



# Cost Reduction for Advanced Integration Heat Exchanger Technology for Microreactors

NEUP Project 21-24226



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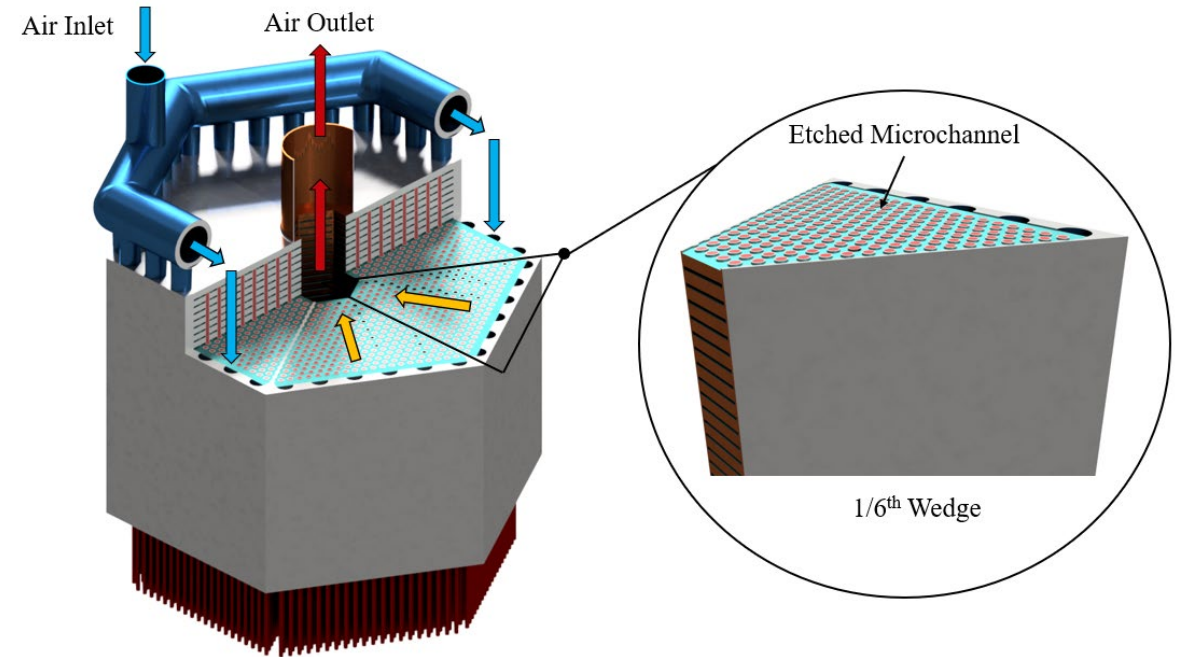


March 11, 2026

# Presentation Overview



- Project background and recap
- sCO<sub>2</sub> test article manufacturing
- WisCO<sub>2</sub> facility integration
- sCO<sub>2</sub> testing
  - Thermohydraulic performance
  - Model comparison and validation
  - Fiber optic temperature sensor results
  - Non-uniform heating testing
- Project conclusions





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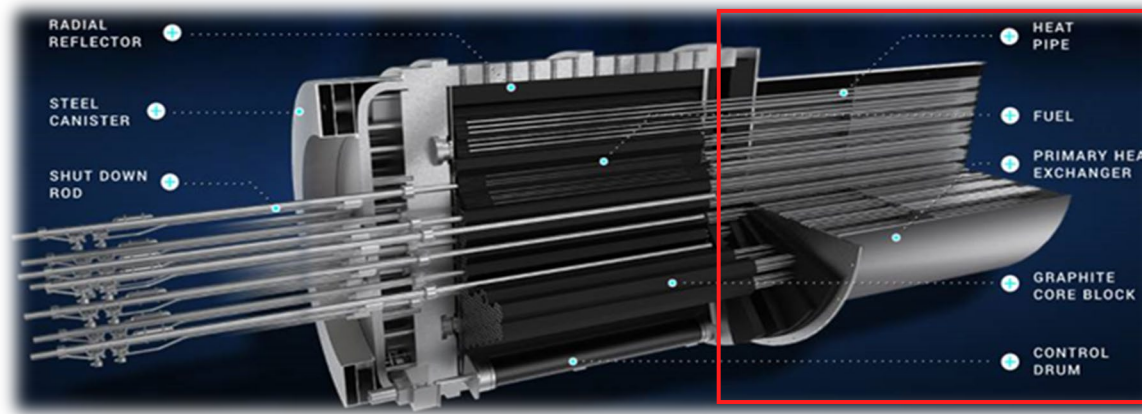
# Project Background

# Interface Heat Exchanger



- Objectives

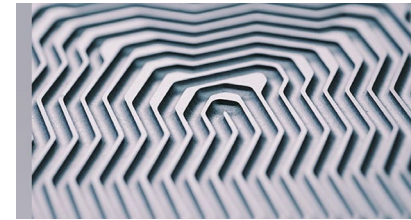
- Development and validation of microreactor integration heat exchanger design tools
- Demonstrate potential cost-reduction/performance improvements in the context of an eVinci™-like microreactor
- Obtain benchmark and validation data
- Demonstrate sub-size PCHE-based integration HX for sCO<sub>2</sub> and air working fluids
- Train several students for nuclear industry



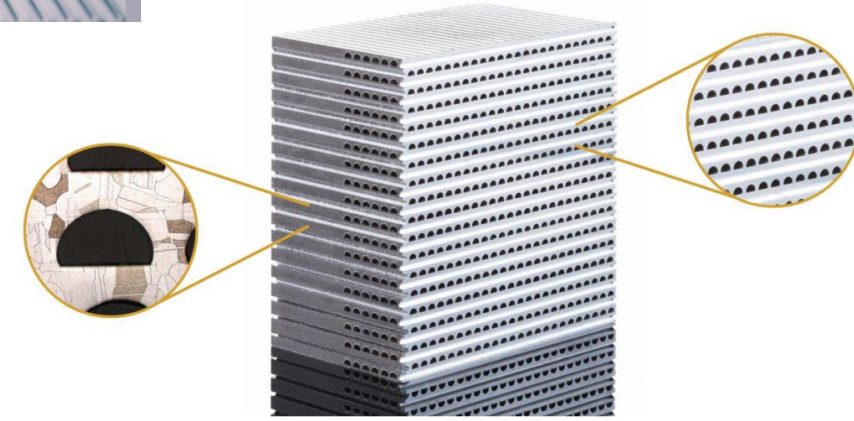
eVinci™ Micro-Reactor, Courtesy of Westinghouse Electric Company LLC

# Printed Circuit Heat Exchanger

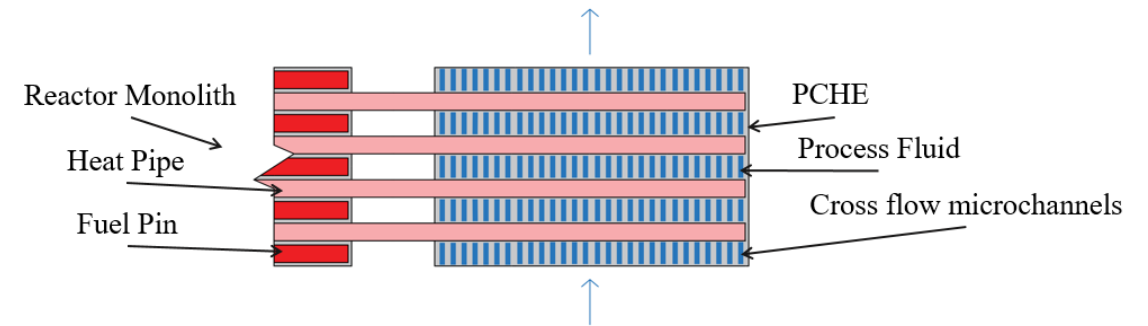
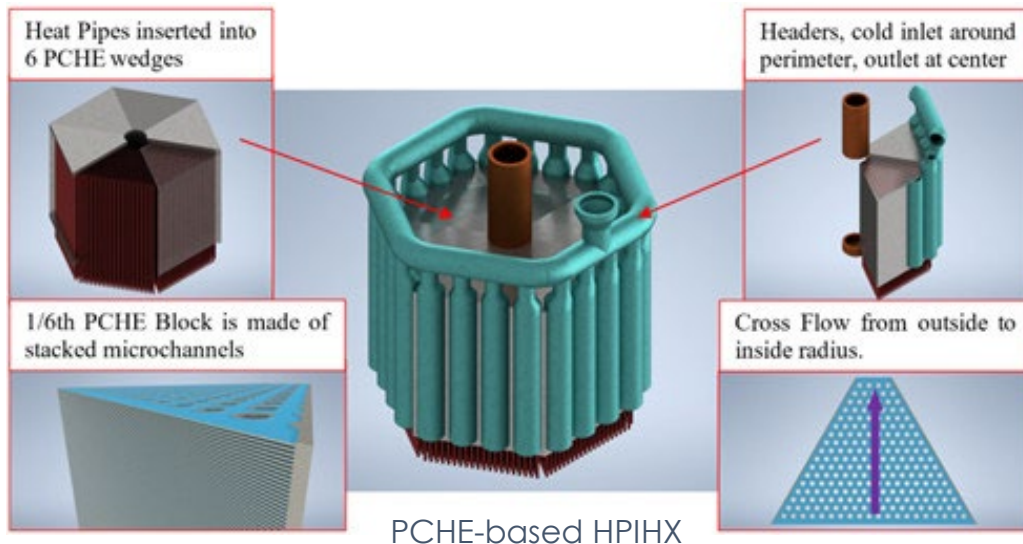
- Printed circuit heat exchanger
  - Thin metal sheets are chemically etched
  - Diffusion bonded together
  - Forms microchannels with high heat transfer area
- HPIHX PCHE
  - Single fluid, cross-flow
  - Add heat pipe holes to plates for HPIHX



Chemically etched microchannel [1]



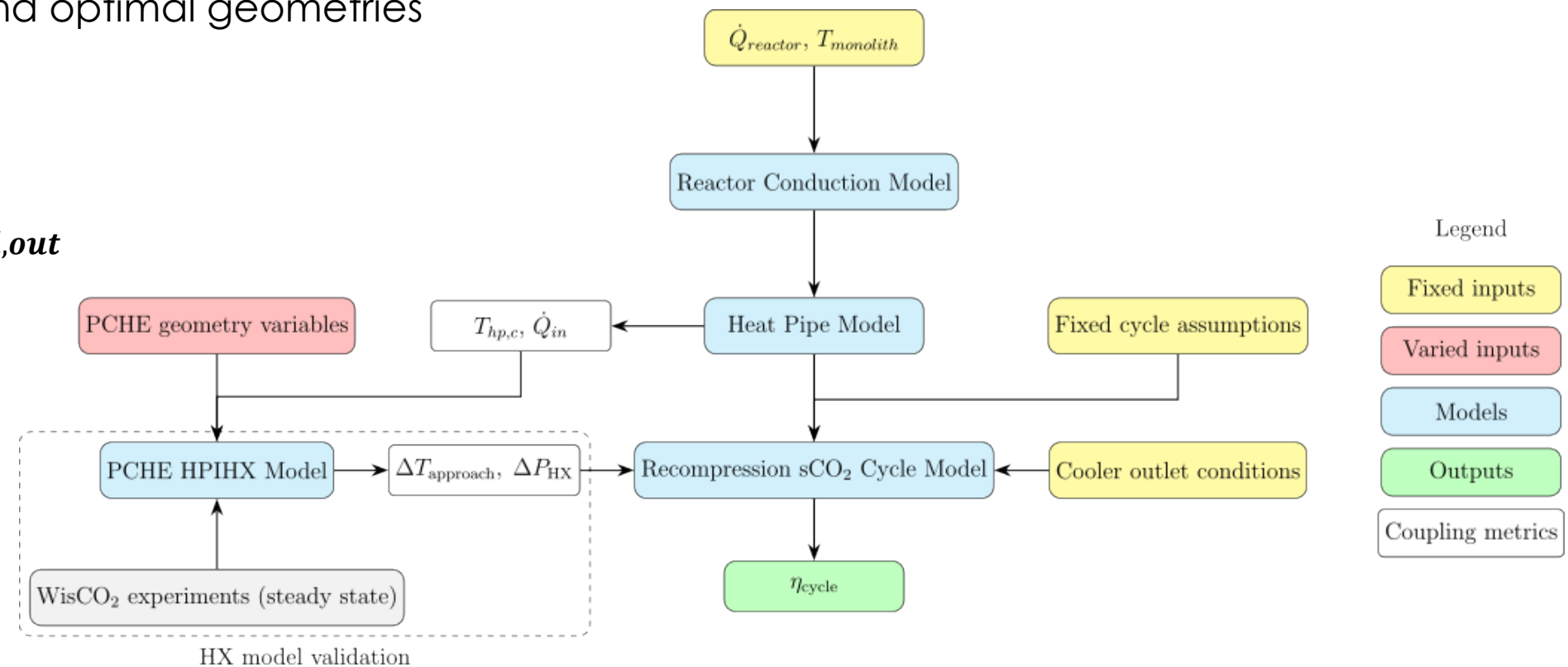
Cutout of diffusion bonded PCHE (VPEI [2])





# Interface Heat Exchanger Optimization

- Optimized using sCO<sub>2</sub> Brayton cycle model
  - Varied HX geometries to maximize cycle efficiency and find optimal geometries
- CFD PCHE model
- $\Delta P = P_{in} - P_{out}$
- $\Delta T = T_{hp,max} - T_{fluid,out}$





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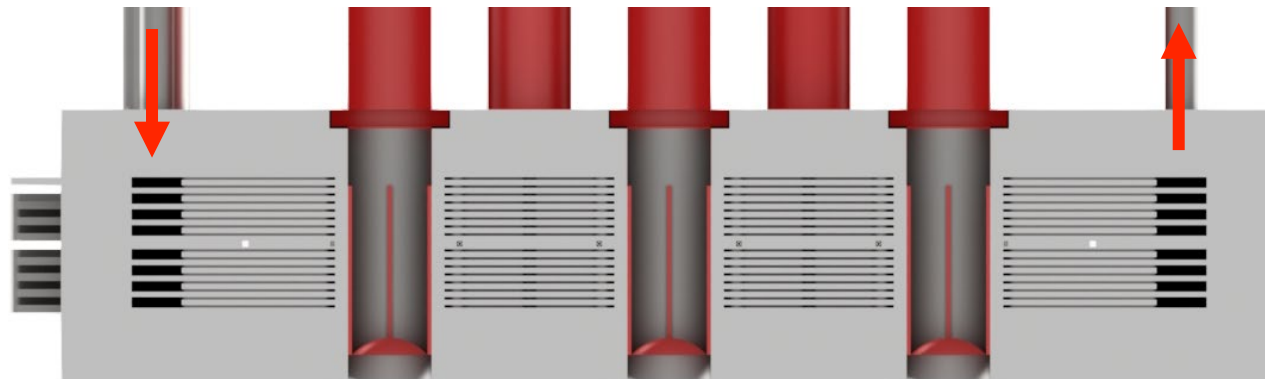
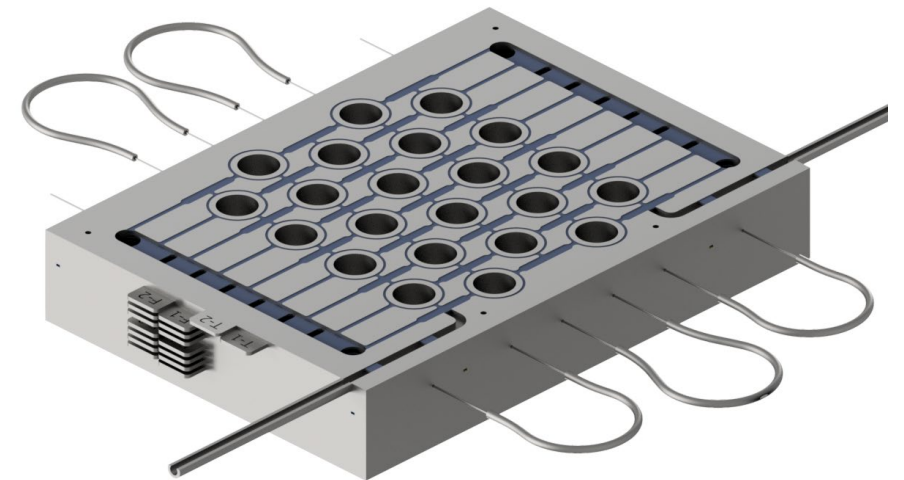
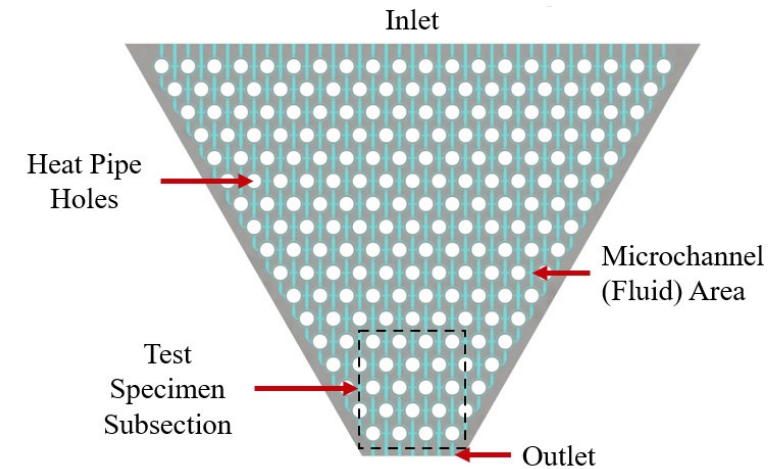
# sCO<sub>2</sub> Test Article

Manufacturing and instrumentation

# sCO<sub>2</sub> Brayton Test Specimen



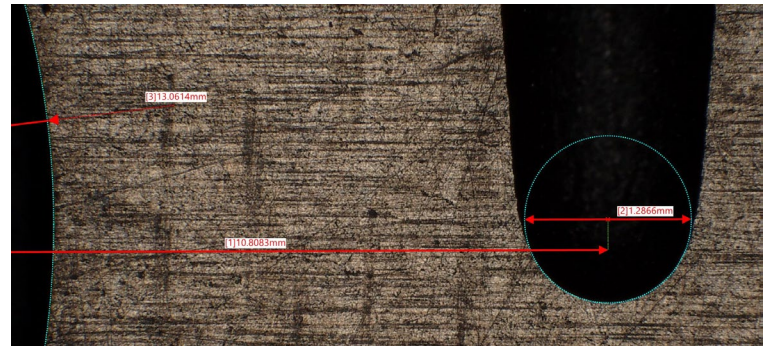
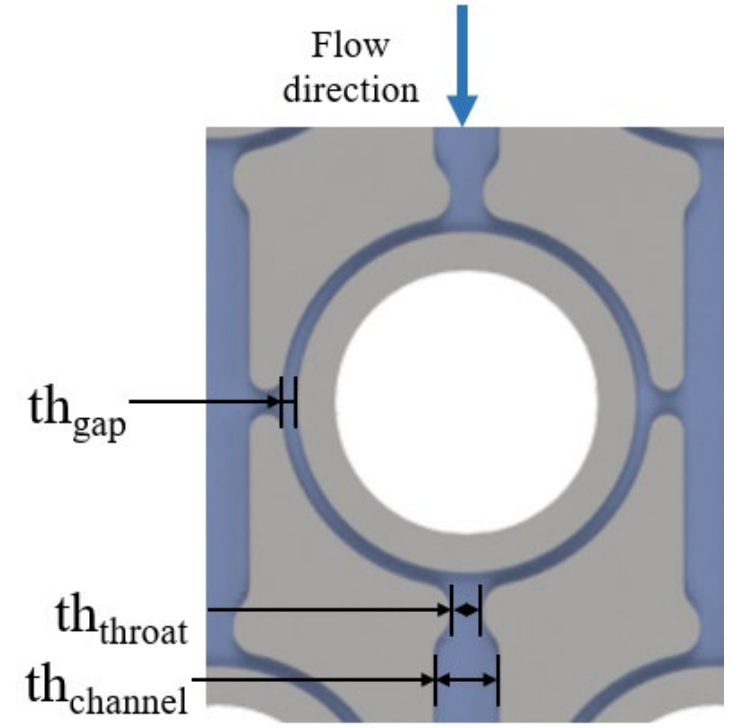
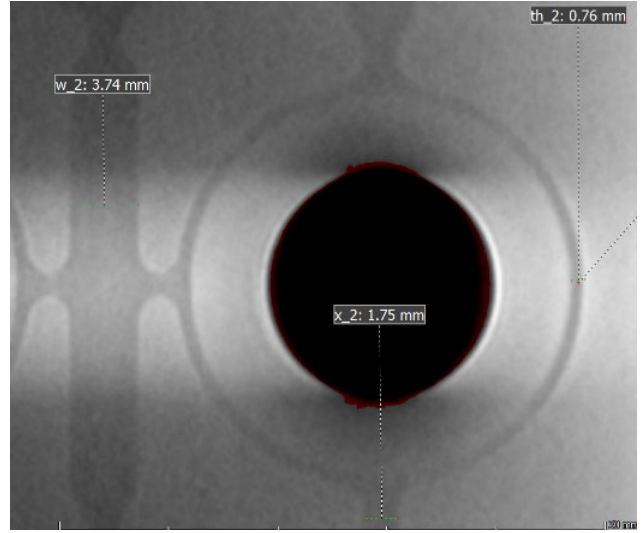
- Design pressure 20 MPa
  - 16 -1.5 mm 316 SS plates
  - 1- instrumentation layer w/ TC's and FOTS
  - 9.5" x 6.5" x 2"
- 22-130 W cartridge heaters
- Can replicate conditions of the full "wedge"



	Test Article	Full Size HX
Power	2.75 kW	5000 kW
Energy density	66 W/in <sup>2</sup>	67 W/in <sup>2</sup>
Cross section	0.08 in <sup>2</sup>	0.08-0.34 in <sup>2</sup>
Mass flow rate	0.06-0.16 kg/s	24.5 kg/s

# sCO<sub>2</sub> Test Specimen Characterization

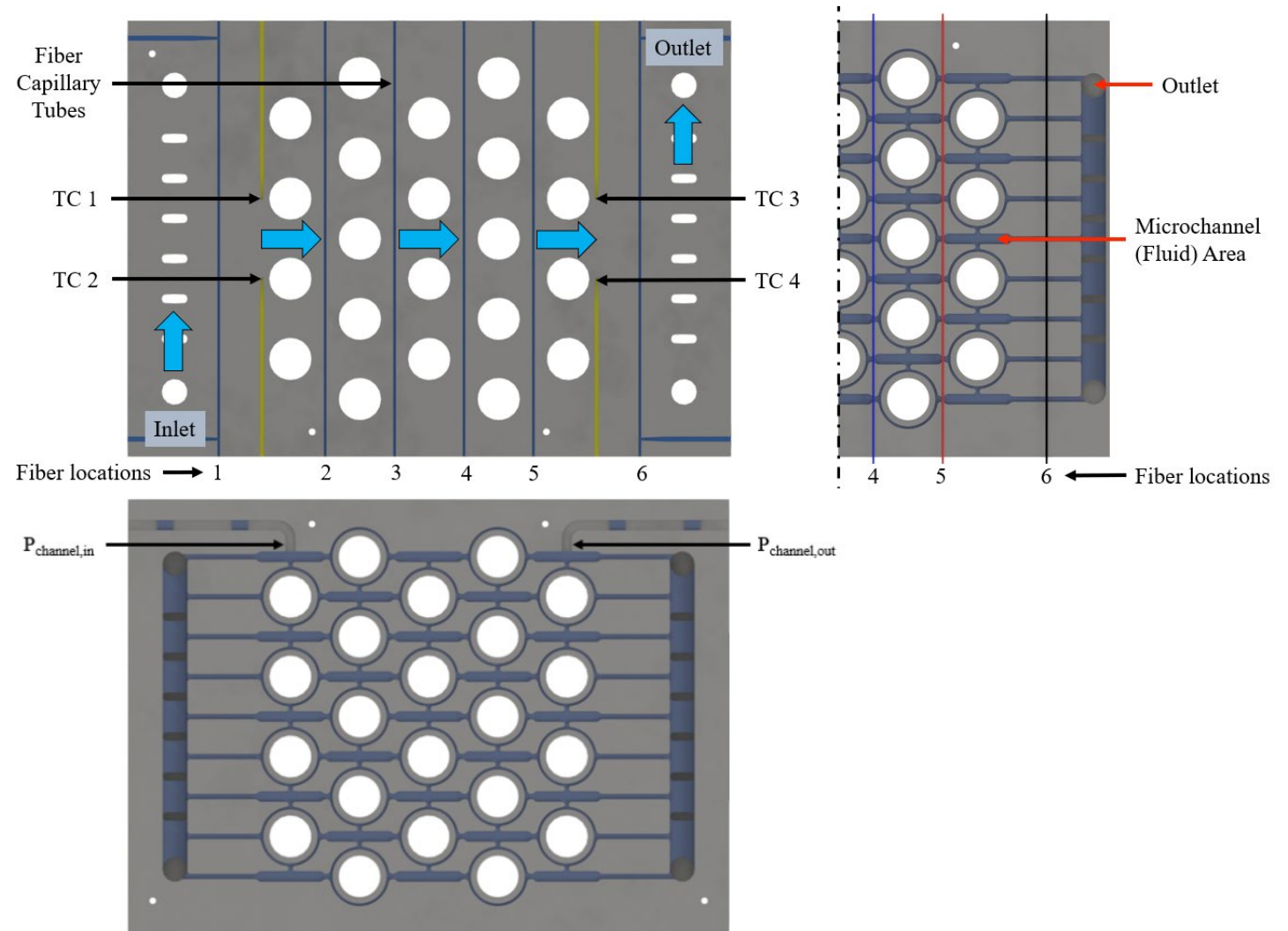
- **CT scan** of channel geometry
- Resulting dimensions
  - **Heater gap ( $th_{gap}$ )** ~ **0.76** vs. 1 mm
  - Relief channel ( $th_{channel}$ ) ~ 3.75 vs. 4 mm
  - Throat thickness ( $th_{throat}$ ) ~ 1.75 vs. 2 mm
- Approximately 0.24 mm narrow
  - 0.12 mm shallow on etch (2x)
  - **Etch depth** ~ **0.38** vs. 0.5 mm
- Etch uncertainty ~ 0.15 mm (10% mat th)
- Optical measurements
  - **TC probe locations**



# Instrumentation



- 4 – embedded TCs
- 6 – fiber optic temperature sensor passes
- Inlet and outlet fluid TCs
- Inlet and outlet pressure
  - Channel and test specimen
- Key parameters
  - $\Delta T_{approach} = \frac{TC_3 + TC_4}{2} - TC_{out}$
  - $\Delta P_{channel} = P_{channel,in} - P_{channel,out}$

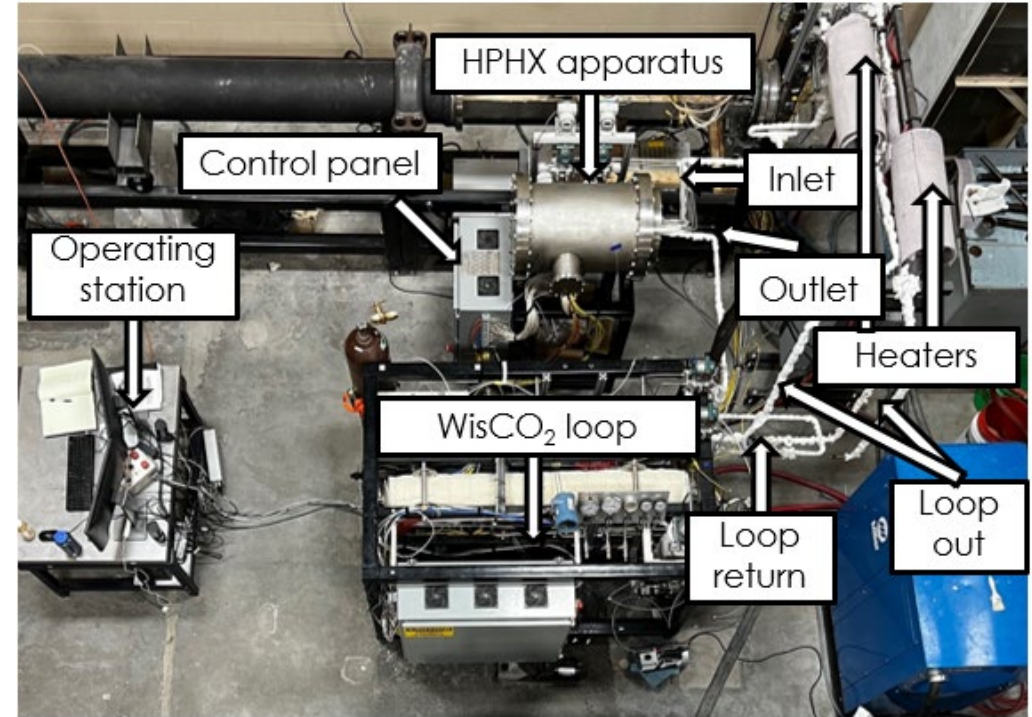
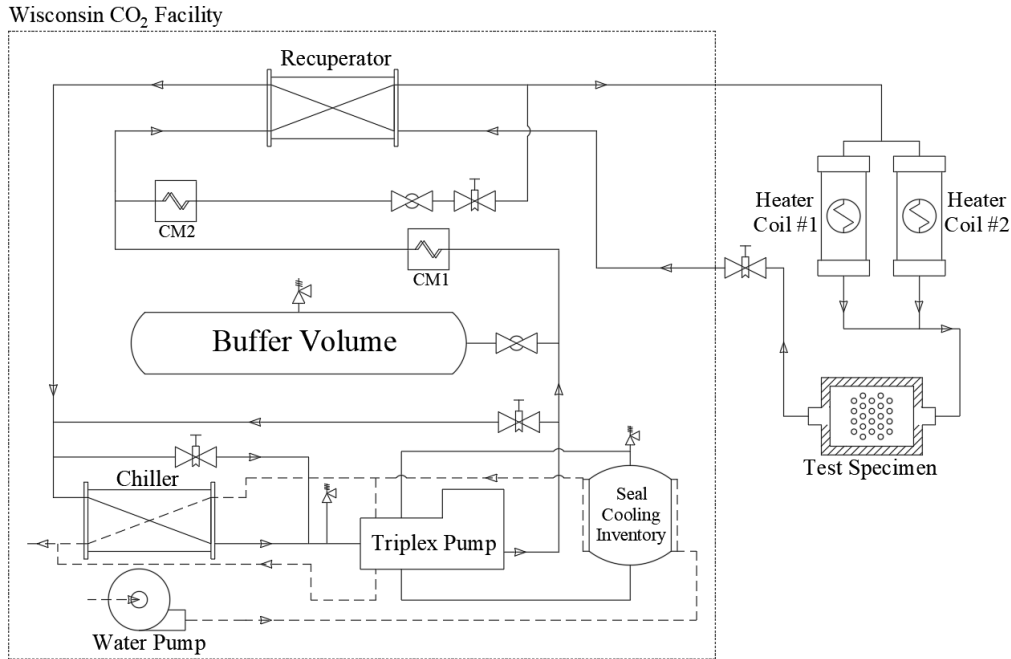




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# WisCO<sub>2</sub> Facility

Facility and integration



WisCO<sub>2</sub> facility layout

- Facility operating region

- Operating pressure up to ~25 MPa
- Mass flow rate ~ 0.02–0.6 kg/s
- Inlet temperature up to ~ 500 C (current configuration)

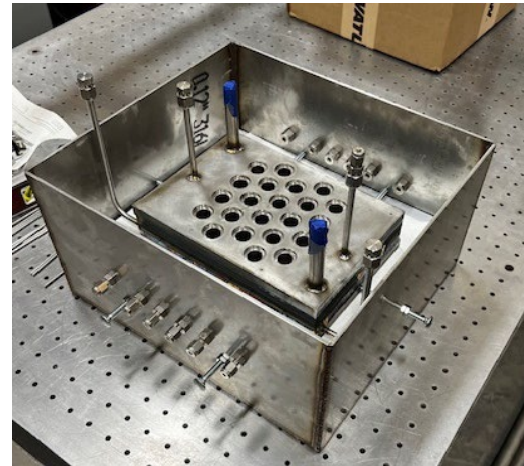
Test	Pressure	Temperature	Mass Flux [kg/m <sup>2</sup> s]	Heater Power
1-7	10 MPa	75°C	260, 400, 530, 660, 800, 930, 1060	100%
8-14		200°C		
15-21		325°C		



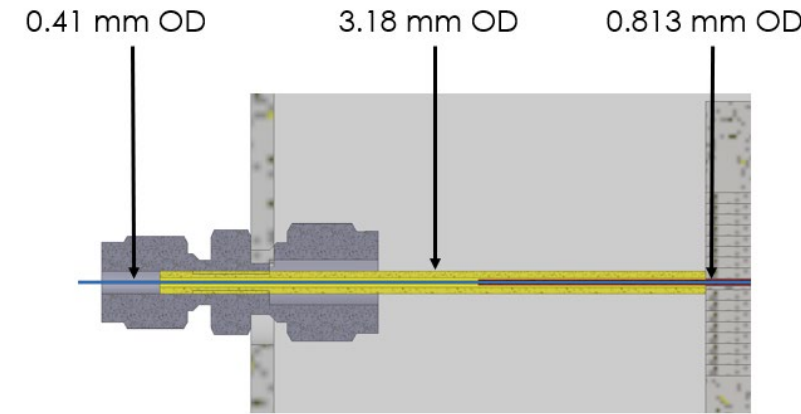
# Test Specimen Integration



- Insulation tray
  - Minimize heat losses
  - Feedthroughs support fiber optic sensor
- Test apparatus
  - SS chamber provides biological shield
  - CO<sub>2</sub> detector monitors for leaks
  - Heater control and DAQ panel



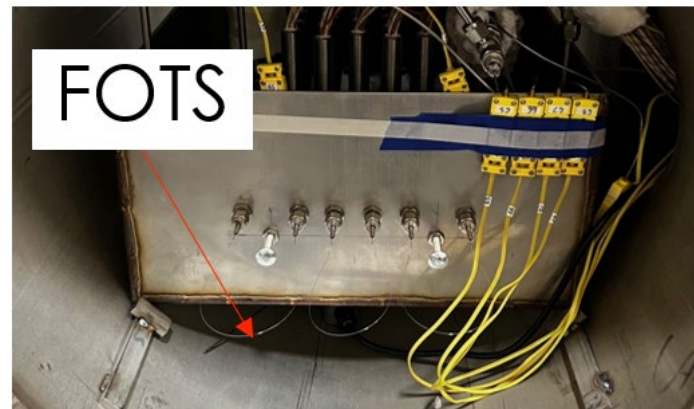
Test specimen insulation tray



FOTS feedthroughs



Instrumented and insulated test specimen



Installed fiber optic temperature sensor



Heat exchanger test apparatus



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# sCO<sub>2</sub> Testing

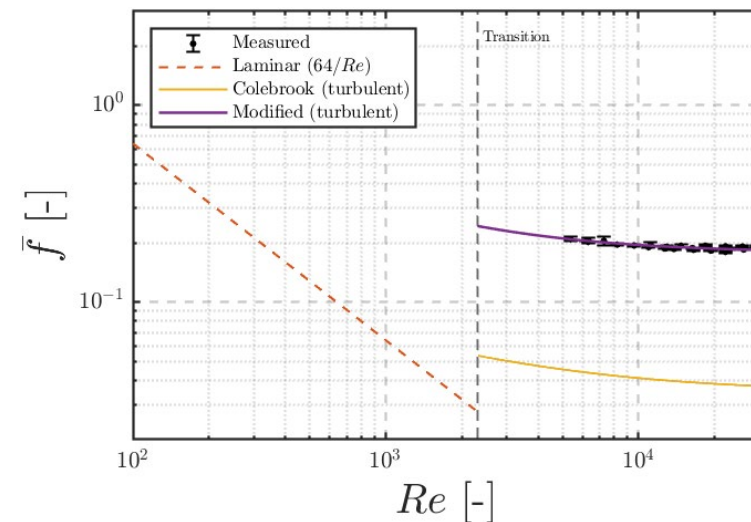
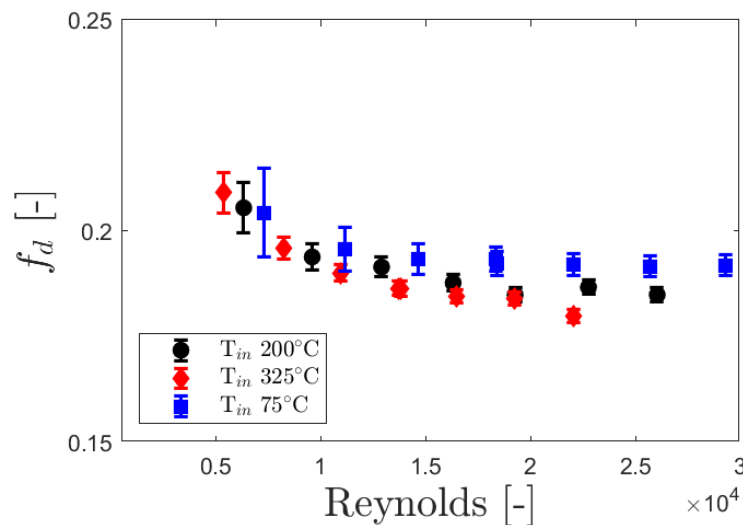
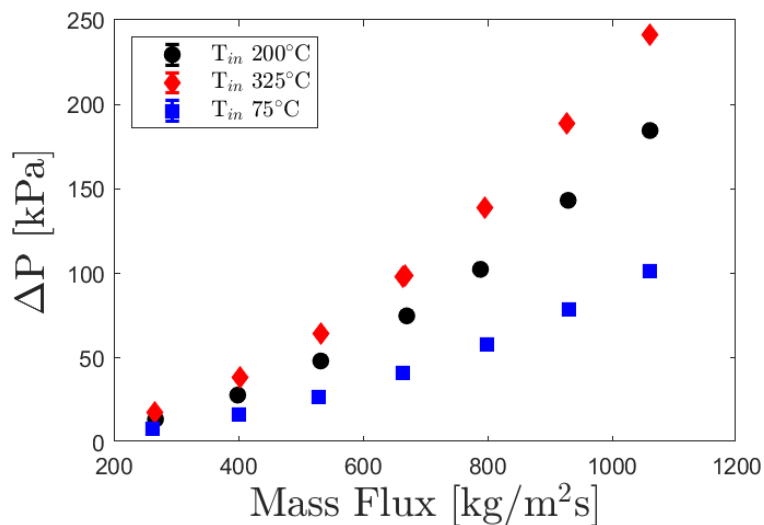
Model validation, FOTS, and non-uniform heating



# sCO<sub>2</sub> Pressure Test Results

- Pressure drop increases with flow rate and temperature
  - Higher flow rates and lower densities result in increased velocities
- An apparent friction factor begins to collapse with Reynolds
- Increased  $\bar{f}$  result from minor-loss contributions (form drag)

- $$\bar{f} = \frac{2\Delta P_{channel} D_h}{\rho L v^2}$$
- $$\bar{f}(Re) = f_0 + f_{cb}(Re)$$
- $$\frac{1}{\sqrt{f_{cb}}} = -A \log_{10} \left( \frac{RR}{B} + \frac{C}{Re \sqrt{f_{cb}}} \right), A = 2, B = 0.1816$$
- $$C = 34.10, f_0 = 0.0410, RR = 0.0081$$

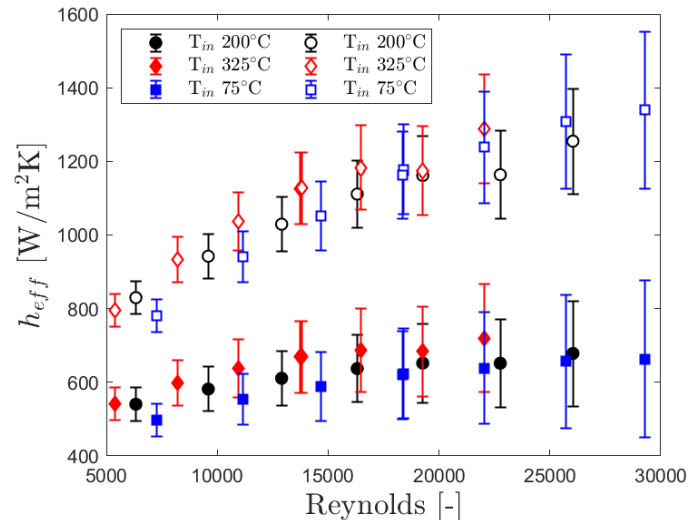
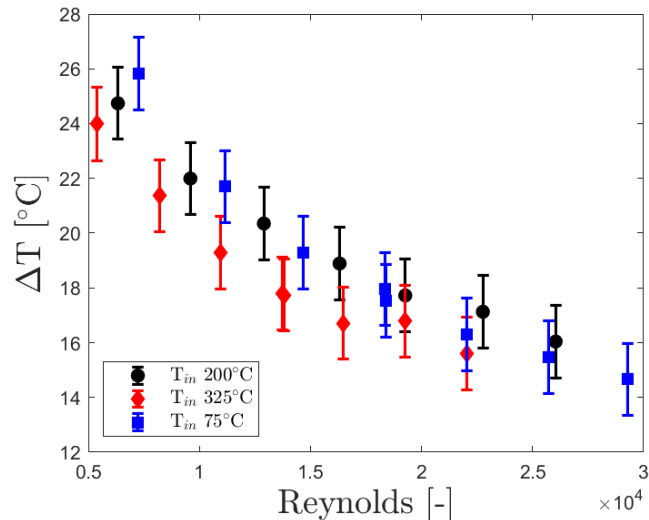


# sCO<sub>2</sub> Thermal Test Results



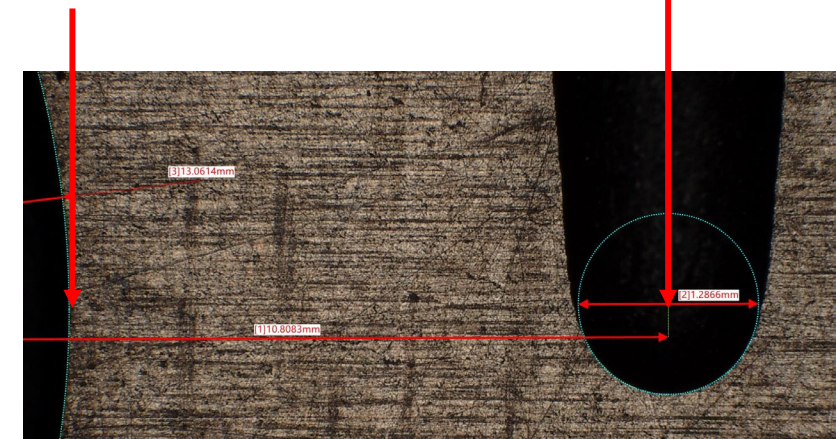
- Approach temperature decreases with Reynolds (HTC increases)
- Temperature increase from 200 to 325°C lowers approach temperature (conductivity increases)
- Non-monotonic from 75 to 200 °C due to property variations near the pseudocritical region below 100°C (Cp is ~2x at 75°C)
- Effective heat transfer coefficient vs Reynolds
  - $\Delta T_{approach} = T_{TC} - T_{out}$  (white markers)
  - $\Delta T_{approach} = T_{interface} - T_{out}$  (radial cond. inferred  $T_{interface}$ )
- TC based  $h_{eff}$  dependent on Reynolds
- Interface temperature shows resistance dominated by conduction through interface solid wall

$$h_{eff,row} = \frac{\dot{Q}_{row}}{A_{s,row} \Delta T_{approach}}$$



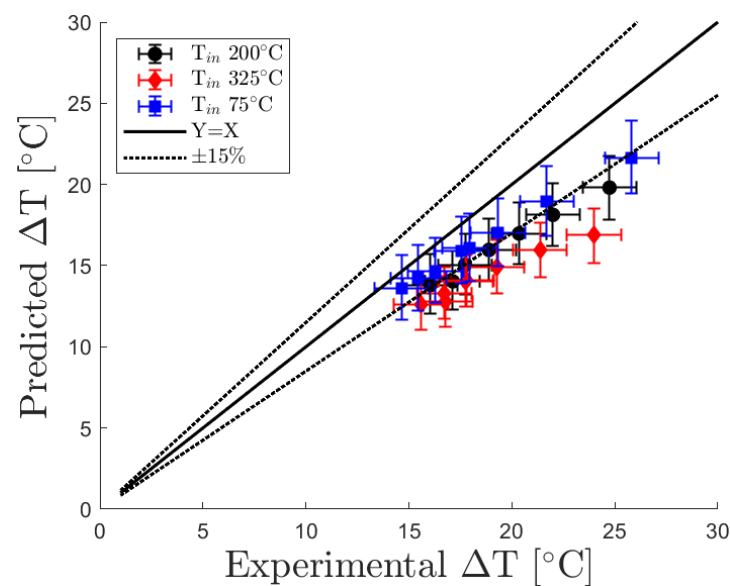
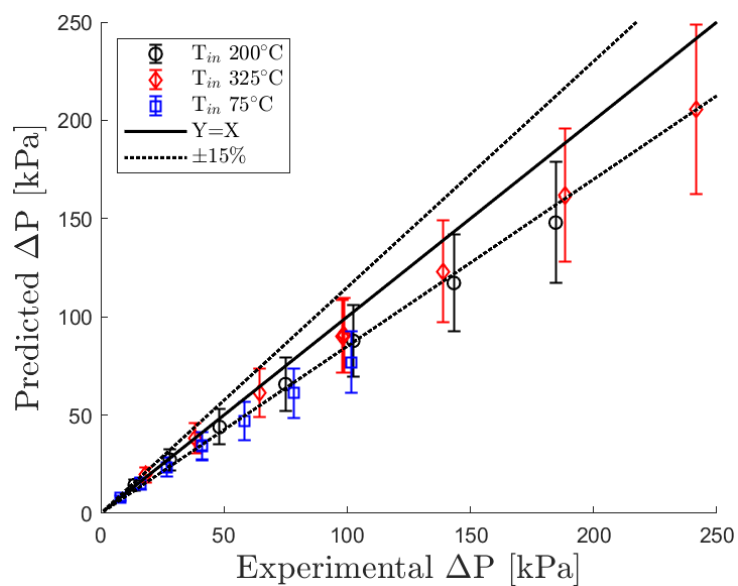
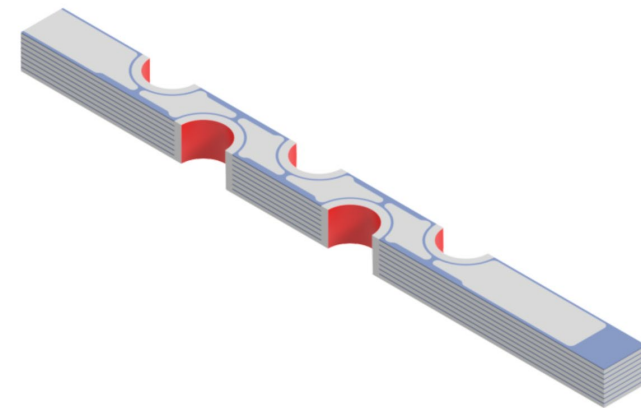
Inferred interface location

TC location



# Model Comparison

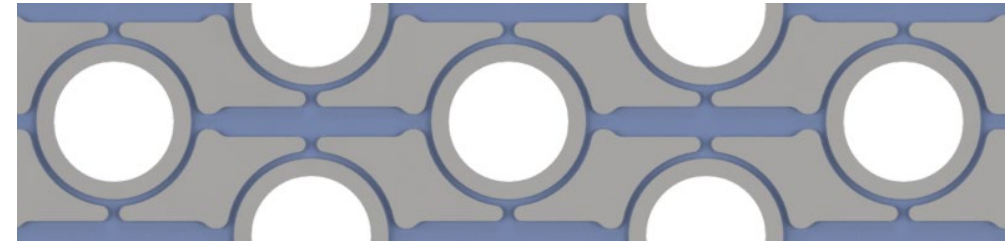
- Model comprised of unit cell with 8 channels and instrument layer
- Uniform heat flux heater boundary condition
- Close agreement with trends and approximately 15% under predicted both pressure drop and approach temperature
- Error bars for predicted values are primarily from etch depth ( $\pm 0.02$  mm) and TC location ( $\pm 0.3$  mm) uncertainty



# sCO<sub>2</sub> Cycle Analysis Impact

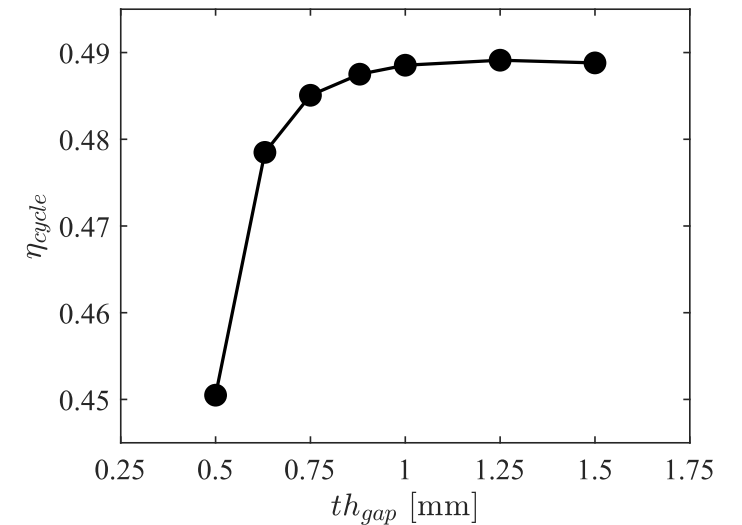


- Correction factors integrated into system-level
  - Pressure drop – 1.15
  - Approach temperature – 1.2
- Cycle efficiency decreased ~0.1%
- Add in in etch uncertainty when selecting optimal channel thickness
- Shift towards 1.3 mm to account for ± 0.3 mm uncertainty



Optimized geometry

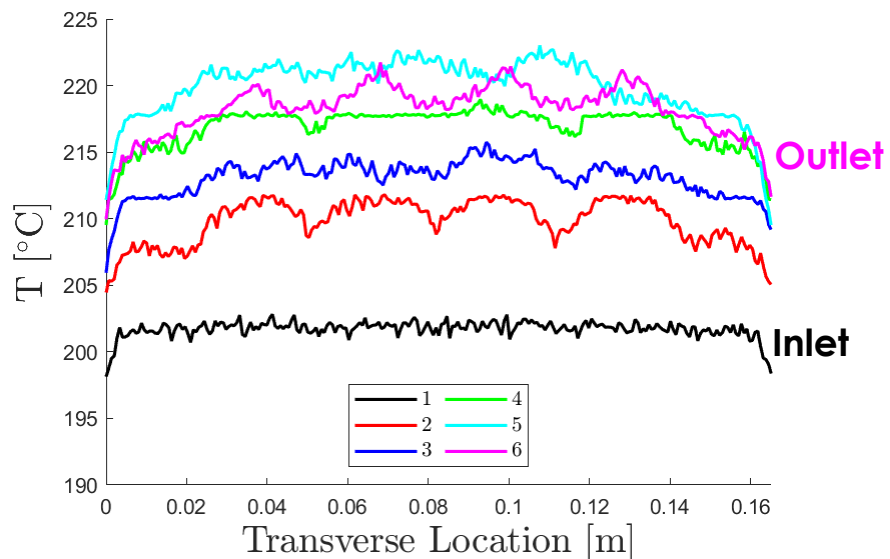
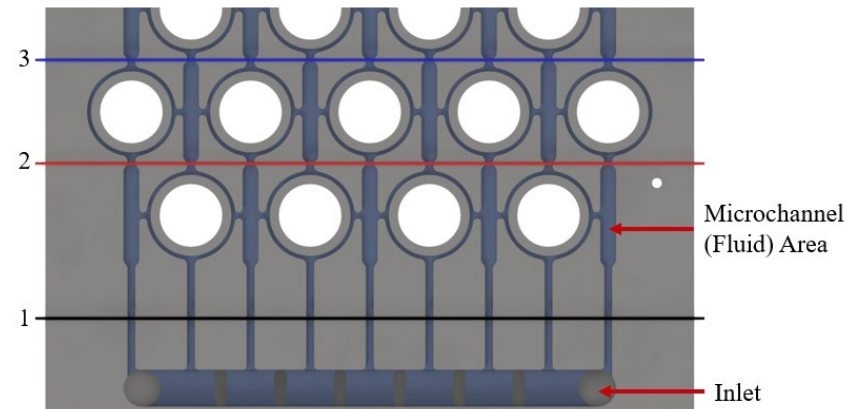
Heat Exchanger	Annular/PCHE Gap	Cycle Efficiency	$\Delta P$ [kPa]	$\Delta T$ [°C]
AFHX (air)	1.9 mm	34.3 %	32.6	51.1
PCHE (air)	1.0 mm	35.3 %	14.2	43.1
PCHE (sCO <sub>2</sub> )	1.2 mm	48.5 %	39.0	15.5



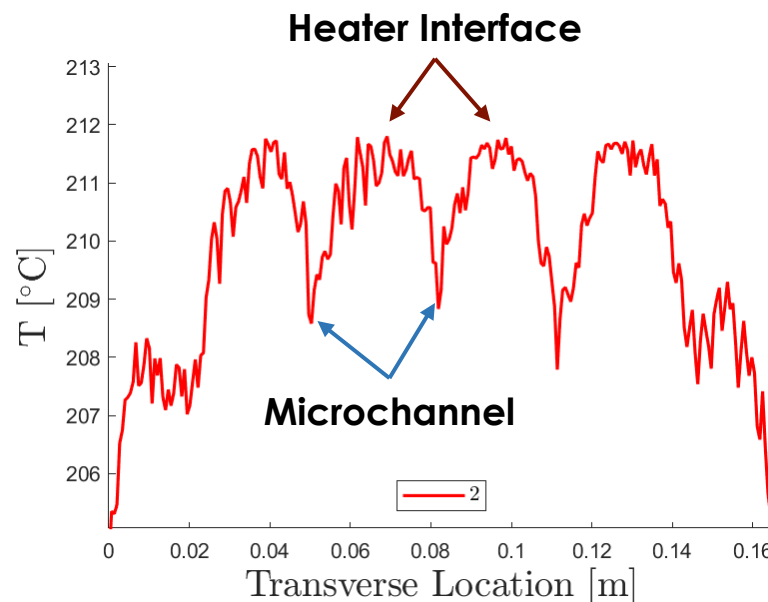
Cycle efficiency vs channel thickness

# Fiber Temperature Measurements

- **Near-inlet** profile nearly **uniform**, confirming minimal upstream thermal influence from downstream heaters
- **Distinct temperature peaks** align with **heater interface locations**, validating spatial alignment and geometric fidelity of the PCHE instrumentation
- Downstream profiles show increasing baseline temperature due to cumulative heat addition, while preserving transverse temperature structure
- Slight peak reduction near outlet, consistent with reduced solid-fluid temperature difference as bulk fluid temperature rises

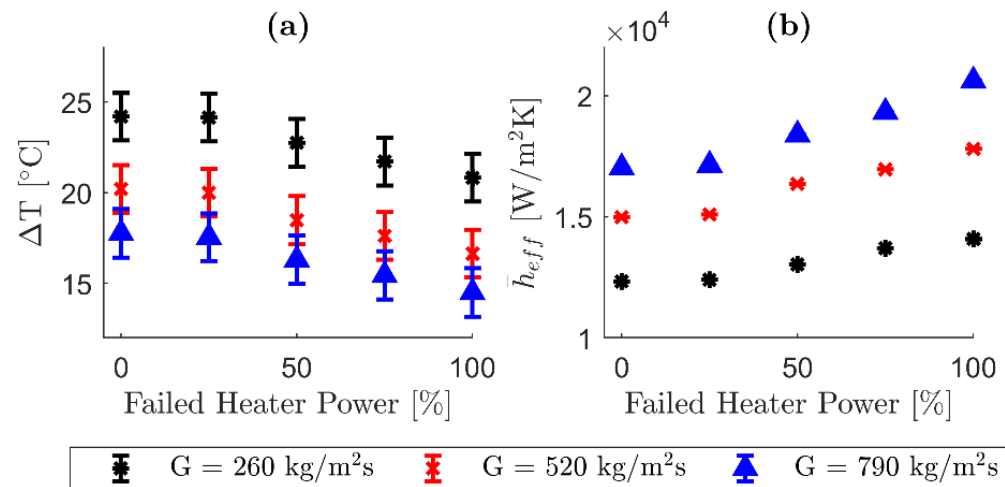
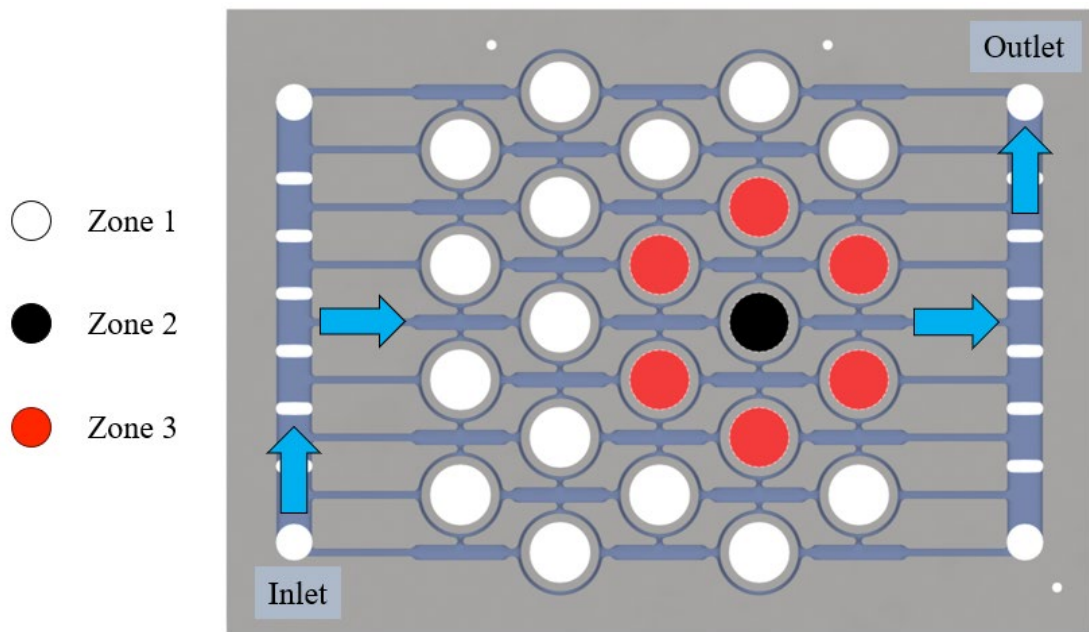


$T_{in} = 200 \text{ C}^\circ$   
 $\dot{m} = 0.16 \text{ kg/s}$



# Non-Uniform Heating (failed heater)

- Ramp down “failed” heater while ramping up surrounding heaters
- Increasing mass flux decreased approach temperature (consistent)
- **Approach temperature increases** as heater degrades due to **increased heat load** of surrounding heaters
- Slight increase in relative increase in approach temperature at lower mass flux

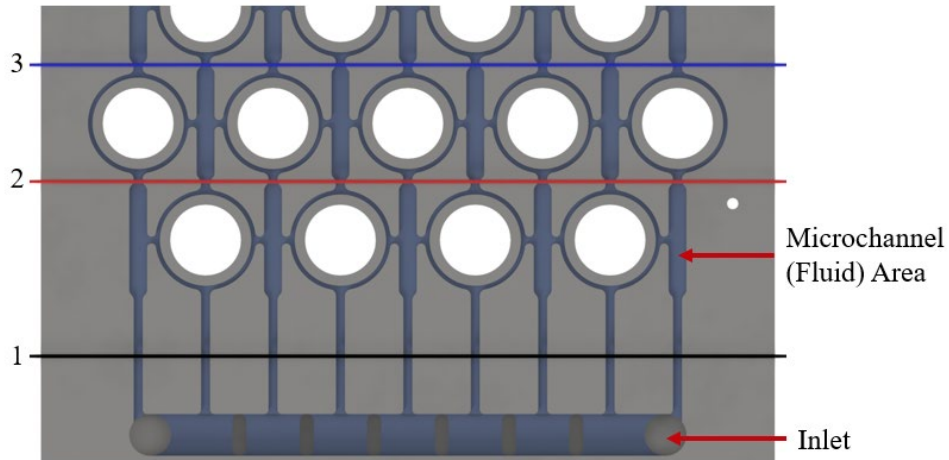


Test	Temperature	Pressure	Zone 1	Zone 2	Zone 3
1	200°C	10 MPa		85.7% (100%)	85.7%
2			85.7%	64.3% (75%)	89.3%
3			85.7%	42.9% (50%)	92.9%
4			21.4% (25%)	96.4%	
5			0% (0%)	100%	

# Fiber Measurements (failed heater)



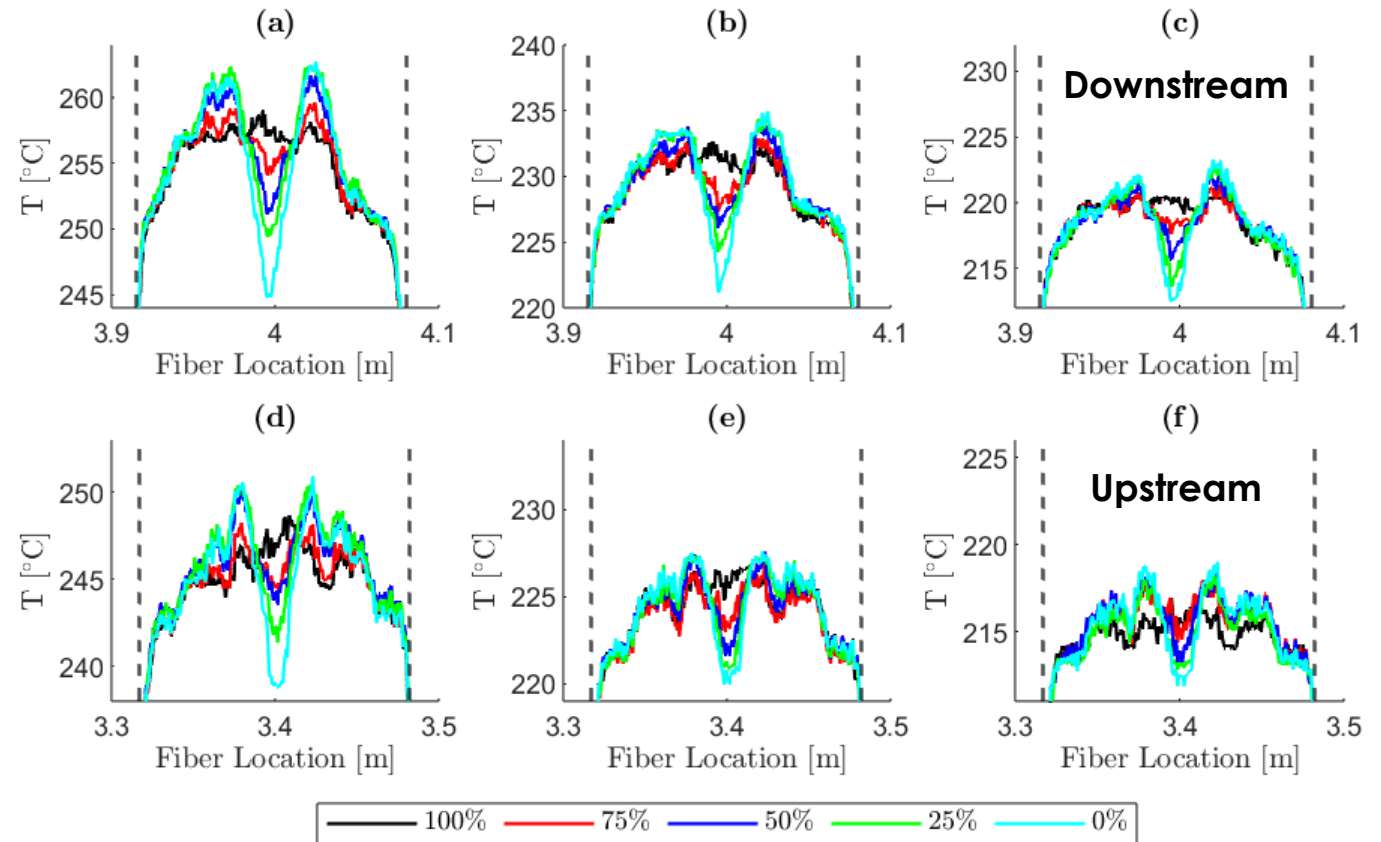
- Reduced heater power produces a central temperature dip, with elevated temperatures on either side, confirming lateral heat redistribution through adjacent regions
- Transverse **temperature variation decreases** with **increasing mass flux**, as enhanced convection smooths solid temperature gradients
- **Lower mass flux** corresponds to outer (wedge) heat pipe failure and **larger thermal gradients**
- Demonstrates the capability of distributed FOTS in detecting heat pipe degradation



$G = 260 \text{ kg/m}^2\text{s}$

$G = 520 \text{ kg/m}^2\text{s}$

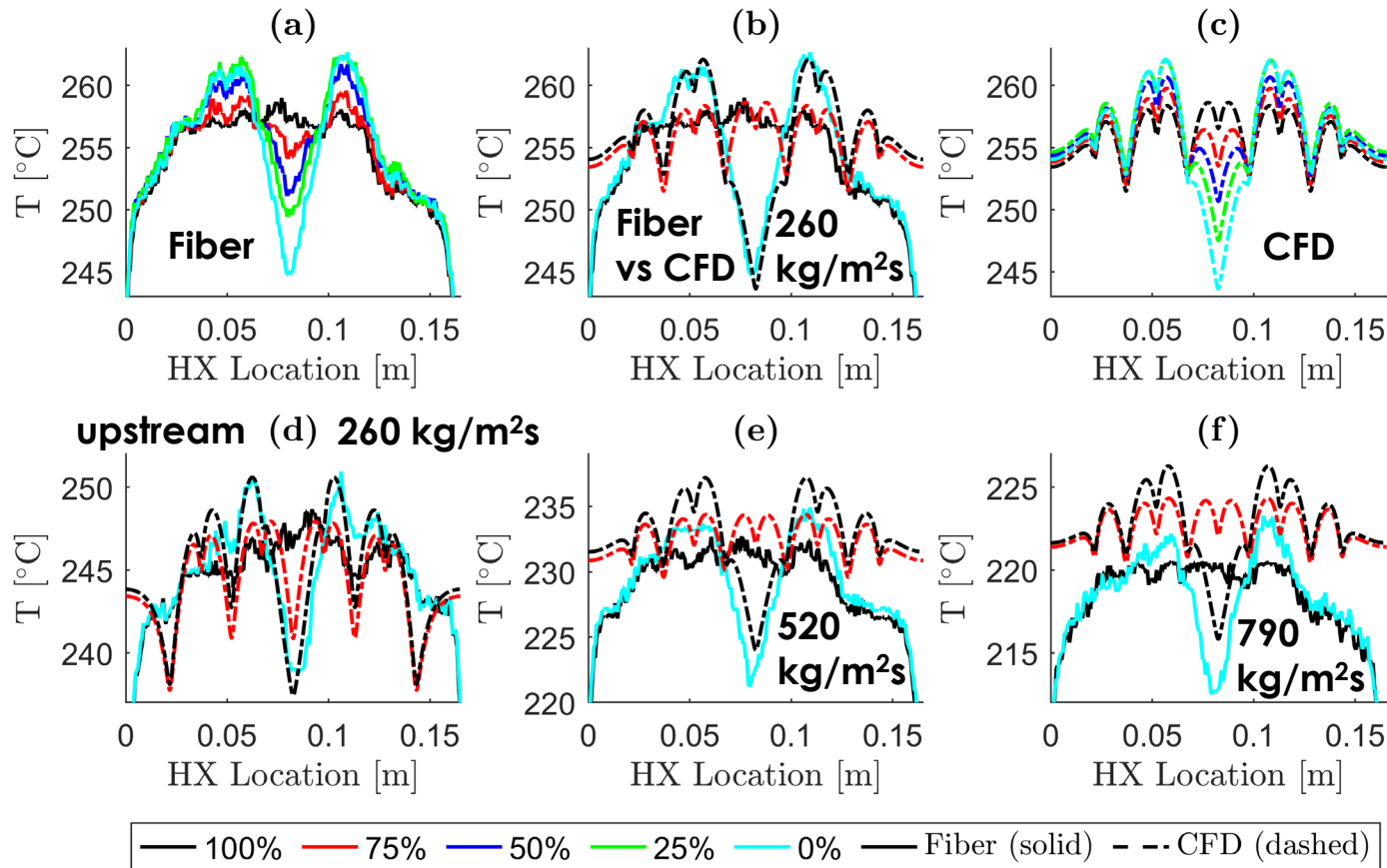
$G = 790 \text{ kg/m}^2\text{s}$



# CFD Comparison



Mass Flux	Fiber $\Delta T_{max}$ [°C]	CFD $\Delta T_{max}$ [°C]	% difference
260 kg/m <sup>2</sup> s	17.9	18.4	2.8%
520 kg/m <sup>2</sup> s	13.1	13.7	4.5%
790 kg/m <sup>2</sup> s	10.4	10.8	3.8%



- CFD model evaluates **single microchannel plate** assuming half symmetry
- CFD underpredicts  $\Delta T_{app}$  by  $\sim 4.5^\circ\text{C}$  due to instrument-layer thickness (fin effect), parasitic heat loss and thermocouple position uncertainty
- After uniform offset correction ( $+4.5^\circ\text{C}$ ), CFD captures **operating trends** and **spatial temperature structure**, including heater footprints and failed-heater depressions
- **Maximum temperature differences** agree within  $\sim 3.8\%$  for failed-heater cases, validating the CFD-predicted **relative temperature distributions**

# CFD Stress Study

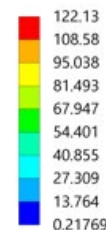


- Higher mass flux reduces thermal stresses, while failed-heater cases show elevated peak stresses due to lateral temperature gradients
- All cases remain below ASME BPVC allowable limits, with no yielding expected in the failed heat pipe cases (118 MPa @ 500°C)
- Pressure loading adds mostly hydrostatic stress and does not increase von Mises stress
- Inlet and outlet **wedge** conditions simulated
  - ~120 kg/m<sup>2</sup>s @ 475°C and 790 kg/m<sup>2</sup>s @ 620°C
  - Failed scenario maximum von Mises stress increases from 106 to 122 MPa
  - Failure in first heat pipe row may further increase localized hot spots

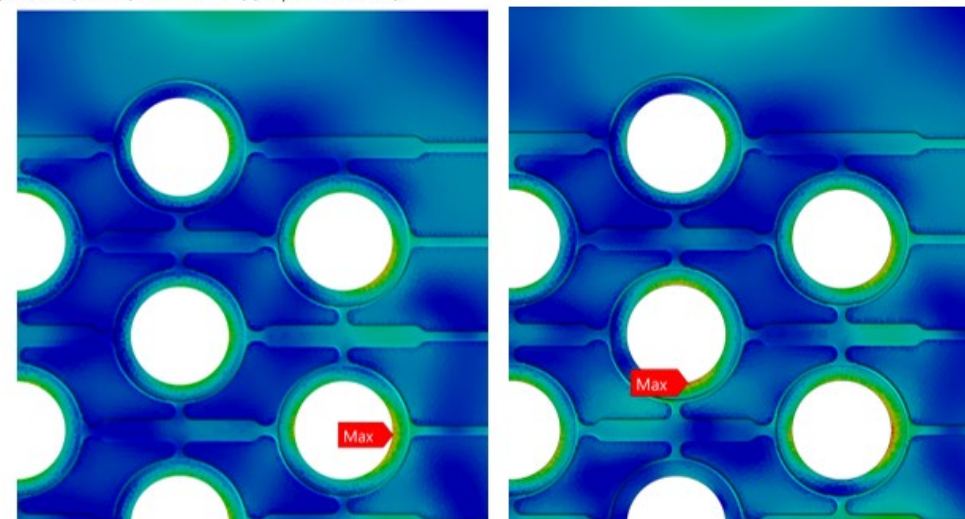
## Test conditions

Mass Flux	Nominal T	Failed T	Nominal T+P	Failed T+P
	$\sigma_{avg}/\sigma_{max}$ [MPa]	$\sigma_{avg}/\sigma_{max}$ [MPa]	$\sigma_{avg}/\sigma_{max}$ [MPa]	$\sigma_{avg}/\sigma_{max}$ [MPa]
260 kg/m <sup>2</sup> s	14.6/92.3	15.9/119	15.8/99.6	17.0/117
520 kg/m <sup>2</sup> s	12.2/77.7	13.4/104	13.6/78.1	14.8/103
790 kg/m <sup>2</sup> s	11.3/72.7	12.2/91.7	12.9/73.1	13.6/90.0

**B: Nominal**  
 Eq stress core T1  
 Type: Equivalent (von-Mises) Stress (Elemental Mean) (Scoped to Elements)  
 Unit: MPa  
 Time: 1 s  
 Max: 105.73  
 Min: 0.21769



## Failed Heat Pipe

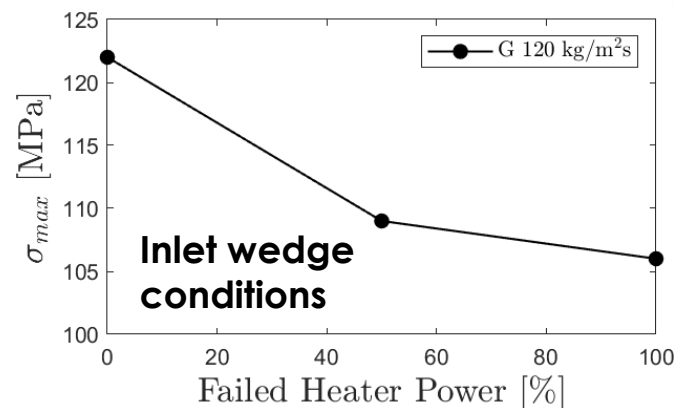
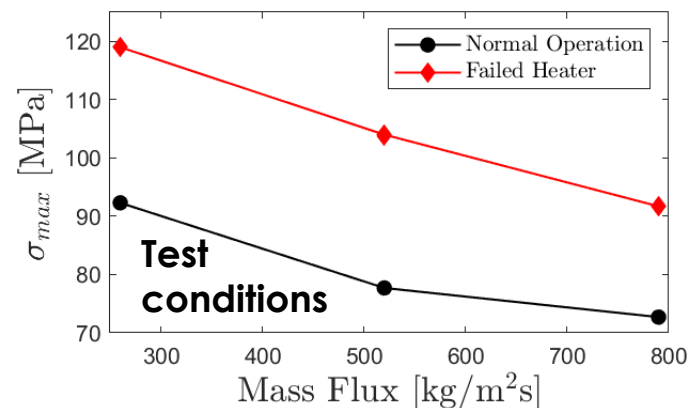


Nominal

0.00 35.00 70.00 (mm)

## Wedge conditions

Mass Flux	Nominal T	50% Failed T	100% Failed T	Failed T+P
	$\sigma_{avg}/\sigma_{max}$ [MPa]	$\sigma_{avg}/\sigma_{max}$ [MPa]	$\sigma_{avg}/\sigma_{max}$ [MPa]	$\sigma_{avg}/\sigma_{max}$ [MPa]
120 kg/m <sup>2</sup> s	17.4/106	17.5/109	18.4/122	19.1/120
790 kg/m <sup>2</sup> s	9.2/55.4	9.4/57.3	9.9/69.5	11.5/67.7



# Conclusions – HPIHX



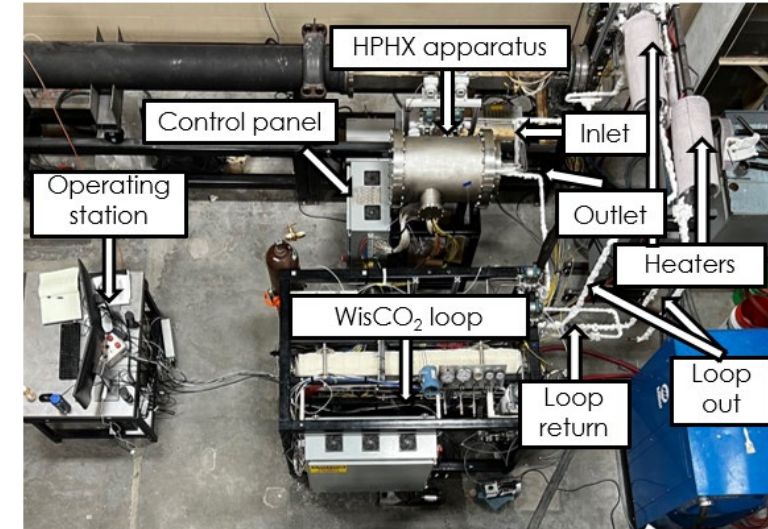
- Conclusions

- Experimental data was used to validate sCO<sub>2</sub> model for PCHE
- With the validated model cycle efficiency decreased ~ 0.1%
- FOTS measurements showed microchannel resolution
- Non uniform heating showed CFD and experimental gradients matched
- Failed heater (heat pipe) resulted in increased approach temperature and stresses



- Recommended future work

- Higher temperature and pressure testing
- Transient investigations
- Heat pipe to PCHE interface improvement





# Thermal Hydraulics Laboratory

## Questions?

