Task 6.0 – Heat Transfer Coefficient Measurements

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Problem Statement

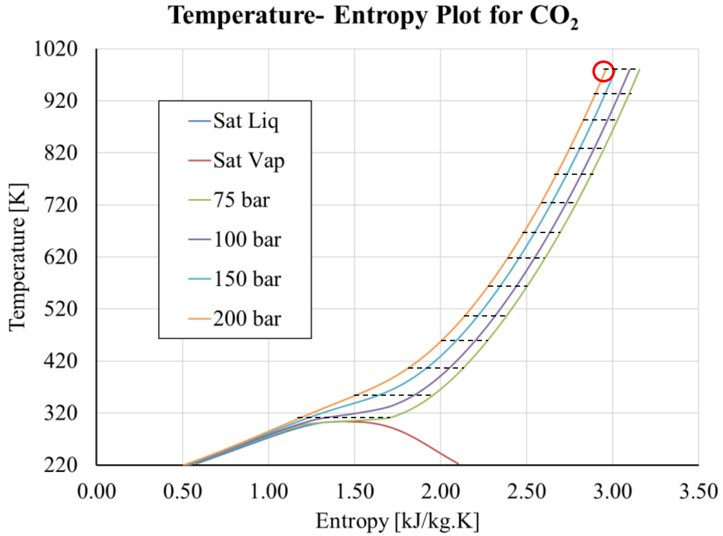
Develop heat transfer coefficient correlations for CO_2 for boiler conditions (200 bar pressure and 32°C to 600°C temperature)

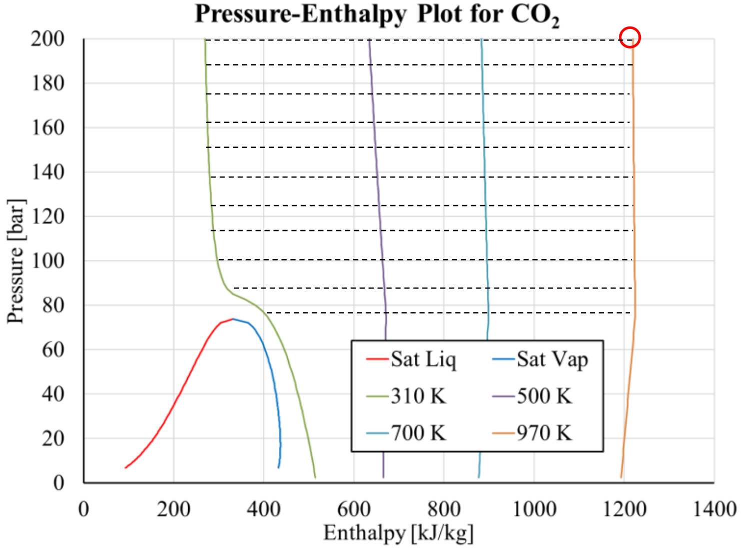
Test section I.D.	**2 mm to 20 mm**
Temperature, T_{in}	**305 to 975 K or (to 702°C)**
Pressure , P_{in}	**100 bar, 200 bar**
Reynolds number	**10,000 to 750,000**
Inclination	**0°, 45°, 90°**

Mass flow rate [kg/s] at 200 bar, 700°C for different tubing sizes

Re	1/8	1/4	1/2	3/4	1
10000	5.73E-04	1.50E-03	3.57E-03	5.42E-03	7.23E-03
60000	3.44E-03	8.98E-03	2.14E-02	3.25E-02	4.34E-02
100000	5.73E-03	1.50E-02			
250000	1.43E-02	3.74E-02			
750000	4.30E-02	1.12E-01			
900000	5.16E-02	1.35E-01			
1.50E+06	8.59E-02	2.24E-01			

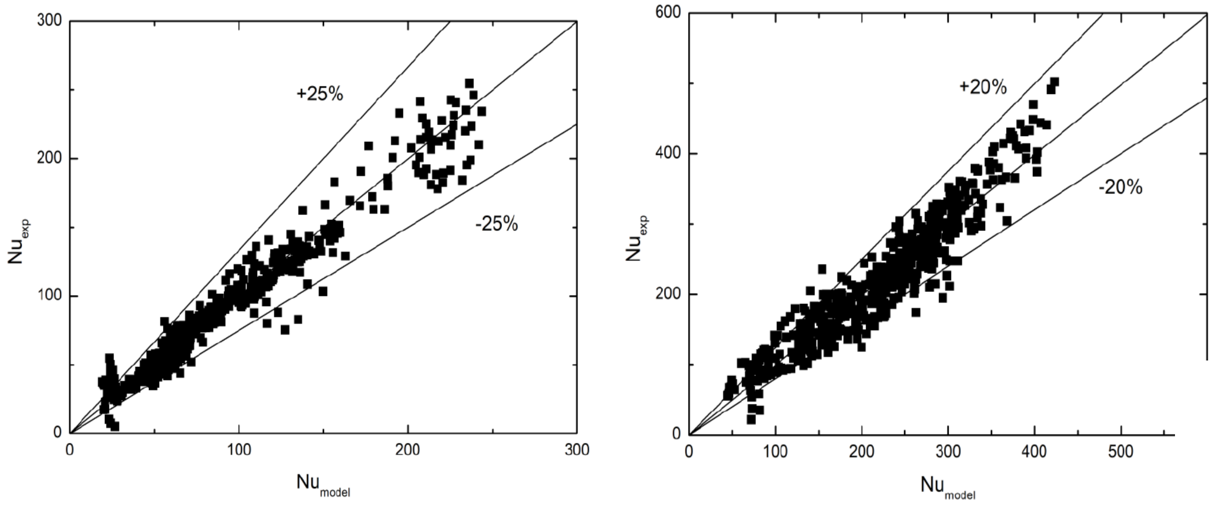
- Light orange cells: High priority
- Dark orange cells: Low priority
- Black cells: Not planned





- STEP HEX inlet conditions circled red
- Shaded region is the domain of interest

Experimental vs Correlations**



(a) top wall (b) bottom wall

Figure 7. Nonlinear fitting results obtained by using our experimental data

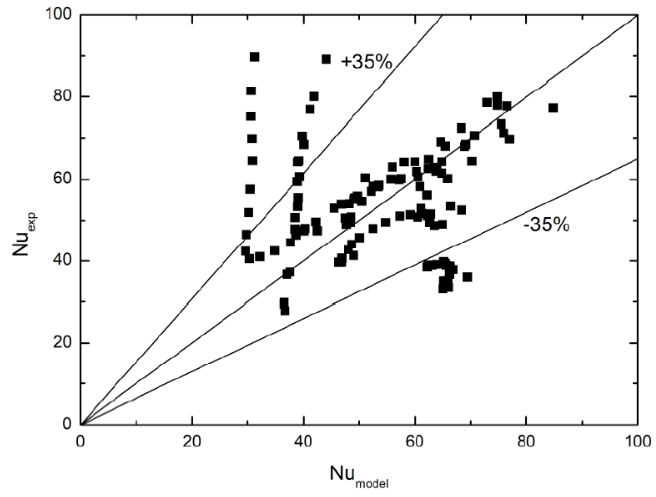
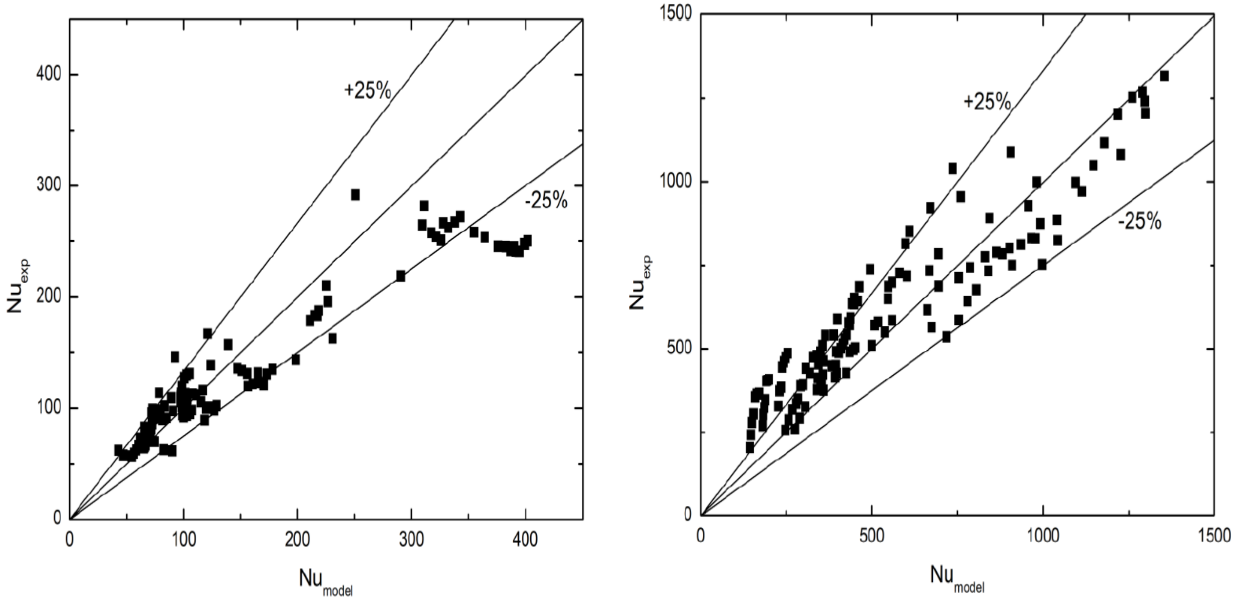


Figure 9. Comparison between the model and Tanimizu and Sadr's data



(a) top wall (b) bottom wall

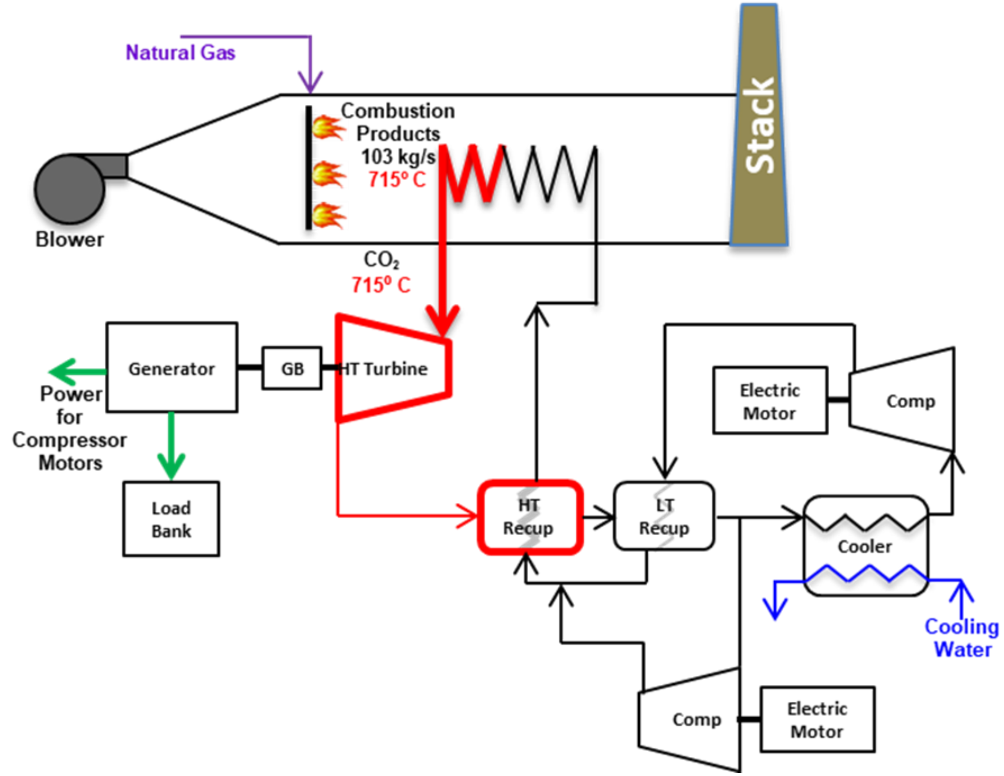
Figure 8. Comparison between the model and Adebiyi and Hall's data

**Kim et. al., "Investigation of heat transfer model for horizontal tubes at supercritical pressures of CO2", 2018 sCO2 symposium

Such large uncertainties are NOT acceptable by the gas turbine OEM's, and may be key for eventual market acceptability in terms of cost.

Motivation

STEP Loop Schematic



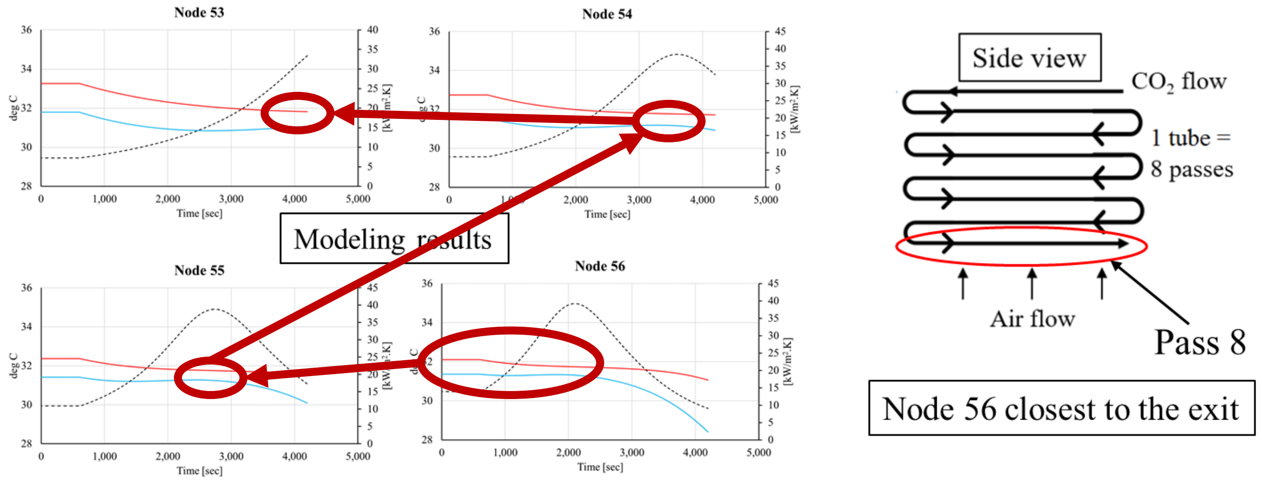
Hot Section Thermal Management

For life estimation of hot section components (e.g. liner, cooled turbine airfoils/platform) with not-so-expensive cooling strategies, accurate knowledge of coolant heat transfer coefficient is needed at ***Reynolds numbers beyond our current experience base*** ***(thermal resistance matching!!)***

Impact on Pre-cooler (& Compressor)

Transient Performance Analysis

- Scenario 3: ***15°C to -10°C swing in ambient temperature over 3600 sec.***



- Pinch point moves from node 56 to node 53 by end of the transient simulations.
- During simulation PP is observed at different nodes during different times of the simulation.
- Pinch point governs size of the HEX
- Thermal designers need to consider this for predicting performance and feasibility

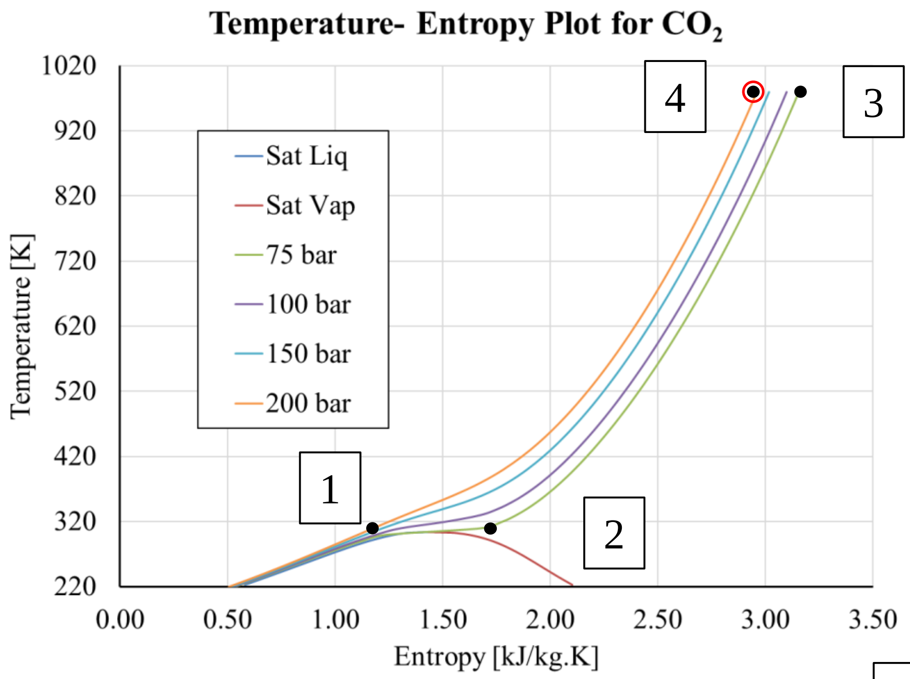
Deshmukh, A., Khadse, A., Kapat, J., 2019, "Transient Thermodynamic Modeling of Air Cooler in sCO_2 Brayton Cycle for Solar Molten Salt Application," ASME Turbo Expo 2019, Paper no. GT2019-91409

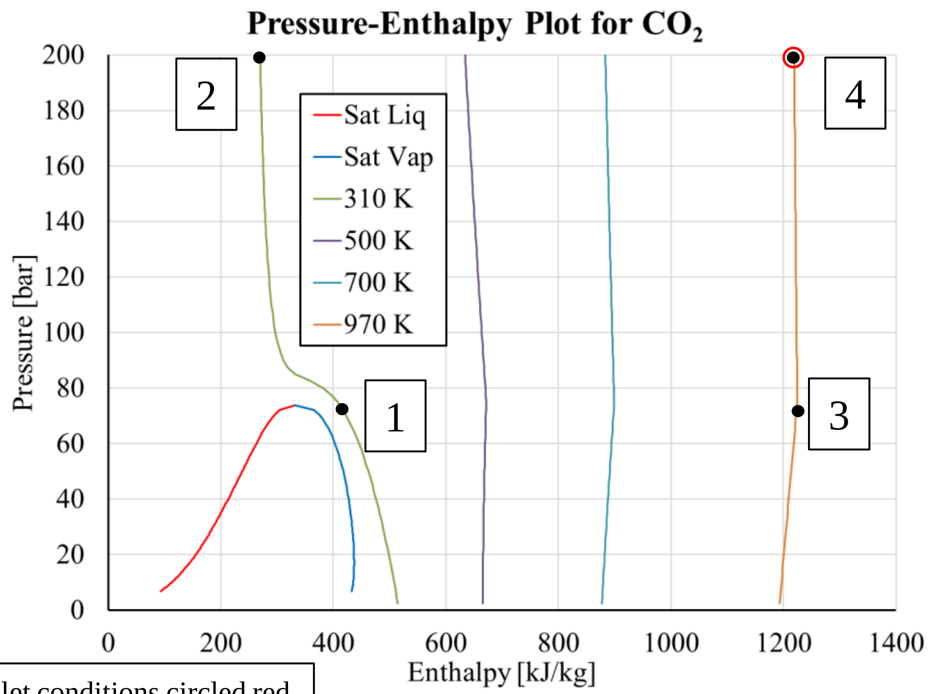
Motivation for the project

Why:

- Property variations
  - In heated wall cases, fluid properties such as Cp, k, ρ and μ can undergo non-linear changes
    - Fluid temperature changes inside the viscous & logarithmic layers, and thermal boundary layer
    - What happens to turbulent models? The models make frequent use of negligible property fluctuations.
- Standard Correlations
  - Conventionally utilized correlations, such as Dittus-Boelter, Petukhov or Gnilienski, are not valid for such severe variations in fluid properties.
  - Large fractional density variations can lead to onset of natural convective recirculation even in nominally forced convection flows
- HOW should we calculate bulk temp?
  - $\dot{m}h_b = \int_{Ac} \rho\, u\, C_p T\, dA_c$ ⟹ T_b ⟹ but will it still satisfy $htc \equiv \frac{q"}{T_w - T_b} \neq fn(q", \mathrm{sgn}(q"))$
  - We have teamed up with Prof Shih of Purdue to answer such and other fundamental questions with computational approach.

Compressibility Factor vs Correlation Uncertainty





STEP HEX inlet conditions circled red

State	Compressibility Factor
1	0.51
2	0.40
3	1.02
4	1.04

- Compressibility factor is less than one for lower temperatures in supercritical region
- Reaches closer to 1 at higher temperatures
- Non-ideal gas behavior near critical temperatures
- Ideal gas behavior at high temperatures

No market-acceptable design can be obtained without accurate uncertainty quantification for given confidence intervals.

Challenges to Instrumentation

Busbar causing problems

- Current density non-uniformity near busbars
- Sufficient contact between busbar and tubing necessary
- Heat generation in thick braided copper transmission lines

Tubing sizes and Pressure rating			
O.D. (in)	Wall thickness (in)	I.D. (mm)	Pressure rating (bar)
1/8	0.028	1.75	592
¼	0.035	4.57	352
½	0.065	9.4	352
7/8	0.109	16.7	324
1	0.12	19.3	324

Pressure derating factor of stainless steel = ***0.77*** at 538°C (1000°F)

ASME B31.3 Pipe Code causing rethinking in the way HTC is calculated from measurements:

Following measurement and setup techniques will not work:

- Local measurements by utilizing electrically heated foils over insulating substrate/wall, where paint-based or IR measurements indicate temp distribution through optical access
  - The setup must be rated for extreme pressure (200 bar). This can only be achieved using high grade materials such as stainless steel or Inconel
- Transient measurements with paints, over a thick insulating substrate, where paint indicate change of temp through some type of optical access
  - Since the heating is done by providing electricity to the metal tubing, the tubing cannot have any machining done. Otherwise this will cause non-uniformity in heat flux
- Segmented, heated copper-blocks with embedded thermocouples to give module-averaged thermocouples
  - Because of high pressure rating requirement the test section cannot be segmented or drilled for TC insertion.

Task 6 sCO_2 Heat Transfer Coefficient Measurements

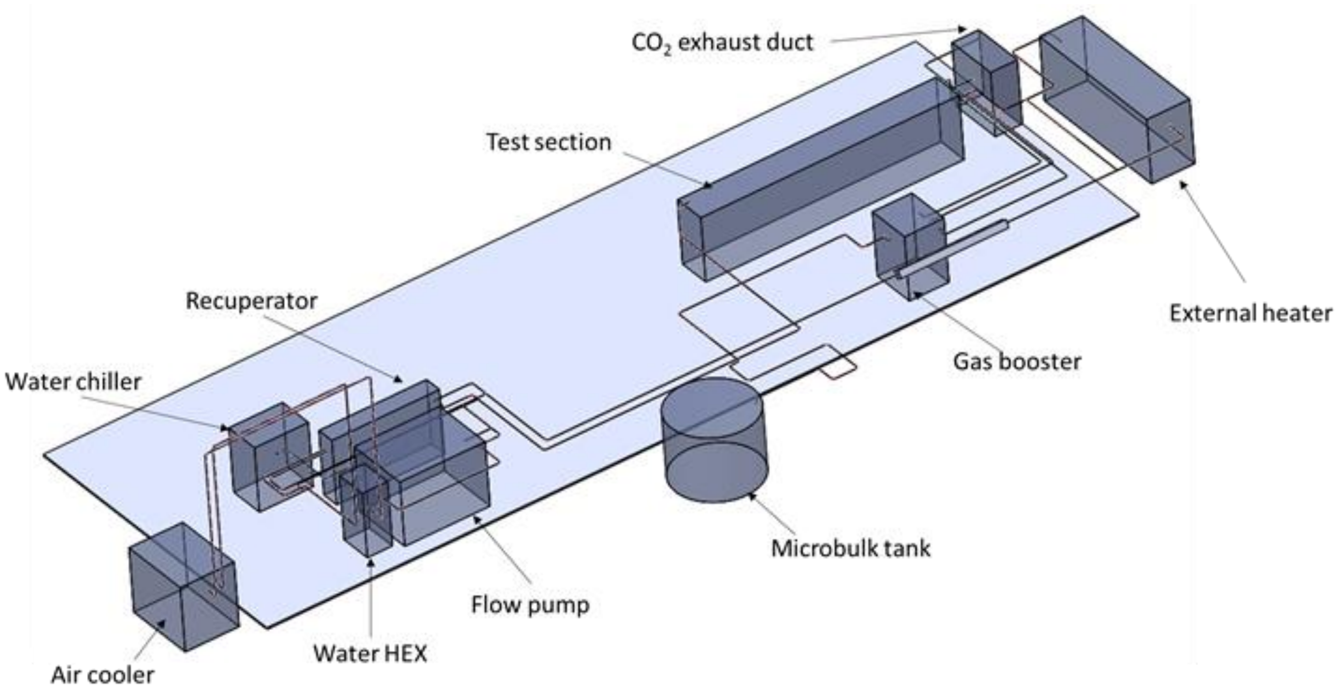
- Experiments are divided into 7 phases with increasing complexity and operating conditions for code compliance and validation for each subsequent phase.
  - Phase 1 is open loop experiments with high pressure air
  - Phase 2 is open loop experiments with sCO_2
  - Phase 3 is closed loop experiments involving Low Re (Re ~250,000) and Low T (420 K)
  - Phase 4 is closed loop experiments involving Low Re and High T (810 K)
  - Phase 5 is closed loop experiments involving High Re (Re ~750,000) and High T (810 K)
  - Phase 6 is closed loop experiments involving High Re and Extreme T (975 K) with Inconel test section

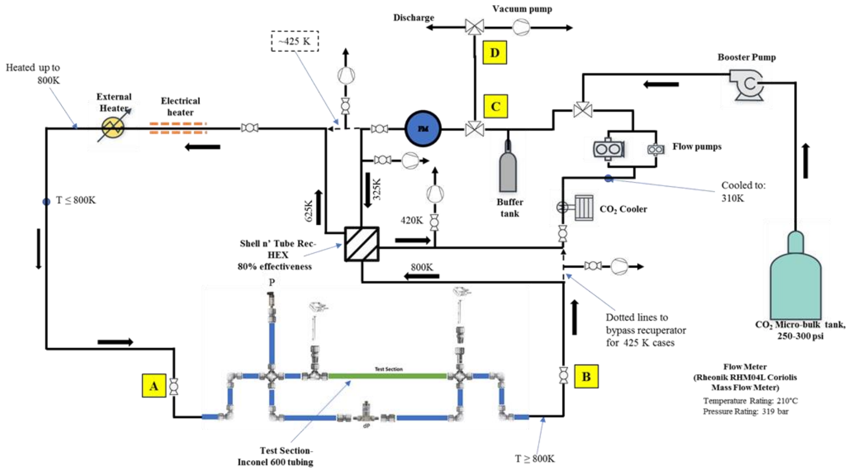
Subtask 6.1 Modification to existing facility

[for GE Film Cooling Expt – e.g. Natsui et al., *ASME J Turbomachinery*, v139(10)}

- Modifications to the existing CO_2 microbulk supply system are necessary for sCO2 flow experiments
- In the proposed setup for the heat transfer experiment, CO_2 is supplied from micro-bulk tank of CO_2, pressurized at 300 psi.
- Open loop operation (for low Re and lower Pressure ~10 MPa) as well as closed loop operation (high Re, high pressure ~20 MPa)

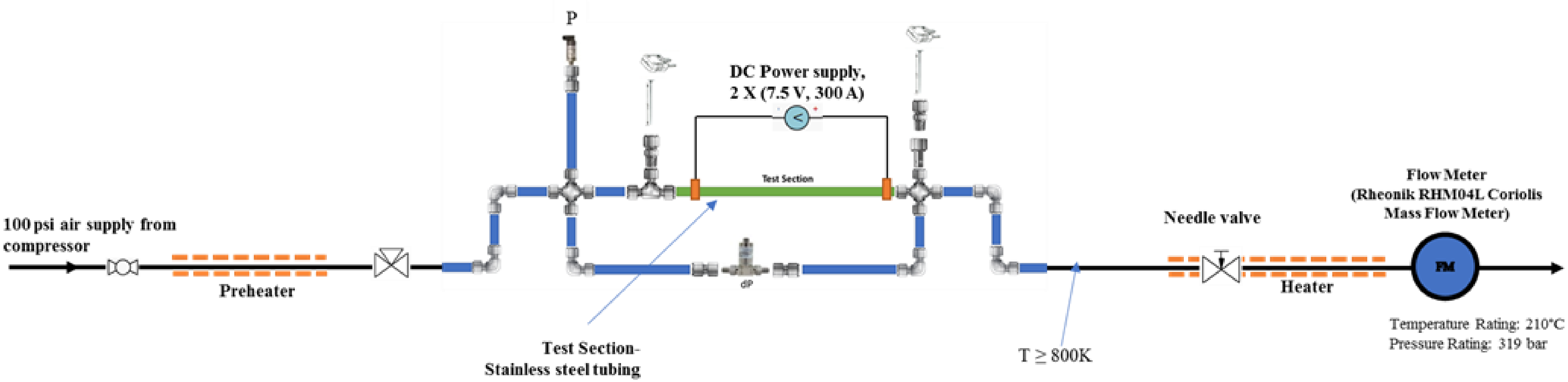
Safety features already in-built to the room: Negative pressure, positive ventilation to scoop out any CO_2 on floor, interlock for CO_2 supply, in addition to a large number of CO_2 alarms



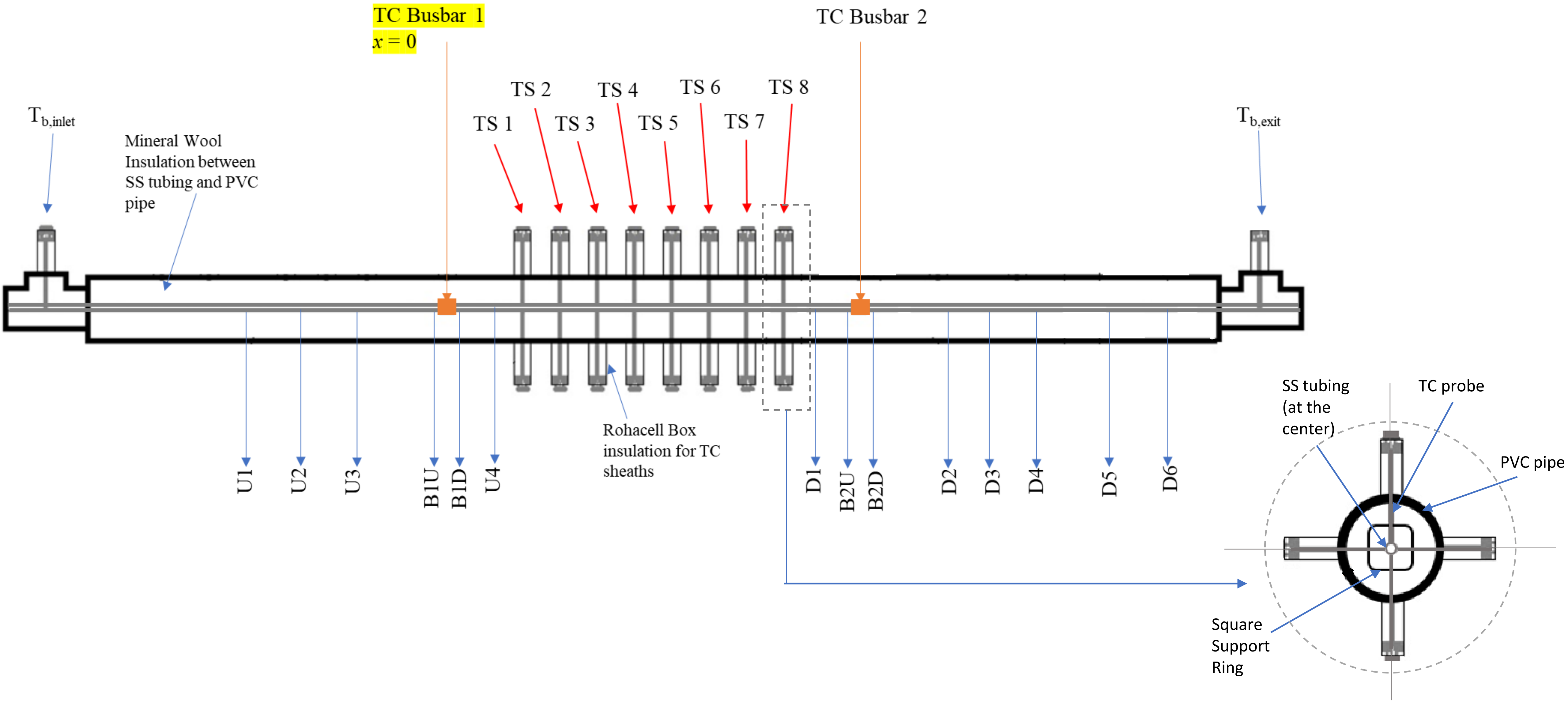


Subtask 6.2 Validation with high pressure air

- For validation of experimental process against atmospheric and high pressure air flow
- To use identical test section design, as to be used for all CO2 tests.
- The results obtained is compared with Dittus-Boelter or Gnilienski correlations for heat transfer
- To establish the baseline confidence interval for the tests to be undertaken in this task.
- To use building compressor
- Maximum temperature = ~370 K; Max pressure = ~6.9 bar (~100 psi)



Instrumented test section schematic



Heat Loss Tests

A. Heat loss/ No-Flow experiments

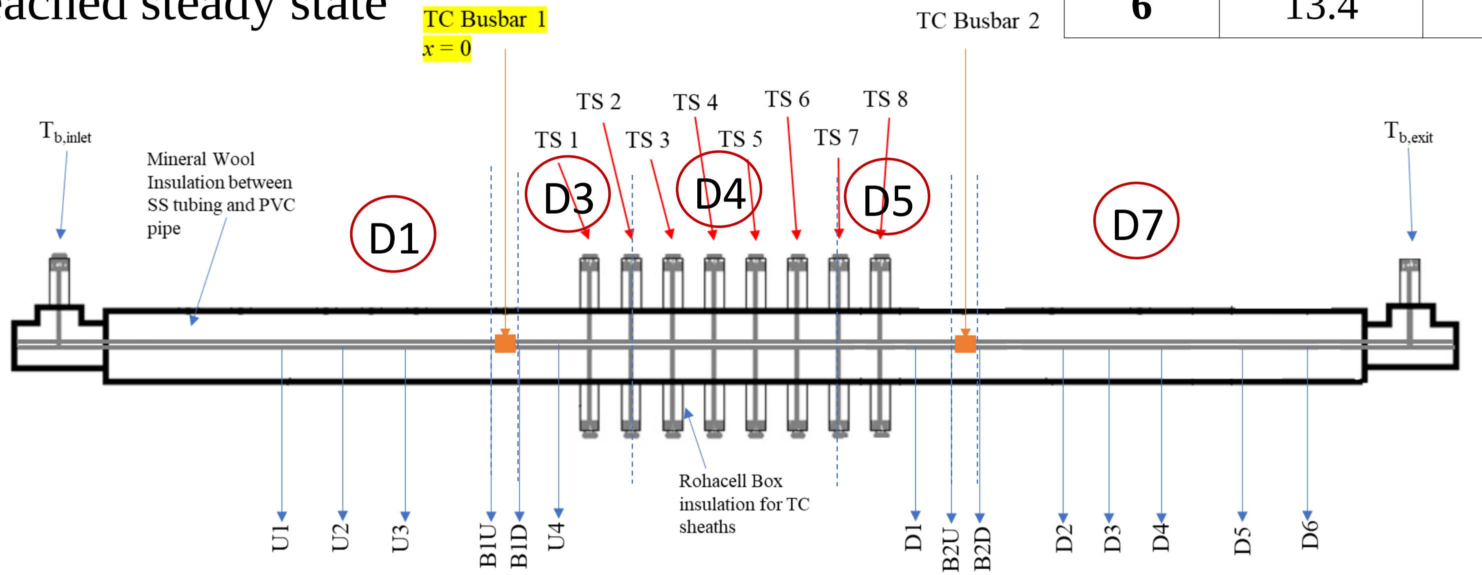
- Five heat loss tests have been carried out with different electrical heat flux and ambient temperature
- Power is supplied to the test section with no flow through the inside of the test tube
- Temperatures were monitored until the system reached steady state

Heat loss test conditions

Test	Power [W]	T_{amb} [^oC]
3	7.4	25.5
4	21.4	28.6
5	10.2	25.7
1	13.2	26.3
6	13.4	29.3

Summary of U_0 results for all domains

Domain	**Average U_o**	**Std dev**
	W/m²K	W/m²K
D1	1.76	0.18
D3	6.16	0.25
D4	5.98	0.23
D5	6.11	0.25
D7	1.67	0.10



Phase I – High Pressure Air Experiments Summary

Case I	$T_{ambient}$	Inlet Reynolds #	Power	Inlet Pressure	$Nu_{Calculated}$	Nu_{DB}	Nu_{Gnlnsk}	Dev Nu_{DB}	Dev Nu_{Gnlnsk}
	[deg C]	[-]	[Watt]	[bar]	[-]	[-]	[-]		
Max Flow #1	27.0	21210	118	6.69	57.2	56.3	52.7	-2%	-9%
Max Flow #2	26.6	21694	120	6.76	56.2	57.3	53.7	2%	-5%
Max Flow #3	26.5	21739	117	6.76	54.5	57.5	53.8	5%	-1%
Max Flow #4	26.0	21814	120	6.77	54.5	57.6	53.9	5%	-1%
Max Flow #5	26.0	21872	120	6.7	58.7	57.7	54.0	-2%	-9%
Medium Flow Case #6	32.1	17491	103	7.2	48.7	48.4	45.5	-1%	-7%
Low Flow Case #7	32.0	12,700	89	3.4	38.1	37.6	35.5	-1%	-7%

Case II	$T_{ambient}$	Inlet Reynolds #	Power	Inlet Pressure	$Nu_{Calculated}$	Nu_{DB}	Nu_{Gnlnsk}	Dev Nu_{DB}	Dev Nu_{Gnlnsk}
	[deg C]	[-]	[Watt]	[bar]	[-]	[-]	[-]		
Max Flow #1	32.5	26008	133	8.4	67.2	66.4	62.0	-1.2%	-8.5%
Max Flow #2	32.0	22323	135	7.1	63.1	58.5	54.7	-7.9%	-15.3%
Max Flow #3	32.1	22266	134	7.1	62.6	58.4	54.6	-7.3%	-14.6%
Max Flow #4	31.8	22282	136	7.1	58.5	58.3	54.6	-0.4%	-7.2%
Max Flow #5	29.9	22535	137	7.1	53.2	58.8	55.1	9.6%	3.4%
Medium Flow Case	32.1	17491	103	7.2	46.9	48.4	45.5	3.1%	-2.9%
Low Flow Case	32.0	12,700	89	3.4	38.1	37.6	35.5	-1.3%	-7.3%

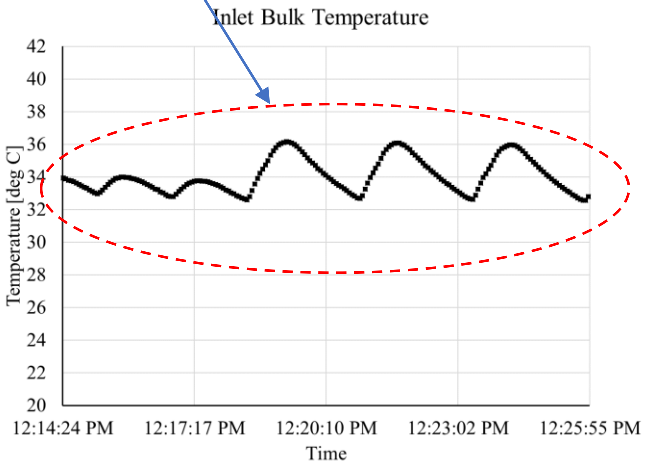
Results with heat loss data when ventilation system in the room is running

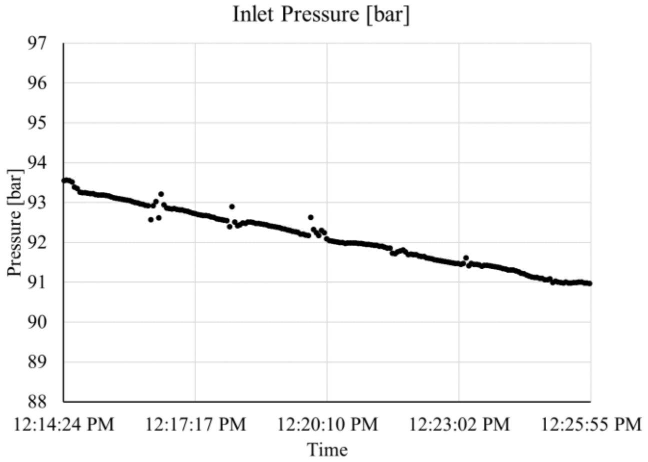
- To check repeatability of the setup, 7 cases are studied with 5 cases with same mass flow rate and power.
- ~21.5k is the maximum Re that can be achieved using the available high pressure air source
- The other two cases have mass flow rate lower than the maximum flow rate case.
- Different results when used heat loss data from different conditions in the room

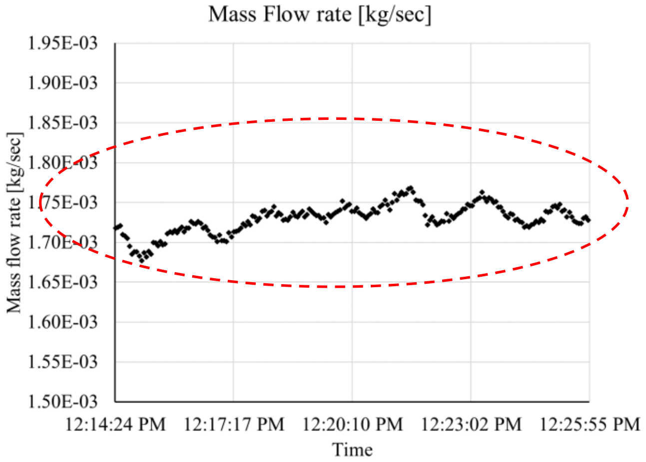
Phase II – Open Loop sCO_2 HT Experiments

Test 1

Initial sCO_2 run observations
- No leaks were observed
- Pressure drops constantly when the CO_2 is flowing out from the cylinder
    - Drop of 2 bar is observed for 10 min run
- Mass flow rate fluctuations are within 3% of mean value
- Inlet bulk temperature also fluctuates because of fluctuations in power from pre-heater
    - ***Needed better strategy for controlling inlet bulk temperature***



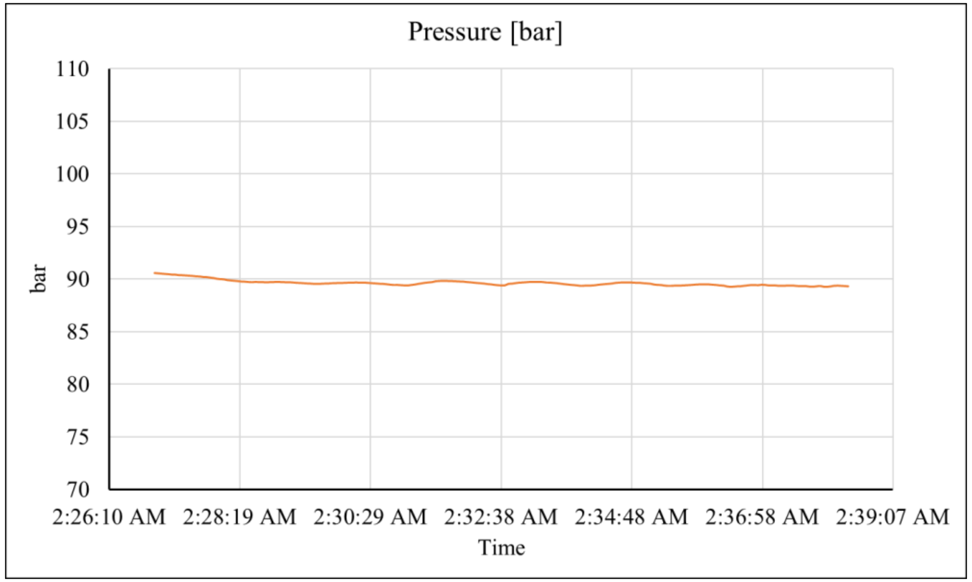




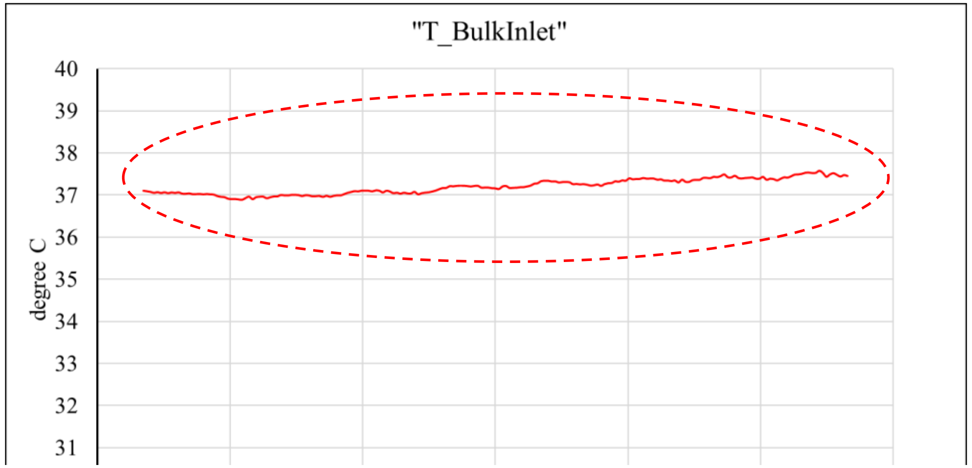
Initial sCO_2 run
- Inlet pressure = 9.21 MPa
- Inlet bulk flow temperature = 34.1 ^{o}C
- Mass flow rate = $1.73 \cdot 10^{-3}$ kg/s

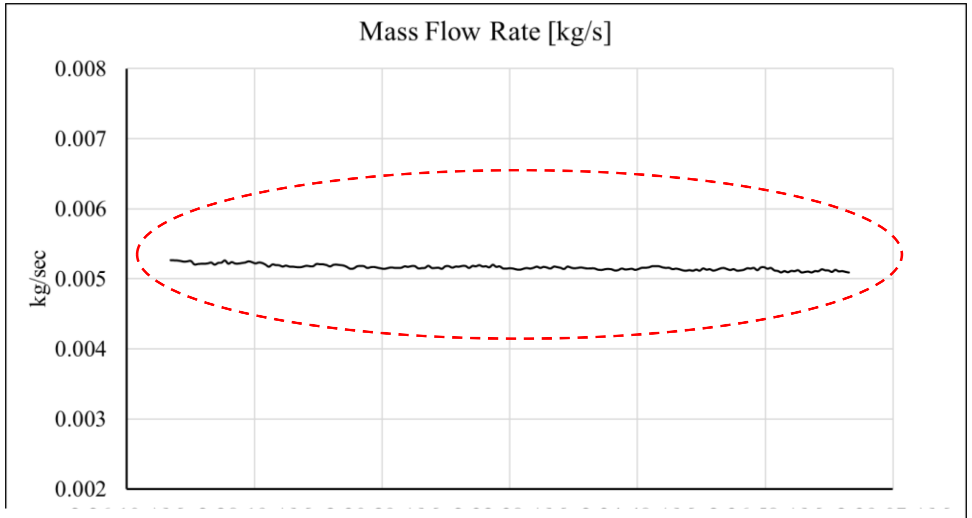
sCO_2 Run: Testing conditions

- Inlet pressure = 8.96 Mpa, Inlet density = 600 kg/m^3
- Inlet bulk temperature = 310.4 K
  - Variation within 0.5 deg C after using the constant power variac transformer compared to 4 deg variation previously
  - Required power is calculated by difference in measured flow temperature without any heating and required inlet bulk temperature
  - Observed variation of 0.5 deg C is due to slight decrease in mass flow rate causing temperature rise of 0.5 deg C
- Mass flow rate = $5.16 \cdot 10^{-03}$ kg/s, Inlet velocity = 0.12 m/s
- Inlet Re = 15673



Test 3





All these fluctuation data PLUS multiple repetitions <u>and</u> replications will lead to accurate quantification of uncertainties for a given, say 95%, confidence interval.

sCO_2 Run: Variations with Bulk, Top wall and Bottom Wall

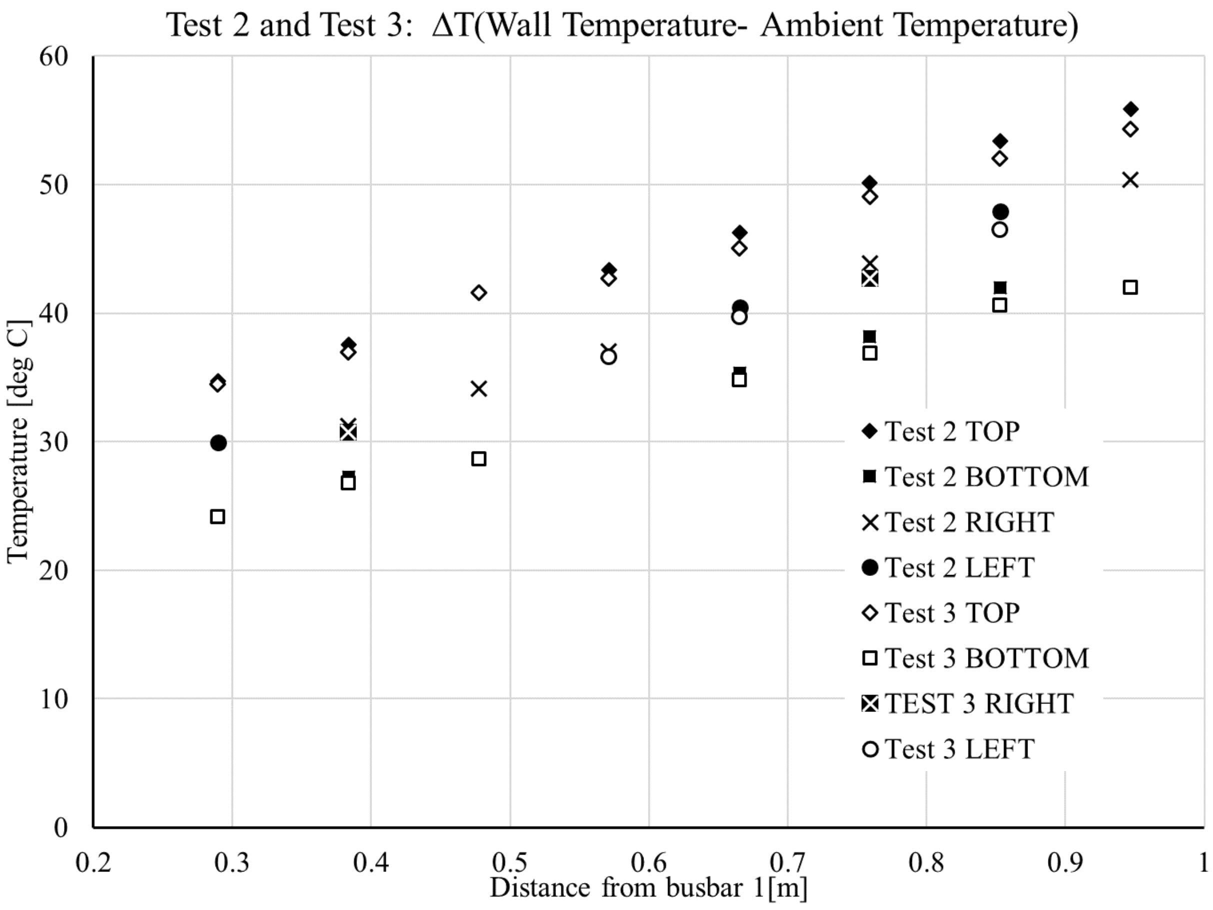
- Two tests are carried out with same test condition to check repeatability with same power to the test section
- Solid marker for Test 2 and hollow marker for Test 3
- Test 3 has top and bottom TCs at more stations

<u>Test 2 conditions</u>

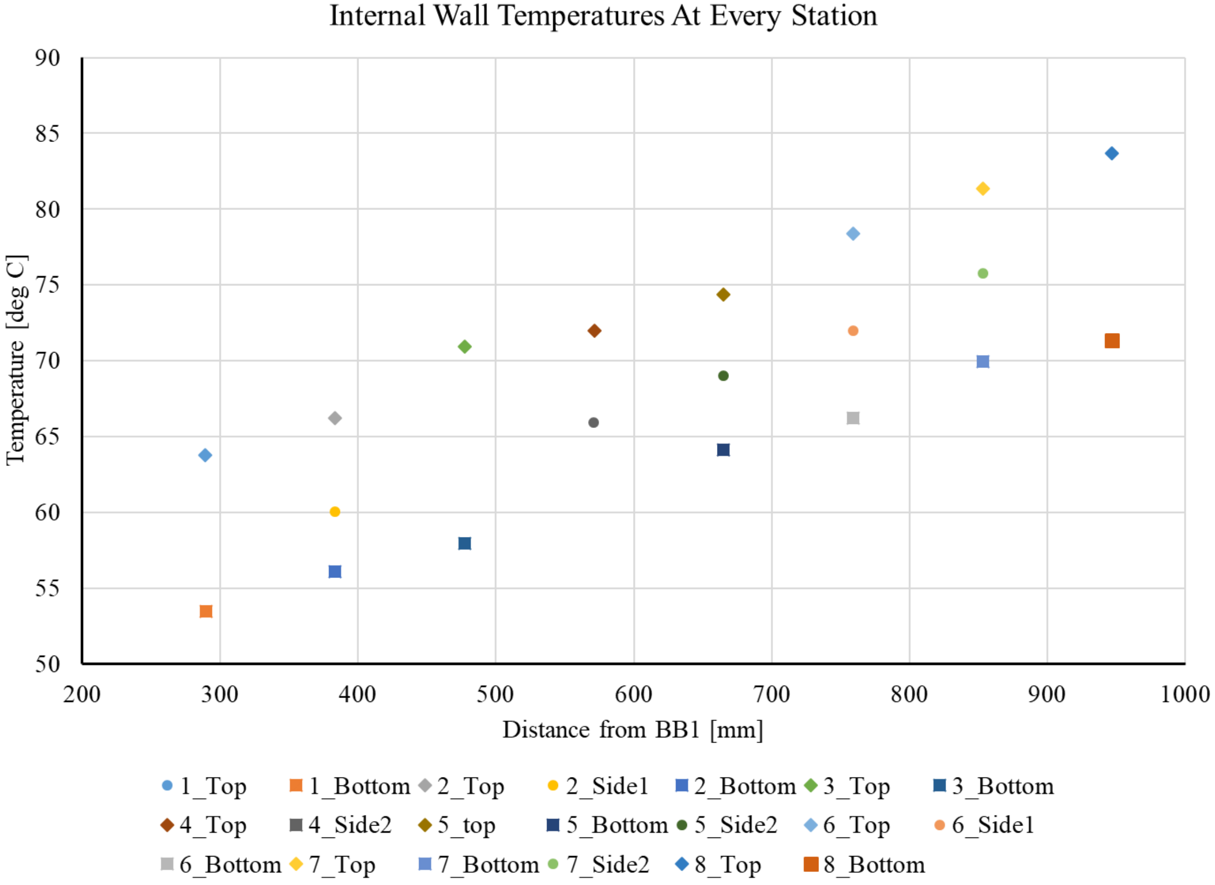
- Inlet pressure = 8.94 MPa
- Inlet bulk flow temperature = 37.8 ^{o}C
- Mass flow rate = 5.30×10^{-3} kg/s

<u>Test 3 conditions</u>

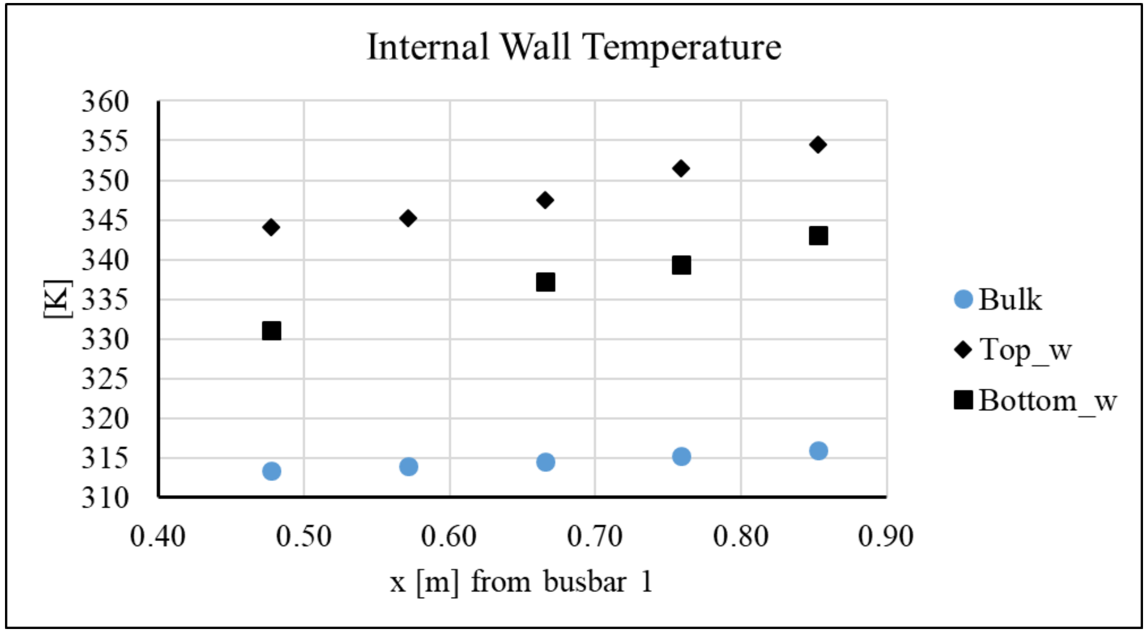
- Inlet pressure = 8.96 MPa
- Inlet bulk flow temperature = 37.2 ^{o}C
- Mass flow rate = 5.16×10^{-3} kg/s



sCO_2 Run: Variations with Bulk, Top wall and Bottom Wall



Test 3



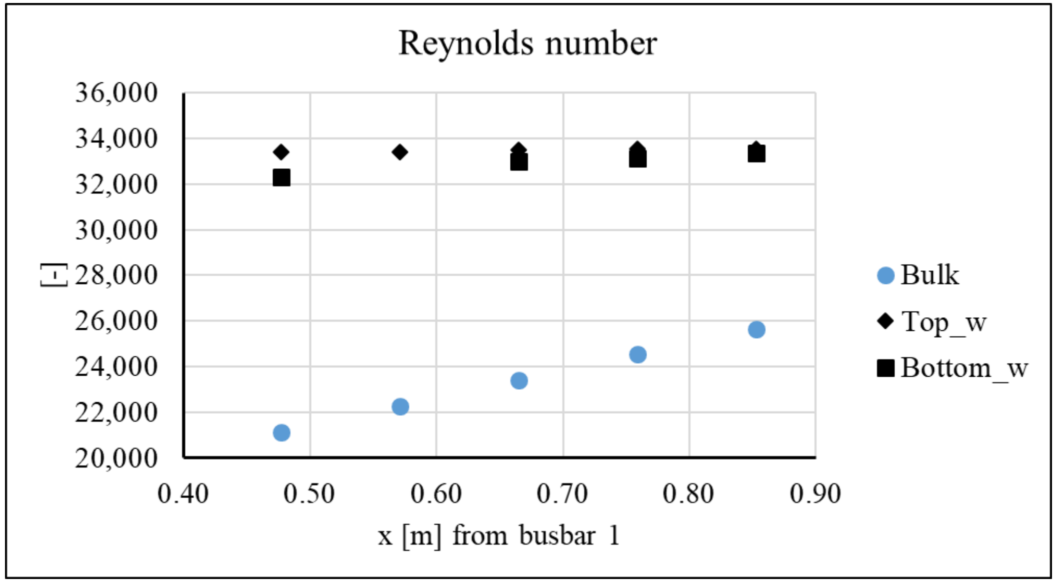
- Bottom TC at station 4 was not connected during the test
- Waiting on delivery of extra DAQ module to measure temperature at all 4 positions at every station
- Difference of ~10 deg C was observed between top and bottom internal wall temperatures
- Strategy of taking average T similar to high pressure air case fails
- Leads to θ-distribution of properties and heat transfer (loss to ambient as well as convective HTC)
- Need to consider θ-distribution in Nu/HTC calculation for “heated” domains

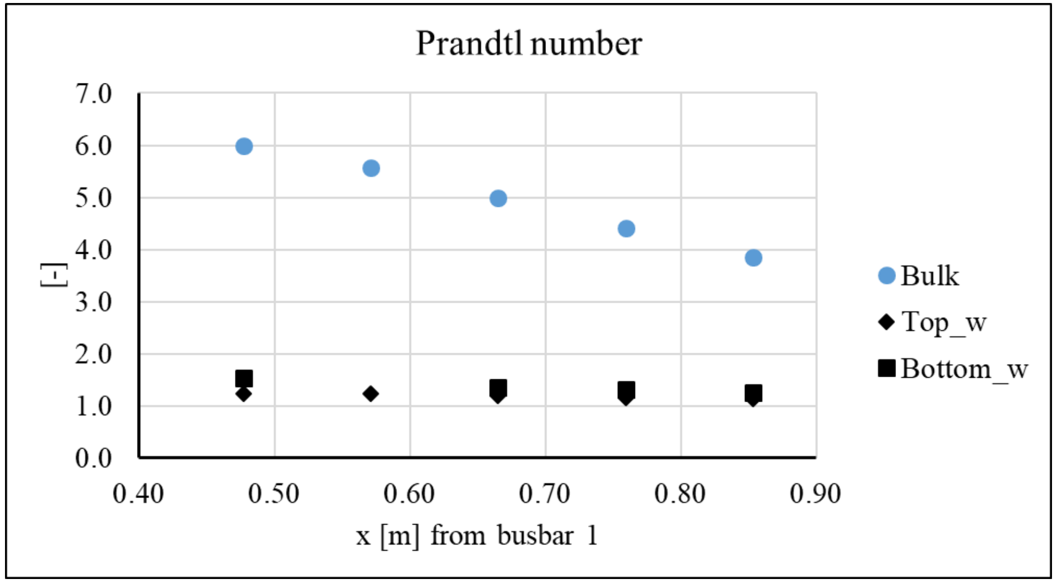
sCO_2 Run: Variations with Bulk, Top wall and Bottom Wall

Test 3

- Parameters plotted here are shown for stations TS3 to TS7 for top and bottom internal wall positions and bulk flow.
- Bottom TC at station 4 was not connected during the test

- Re and Pr are different when used viscosity from bulk flow and wall measurement
- Nu correlation should involve wall as well as bulk parameters

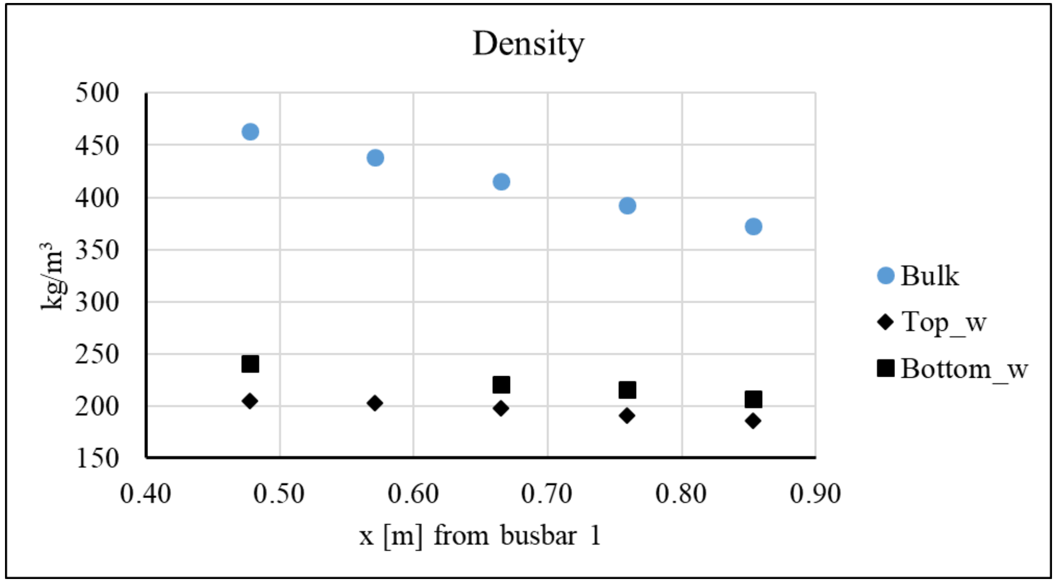


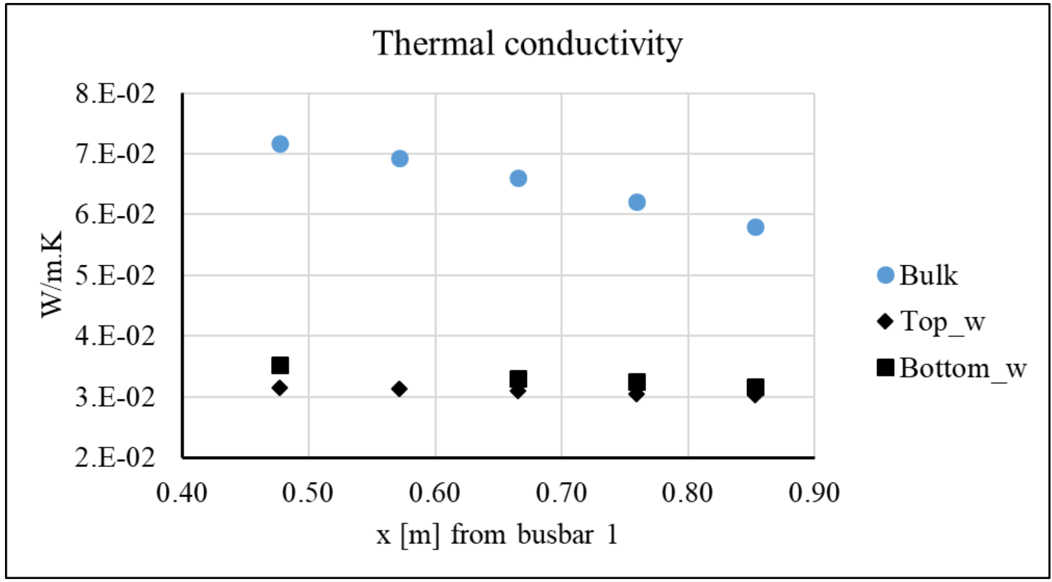


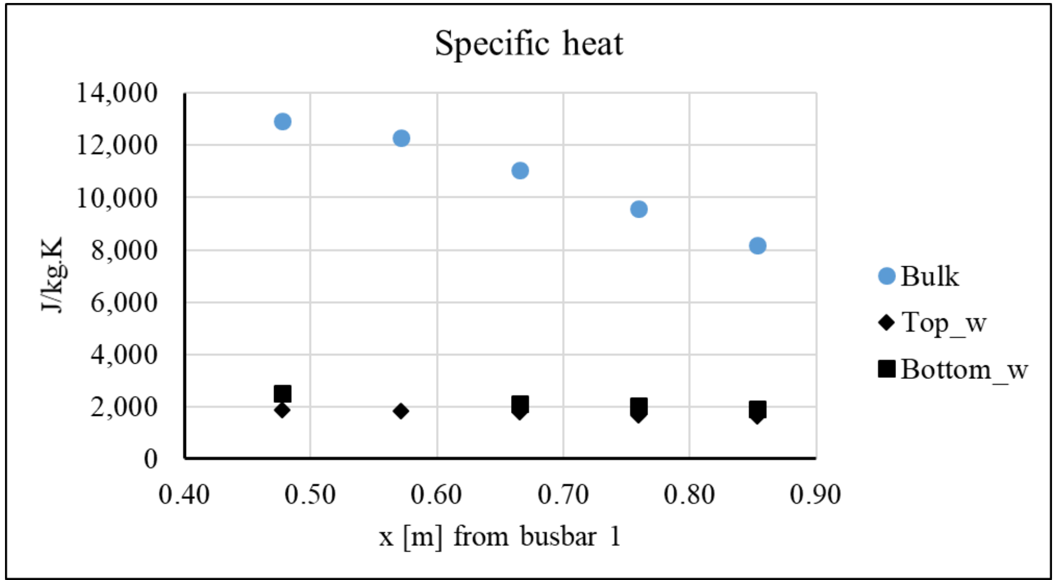
sCO_2 Run: Property variation along length at Bulk, Top wall and Bottom Wall

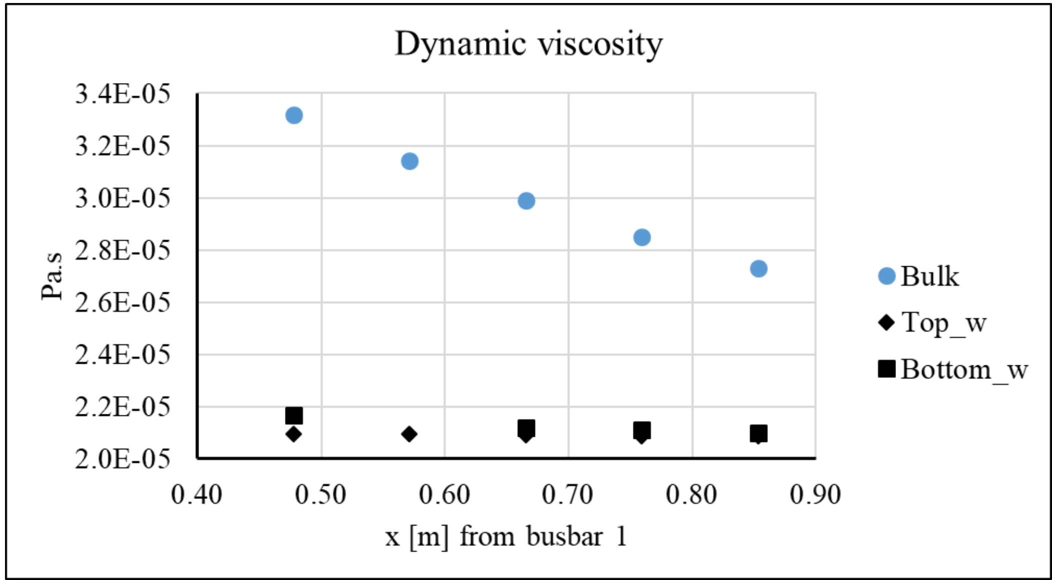
Test 3

- Parameters plotted here are shown for stations TS3 to TS7 for top and bottom internal wall positions and bulk flow.
- Bottom TC at station 4 was not connected during the test





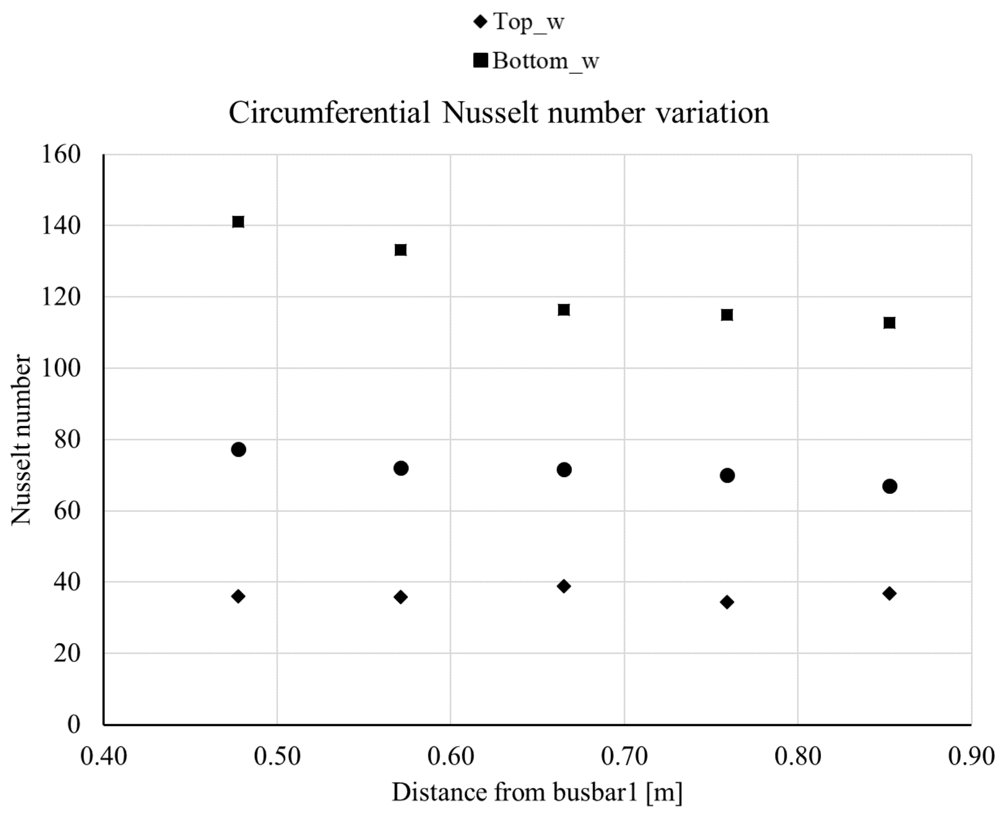




sCO_2 Run: Heat transfer at Top wall and Bottom Wall

Test 3

- Nu variation here is shown for Station 3 to Station 7
- *Circumferential conduction* within a cross section is also considered for the HTC calculations
- k_{fluid} for Nu calculation is taken from bulk flow value
- Mass flux = 74.54 kg/m^{2}s
- Heat flux = 9.87 kW/m^2
- T_{in} = 37.2 ^oC
- Higher Nu at bottom wall due to additional convection by buoyancy



<u>As percentage of heat addition (V·I)</u>

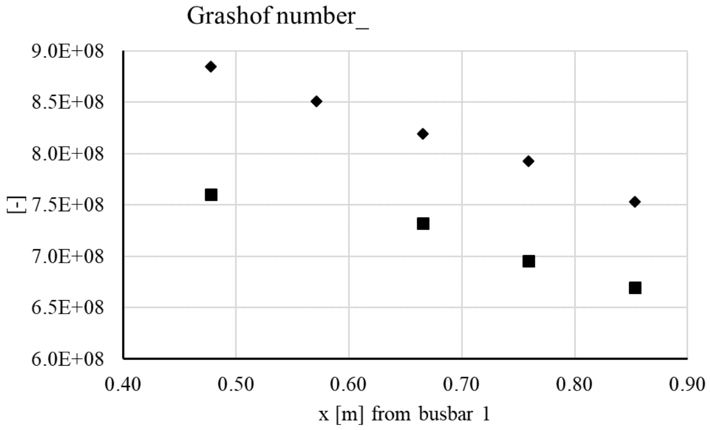
- $Q_{loss,upstream}$ (before TS1) = 0.04%
- $Q_{gain,upstream\ busbar}$ = 0.41%
- T at TS2 changes by about 3K (or 10% of driving temp difference)
- $Q_{radial\ loss}$ = 6.24%

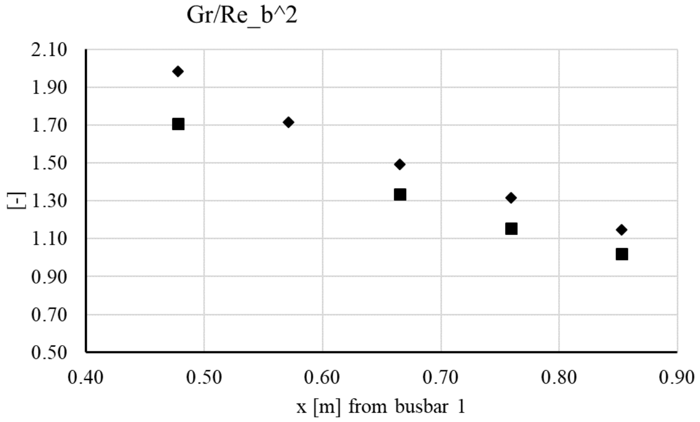
sCO_2 Run: Buoyancy effects in horizontal flow

Test 3

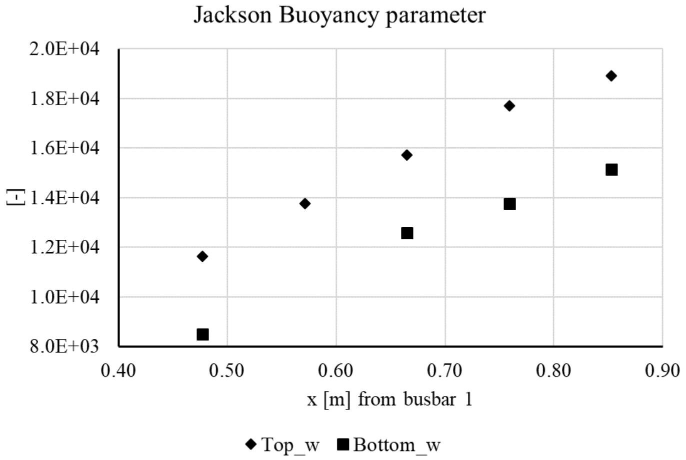
Grashof number

- Measure of the ratio of buoyancy forces to viscous forces
  - From Jackson 1975 paper *
  - $Gr/Re_b^2 < 10^{-3}$ buoyancy effects becomes negligible**
- For our case, this value is very high compared to 1/1000
- ***Jackson*** proposed Buoyancy parameter used by Adebiyi and Hall
- For absence of buoyancy $Gr_b Re_b^{-2} \left(\frac{\rho_b}{\rho_w}\right)\left(\frac{x}{D}\right)^2 < 10$
- Observed very high values of buoyancy parameter
- Buoyancy effects are considerable





$$Gr = \frac{(\rho_w - \rho_b) g d^3}{\rho_b \nu_b^2} = \frac{(\rho_w - \rho_b)\rho_b g d^3}{\mu_b^2}$$



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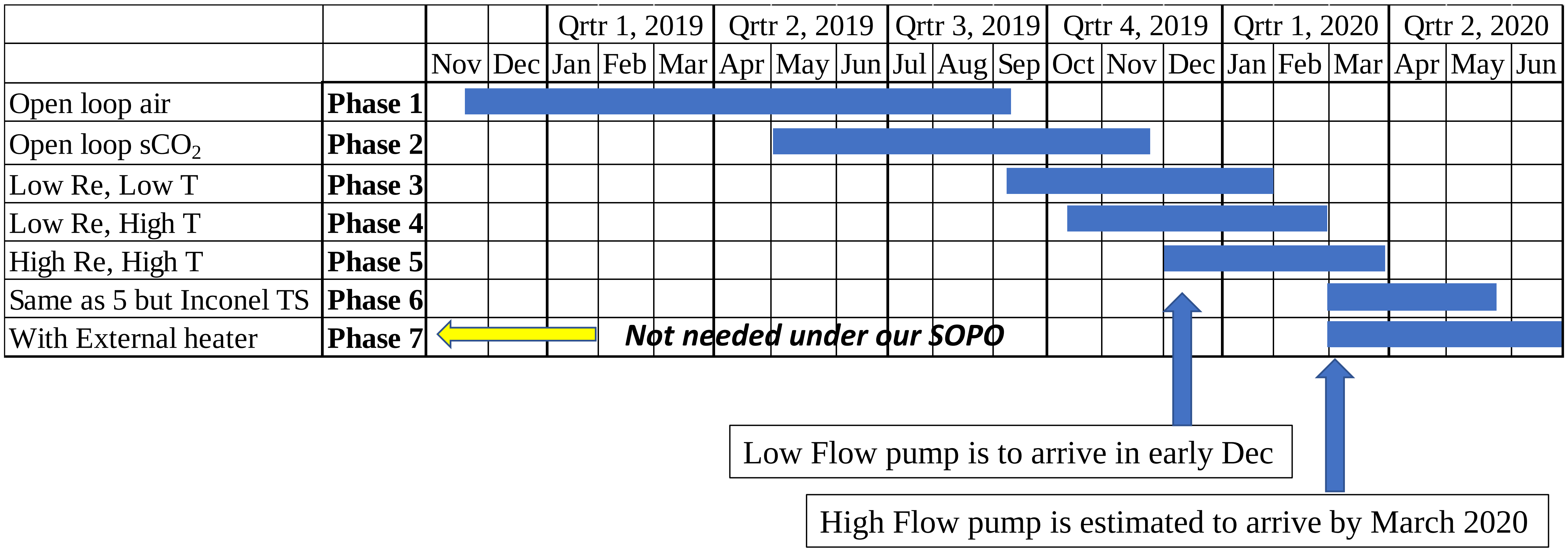
- *Jackson, J. D., Hall, W. B., Fewster, J., Watson, A., and Watts, M. J., 1975, ‘‘Heat Transfer to Supercritical Pressure Fluids,’’ U.K.A.E.A. A.E.R.E.-R 8158, Design Report 34.
- Lee, S. H., and Howell, J. R., 1998, ‘‘Turbulent Developing Convective Heat Transfer in a Tube for Fluids Near the Critical Point,’’ Int. J. Heat Mass Transf., 41, No. 10, pp. 1205–1218.
- Petrov, N. E., and Popov, V. N., 1985, ‘‘Heat Transfer and Resistance of Carbon Dioxide Cooled in the Supercritical Region,’’ Thermal Engineering, 32, No. 3, pp. 131–134.
- Zhou, N., and Krishnan, A., 1995, ‘‘Laminar and Turbulent Heat Transfer in Flow of Supercritical CO2 ,’’ Proceedings of the 30th National Heat Transfer Conference, Vol. 5, ASME, Portland, OR, pp. 53–63.
- Kakac, S., 1987, ‘‘The Effect of Temperature-Dependent Fluid Properties on Convective Heat Transfer,’’ Handbook of Single-phase Convective Heat Transfer, S. Kakac et al., eds., John Wiley & Sons, New York, pp. 18.1–18.56.
- Adebiyi, G. A., & Hall, W. B. (1976). Experimental investigation of heat transfer to supercritical pressure carbon dioxide in a horizontal pipe. International journal of heat and mass transfer, 19(7), 715-720.

Task 6 Timeline (Updated)



Thank You.