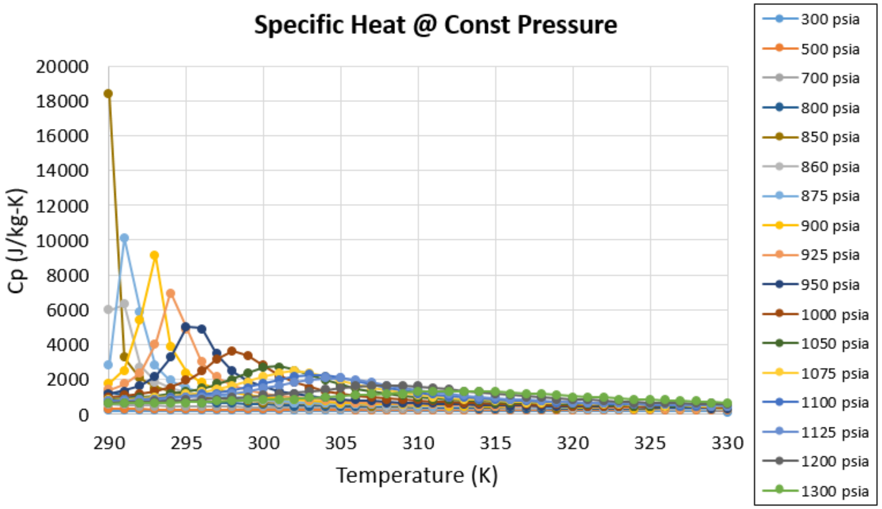
Xenon Fluid Properties

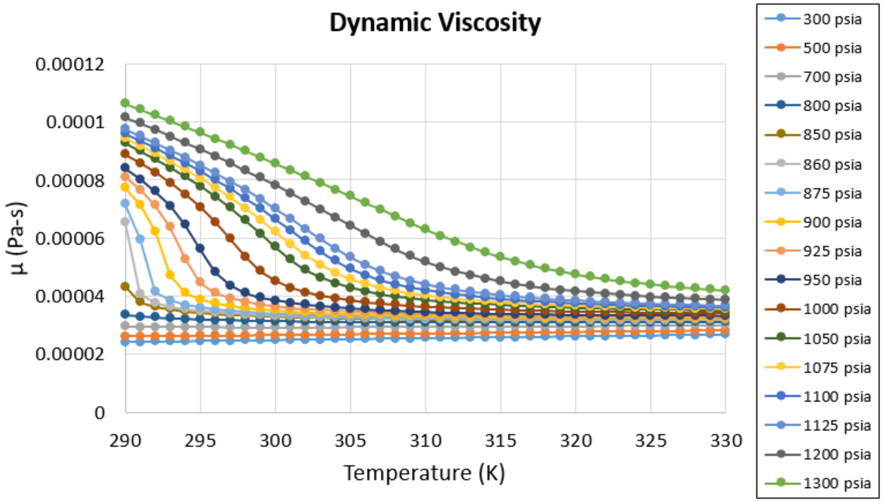
Xenon Critical Point:
T_{crit} = 289.7 K
P_{crit} = 847 psia

via NIST REFPROP


Density

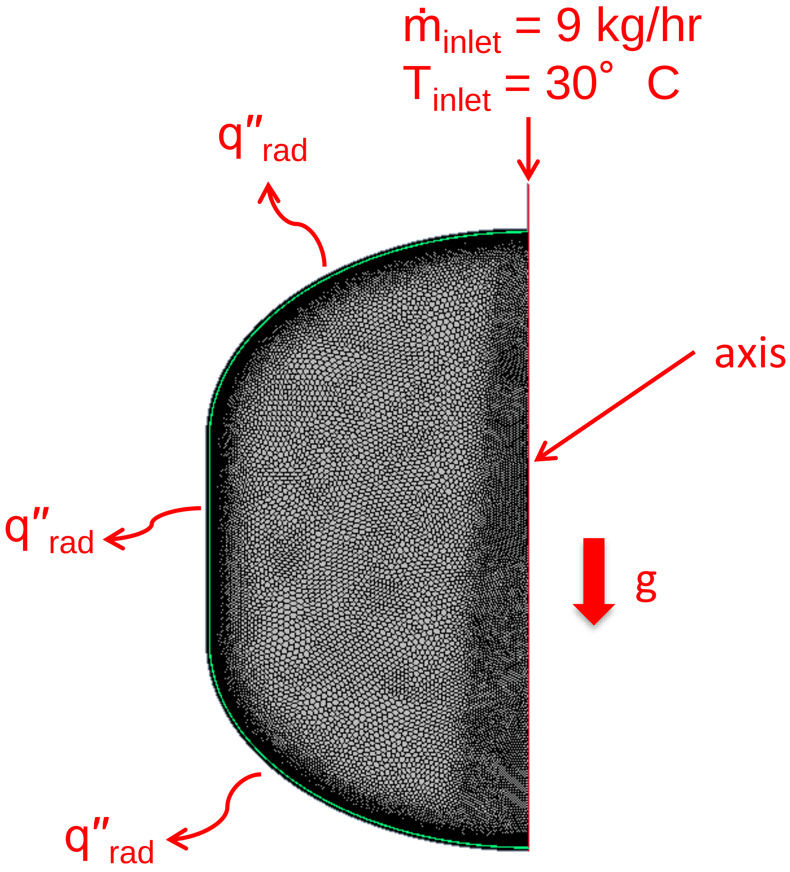

Thermal Conductivity


Specific Heat @ Const Pressure


Dynamic Viscosity

BCs and ICs

BCs



$$q''_{rad} = \frac{Q_{rad}}{A} = F\sigma\epsilon\left(T_{wall}^4 - T_{surr}^4\right) \quad \text{where}$$

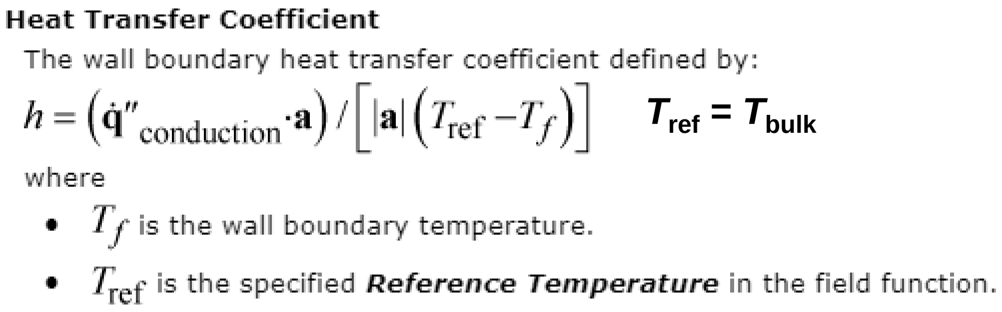
ε=0.9 (w/o MLI) & 0.03 (w/ MLI)
F=1
T_{surr} = 20°C

ICs

$T_{initial} = 30^\circ$ C (uniform)
$P_{initial}$ = 300 & 900 psia
$\mathbf{U}_{initial}$ = <0, 0> m/s

$M_{initial,\,300}$ = 103.4 kg
$M_{initial,\,900}$ = 497.2 kg

HTC as defined by STAR-CCM+



Heat Transfer Coefficient

The wall boundary heat transfer coefficient defined by:

$$h = \left(\dot{\mathbf{q}}''_{conduction} \cdot \mathbf{a}\right) / \left[|\mathbf{a}| \left(T_{ref} - T_f\right)\right]$$

where

- T_f is the wall boundary temperature.
- T_{ref} is the specified ***Reference Temperature*** in the field function.

$\boldsymbol{T_{ref} = T_{bulk}}$

From literature, internal flows typically use bulk fluid temperature (T_{bulk}) as the reference temperature for HTC calcs

CFD Simulation Cases

Case #	Starting Gas Temp (°C)	Inlet Flow Constant Temperature (°C)	Bus Heat Rejection Temp (°C)	Mass Flow Rate (kg/hr)	Initial Pressure (psia)	Tank Emissivity
1	30	30	20	9	300	0.9
2	30	30	20	9	900	0.9
3	30	30	20	9	300	0.03
4	30	30	20	9	900	0.03

ε=0.9 (w/o MLI) & 0.03 (w/ MLI)

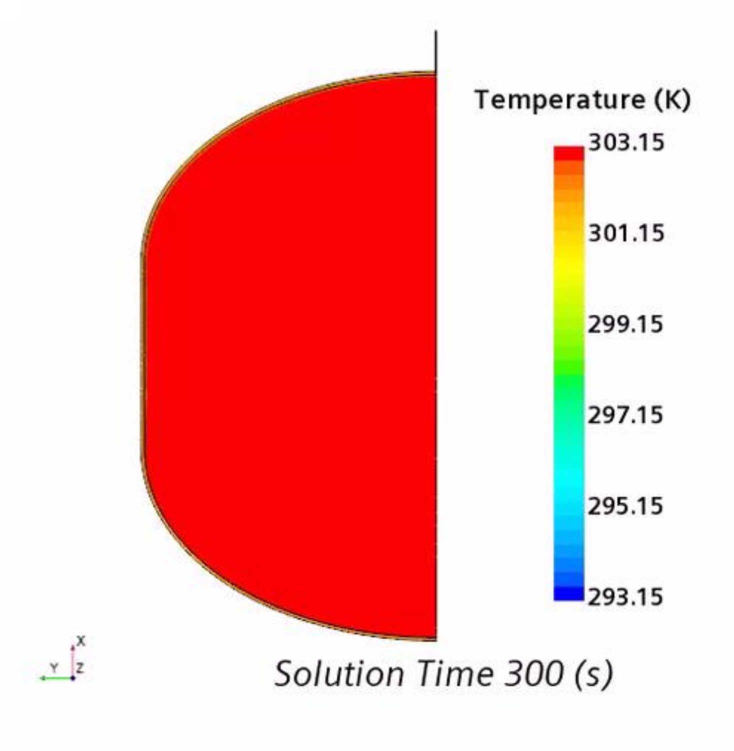
- Two Thermal Desktop (Multi-Node) simulations were run for each Case listed above:
  1. Assuming pure conduction, Nu# = 1 (Multi-Node Cond)
  2. Time-dependent HTCs predicted by CFD implemented into TD using curve fit (Multi-Node Conv)
- Pure conduction was assumed since very little natural convection is anticipated in microgravity and convective environment was unknown
- TD and CFD should produce similar results with the same HTCs

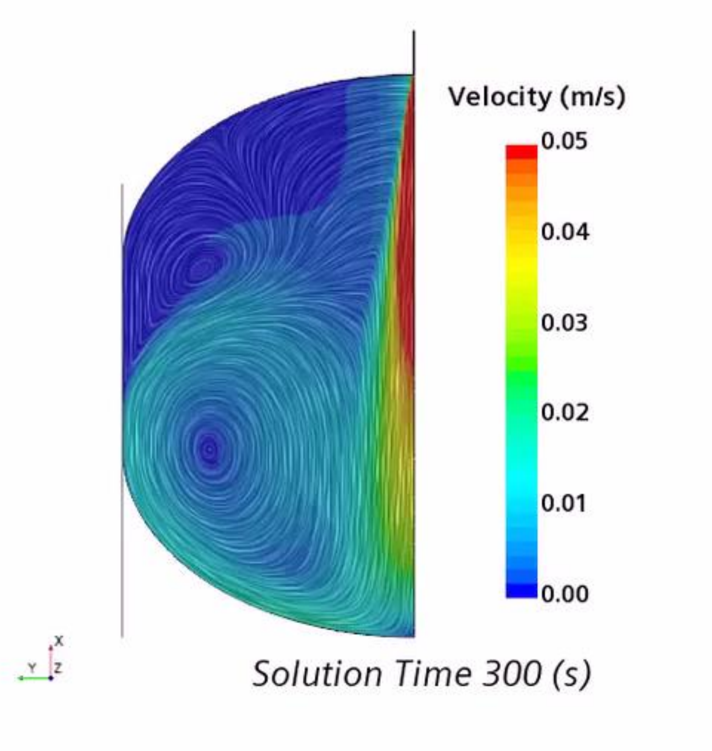
400 kg refuel at 300 psia NO MLI (ε = 0.9)

CASE 1

Temperature and Flowfield Evolutions

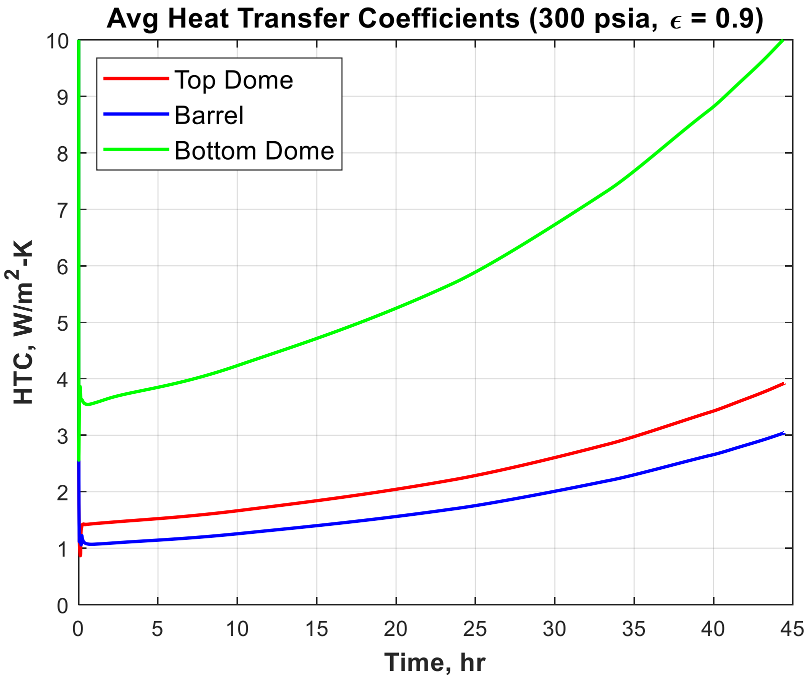
- Video frames exported every 5 min over 44.4 hours
- Injected fluid impinges on bottom dome providing warm fluid to the wall and forms primary vortex, inducing secondary vortex that provides heat transfer to top dome
- Jet penetration depth slowly retreats as injection velocity decreases with increasing pressure/density (constant mass flow rate)
- Although natural convection is negligible, forced convection is dominant

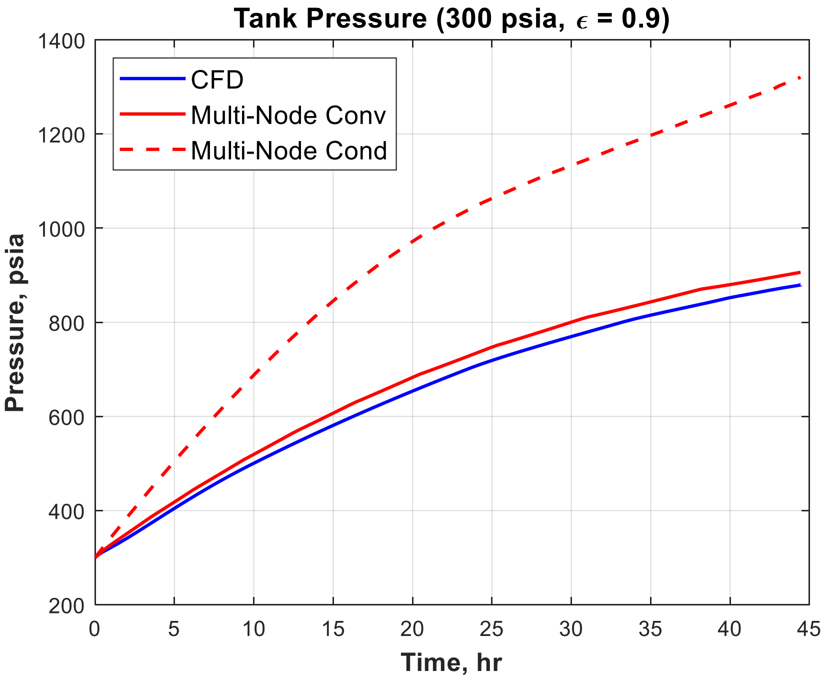




Refuel at Low Pressure NO MLI

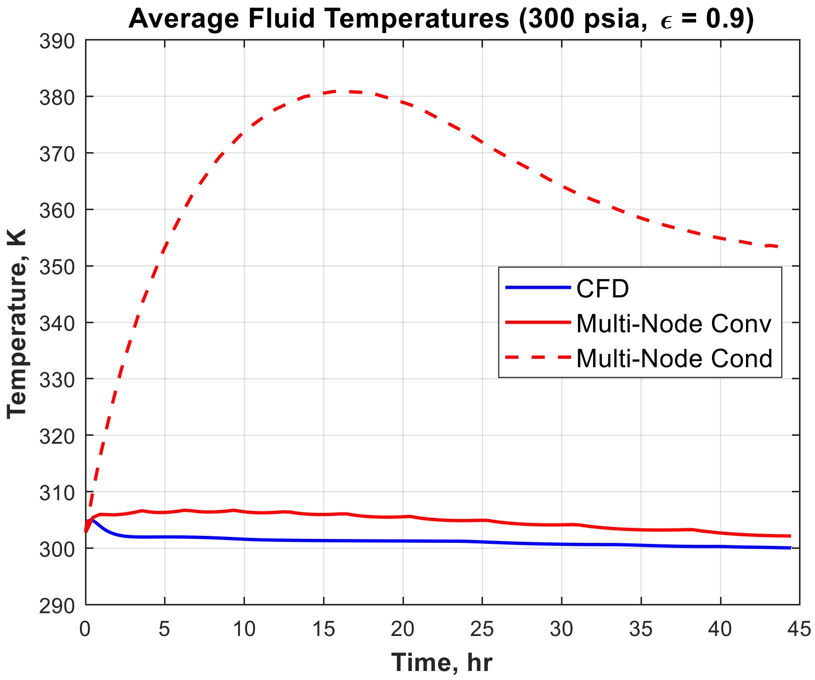
- HTCs increase over refueling duration due to increased thermal conductivity at higher pressures (↑ k, ↑ q_{cond}, ↑ HTC)
- Pure conduction over-predicts tank pressure (too conservative)
- CFD and TD convection results predict similar tank pressures with max difference ~30 psia (TD is slightly conservative)

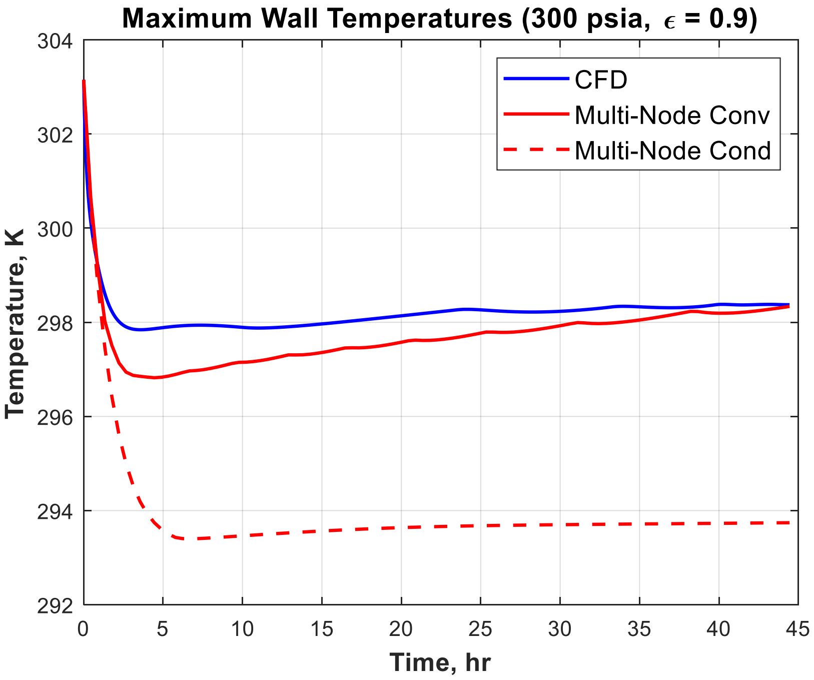




Refuel at Low Pressure NO MLI

- Pure conduction over-predicts fluid temp, while slightly under-predicting wall temperature (not conservative)
- TD convection and CFD temperatures agree within a few Kelvin
- Injected turbulent jet mixes tank and lowers fluid temp and pressure compared to pure conduction
- Pure conduction predicts lower wall temp – no convective heat transport to the wall while radiating to surroundings



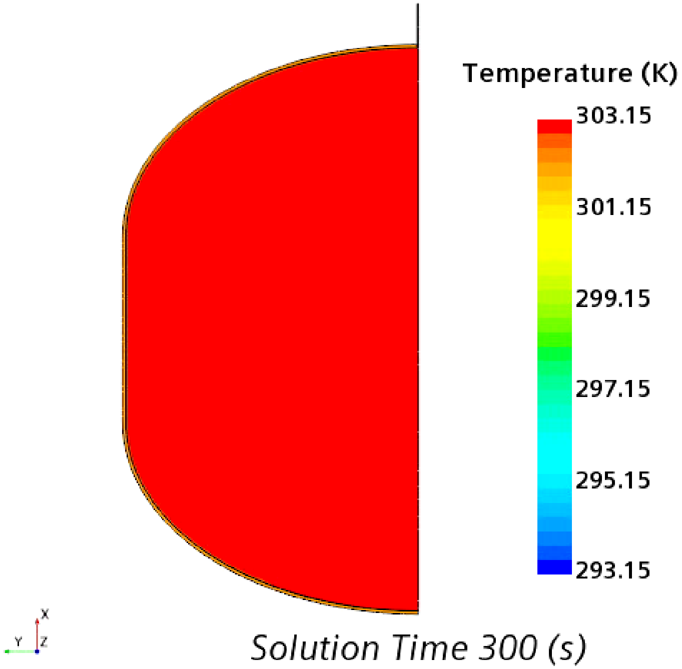


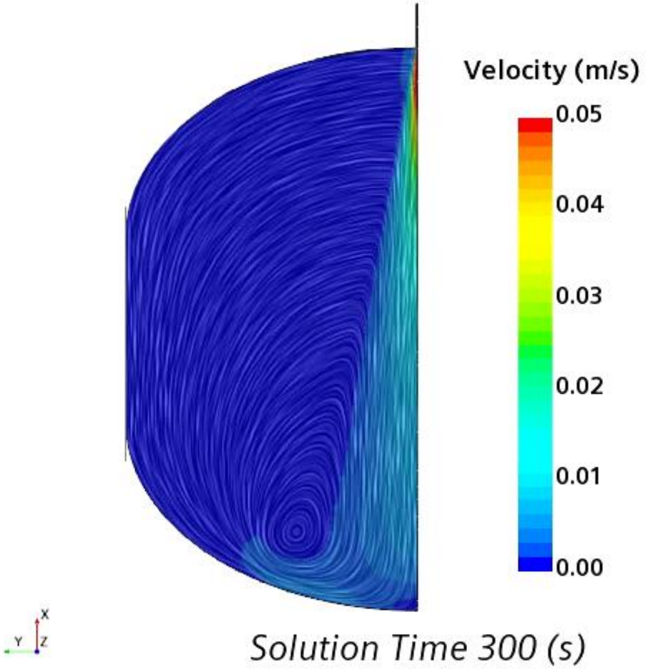
400 kg refuel at 900 psia NO MLI ($\varepsilon = 0.9$)

CASE 2

Temperature and Flowfield Evolutions

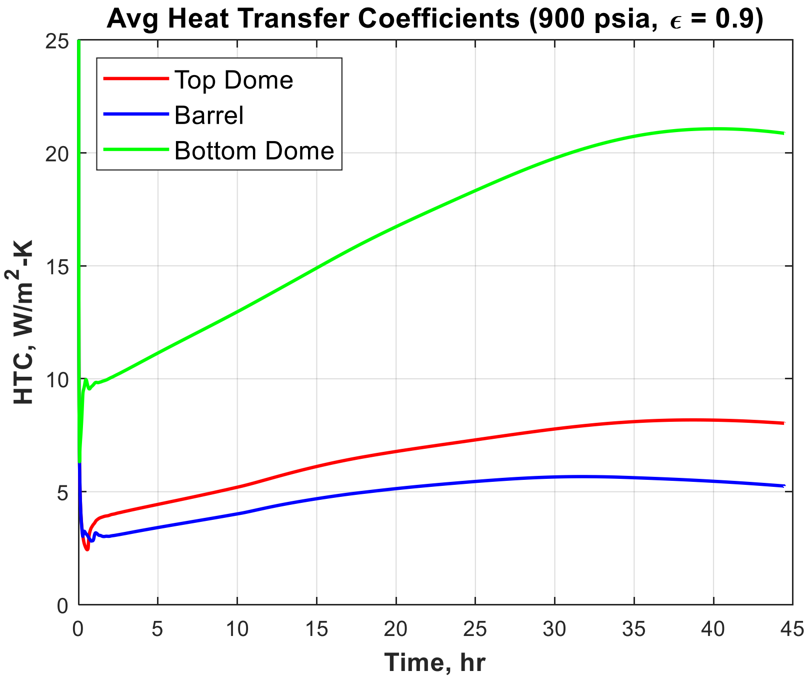
- Video frames exported every 5 min over 44.4 hours
- Tank temperature approaching isothermal condition with the coldest fluid on barrel wall, warmest at the tank inlet, and elevated temperature in the vortex cores
- Jet velocity reduction much less than 300 psia case; less change in pressure and density

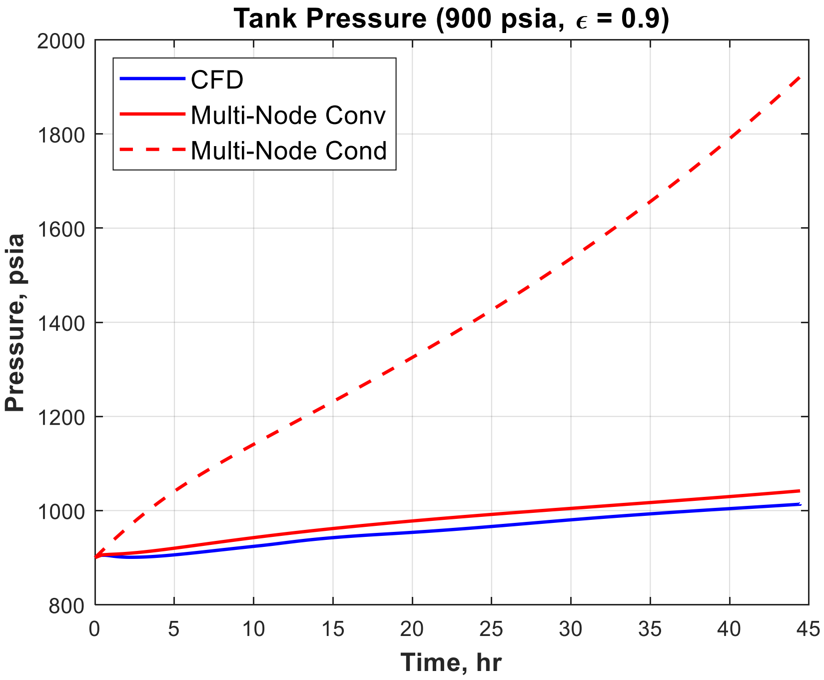




Refuel at High Pressure NO MLI

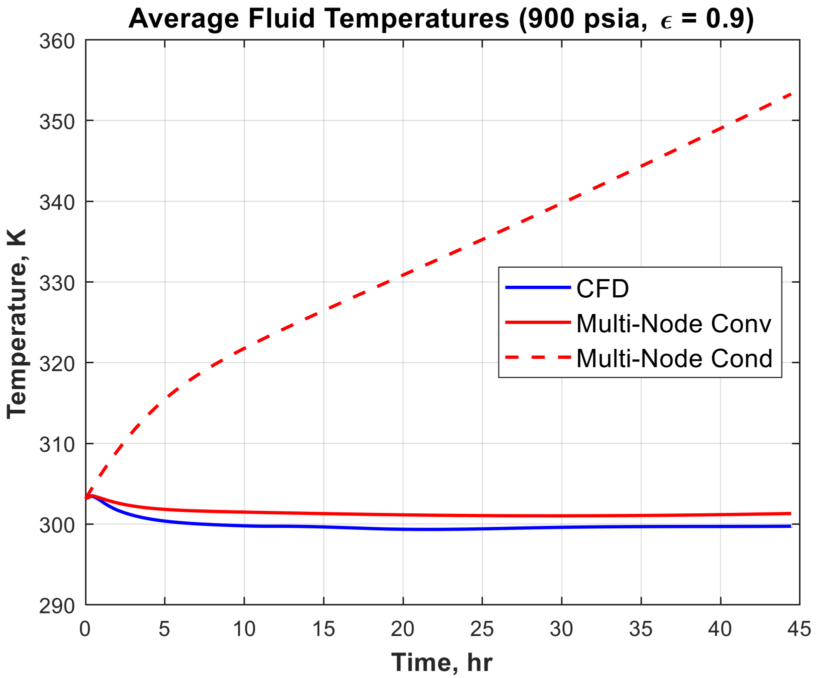
- HTCs are higher at elevated pressure compared to 300 psia case
- Pure conduction case significantly over-predicts CFD and TD convection
- CFD and TD convection results predict similar tank pressures with max difference ~30 psia (TD is slightly conservative)

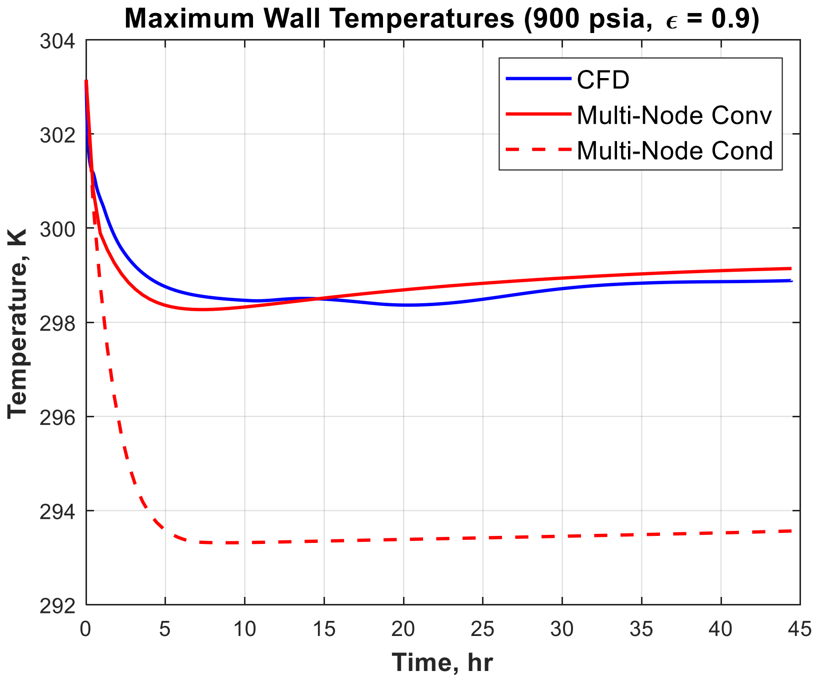




Refuel at High Pressure NO MLI

- Very similar conclusions as 300 psia case:
- Pure conduction over-predicts fluid temp, while slightly under-predicting wall temperature
- TD convection and CFD yield very similar temperatures



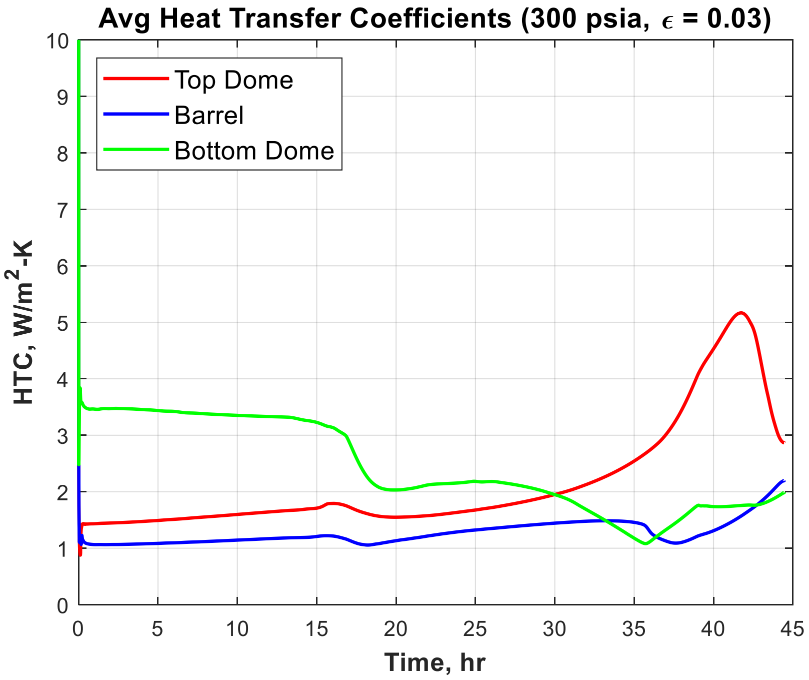


400 kg refuel at 300 psia w/ MLI (ε = 0.03)

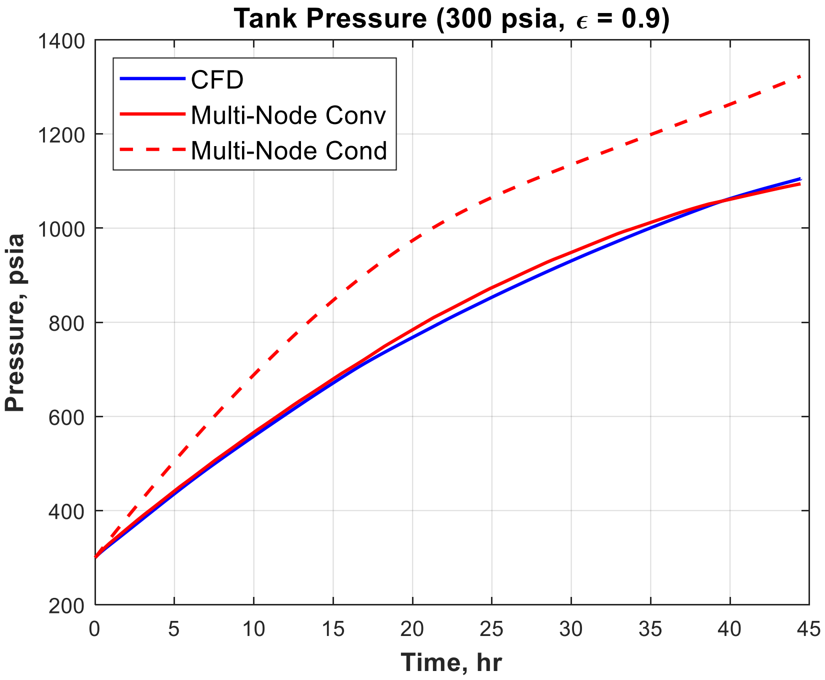
CASE 3

Refuel at Low Pressure w/ MLI

- With MLI yields much lower HTCs and dynamic time history
  - Fluid’s highly nonlinear properties near critical point
  - Wall heat transfer is not in equilibrium (Compressive Heating Rate > Radiative Cooling Rate)
- CFD and TD convection results agree well, pure conduction over-predicts



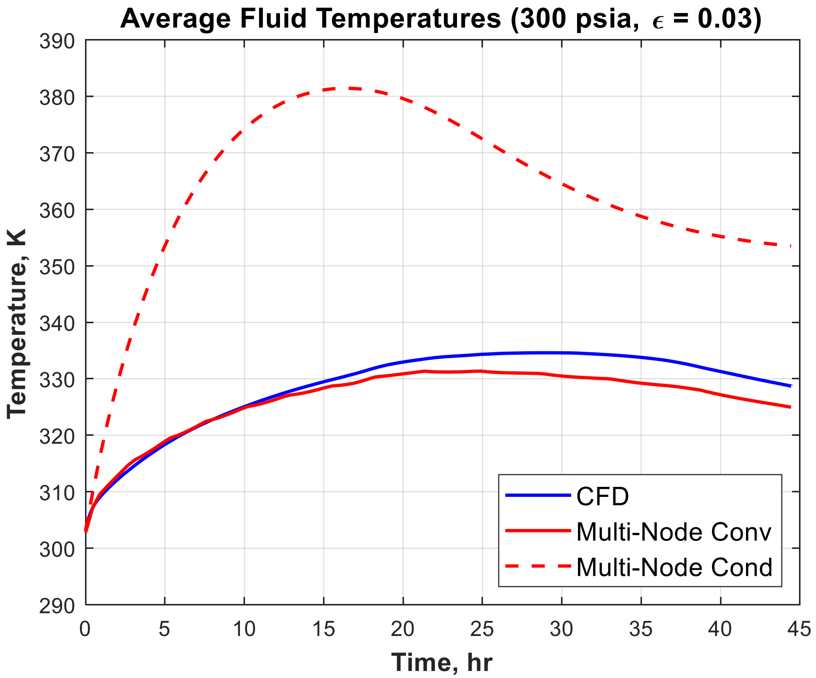
Avg Heat Transfer Coefficients (300 psia, ϵ = 0.03)

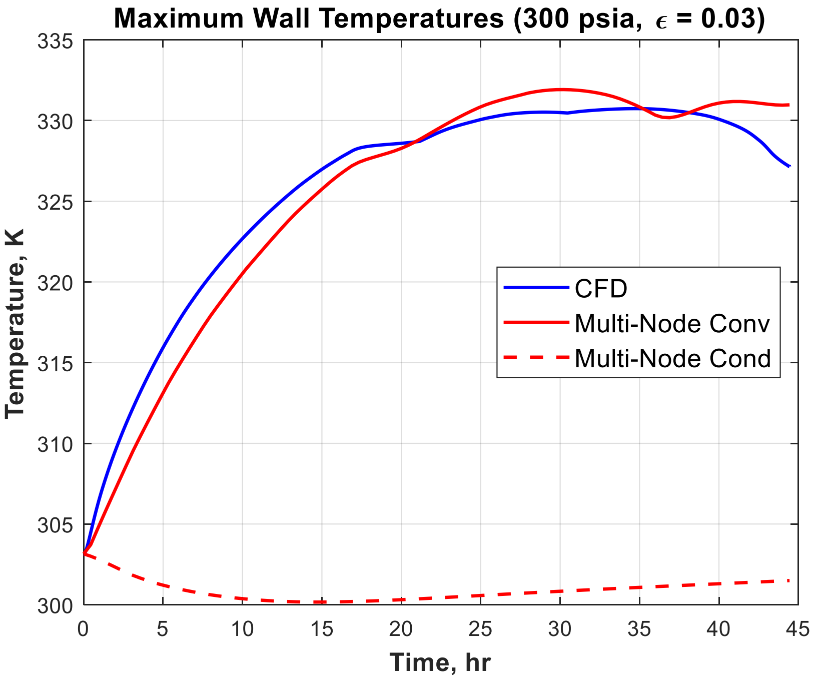


Tank Pressure (300 psia, ϵ = 0.9)

Refuel at Low Pressure w/ MLI

- TD convection and CFD yield very similar temperatures while conduction only results over-predict fluid temp, under-predict wall temp
- Fluid and wall temperatures increase above inlet temperature whereas temperatures dropped below inlet temperature for NO MLI case
  - Compressive Heating Rate > Radiative Cooling Rate due to reduced Emissivity



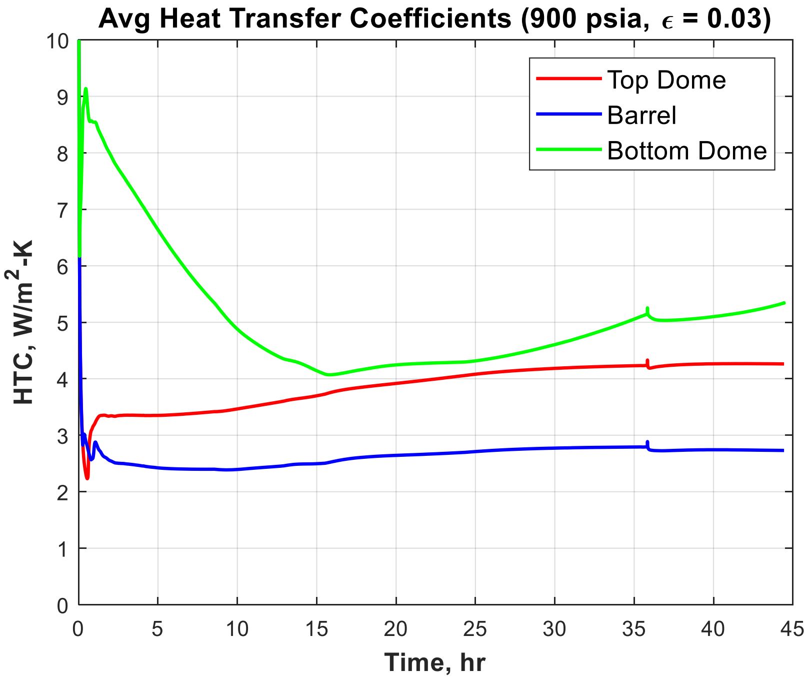


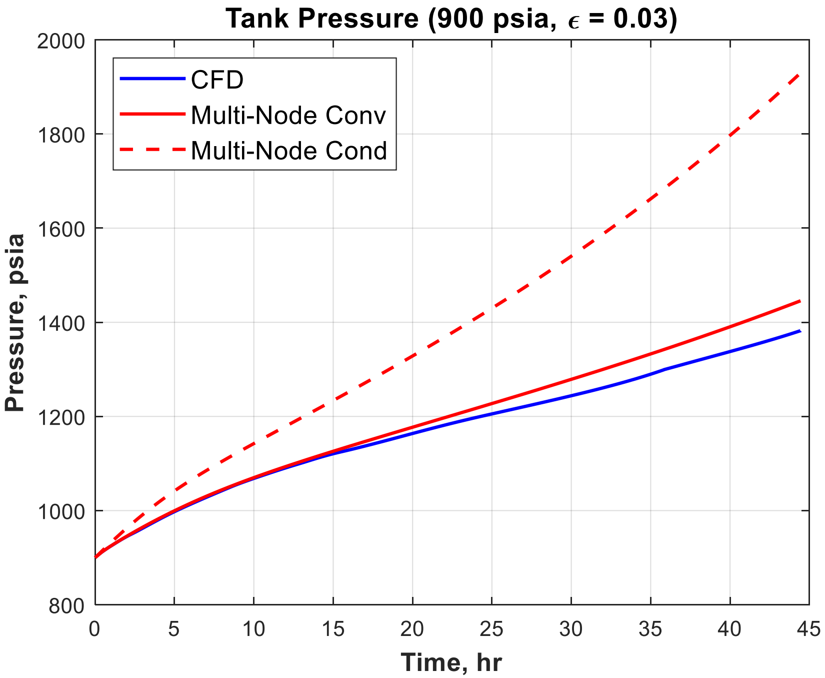
400 kg refuel at 900 psia w/ MLI (ε = 0.03)

CASE 4

Refuel at High Pressure w/ MLI

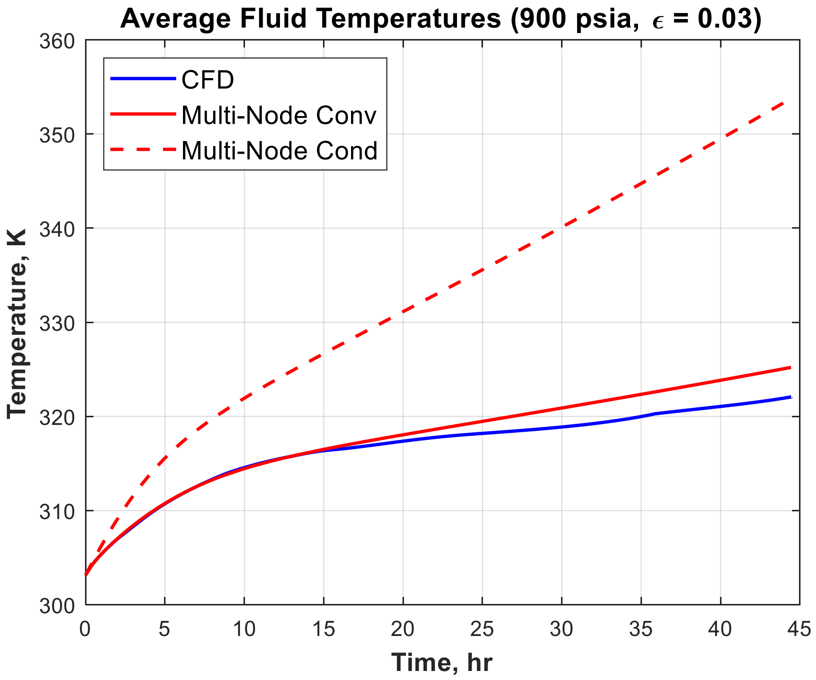
- With MLI yields much lower HTCs and dynamic time history
- Small blip in HTCs at hour 36 due to numerical instability potentially caused by fluid property interpolation
- CFD and TD convection results compare well for first 15 hours then begin to diverge

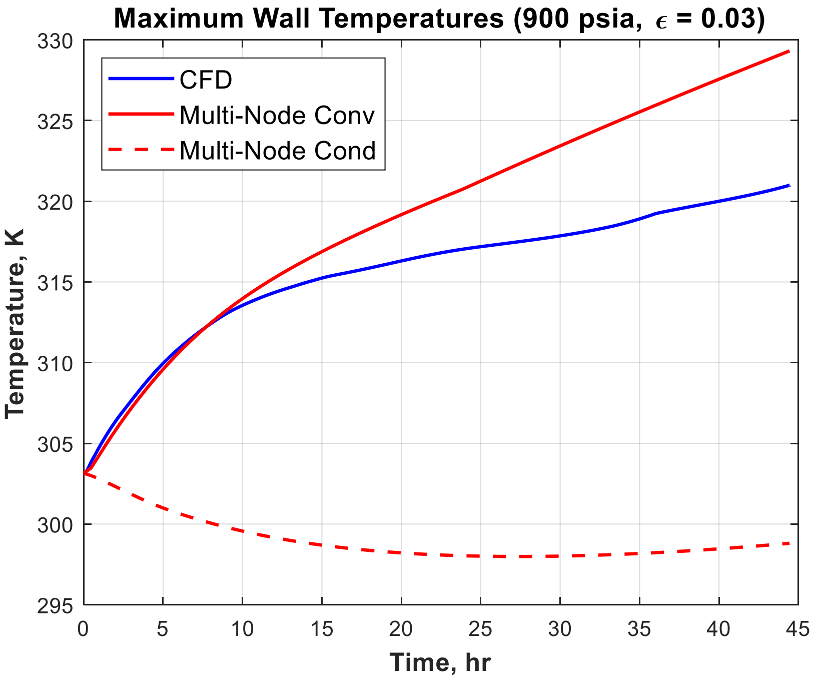




Refuel at High Pressure w/ MLI

- Same trends with pure conduction case
- CFD and TD convection results compare well for first 10-15 hours then begin to diverge





CONVECTIVE REGIME AND NU# CORRELATION

Convective Regime

$\frac{Ra}{PrRe^2} \gg 1$ → natural convection dominates

$\ll 1$ → forced convection dominates

≈ 1 → mixed convection

Pressure (psia)	Re_D	Pr	Ra	$\frac{Ra}{PrRe^2}$
300	25839	0.79	1.27e6	**0.002 << 1**
900	18999	1.88	1.72e8	**0.253 < 1**
1300	8208	2.68	2.98e8	**1.65 ≈ 1**

Forced convection dominates at lower pressure and regime approaches mixed convection at higher pressure

Mixed Convection Correlation

- Woodfield et al. published a mixed convection correlation for filling of tanks
- CFD HTCs and HTC via Nu# correlation are compared using quantities from the end of simulation (Case1)

Wall Boundary	CFD HTC (W/m²-K)	HTC via Nu# (W/m²-K)
Barrel	3.0	17.2
Bot Dome	10.0	16.6
Top Dome	3.9	16.7

$$h = \frac{k}{H}\left(\underbrace{0.56Re_d^{0.67}}_{\text{forced}} + \underbrace{0.104Ra_H^{0.352}}_{\text{free}}\right)$$

Source: *Woodfield et al. “Measurement of Averaged Heat Transfer Coefficients in High-Pressure Vessel during Charging with Hydrogen, Nitrogen or Argon Gas,” 2007*

- **Nu# correlation uses “global” Reynolds Number for HTC calcs**
- **CFD HTCs are computed by local flow conditions**
- **Tank dimensions may be out of valid range for correlation to be applicable**

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Tank	H/D	D/d
Test Tank	2.8	7.5
PPE Xenon Tank	0.96	231.3

This indicates PPE tank diameter is much wider than test tank

H = tank height
D = tank diameter
d = injector tube diameter

Conclusions and Future Work

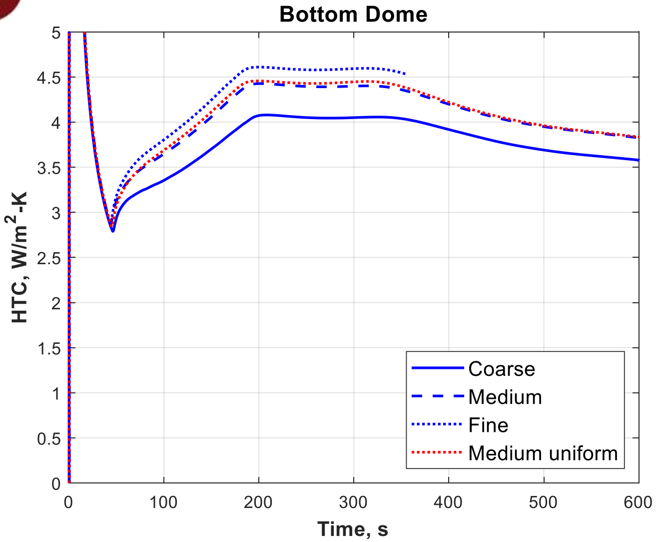
- Conclusions:
  - Xenon is weird – nonlinear fluid properties make it difficult to understand the heat transfer trends particularly when the wall heat transfer is not in equilibrium
  - User-defined Equation of State model performed better than anticipated but interpolation error is noticeable
  - Multi-Node model shows good agreement to CFD when using HTCs, pure conduction is too conservative
  - CFD simulation took 70 hours to complete running on 4 cores on the Pleiades supercomputer, TD finishes in <20 s
- Future Work:
  - How accurate do the HTCs need to be to yield similar thermal response predictions in Thermal Desktop compared to using CFD HTCs?
  - Can we get away with using the Mixed Convection Correlation by Woodfield et al.?
  - Validate models with future flight test data 🤞

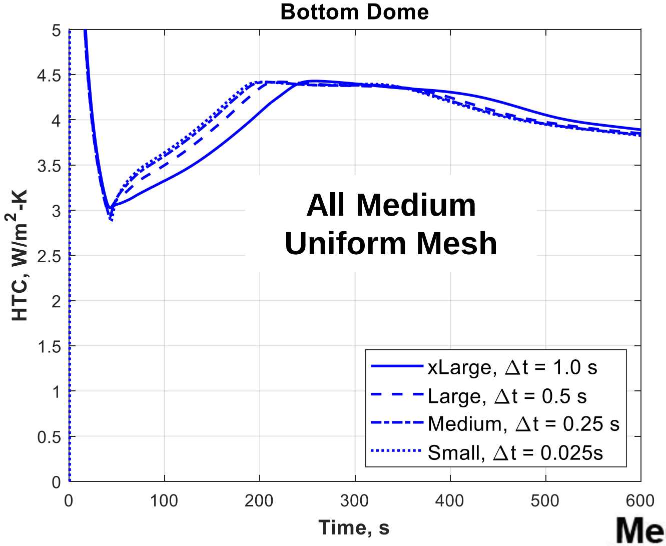
Acknowledgments

- Maxar Technologies
- NASA Advanced Supercomputing (NAS)
- Daniel Hauser (GRC/LTF) for collaboration on Thermal Desktop modeling

BACKUP

Mesh and Time Step Independence Study





All Medium Uniform Mesh

Coarse

20.8k cells

Medium

54.8k cells

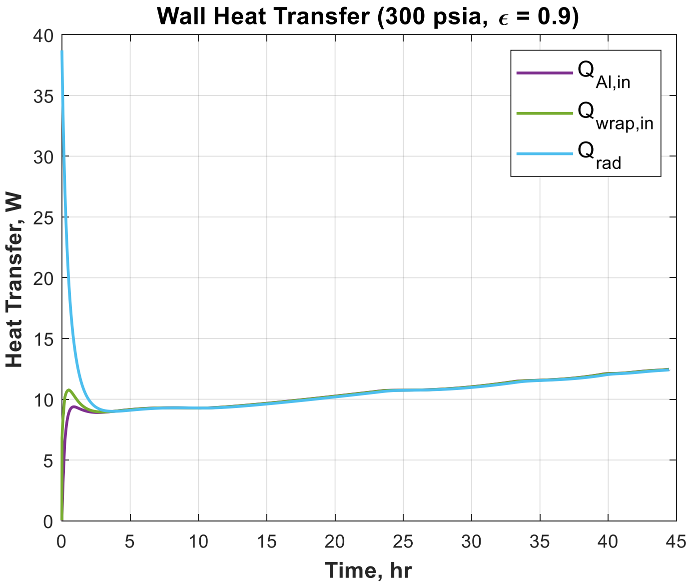
Fine

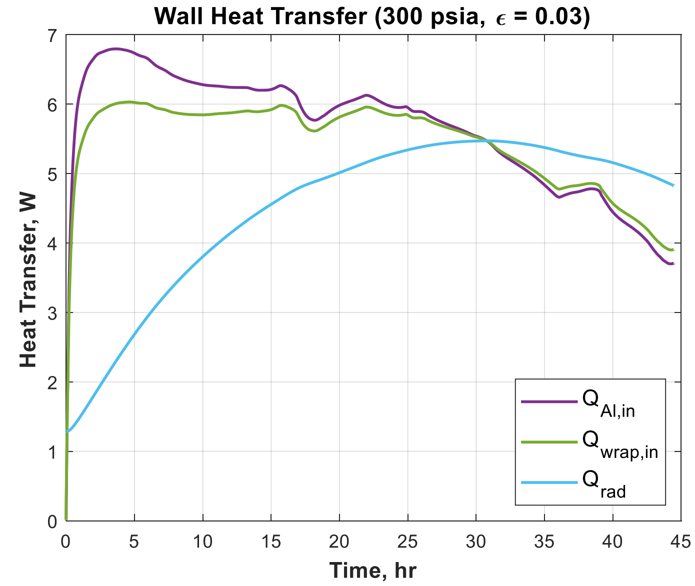
177.4k cells

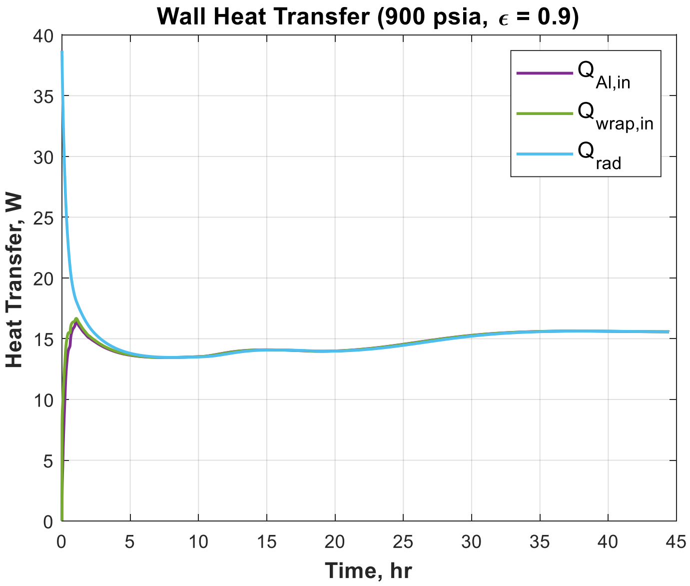
Medium Uniform

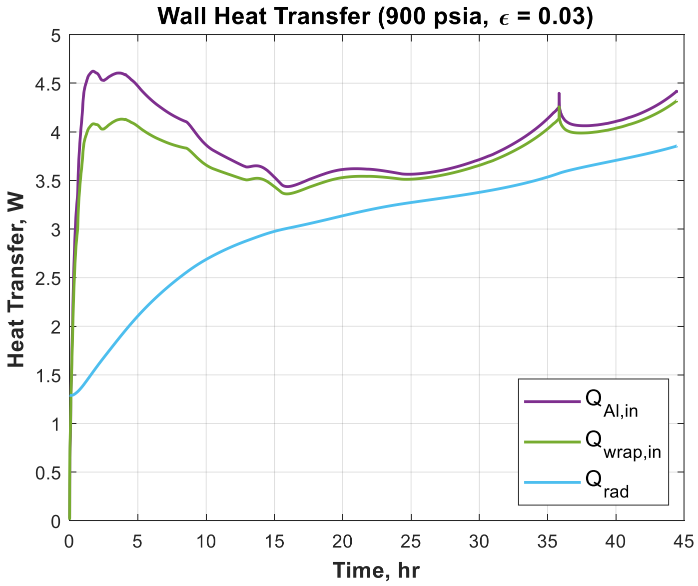
62.2k cells

Wall Heat Transfer

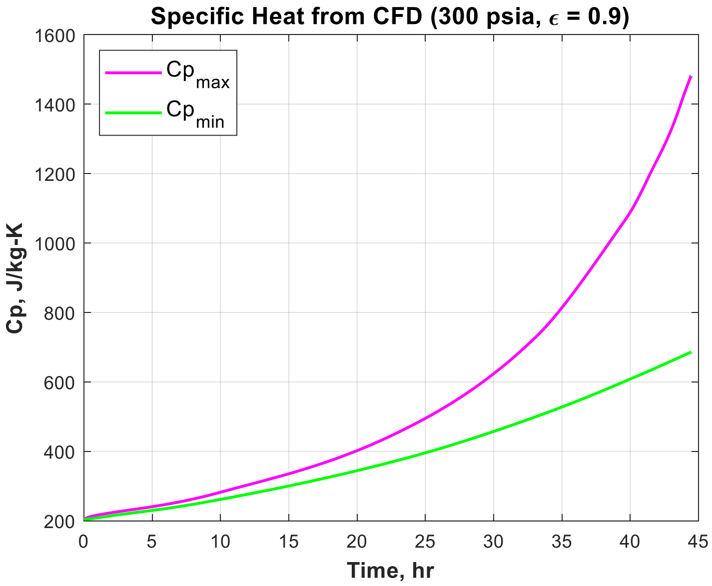

Wall Heat Transfer (300 psia, ϵ = 0.9)

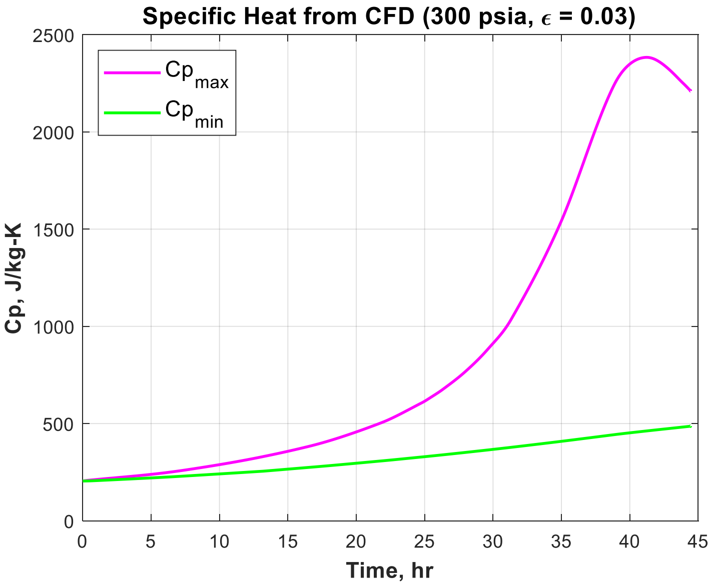

Wall Heat Transfer (300 psia, ϵ = 0.03)


Wall Heat Transfer (900 psia, ϵ = 0.9)


Wall Heat Transfer (900 psia, ϵ = 0.03)

Specific Heat Evolution


Specific Heat from CFD (300 psia, ϵ = 0.9)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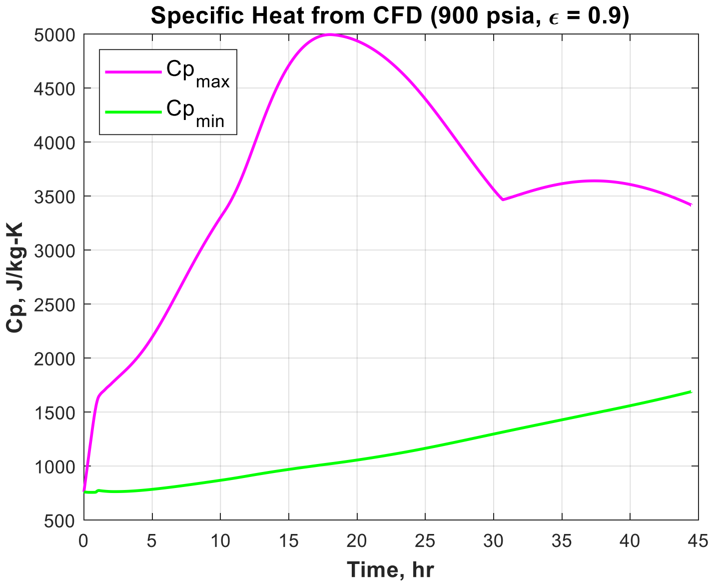
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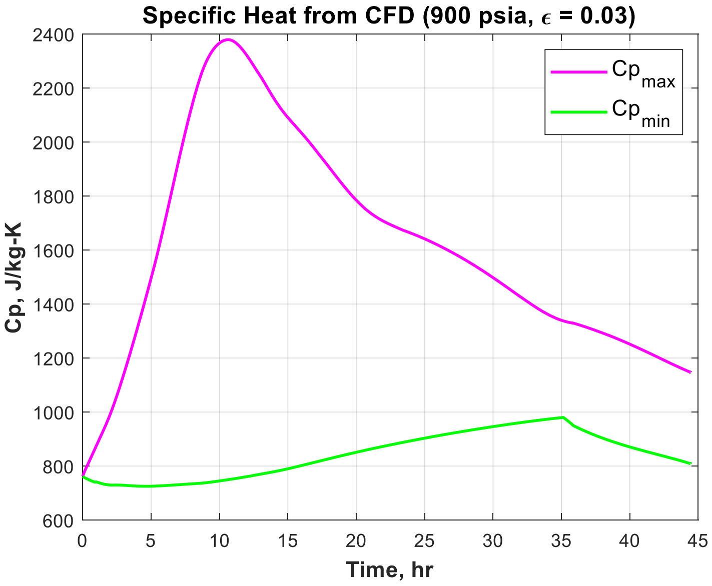
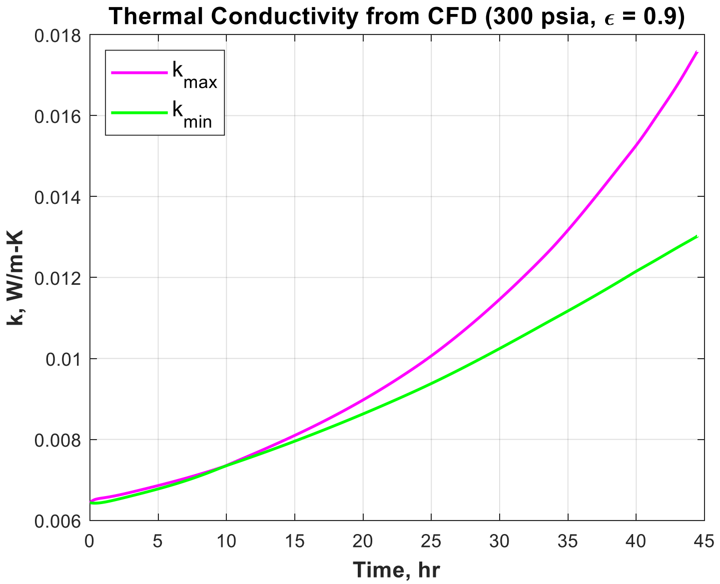
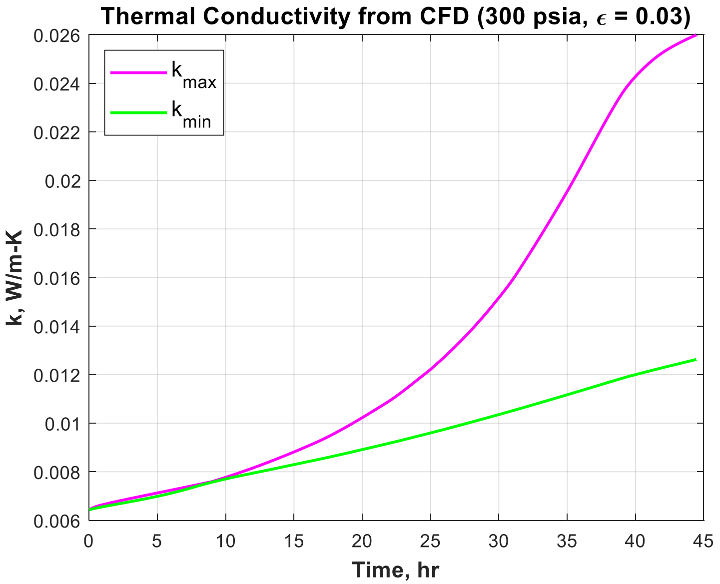

Specific Heat from CFD (900 psia, ϵ = 0.03)

Thermal Conductivity Evolution


Thermal Conductivity from CFD (300 psia, ϵ = 0.9)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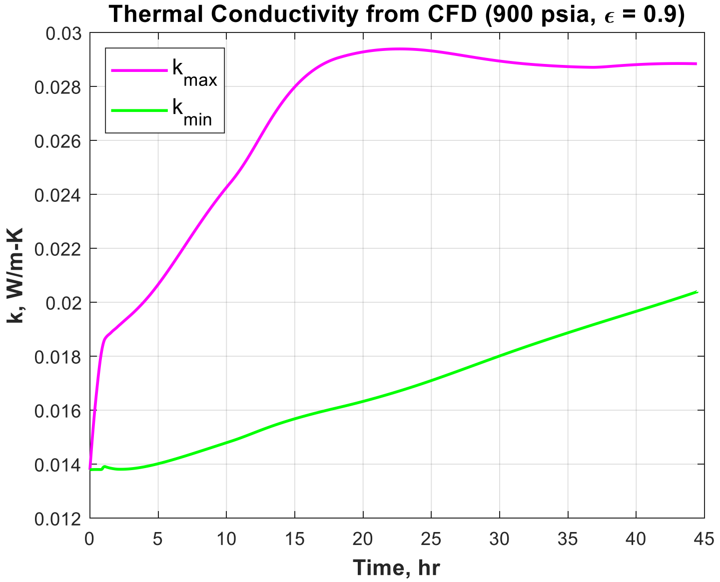
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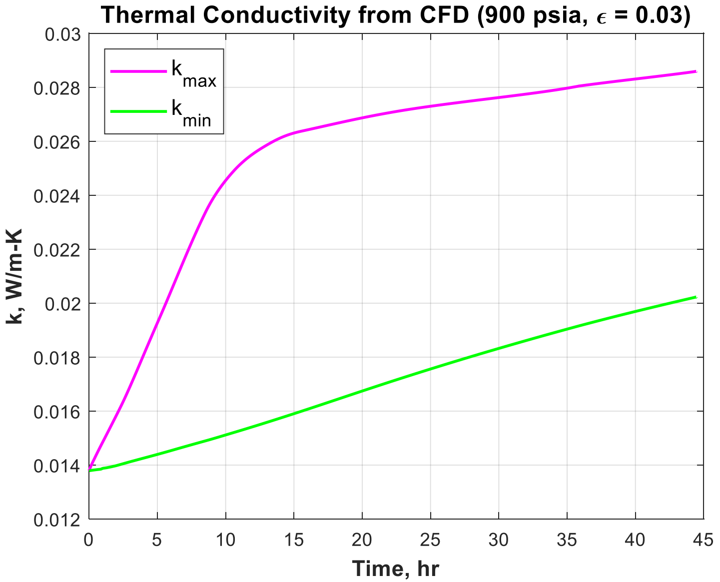
Thermal Conductivity from CFD (900 psia, ϵ = 0.03)

References

1. Woodfield et al., Measurement of Averaged Heat Transfer Coefficients in High-Pressure Vessel during Charging with Hydrogen, Nitrogen or Argon Gas, *Journal of Thermal Science and Technology*, Vol. 2, No. 2, 2007

2. D. E. Daney, Turbulent natural convection of liquid deuterium, hydrogen and nitrogen within enclosed vessels, *International Journal of Heat and Mass Transfer*, Vol. 19 (1976) pp. 431-441