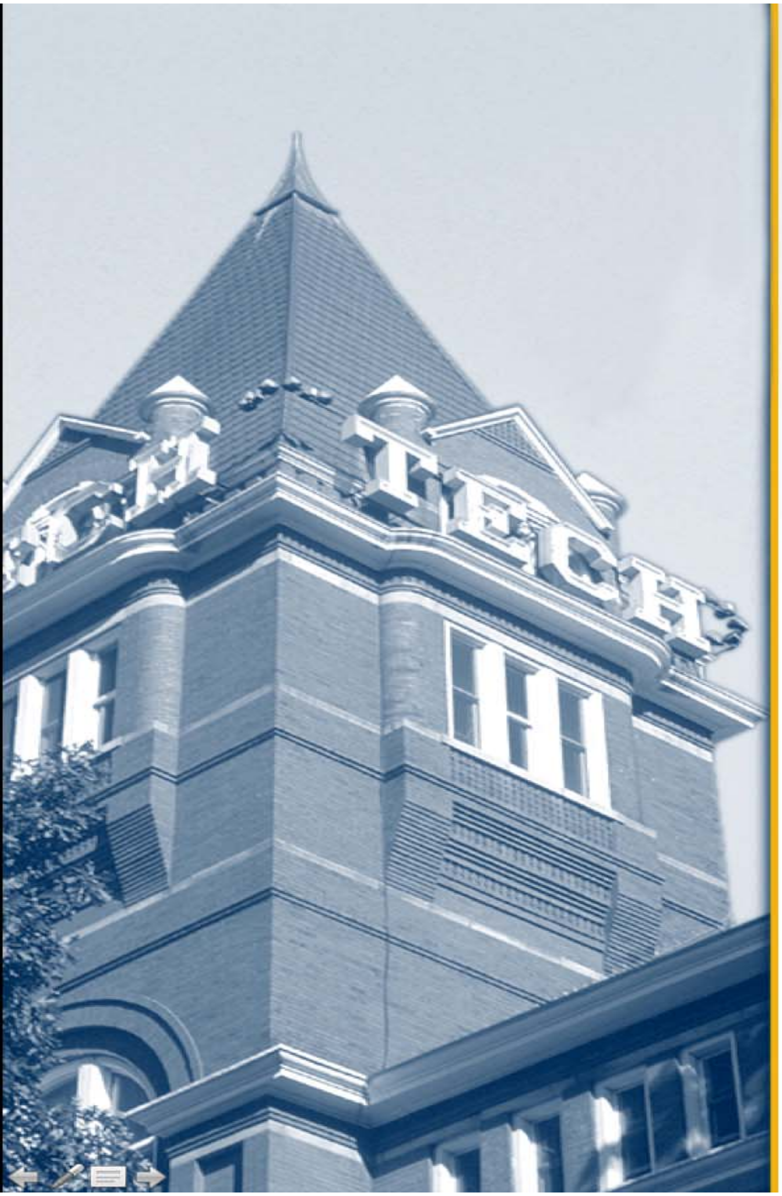

Thermal-hydraulic performance of discontinuous fin heat exchanger geometries using Supercritical CO_2 as the working fluid

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The 6th International Symposium – Supercritical CO_2 power cycles

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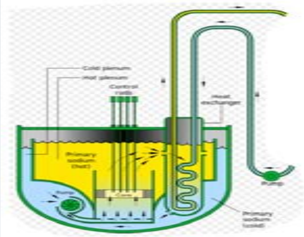
Georgia Institute of Technology

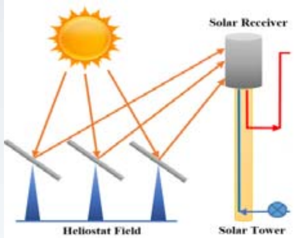
Outline

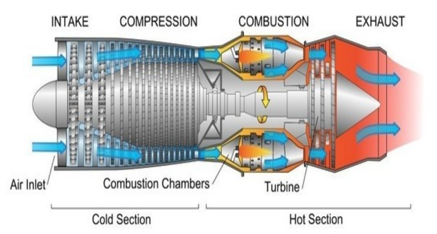
- Introduction
- Experimental test facility
- Data analysis procedure
- Results and discussion
  - Correlation development and comparison with existing geometries
- Summary/Conclusions

Supercritical CO_2 Power Cycles – Overview

- U.S. Department of Energy (DOE) initiative for clean and efficient energy conversion.
- Supercritical CO_2 power cycles are gaining increasing attention compared to the widely-used steam Rankine cycles and gas Brayton cycles.



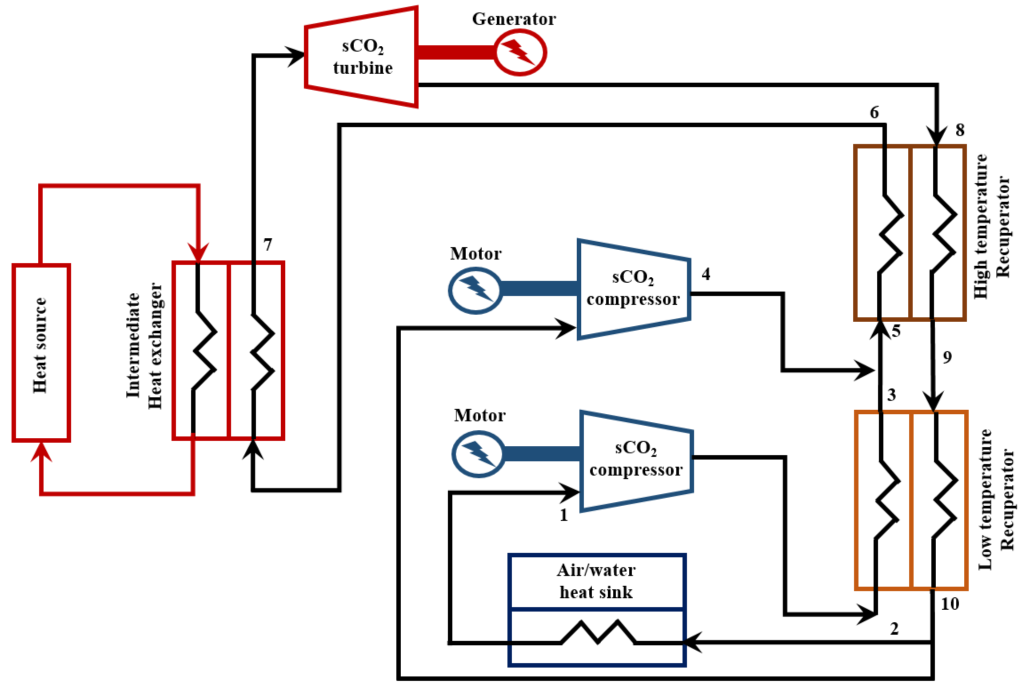




DOE NE (Gen IV NPPs)

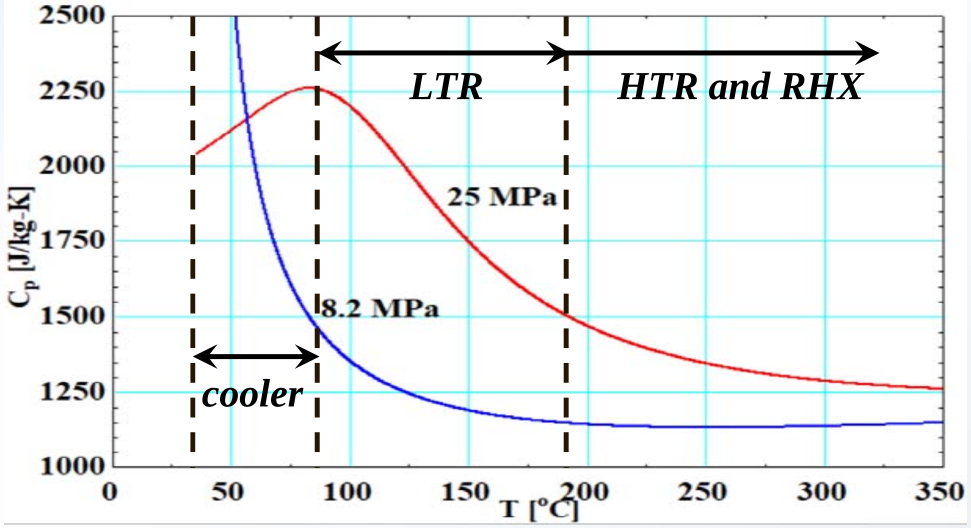
DOE EERE (Gen III CSPs)

DOE FE



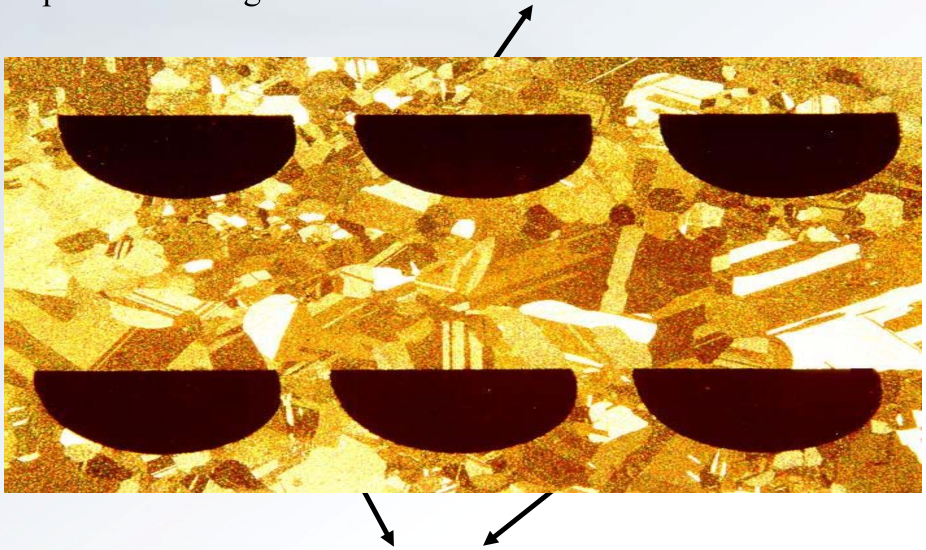
Supercritical CO_2 Power Cycles – Overview

- Moderate critical pressure (7.38 MPa) and critical temperature ~ 31.1°C is near ambient temperature (Dry air cooling feasible)
- Density resembles that of liquid near the critical point
    - Reduced compression work, larger W_{net}
- Higher energy density compared to H_2O and He
    - Allows Compact turbomachinery to achieve same power
- Specific heat mismatch between high and low pressure sides requires two stage recuperation and split compression to increase recuperation effectiveness (> 2/3rd of heat is recuperated)



Printed Circuit heat exchangers for sCO_2 power cycles

- Channels/patterns are photo chemically etched on to a plate.
- Semicircular channels are convenient to etch because of the inherent process etching corner radius



- Multiple plates are diffusion bonded by applying pressure at high temperature (50-80% of the melting temperature) to form monolithic core
- Promotes grain growth at the interface (No foreign material as in the case of brazing)

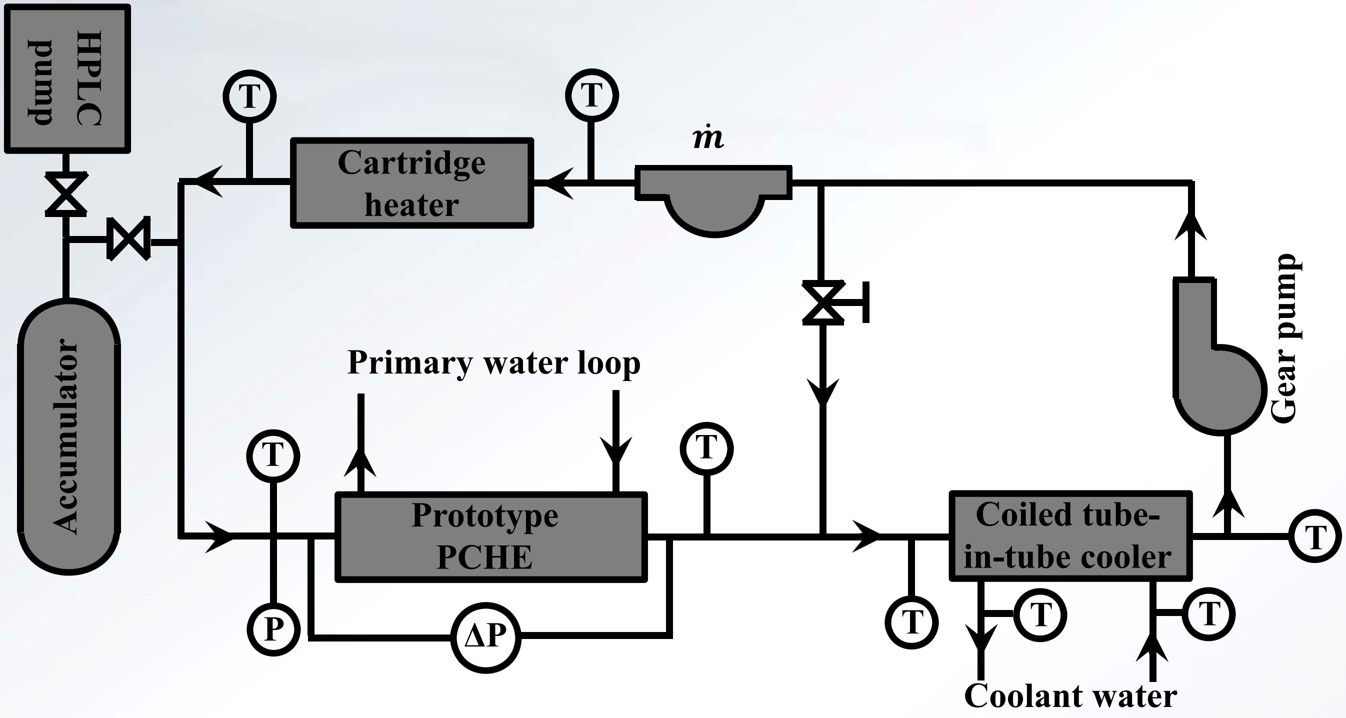
Chemical etching and diffusion bonding

- Typical size of heat exchanger core, 1.5 x 0.6 x 0.6 [m]
- Multiple cores are welded together to form a heat exchanger without headers

Core Welding

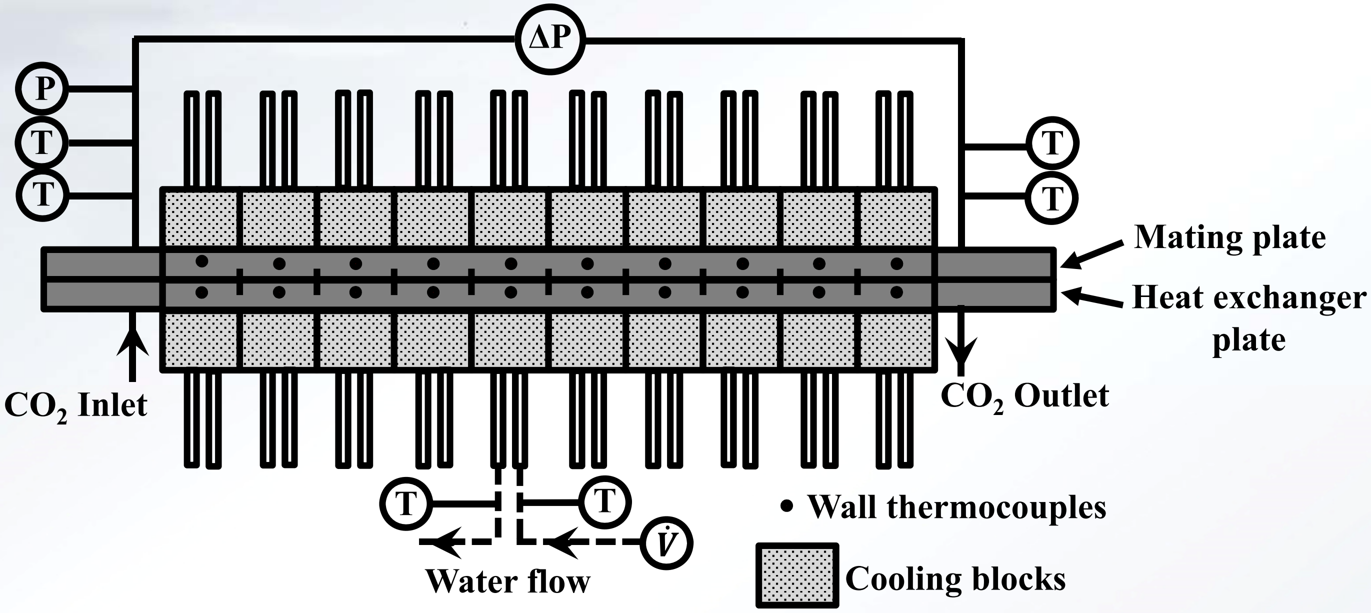
Source: Heatric

Experimental facility



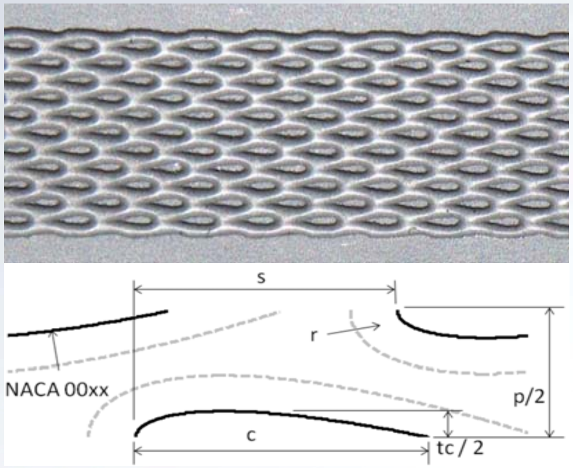
Component	Capabilities
HPLC pump	Up to ~ 10,000 psi
Circulation pump	0.6 – 7.0 GPM
Coriolis flow meter	0 – 0.27 Kg/sec
Pre-heater	Maximum 5.5 KW_{th}
Water chiller	Maximum 25 KW_{th}
Pre-cooler	Double tube HEX
Accumulator	~ 13 Gallons (49 L)

Test section

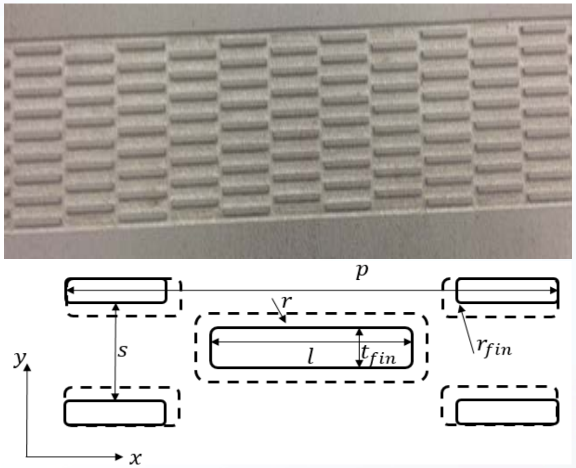


- RTD probes are used to measure inlet and outlet temperature to the Heat exchanger
- Wall temperatures are measured using 20 type K-thermocouples (10 on each plate)
- Volumetric flow rate of water to each cooling block measured using turbine type flowmeters
- Inlet and outlet temperatures of water to each cooling block are measured using type K-thermocouples
- Inlet pressure measured using gage pressure transmitter and pressure drop measured using differential pressure transmitter.

Heat exchanger test plates



	4mmNACA0020	
	Design	Measured
Chord width, *c* (mm)	4	3.566
Thickness/Chord length	0.2	0.202
Fillet radius, *r* (mm)	0	0.795
Channel depth, *h* (mm)	0.95	0.685
Axial pitch, *s* (mm)	3.5	3.466
Lateral pitch, *p* (mm)	3.6	3.657
Plate thickness, *t* (mm)	6.3	
Number of Rows (N_x)	144	
Airfoils per Row (N_y)	6	
Hydraulic diameter, D_h (mm)	1.205	1.112
Unit cell heat transfer area, A_s (mm^2)	30.18	24.94
Cross-sectional area, A_c (mm^2)	15.96	12.07
Measured Relative roughness	7.259e-3	



	Design	Measured
Fin thickness, t_{fin} (mm)	0.65	0.65
Fillet radius, r (mm)	0	0.47
Fillet radius, r_{fin} (mm)	0	0.18
Fin depth, *h* (mm)	0.65	0.65
Fin spacing, *s* (mm)	1.95	1.95
Fin length, *l* (mm)	9.025	7.69
Lateral pitch, *p* (mm)	18.05	17.68
Plate thickness, *t* (mm)	6.3	
Number of unit cells along length (N_x)	28	
Number of unit cells per row (N_y)	9	
Hydraulic diameter, D_h (mm)	0.9502	0.9973
Unit cell heat transfer area, A_s (mm^2)	82.01	91.133
Cross-sectional area, A_c (mm^2)	11.43	11.567
Measured Relative roughness	7.4e-3	

Test matrix

Data recorded for 500s @ 1Hz after reaching steady state

Range of experimental parameters	
Inlet pressure (bar)	75, 81, 102
CO_2 inlet temperature (°C)	50 – 200°C (In increments of 10°C) 20 – 50°C (In increments of 5°C)
CO_2 flowrate, $\dot{m}_{CO_2}$ (kg/h)	8.8 – 28.8 kg/h (In increments of 2.9 kg/h)
Water inlet temperature (°C)	10 – 20°C
Water flow rate (GPM)	0.05 – 0.1

Uncertainty of measured variables	
$\dot{m}_{CO_2}$ (kg/s)	±1% of measured value
CO_2 T_{in}, T_{out}	±0.15°C
P_{in}	±0.025% of full scale (0-3000 psig)
ΔP	±0.025% of full scale (0-15 psid)
$\dot{V}_{dot}$ (GPM)	±1.5% of measured value
ΔT_{water}	±0.15°C
T_{wall}	±0.15°C

Data analysis procedure – Frictional Pressure drop

$$\Delta P_{measured} = \Delta P_{friction} + \Delta P_{local} + \Delta P_{accel} + \cancelto{0}{\Delta P_{gravity}}$$

$$\Delta P_{accel} = G^2\left(\frac{1}{\rho_{out}} - \frac{1}{\rho_{in}}\right)$$

$$\Delta P_{gravity} = \pm g\left(\frac{i_{out}\rho_{out} + i_{in}\rho_{in}}{i_{out} + i_{in}}\right) L sin\theta$$

$$\Delta P_{local} = \underbrace{\left[1 - \frac{A_c}{A_{manifold}}\right]^2 \rho_{out}\frac{v_{out}^2}{2}}_{\Delta P_{expansion}} + 0.5\underbrace{\left[1 - \frac{A_c}{A_{manifold}}\right]^{0.75} \rho_{in}\frac{v_{in}^2}{2}}_{\Delta P_{contraction}}$$

Data analysis procedure - Test section Heat duty

Test section heat duty

$$Q_{CO_2} = \dot{m}_{CO_2}(i_{in} - i_{out})$$

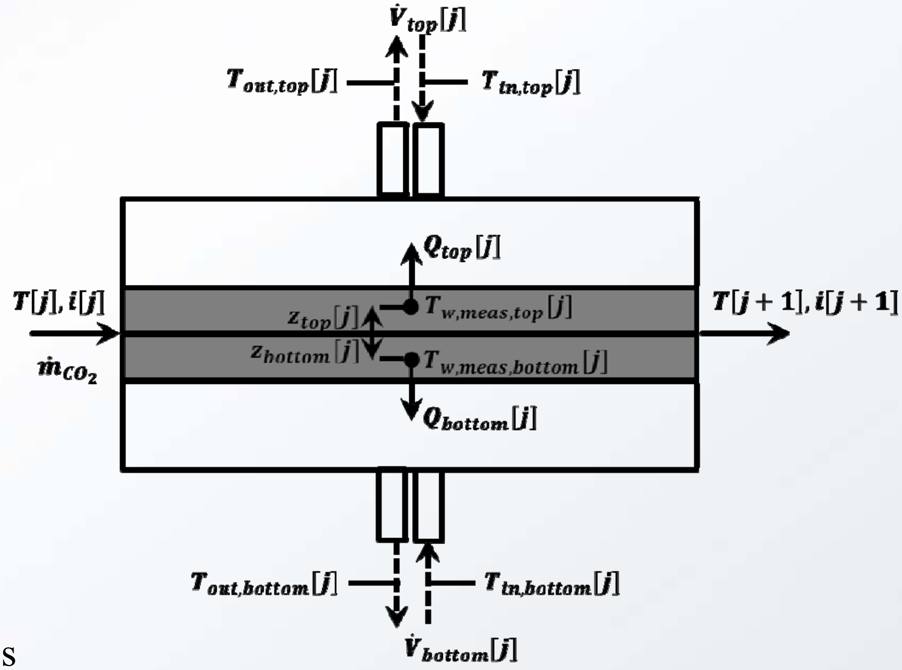
$$Q_{top}[j] = \dot{V}_{top}[j]\rho|_{T_{avg}}C_p|_{T_{avg}}\left(T_{out,top}[j] - T_{in,top}[j]\right)$$

$$Q_{bottom}[j] = \dot{V}_{bottom}[j]\rho|_{T_{avg}}C_p|_{T_{avg}}\left(T_{out,bottom}[j] - T_{in,bottom}[j]\right)$$

$$Q_{water} = \sum_{j=1}^{10} Q_{top}[j] + \sum_{j=1}^{10} Q_{bottom}[j]$$

$$\bar{Q} = 0.5(Q_{CO_2} + Q_{water})$$

- Maximum $\%Difference$ between water side and CO_2 side heat duties is <10% when the CO_2 inlet/outlet temperature is near the pseudo-critical temperature



Data analysis procedure – Local bulk and wall temperatures

Bulk fluid temperature

$$i[j+1] = i[j] - \frac{Q_{top}[j] + Q_{bottom}[j]}{\dot{m}_{CO_2}}$$

$$i_b[j] = 0.5(i[j] + i[j+1])$$

Assuming linear pressure drop, $P_b[j] = P_{in} - \frac{\Delta P}{L} x[j]$

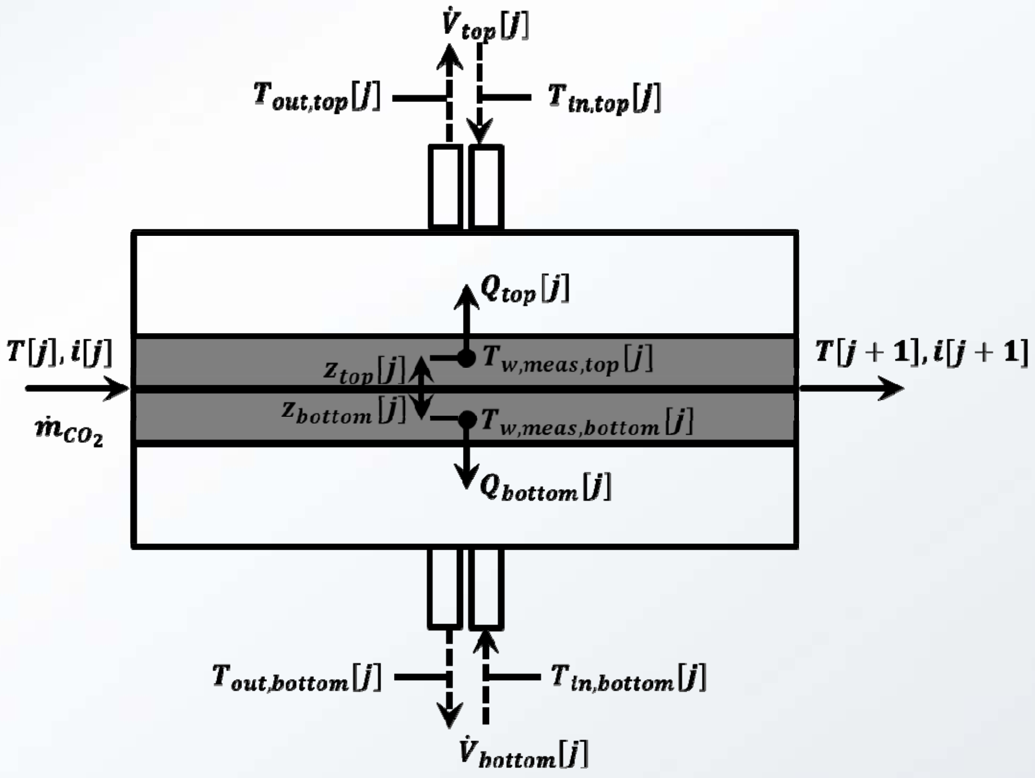
$$T_b[j] = f(i_b[j], P_b[j])$$

Wall temperature

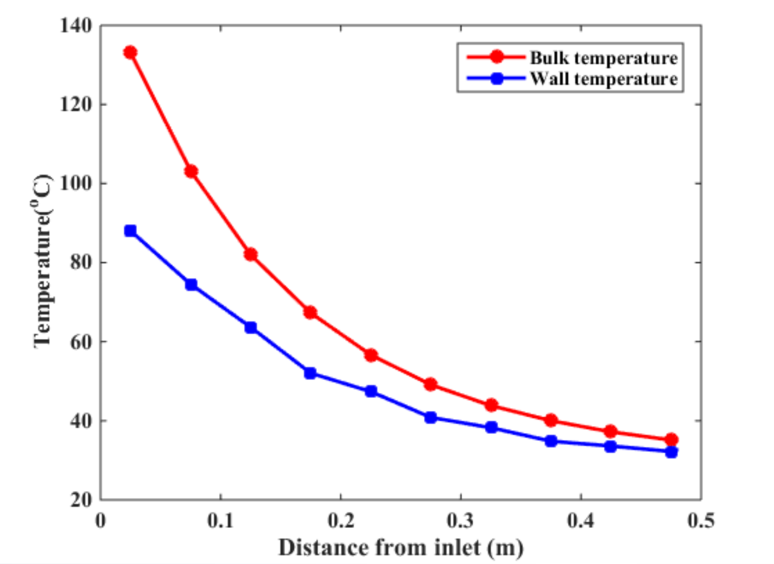
$$T_{w,calc,top}[j] = T_{w,meas,top}[j] + \frac{Q_{top}[j].z_{top}[j]}{k_{ss316}.A_{cb}}$$

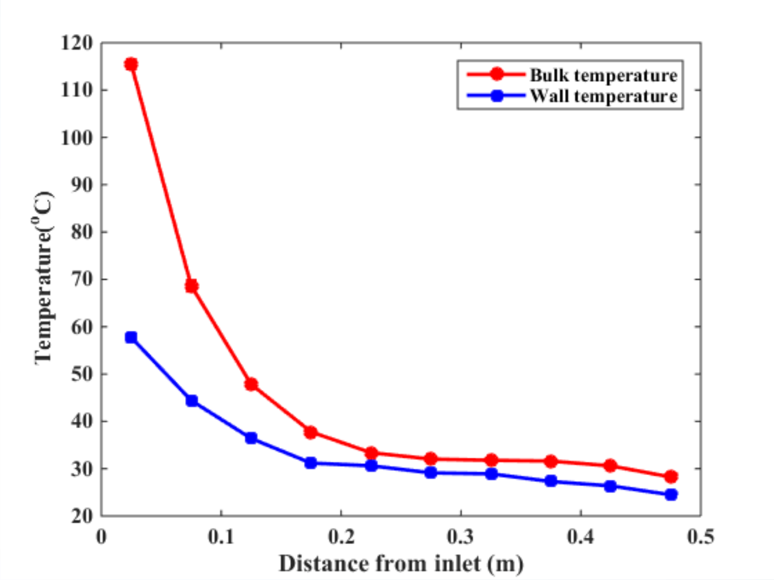
$$T_{w,calc,bottom}[j] = T_{w,meas,bottom}[j] + \frac{Q_{bottom}[j].z_{bottom}[j]}{k_{ss316}.A_{cb}}$$

$$T_w[j] = 0.5(T_{w,calc,top}[j] + T_{w,calc,bottom}[j])$$



Data analysis procedure – Average bulk and wall temperatures





$$\overline{T_b} = \frac{1}{L}\int_0^L T_b(x).\,dx = \frac{1}{N}\sum_{j=1}^{N} T_b[j]$$

$$\bar{\rho}_b = \frac{1}{L}\int_0^L \rho(T_b).\,dx$$

$$\bar{\mu}_b = \frac{1}{L}\int_0^L \mu(T_b).\,dx$$

$$\overline{Pr}_b = \frac{1}{L}\int_0^L Pr(T_b).\,dx$$

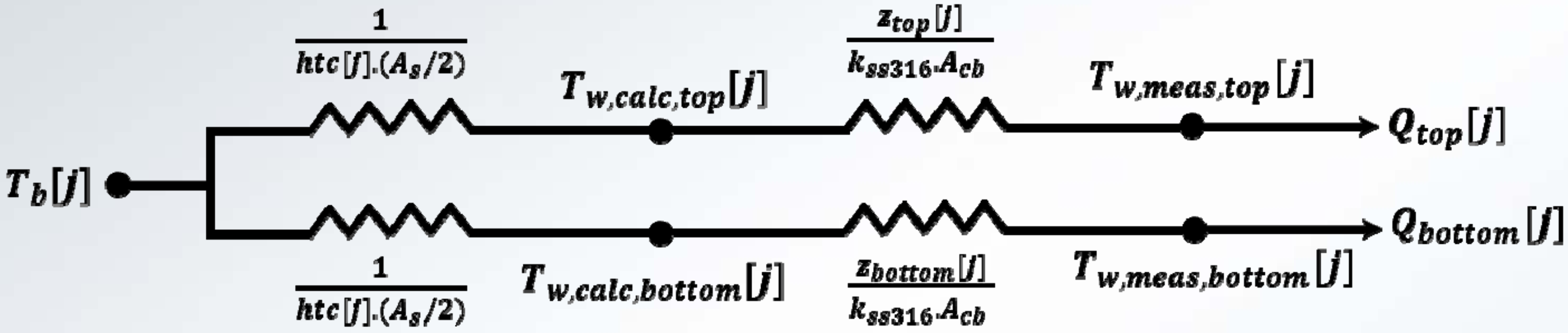
$$\overline{T_w} = \frac{1}{L}\int_0^L T_w(x).\,dx = \frac{1}{N}\sum_{j=1}^{N} T_b[j]$$

$$\bar{\rho}_w = \frac{1}{L}\int_0^L \rho(T_w).\,dx$$

$$\bar{\mu}_w = \frac{1}{L}\int_0^L \mu(T_w).\,dx$$

$$\overline{Pr}_w = \frac{1}{L}\int_0^L Pr(T_w).\,dx$$

Data analysis procedure – Local and average heat transfer coefficients



$$htc[j] = \frac{Q_{top}[j]+Q_{bottom}[j]}{A_s.(T_b[j]-T_w[j])}$$

$$\overline{htc} = \frac{\bar{Q}}{N.A_s(\bar{T}_b - \bar{T}_w)}$$

$$Nu[j] = \frac{htc[j].D_h}{k_b}$$

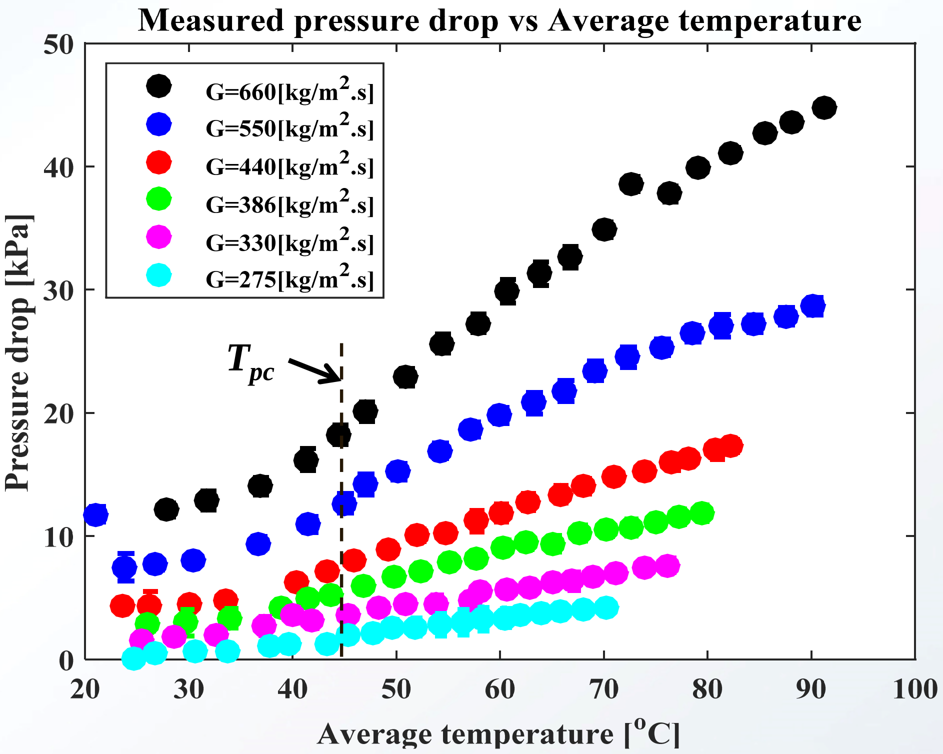
$$\overline{Nu} = \overline{htc}.\frac{D_h}{\bar{k}_b}$$

Pressure drop data – Example

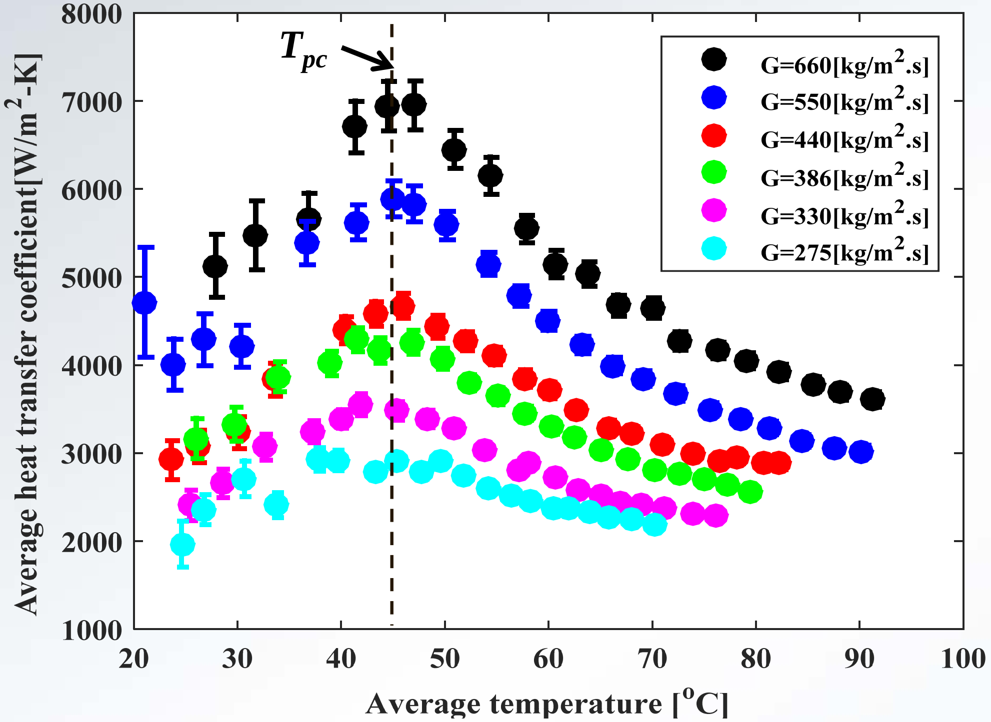
- Test condition
  - Operating pressure 10.2 MPa
  - 4mm NACA0020 airfoil Offset fin test plate

- Pressure drop decreases rapidly in the vicinity of pseudo-critical point → Due to increase in viscosity and density leading to lower velocities and Re for constant mass flux

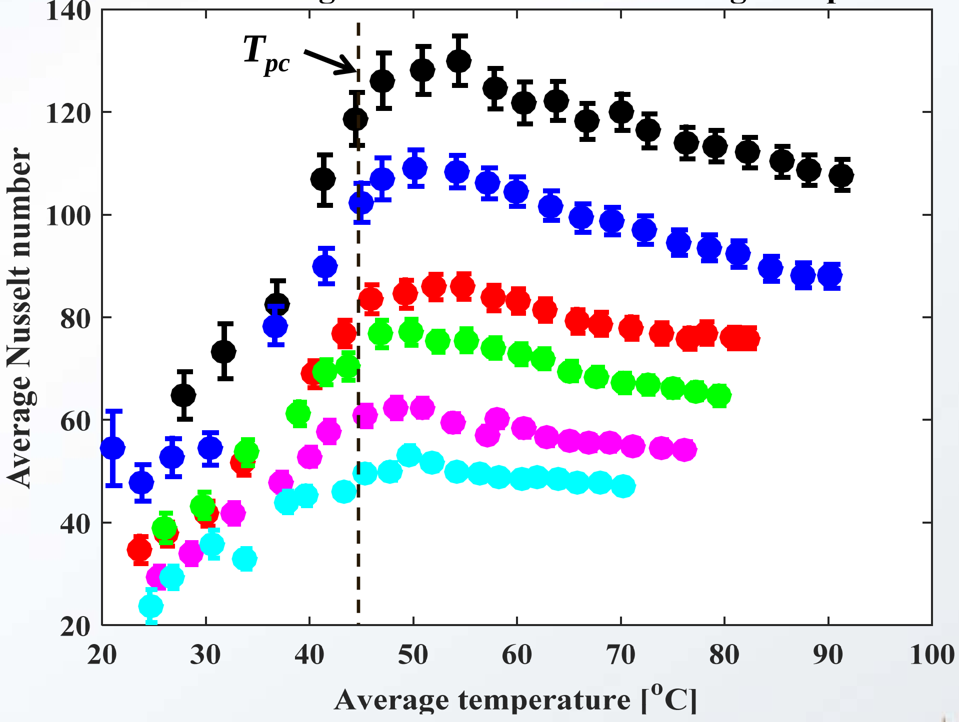
- Pressure drop increases with increase in mass flux (and Re)



Heat transfer data – Example



Measured average heat transfer coefficient vs Average temperature



Measured average Nusselt number vs Average temperature

Friction factor – Correlation development

- Calculated frictional pressure drop for each case can be written as,

$$\Delta P_{calc} = \sum_{i=1}^{N} 2 \left(\frac{L}{N.D_h}\right) \frac{G^2}{\rho_i} f_i$$

Where, ρ_i and f_i represent local density and friction factor. Assuming a friction factor of the form,

$$f_i = aRe_i^b$$

Coefficients a and b are found out using least squares curve fitting approach

$$\sum_{i=1}^{N_{expt}} \left(\Delta P_{calc}^i - \Delta P_{exp}^i\right) \rightarrow minimum$$

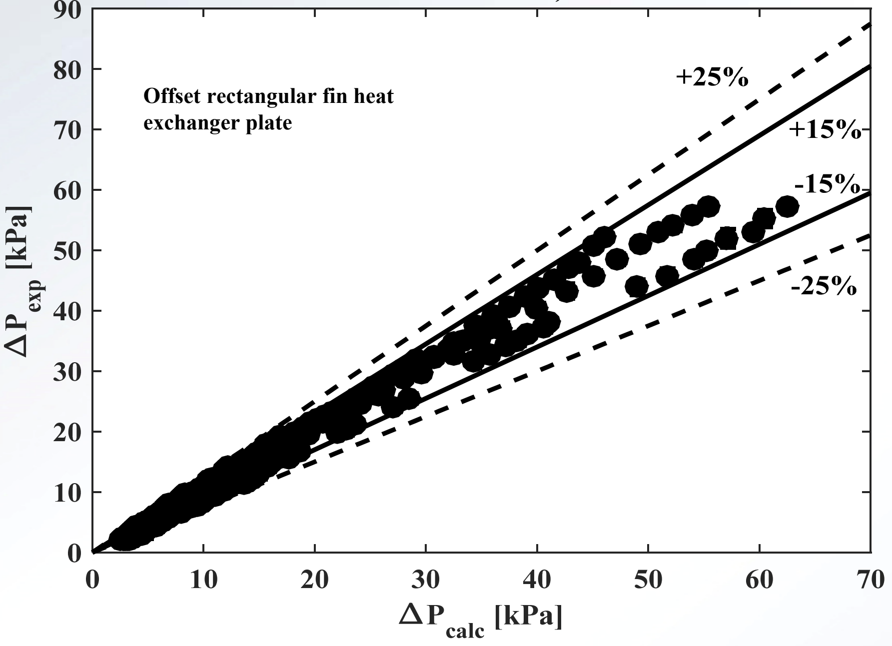
$$Error = \frac{\Delta P_{calc} - \Delta P_{exp}}{\Delta P_{exp}} .100$$

Heat exchanger plate	a	b
Offset rectangular fin plate	0.0276	-0.002
Offset NACA0020 airfoil fin plate	0.0077	0.1201

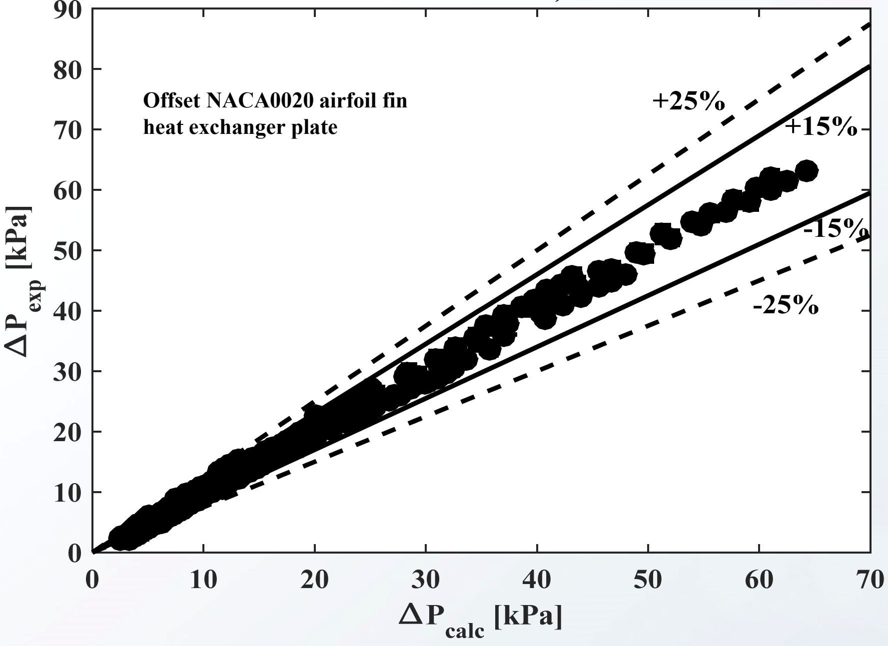
Friction factor – Correlation development

Heat exchanger plate	MAD Error	σ Error	Points with |Error|<15%	Points with |Error|<25%
Offset rectangular fin plate	11.2%	13.7%	80%	91.7%
Offset NACA0020 airfoil fin plate	8.1%	11%	88.6%	94.7%

Calculated vs measured frictional pressure drop
2700 < Re < 32,500



Calculated vs experimental frictional pressure drop
3800 < Re < 38,000



Nusselt number – Correlation development

- Calculated average Nusselt number for each case can be written as the Dittus-Boelter form ($a\overline{Re}^{b}\overline{Pr}^{c}$) along with additional wall to bulk property ratios to take into account non-linear variation of properties with temperature.

$$Nu_{calc} = a\overline{Re}^{b}\overline{Pr}^{c}\left(\frac{\rho_b}{\rho_w}\right)^{d}\left(\frac{\overline{C_{pb}}}{\overline{C_p}}\right)^{e}$$

Where, $\overline{C_p} = \frac{\overline{i_w} - \overline{i_b}}{\overline{T_w} - \overline{T_b}}$

Coefficients a through e are found out using least squares curve fitting approach,

$$\sum_{i=1}^{N_{expt}}\left(Nu_{calc}^{i} - Nu_{exp}^{i}\right) \rightarrow minimum$$

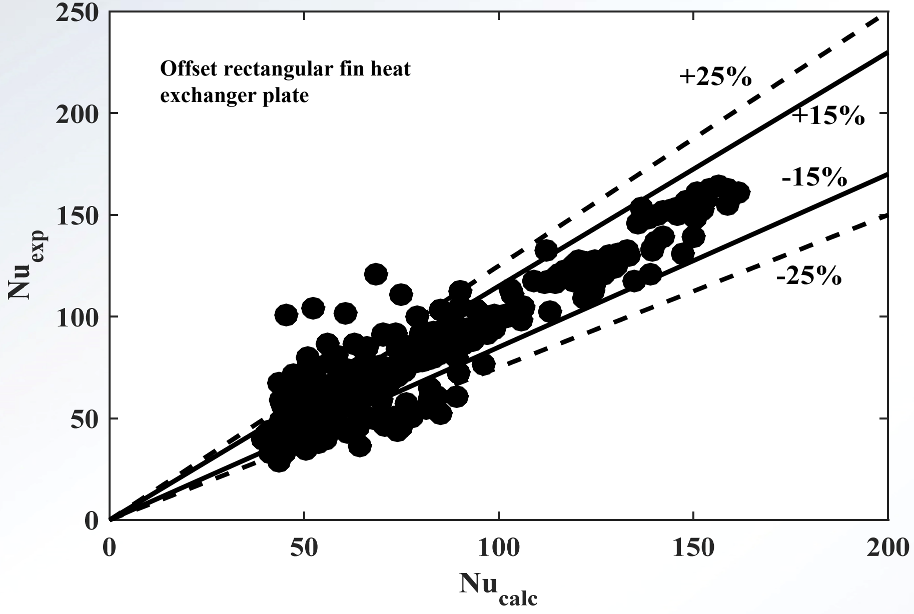
$$Error = \frac{Nu_{calc} - Nu_{exp}}{Nu_{exp}} .100$$

Heat exchanger plate	**a**	**b**	**c**	**d**	**e**
Offset rectangular fin plate	0.1034	0.7054	0.3489	0.9302	-0.366
Offset NACA0020 airfoil fin plate	0.0601	0.7326	0.3453	0.4239	-0.3556

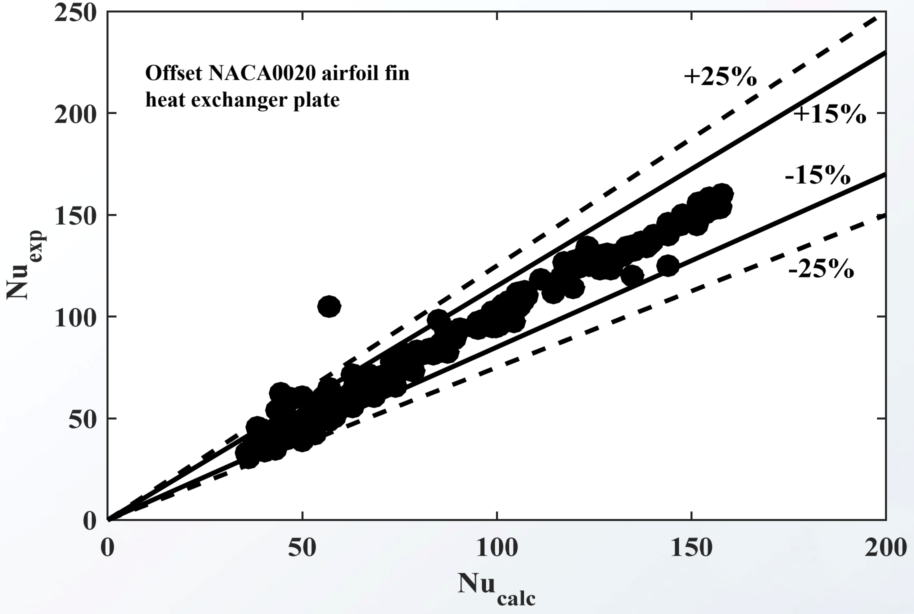
Nusselt number – Correlation development

Heat exchanger plate	MAD Error	σ Error	Points with |Error|<15%	Points with |Error|<25%
Offset rectangular fin plate	9.1%	15.4%	90%	93%
Offset NACA0020 airfoil fin plate	5.2%	8%	96%	99%

Calculated vs experimental average Nusselt numbers
2800 < Re < 32,000
1.5 < Pr < 24



Calculated vs measured average Nusselt numbers
4000 < Re < 37,000
1.35 < Pr < 25



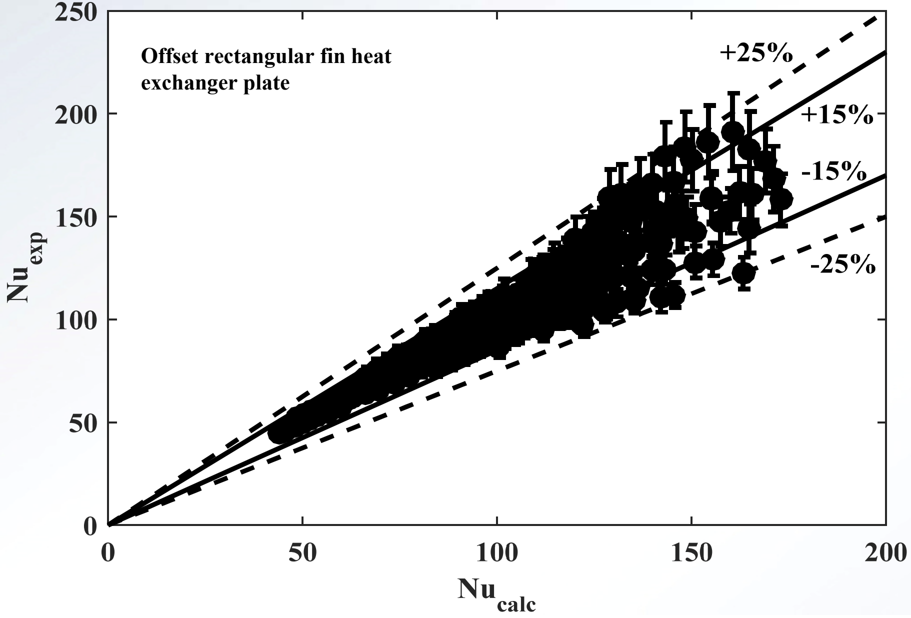
Nusselt number correlation – Gas like regimes

Heat exchanger plate	a	b	c	d	e
Offset rectangular fin plate	0.1034	0.7054	0.3489	0.9302	-0.366
Offset NACA0020 airfoil fin plate	0.0601	0.7326	0.3453	0.4239	-0.3556

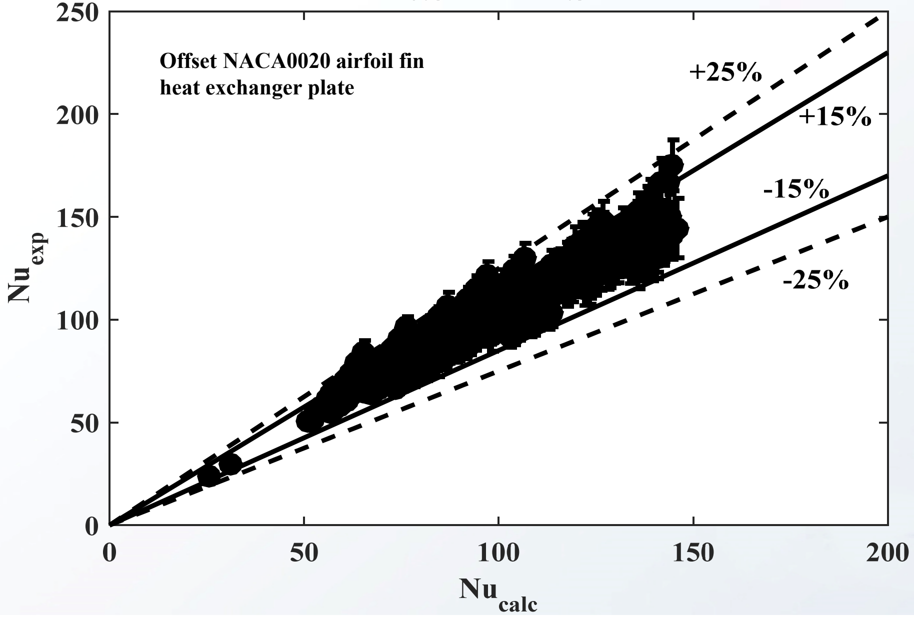
$$T_{GL} = 0.0034P^3 - 0.3284P^2 + 15.963P - 43.85$$

$$Nu_{calc} = a\overline{Re}^b\,\overline{Pr}^c\left(\frac{\overline{\rho_b}}{\rho_w}\right)^d\left(\frac{\overline{C_{pb}}}{C_p}\right)^e$$

Calculated vs measured local Nusselt numbers for Gas like regime
9000 < Re < 32,600
0.8 < Pr < 1.3



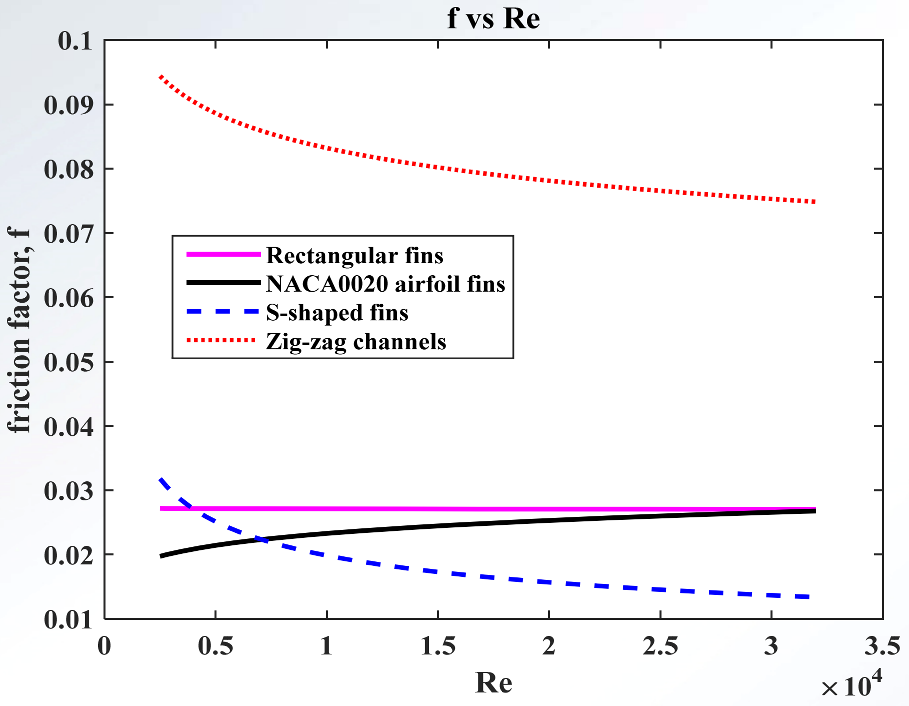
Calculated vs measured local Nusselt numbers for Gas like regime
4000 < Re < 38,000
0.8 < Pr < 1.3

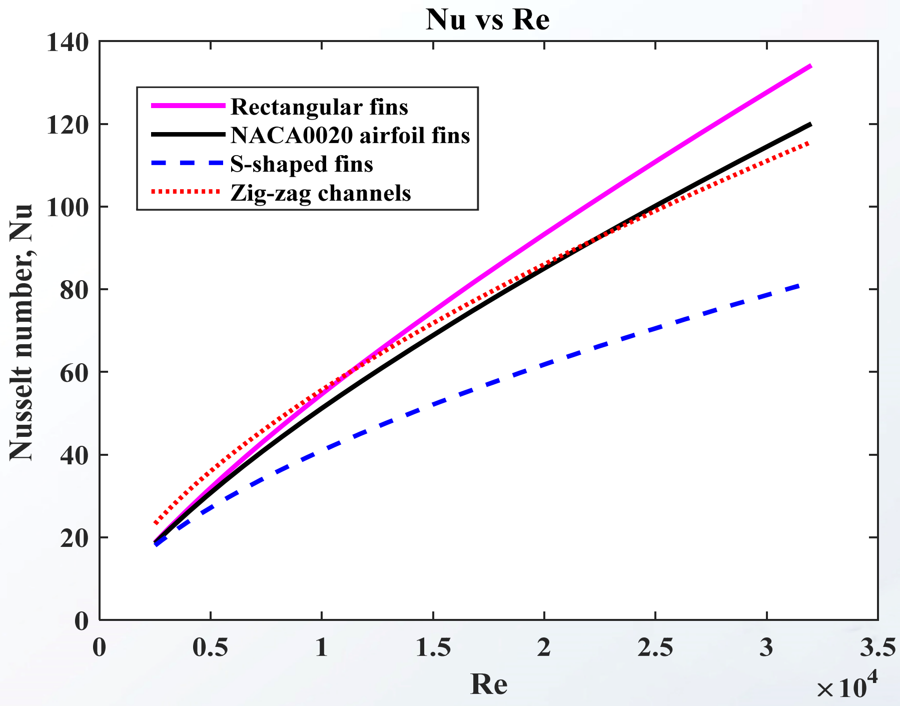


Comparison with existing correlations

S-shaped fins from Ngo et al. (2007) → $f_{SS} = 0.4545Re^{0.43}; Nu_{SS} = 0.174Re^{0.593}Pr^{0.43}$

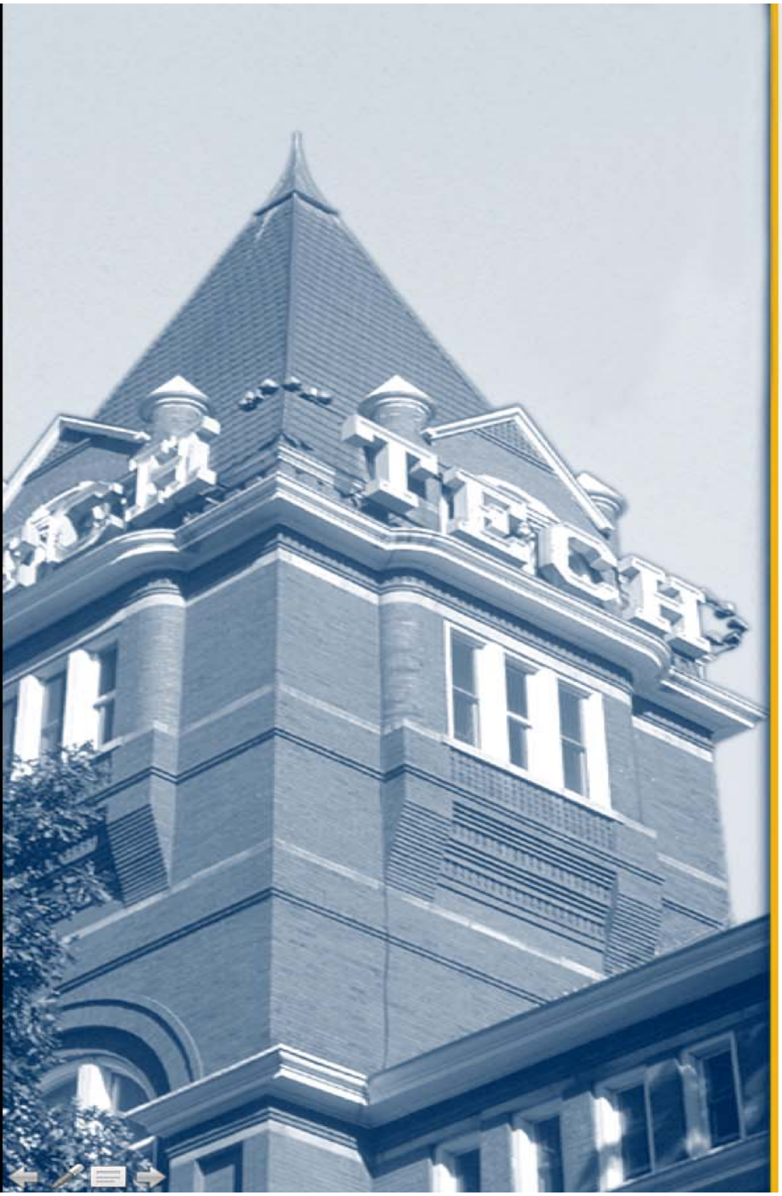
Zig-zag channels from Ngo et al. (2007) → $f_{ZZ} = 0.1924Re^{-0.091}; Nu_{ZZ} = 0.629Re^{0.629}Pr^{0.317}$





Conclusions

- Heat transfer and pressure drop characteristics of sCO_2 flow through discontinuous offset NACA0020 airfoil and rectangular fins was investigated experimentally
- Correlations to predict average and local Nusselt numbers as well as frictional pressure drop are proposed based on least squares fitting to the experimental data
- Both the heat exchanger plates offered significantly lower pressure drop compared to zig-zag channel whereas the Nusselt numbers are almost similar based on the correlations of Ngo et al. (2007)
- Mechanical Integrity of such discontinuous fins geometries needs to be verified and design procedures needs to be established.
  - Offset Rectangular fin geometries could still potentially use ASME Sec VIII procedures



Thank you for your time!

Questions?

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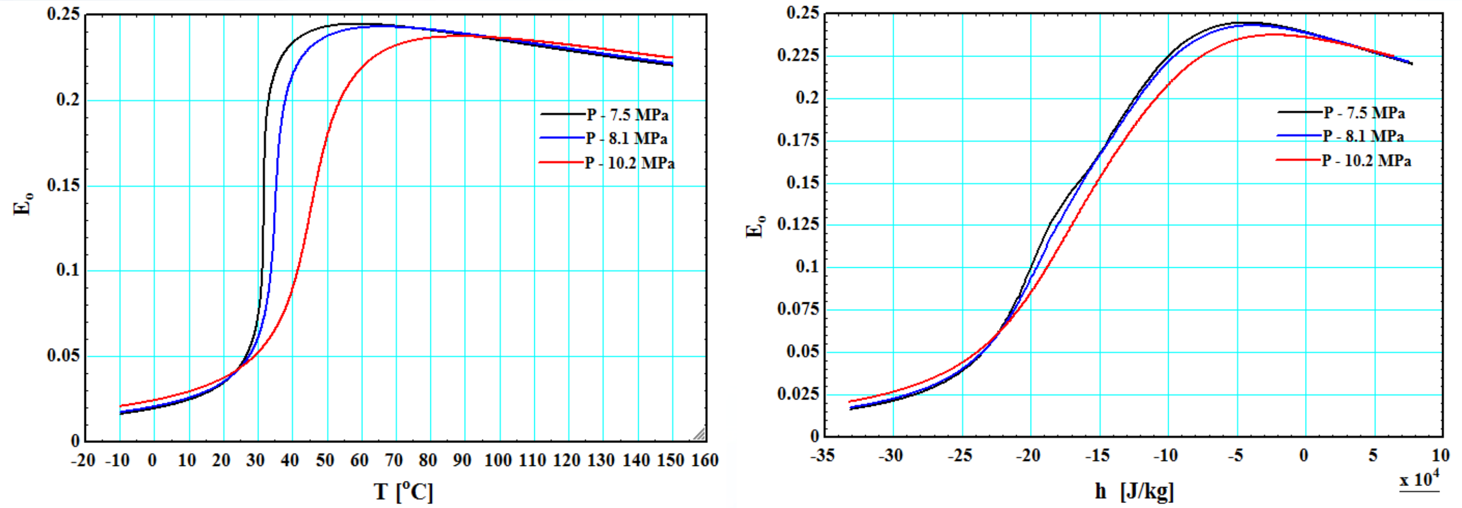
Nusselt number correlation – Gas like regimes

- Data was divided into three regimes – Liquid like, pseudo-critical transition, and gas like regime based on specific work of thermal expansion/contraction

$$E_o = P.\beta/(\rho C_p)$$

$$T_{GL} = 0.0034P^3 - 0.3284P^2 + 15.963P - 43.85$$



Pressure [MPa]	Liquid-like regime	Pseudo-critical transition regime	Gas-like regime
7.5	T < 26.46^{o}C	26.46^{o}C ≤ T ≤ 58.79^{o}C	T > 58.79^{o}C
8.1	T < 27.25^{o}C	27.25^{o}C ≤ T ≤ 65.76^{o}C	T > 65.76^{o}C
10.2	T < 28.82^{o}C	28.82^{o}C ≤ T ≤ 88.44^{o}C	T > 88.44^{o}C