Managing Thermal Gradients on a Supercritical Carbon Dioxide Radial Inflow Turbine

David W. Stevens

dstevens@peregrineturbine.com

PEREGRINE
TURBINE TECHNOLOGIES

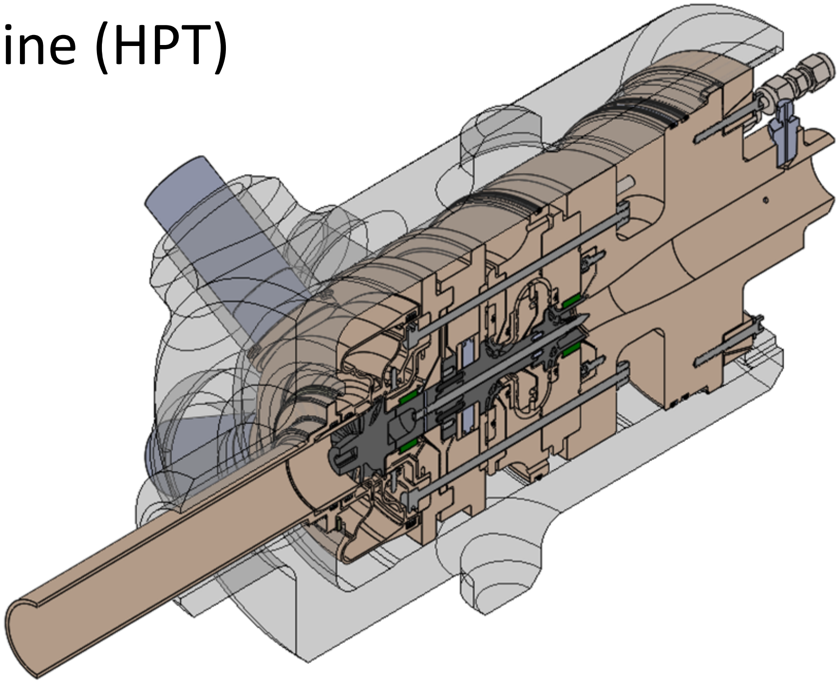
6th International Supercritical Power Cycles Symposium
March 27-29, 2018, Pittsburgh, Pennsylvania

Turbine Cooling

- Historically, a consistent engineering challenge
- Machine durability
- Traditional gas turbines:
  - Turbine inlet temperatures as high as 1600°C (2912°F)
  - Beyond melting point of metal turbine materials
  - Secondary flows are routinely used to cool components
    - Sophisticated air cooling passages employed on blades
- SCO2 Turbomachines:
  - Turbine inlet temperatures are currently limited to 750°C (1382°F)
    - Primary heat exchanger
  - Challenges originate from the unique thermo-physical heat transfer properties of SCO2

Peregrine 1MW Turbo Pump

- Two stage centrifugal compressor
- Single stage radial high pressure turbine (HPT)
- W0 = 12.1 lbm/s
- T2 = 194°F
- P2 = 6220 psi
- T3 = 1382°F
- P3 = 6000 psi



Peregrine 1MW Turbo Pump

- Steady state and transient thermal analysis performed
- Results revealed engineering challenge contrary to traditional air cooled machines
- Strong thermal gradients through back face of HPT caused by overcooling
- Overcooling caused by unique thermo-physical properties of SCO2
- Resulting structural analysis revealed stresses beyond fatigue design criteria
- Unique secondary flow cooling method was developed to correct the problem

Air vs. SCO2 Turbomachines

- Typical temperature and pressures of air and SCO2 machines are as different as the physical properties themselves
  - Traditional gas turbine:
    - T1, P1 ≈ Ambient
    - P2 ≈ 400 psi
    - T2 ≈ 1000°F
    - T3 ≈ 2900°F (Max)
  - Peregrine 1 MW turbo pump:
    - P1 = 1100 psi
    - T1 = 90°F
    - P2 = 6220 psi
    - T2 = 194°F
    - T3 = 1382°F

Thermo-Physical Properties and Convective Heat Transfer

- Root of the problem is high heat transfer coefficients produced by the unique thermo-physical properties of SCO2
  - Examine how differences in thermo-physical properties effect fundamental convective heat transfer equations and dimensionless quantities
  - In context of typical operating temperatures and pressures of each machine type.
- Properties:
  - dynamic viscosity μ
  - thermal conductivity k
  - density ρ
  - specific heat c_p
- Dimensionless Quantities:
  - Reynolds Number Re
  - Prandtl Number Pr
  - Nusselt Number Nu

Relevant Equations

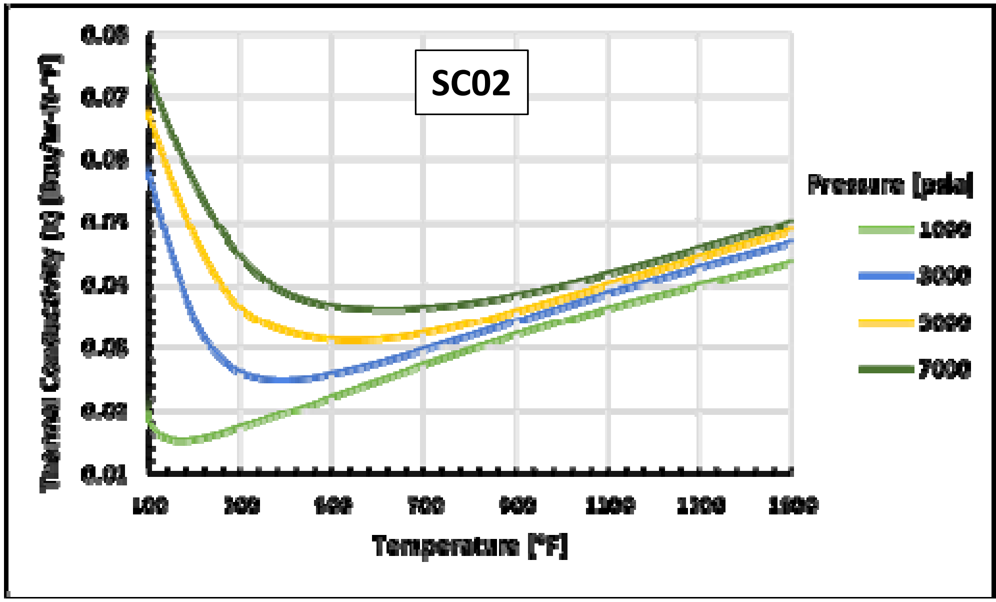
$$\boldsymbol{Reynolds\ Number}: \quad \boldsymbol{Re} = \frac{\boldsymbol{Inertial\ forces}}{\boldsymbol{Viscous\ forces}} = \frac{\boldsymbol{\rho v L_c}}{\boldsymbol{\mu}}$$

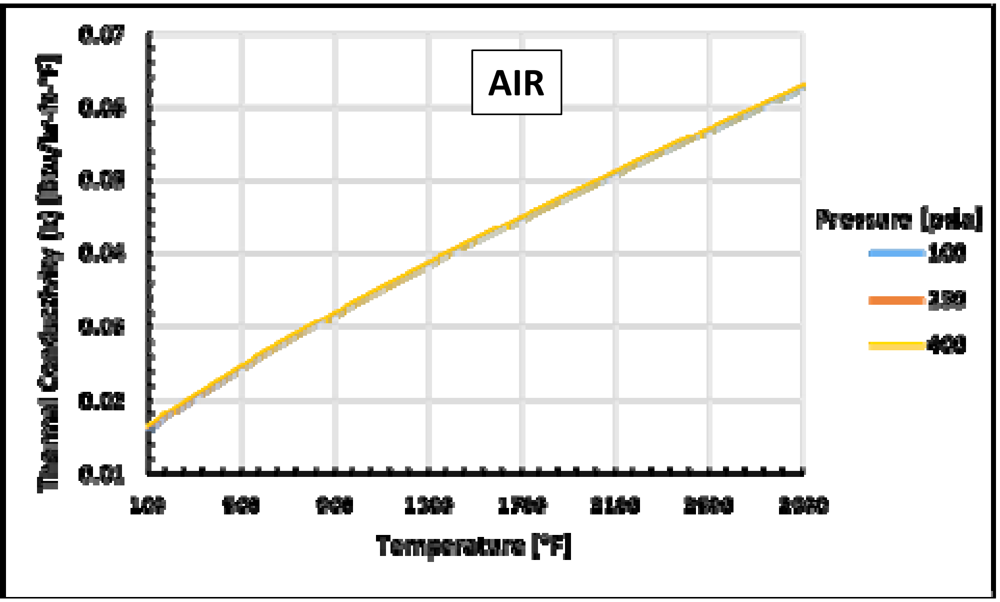
$$\boldsymbol{Prandtl\ Number}: \quad \boldsymbol{Pr} = \frac{\boldsymbol{Molecular\ diffusivity\ of\ momentum}}{\boldsymbol{Molecular\ diffusivity\ of\ heat}} = \frac{\boldsymbol{\mu c_p}}{\boldsymbol{k}}$$

$$\boldsymbol{{*}Nusselt\ Number}: \quad \boldsymbol{Nu} = \frac{\boldsymbol{hL_c}}{\boldsymbol{k}} = 0.023\boldsymbol{Re}^{0.8}\boldsymbol{Pr}^{1/3}\ \boldsymbol{for} \begin{Bmatrix} 0.5 \le \boldsymbol{Pr} \le 160 \\ \boldsymbol{Re} \ge 10^4 \end{Bmatrix}$$

**Chilton Colburn equation used for comparison purposes only.
Actual correlations used in thermal model beyond scope of paper*

Thermal Conductivity



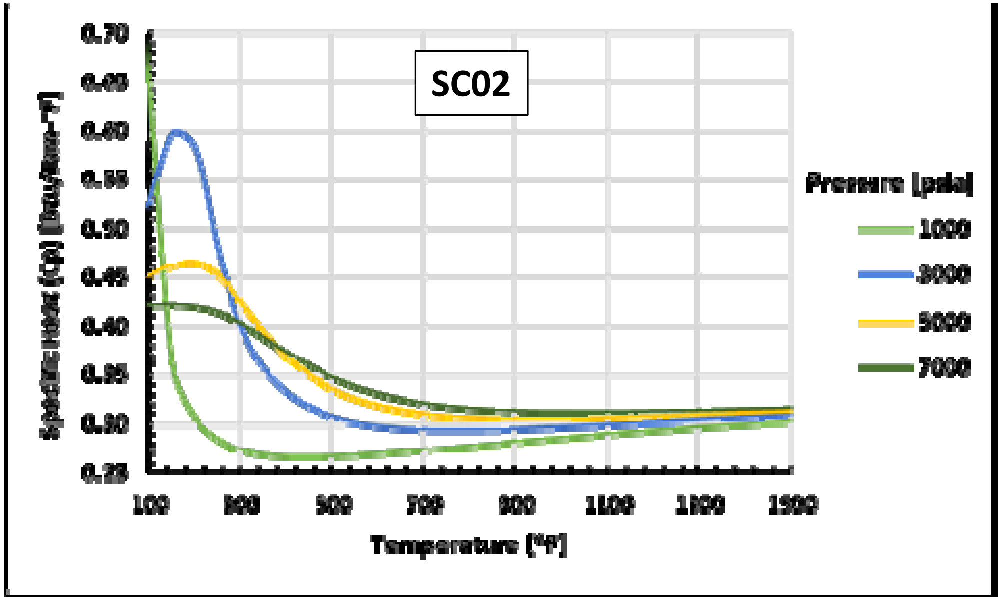


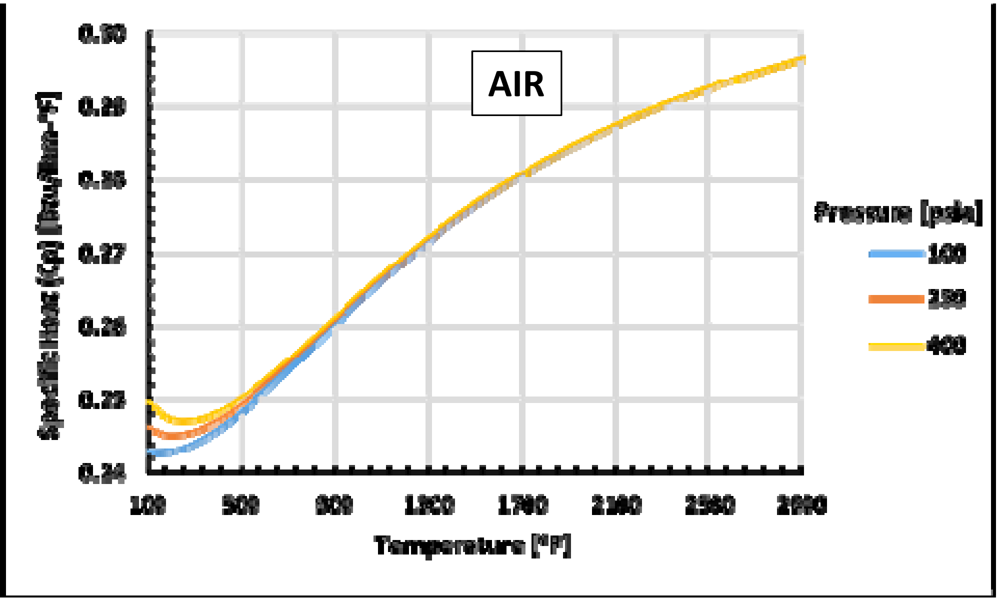
Compressor Discharge Conditions

	Pressure [psia]	Temperature [°F]	Thermal Conductivity (k) [Btu/hr-ft-°F]
PTT 1MW Turbo Pump	6220	194	0.054
Air Gas Turbine	400	1000	0.034

Thermal conductivity of SCO2 1.5x greater than air

Specific Heat





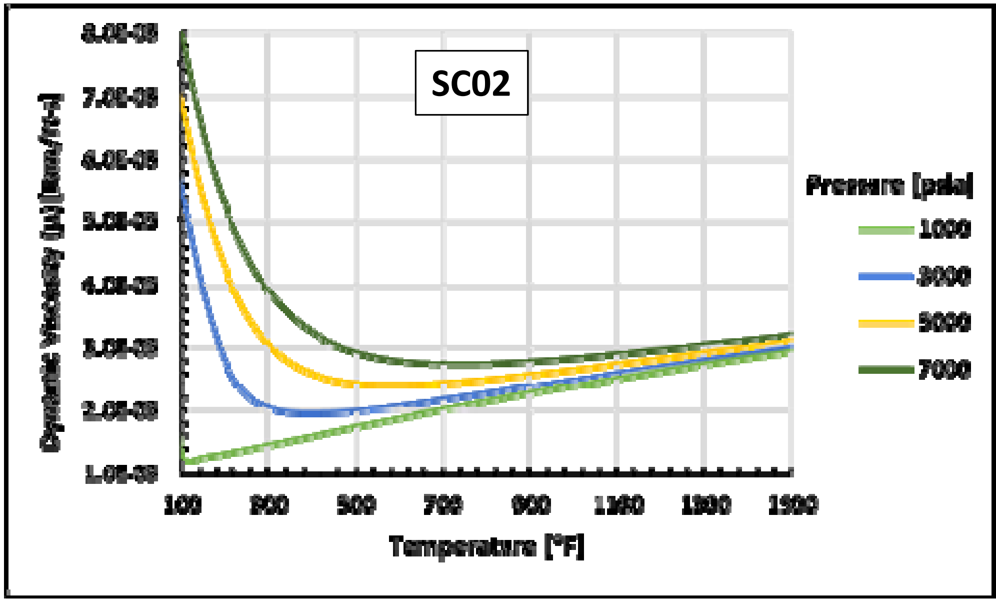
Compressor Discharge Conditions

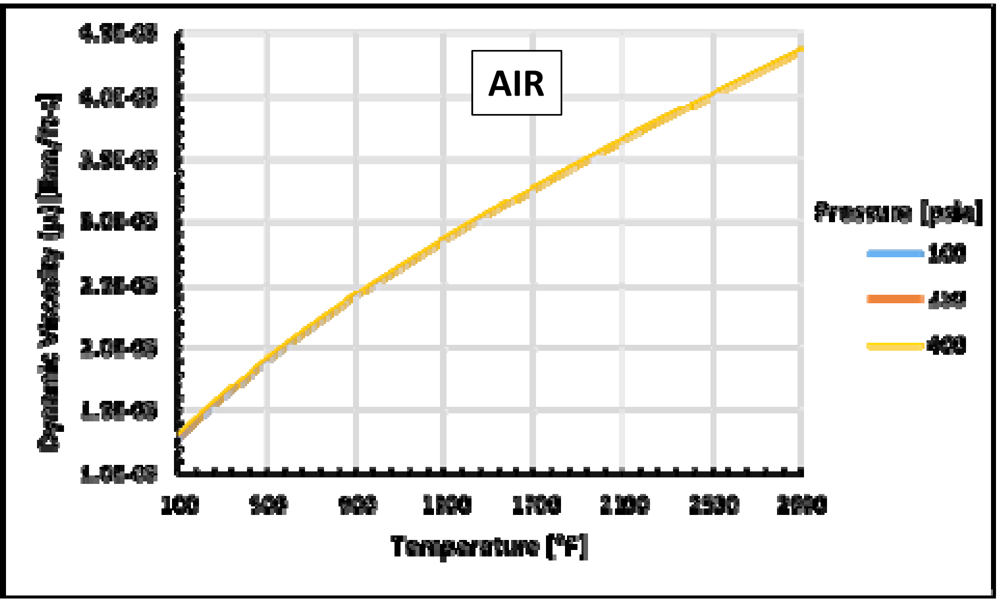
	Pressure [psia]	Temperature [°F]	Specific Heat (Cp) [Btu/lbm-°F]
PTT 1MW Turbo Pump	6220	194	0.433
Air Gas Turbine	400	1000	0.264

Specific heat of SCO2 2x greater than air

Higher heat transfer capacity of fluid

Dynamic Viscosity





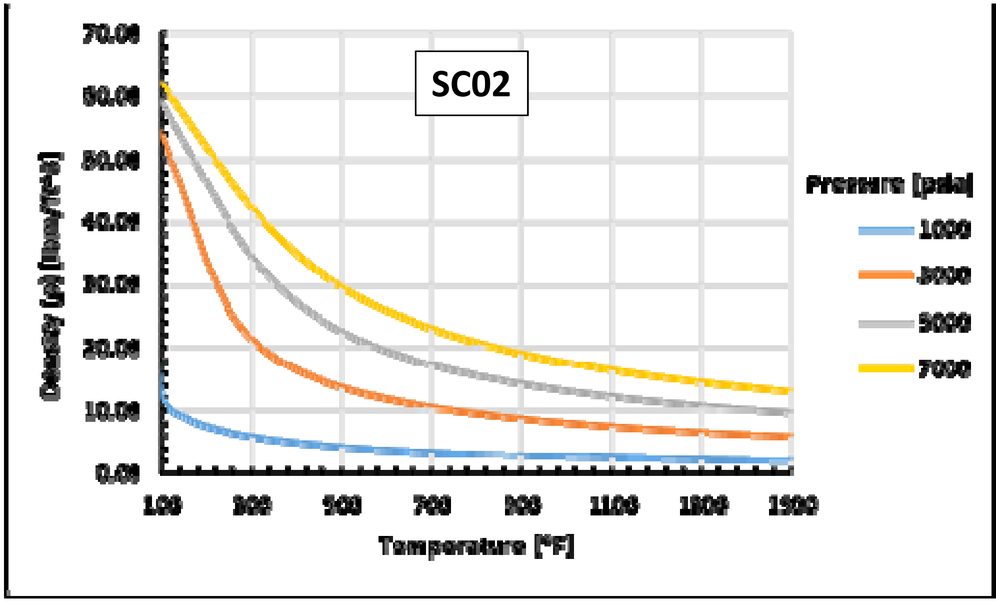
Compressor Discharge Conditions

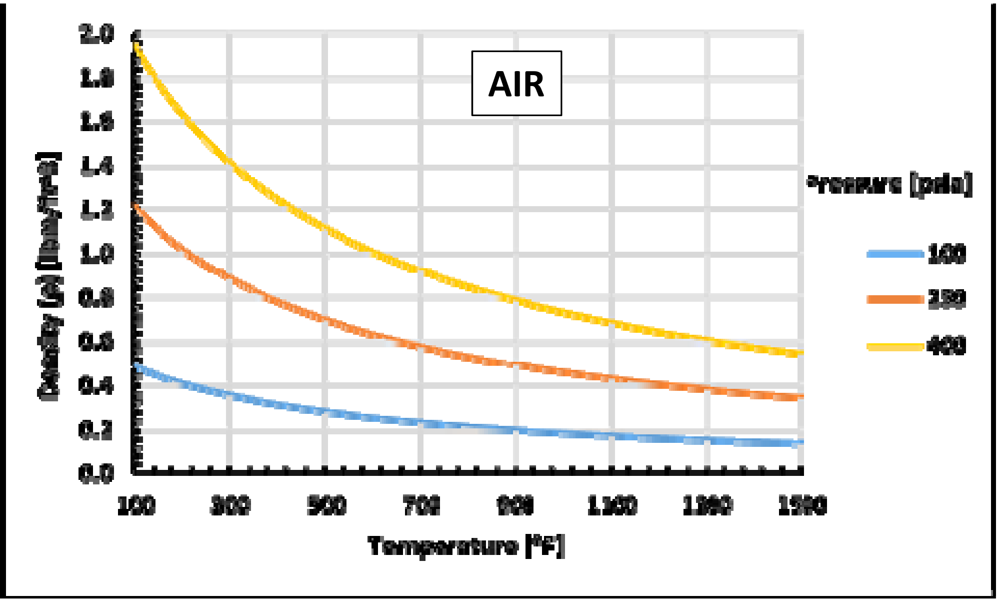
	Pressure [psia]	Temperature [°F]	Dynamic Viscosity (μ) [lbm/ft-s]
PTT 1MW Turbo Pump	6220	194	4.97E-05
Air Gas Turbine	400	1000	2.55E-05

Dynamic viscosity of SCO2 2x greater than air

Decrease in Reynolds Number
Increase in Prandtl Number

Density





Compressor Discharge Conditions

	Pressure [psia]	Temperature [°F]	Density (ρ) [lbm/ft^3]
PTT 1MW Turbo Pump	6220	194	50.460
Air Gas Turbine	400	1000	0.733

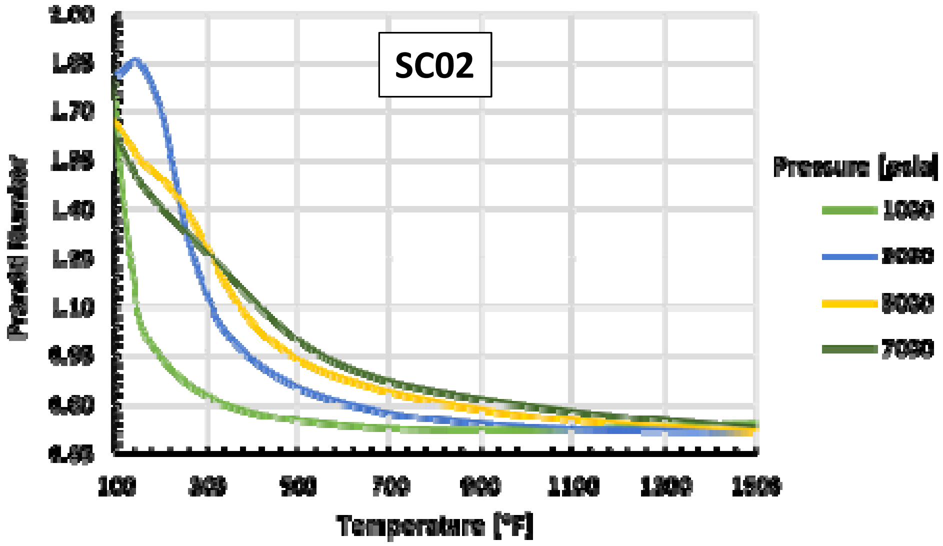
Density of SCO2 70x greater than air

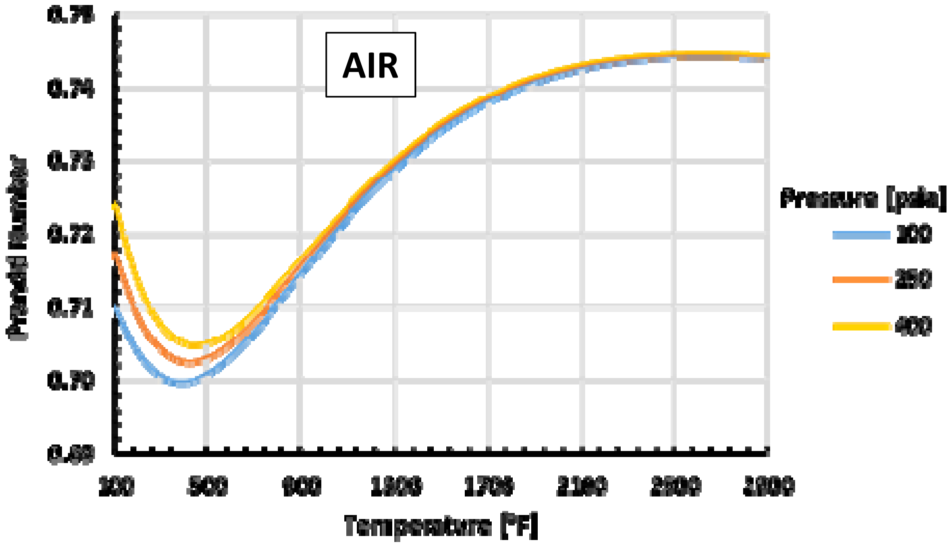
Higher Reynolds Numbers

PEREGRINE
TURBINE TECHNOLOGIES

Prandtl Number

6th International Supercritical
Power Cycles Symposium
March 27-29, 2018
Pittsburgh, Pennsylvania





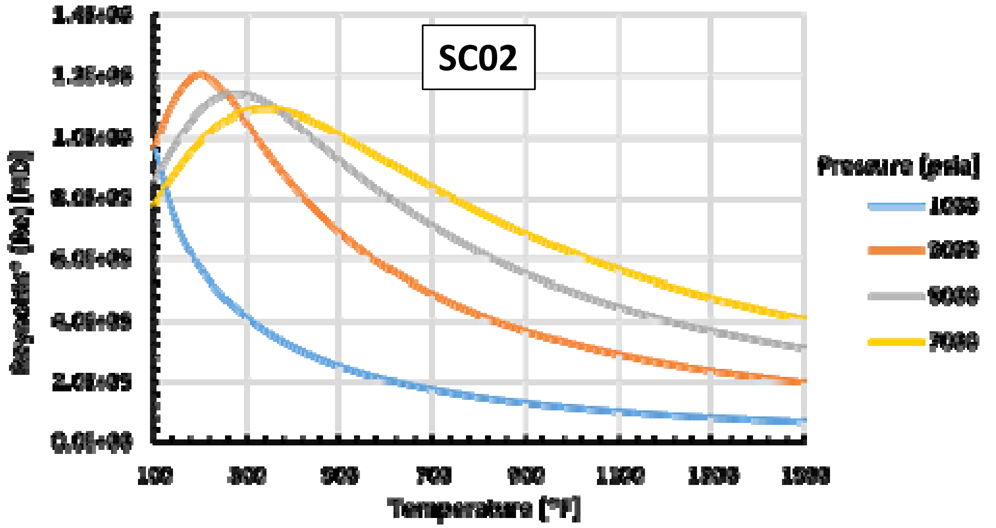
Compressor Discharge Conditions

	Pressure [psia]	Temperature [°F]	Prantl (Pr) [ND]
PTT 1MW Turbo Pump	6220	194	1.44
Air Gas Turbine	400	1000	0.72

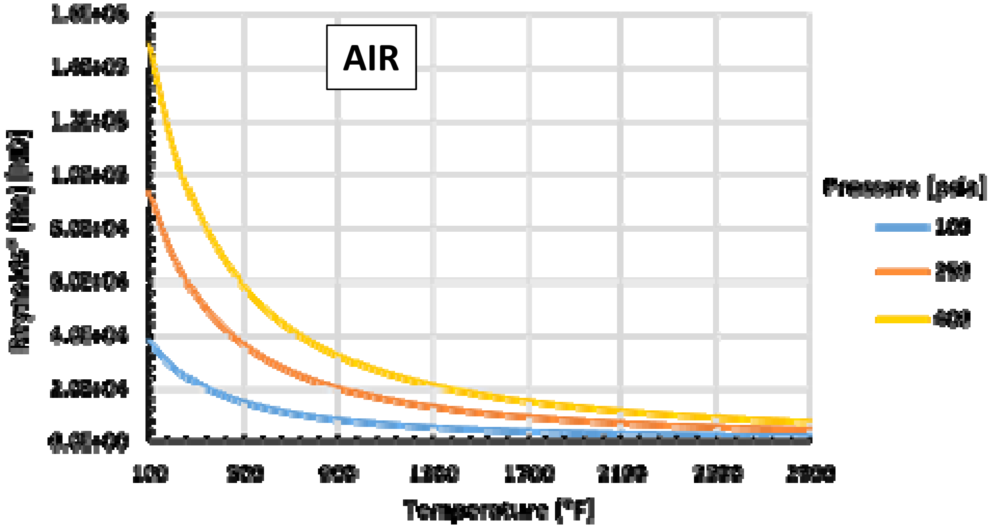
SCO2 will have relatively thinner thermal boundary layer
Pr > 1: Convection Dominated
Pr < 1: Conduction Dominated

peregrineturbine.com

Reynolds Number



*Assuming ε and L_c are constant and equal to 1 respectively



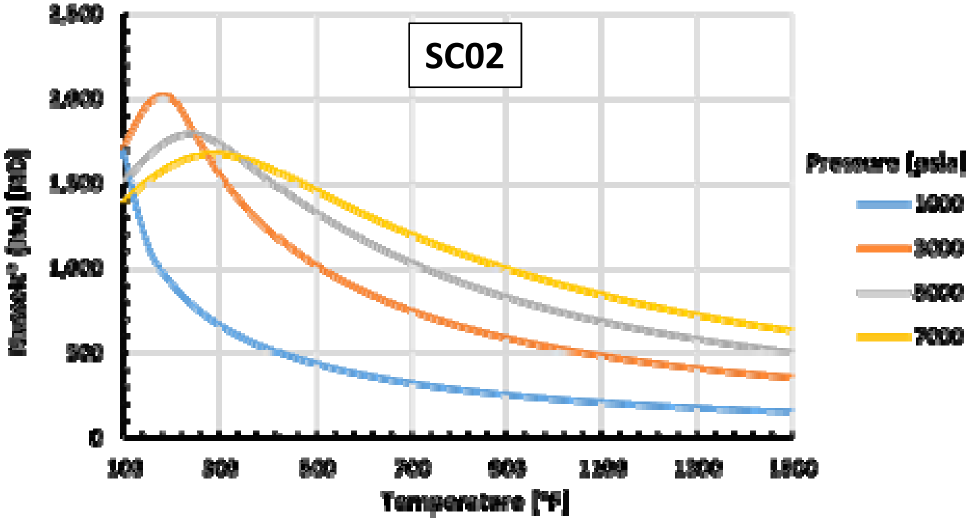
*Assuming ε and L_c are constant and equal to 1 respectively

Compressor Discharge Conditions

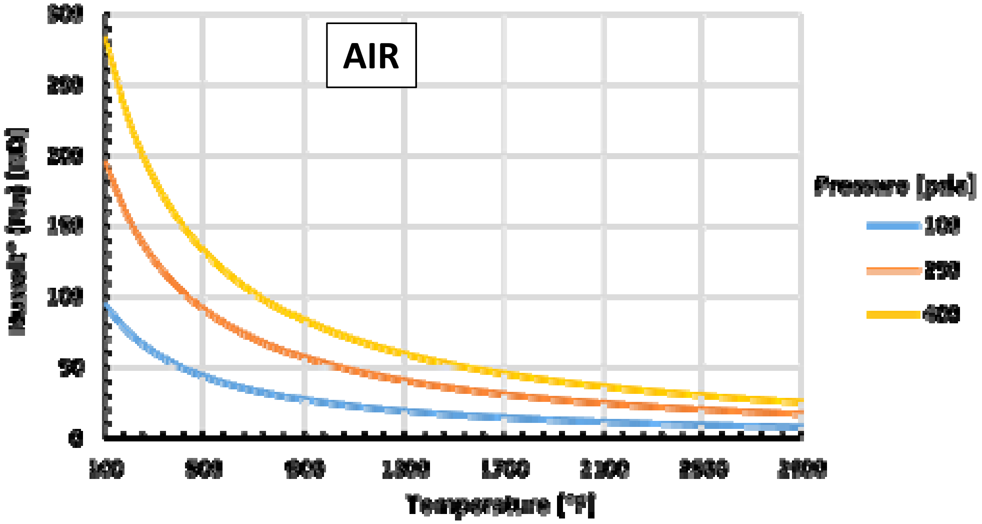
	Pressure [psia]	Temperature [°F]	Reynolds (Re) [ND]
PTT 1MW Turbo Pump	6220	194	1.02E+06
Air Gas Turbine	400	1000	2.88E+04

→ **Reynolds Number of SCO2 35x greater than air**

Nusselt Number



*Assuming μ and f_{sc} are constant and equal to 1 respectively



*Assuming μ and f_{sc} are constant and equal to 1 respectively

Compressor Discharge Conditions

	Pressure [psia]	Temperature [°F]	Nusselt (Nu) [ND]
PTT 1MW Turbo Pump	6220	194	1659
Air Gas Turbine	400	1000	76

Nusselt Number of SCO2 22x greater than air

Result?

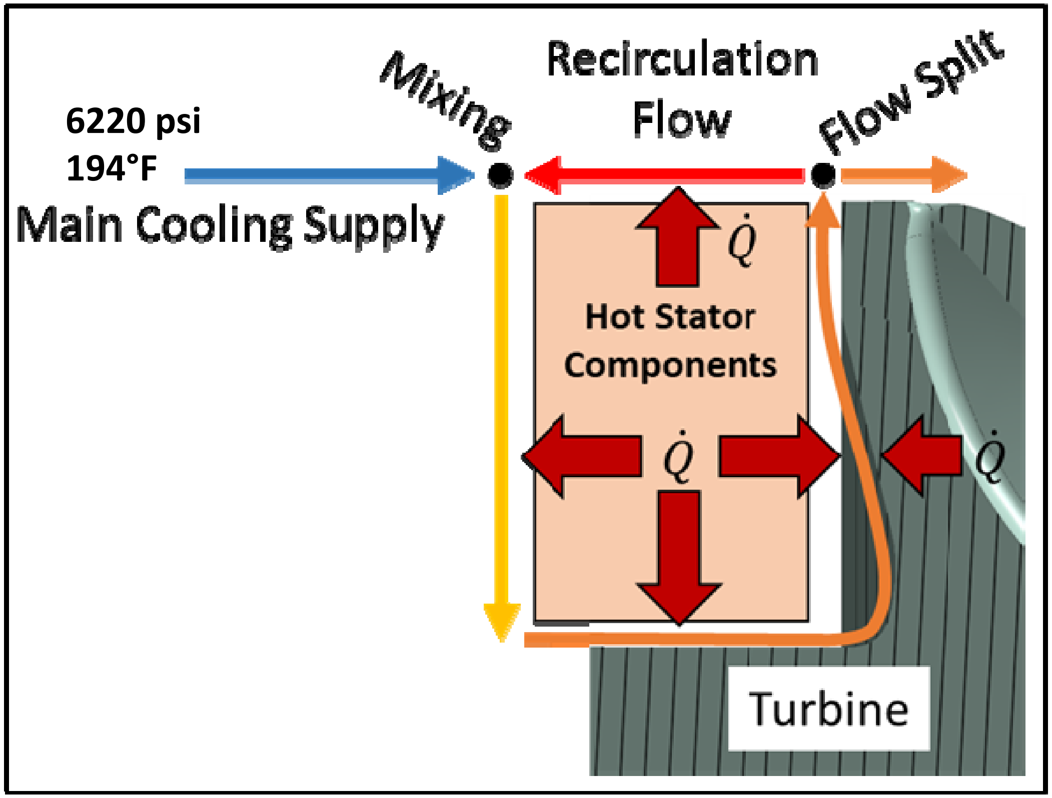
- Significantly higher Nusselt numbers in SCO2 indicate highly convection dominated heat transfer coefficients.
- Surface temperatures on HPT will be driven close to mean fluid temperatures
- Thermal tug of war between heating of HPT primary flow surfaces and turbine back face
- Steep thermal gradient on turbine back face

Mitigation

- Reduce cooling mass flow rate?
  - Very low already
  - Bearing cooling concerns
- Reduce Nusselt number/ HTC?
  - Inherent to fluid properties

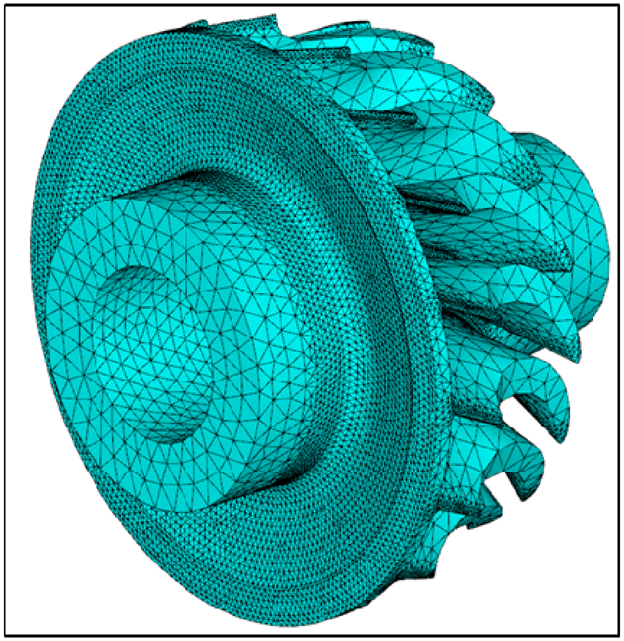
✓Raise temperature of cooling flow

  - Preheat cooling flow using high temperature stator components

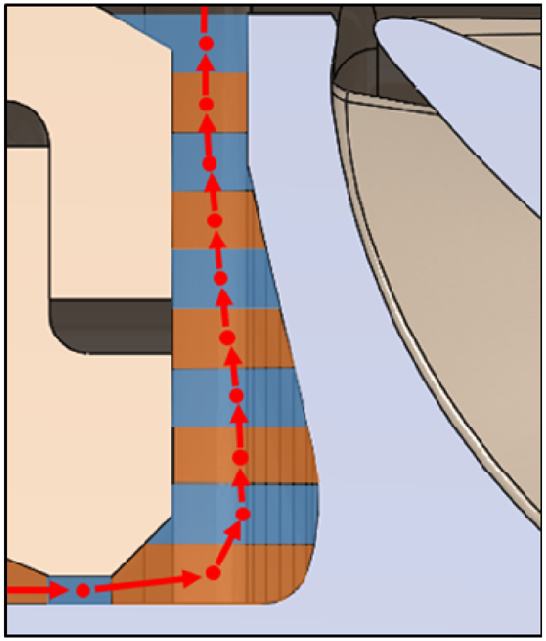


Thermal Finite Element Model

- ANSYS APDL
- Ten node tetrahedral solid elements
- FLUID 116 Elements
- Model Inputs:
  - Aerodynamic analysis results
  - Secondary flow analysis results
- Common model of transient and steady state



HPT solid body thermal mesh

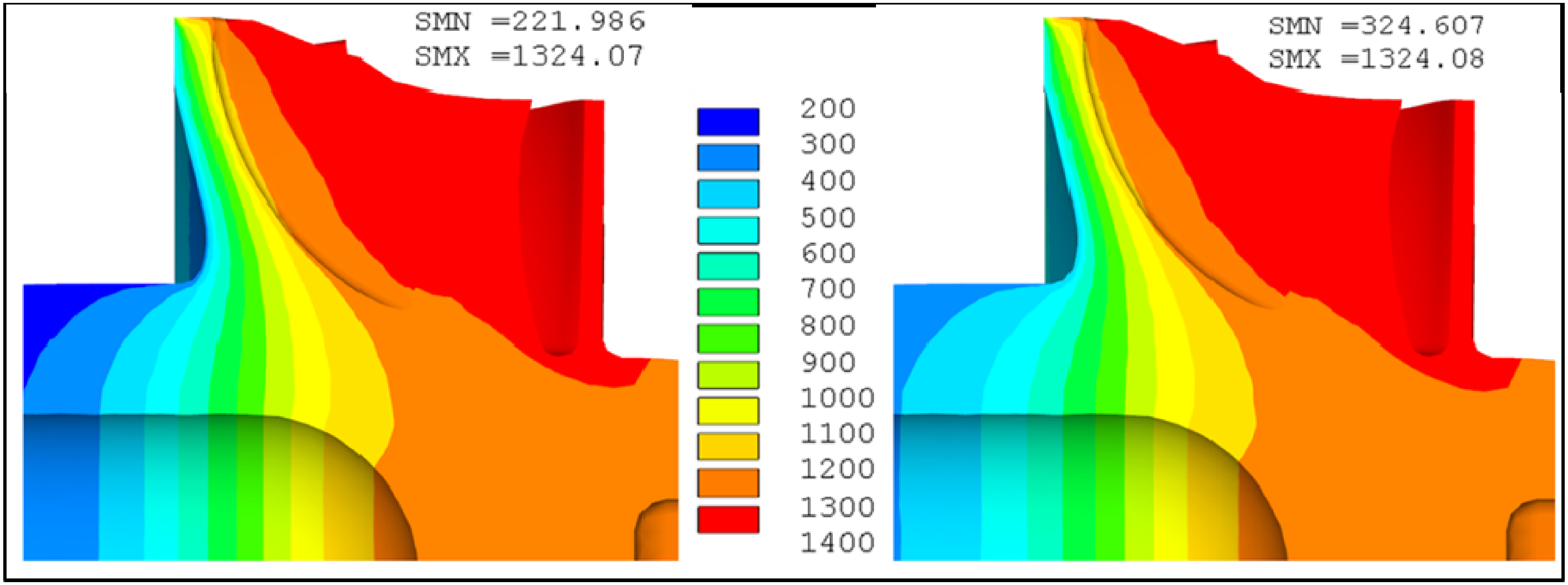


HPT back face fluid network

Structural Finite Element Model

- ANSYS
- Ten node tetrahedral solid elements
- Model Inputs:
  - Thermal model results
  - Angular velocity
  - Surface pressures
  - Assembly loads

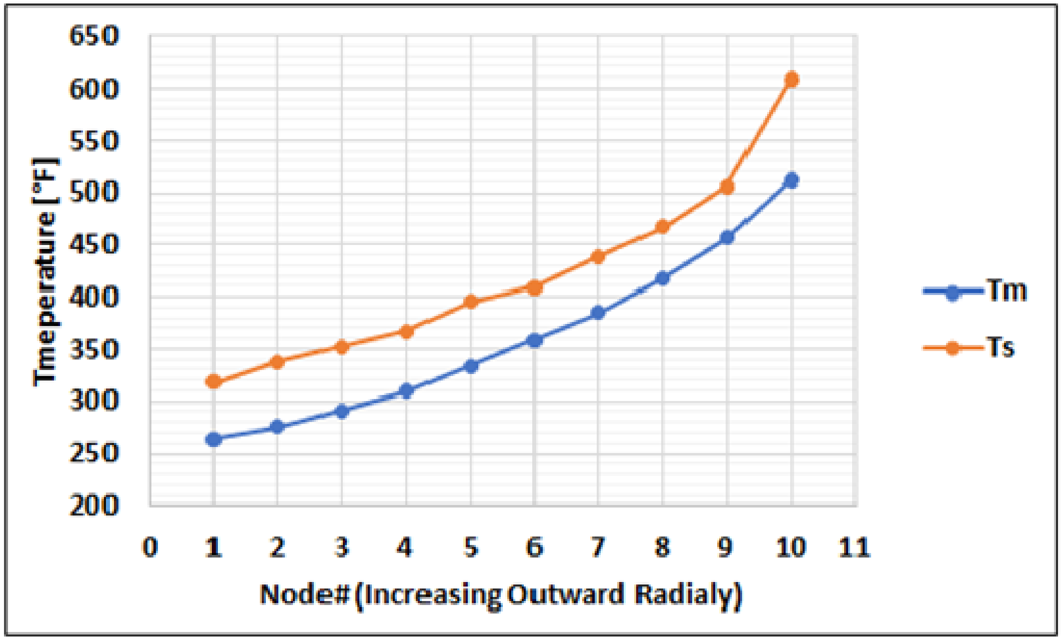
Thermal Model Results



SS max HPT isothermal Plot (°F). No recirculation flow left. With recirculation flow right. (SMN and SMX are maximum and minimum temperatures)

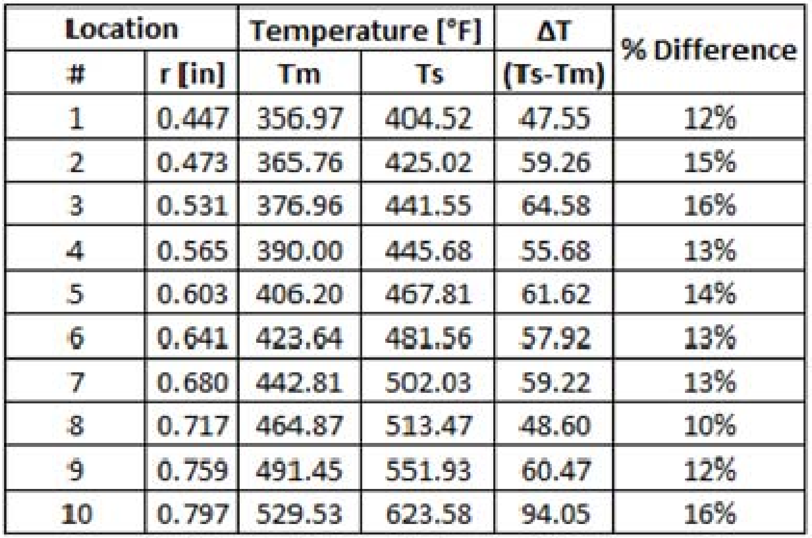
Thermal Model Results

Location		Temperature [°F]		ΔT	% Difference
#	r [in]	Tm	Ts	(Ts-Tm)	
1	0.447	264.13	317.26	53.13	18%
2	0.473	275.80	338.00	62.21	20%
3	0.531	291.64	352.95	61.31	19%
4	0.565	309.71	367.36	57.65	17%
5	0.603	333.88	395.54	61.66	17%
6	0.641	358.25	409.50	51.25	13%
7	0.680	385.43	438.13	52.70	13%
8	0.717	417.47	467.26	49.79	11%
9	0.759	456.08	505.93	49.85	10%
10	0.797	511.65	609.01	97.36	17%

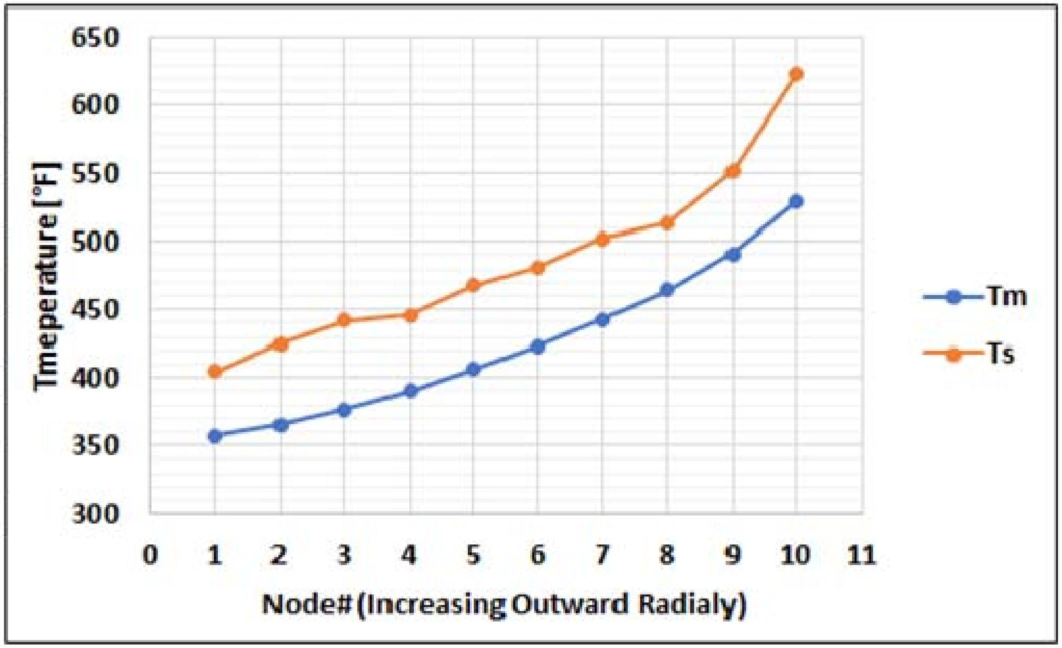


SS max HPT back face temperatures fluid vs. surface (no recirculation flow)

Thermal Model Results

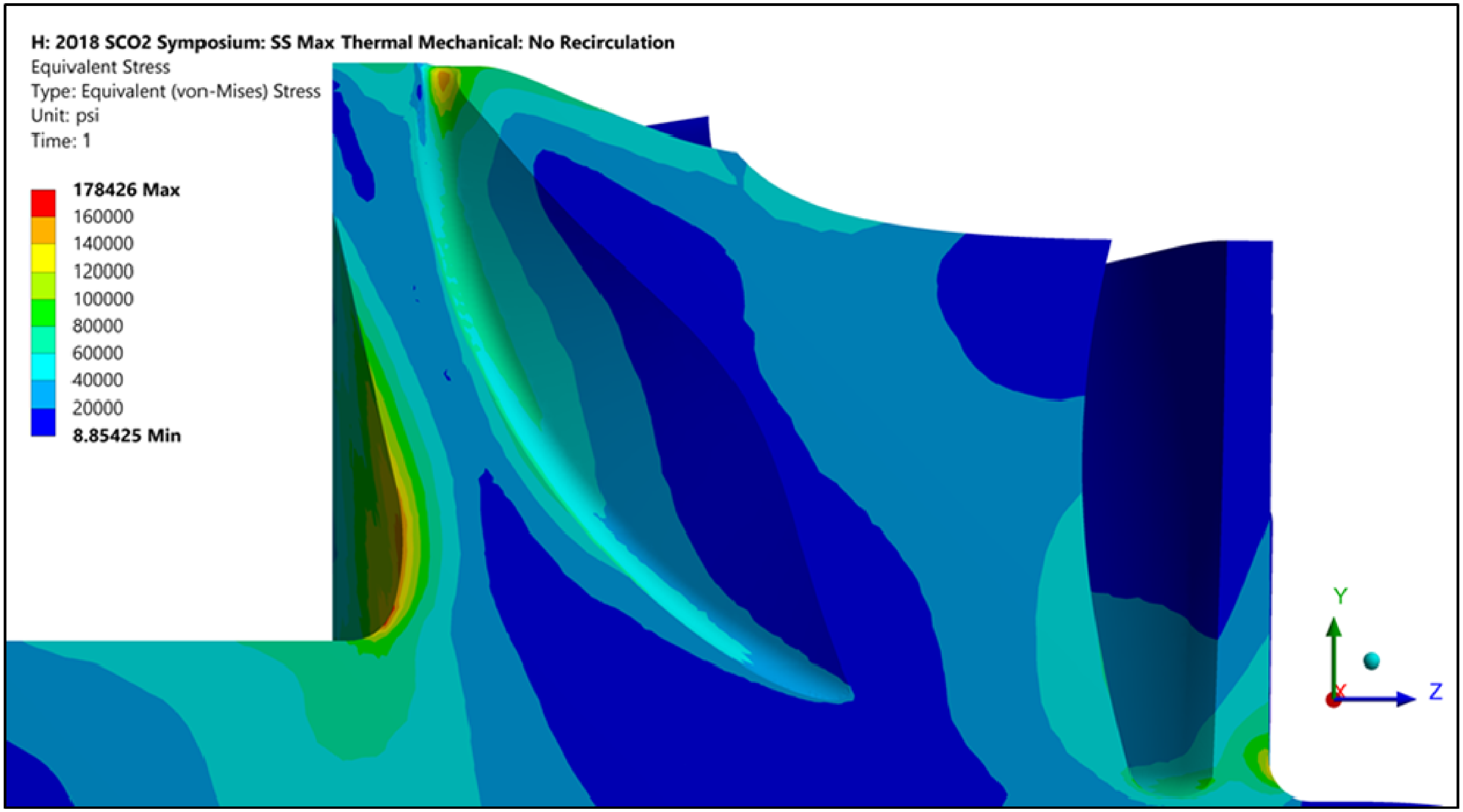


Location		Temperature [°F]		ΔT	% Difference
#	r [in]	Tm	Ts	(Ts-Tm)	
1	0.447	356.97	404.52	47.55	12%
2	0.473	365.76	425.02	59.26	15%
3	0.531	376.96	441.55	64.58	16%
4	0.565	390.00	445.68	55.68	13%
5	0.603	406.20	467.81	61.62	14%
6	0.641	423.64	481.56	57.92	13%
7	0.680	442.81	502.03	59.22	13%
8	0.717	464.87	513.47	48.60	10%
9	0.759	491.45	551.93	60.47	12%
10	0.797	529.53	623.58	94.05	16%

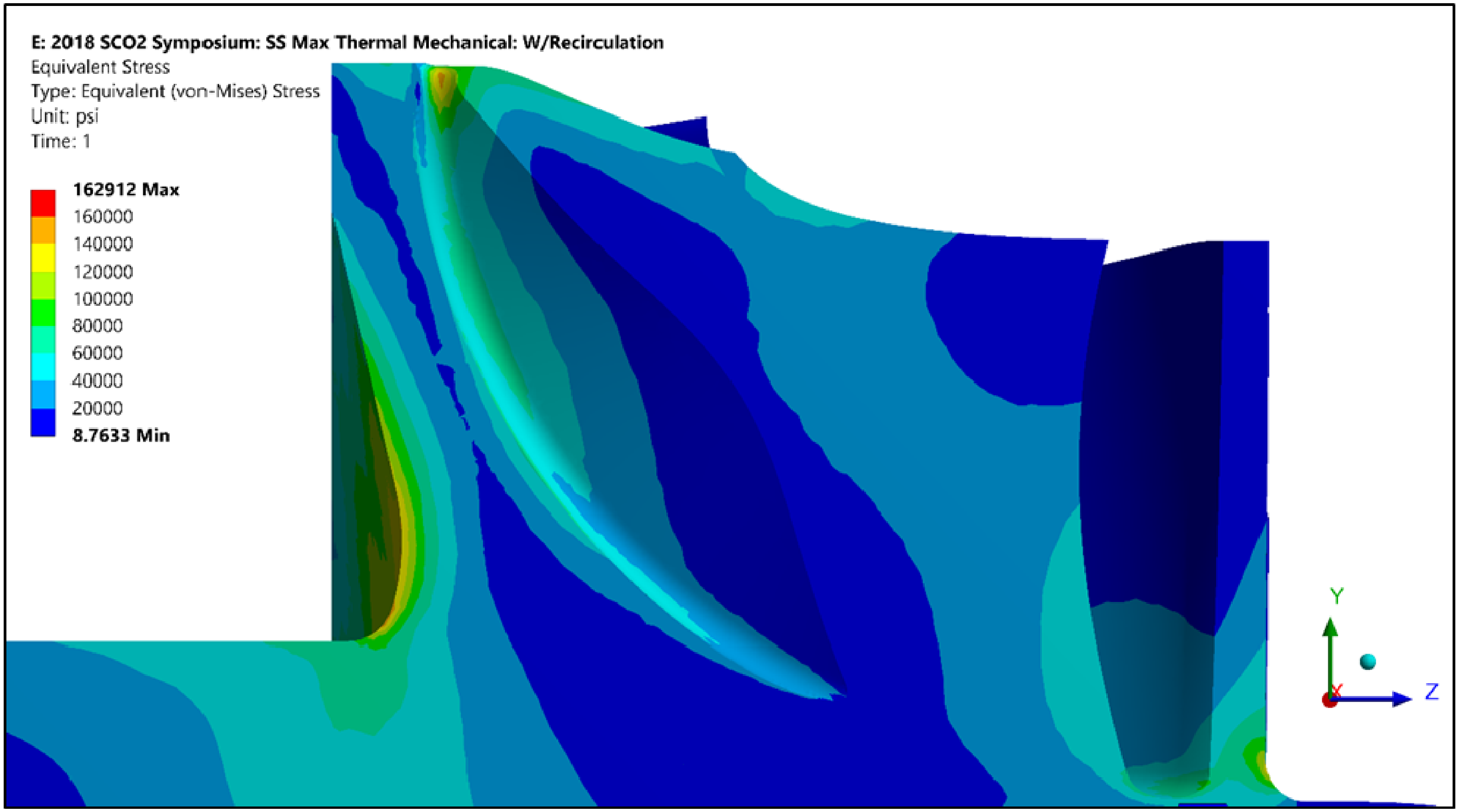


SS max HPT back face temperatures fluid vs. surface (with recirculation flow)

Structural Model Results



Structural Model Results



Conclusions

- Problem and solution were contrary to traditional air cooled radial turbine wheels
- The root of the problem was high thermal gradients caused by high convective heat transfer on the primary flow and back face surfaces of the HPT.
- By examining and discussing how sCO_2 and air perform very differently as heat transfer media in turbomachinery the problem was directly linked to the unique thermo-physical properties of sCO_2.
- Examination revealed Nusselt numbers in typical sCO_2 turbomachinery can be over 20x greater than in typical air breathing machines.
- A unique and successful secondary flow solution comprised of a preheating recirculation loop was described.
- Stresses were reduced to acceptable fatigue design criteria.