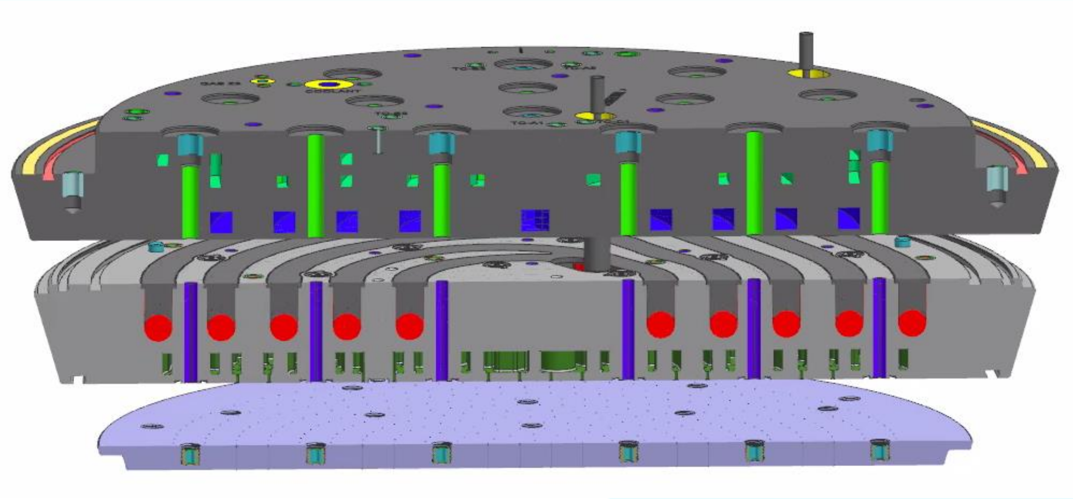
FEA ANALYSIS OF SHOWERHEAD

introduction

SHOWERHEAD:

- This study focuses on the FEA analysis of a plasma-producing showerhead, aiming to improve its efficiency and reliability.
- The primary goal is to maintain a constant temperature throughout the wafer from the shower head.
- In this study, varying critical design factors are heater coil length, cooling channel path, coolant flow rate, thermal gasket material, gasket diameter, and power input conditions.



DESIGN OPTIMIZATION OF SHOWERHEAD

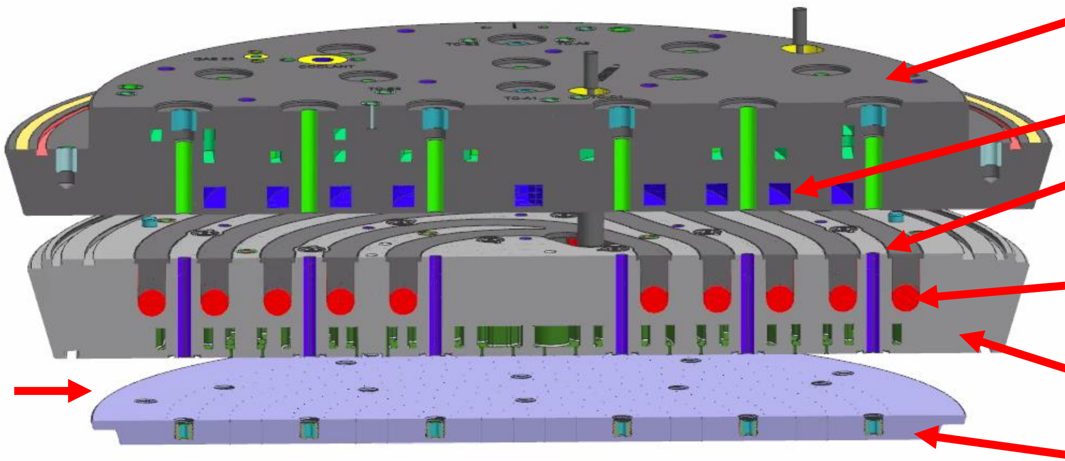
Objective:

- Optimize the heater coil and cooling channel design.
- Limit temperature variation on the showerhead to 5^0C.

Methodology:

- Performed a steady-state thermal analysis for the showerhead.
- Performed a parametric study by varying:
  a) Heater coil length
  b) Cooling channel path
  c) Coolant flow rate
  d) Thermal gasket material
  e) Gasket diameter
  f) Power input conditions
- The problem can be solved either by FEA or CFD approach.
- FEA is a cost and time-saving approach while the CFD approach involves greater time and cost investment.
- The challenge with FEA is to compute the heat transfer coefficients for the cooling channel analytically.
- Both FEA and CFD simulations were performed for one design configuration, and results were compared.
- Analytically calculated heat transfer coefficients(HTC) were in close correlation with computed values by CFD.
- FEA approach was used for other design configurations.
- The cooling channel was divided into 4 equal segments of length and HTC for each segment is defined.
- Ansys solver is used for the simulation.

SHOWERHEAD ANATOMY



Thermal Gasket (diameter = 12.57in)
Thermal Impedance = .110 C-in^2/W

Thermal Gasket (diameter = same as showerhead)
Thermal Impedance = .244 C-in^2/W

Chillplate (AL6061)

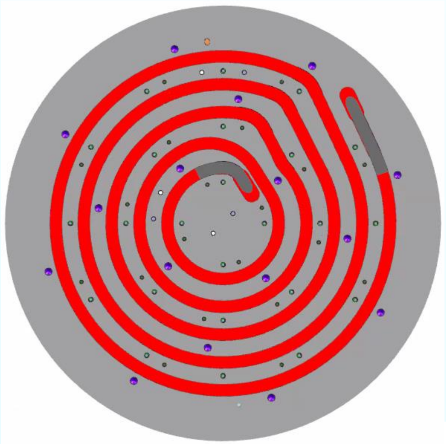
Cooling Channel

E-beam Welded Heater Cap (AL6061)
(weld dimensions: .63” long X .02” wide X .08” deep)

Heater (SST sheath)

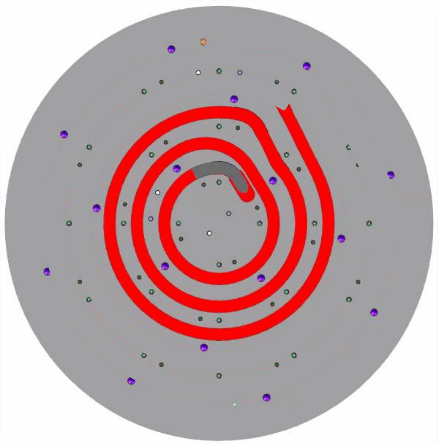
Heater Backplate (AL6061)

Showerhead (Si)



Heater (standard)

- 1 zone
- 6kW max



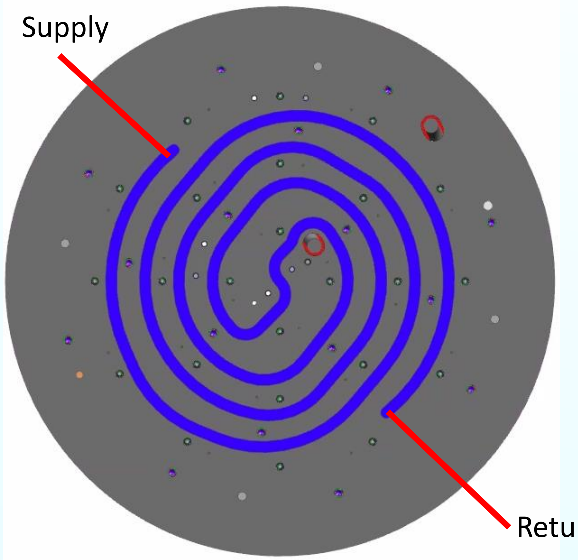
Heater (reduced length)

- 1 zone
- 4.6kW max

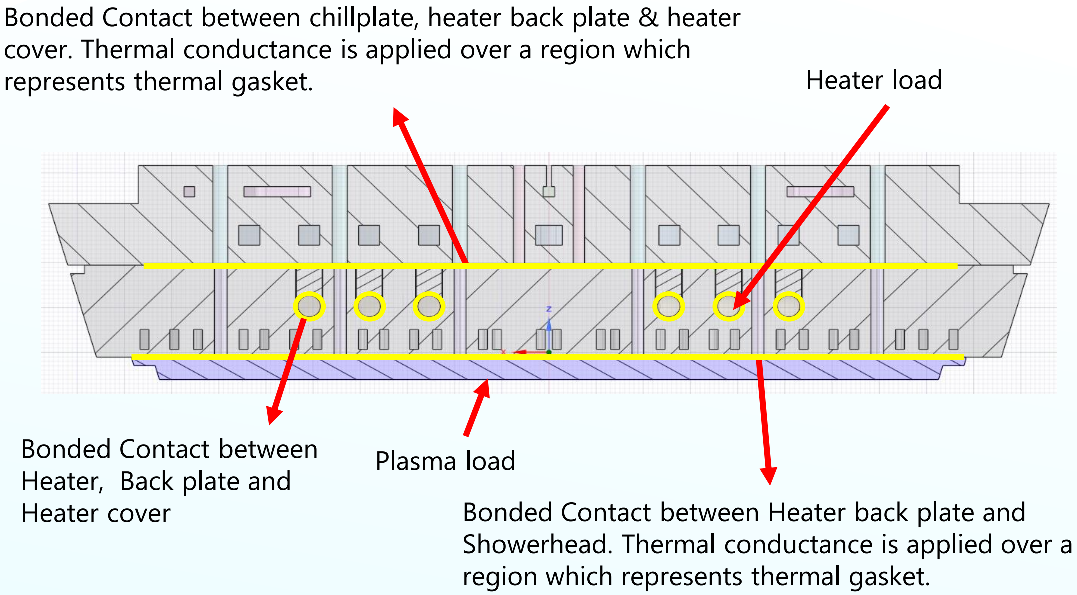
Heater coil configurations

Cooling Channel

- 1 zone
- Coolant: HT200
- Max Flow: 4GPM & 6 GPM
- Coolant Temp: 25-65C



LOADS AND BOUNDARY CONDITIONS

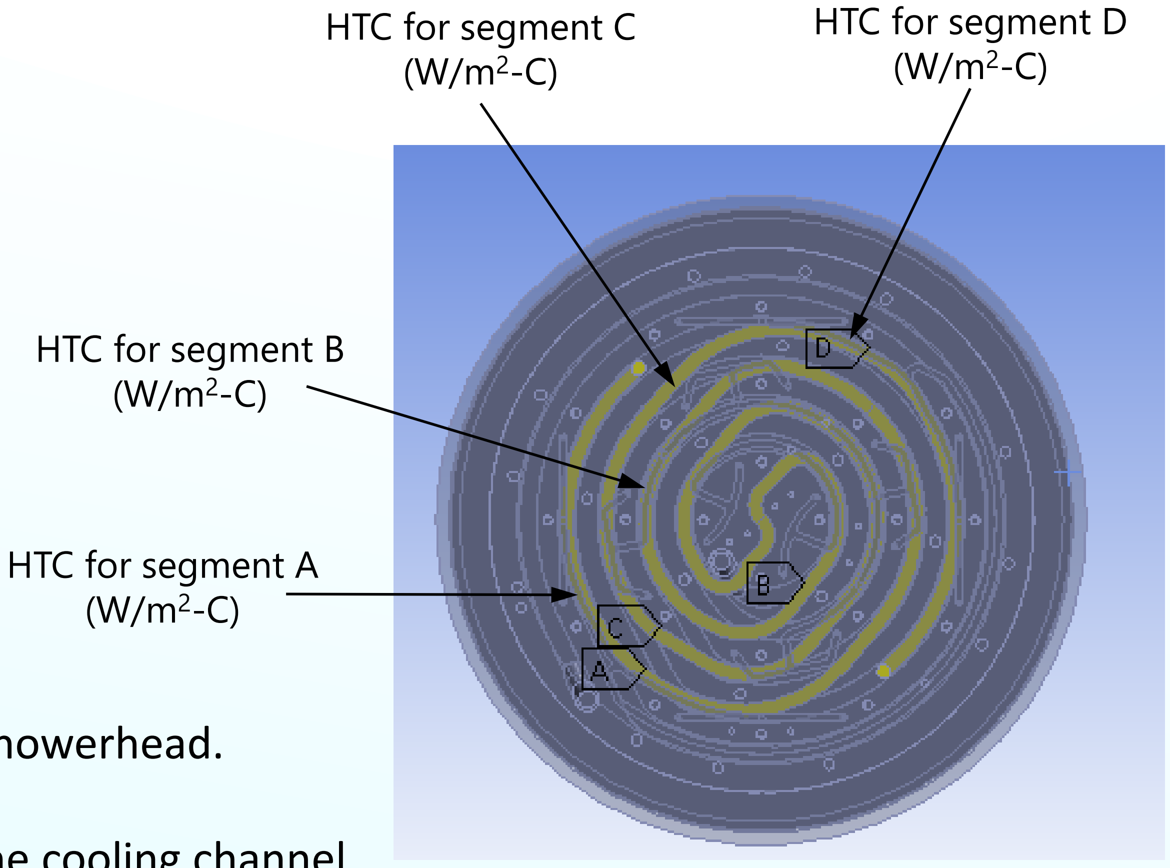

Bonded Contact between chillplate, heater back plate & heater cover. Thermal conductance is applied over a region which represents thermal gasket.

Heater load

Bonded Contact between Heater, Back plate and Heater cover

Plasma load

Bonded Contact between Heater back plate and Showerhead. Thermal conductance is applied over a region which represents thermal gasket.


HTC for segment C (W/m^2-C)

HTC for segment D (W/m^2-C)

HTC for segment B (W/m^2-C)

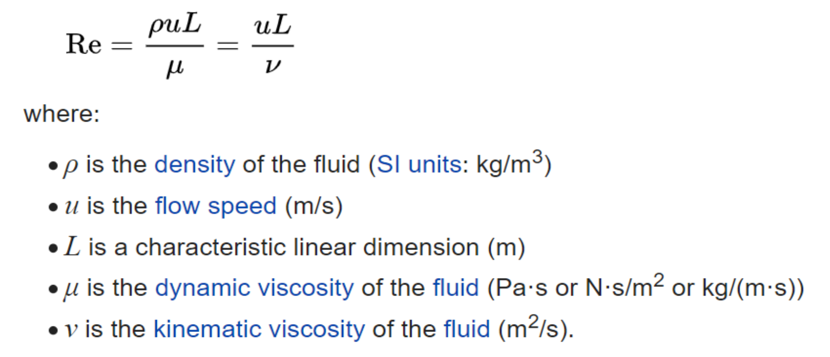
HTC for segment A (W/m^2-C)

Cooling channel segments A, B, C, & D

- Thermal plasma load of 3kW on the bottom surface of the showerhead.
- Heater load of 0.46 to 6kW is applied to heater volume.
- Heat transfer coefficients (HTC) are applied to the walls of the cooling channel.
- Inlet fluid temperature = 25 to 65 ^{0}C
- Inlet fluid volume flow rate = 4 or 6 GPM

HEAT TRANSFER COEFFICIENT CALCULATIONS

Step 1: Reynolds number computation

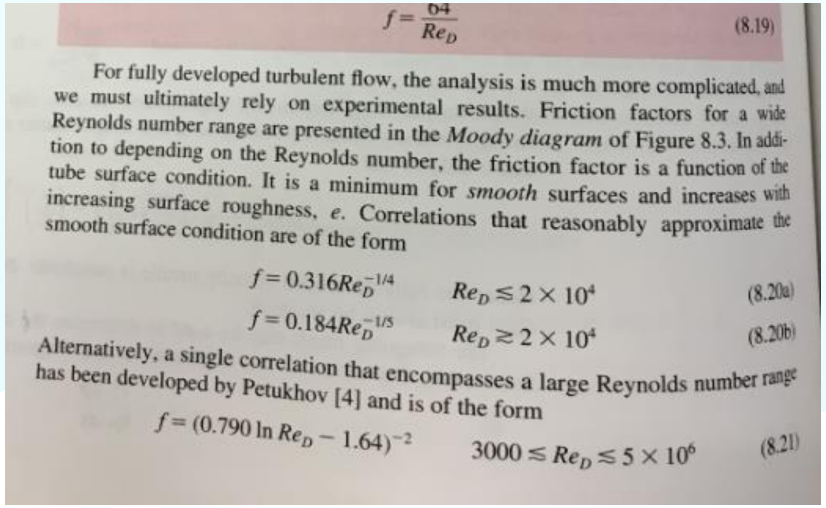


$$\mathrm{Re} = \frac{\rho u L}{\mu} = \frac{u L}{\nu}$$

where:

- ρ is the density of the fluid (SI units: kg/m^3)
- u is the flow speed (m/s)
- L is a characteristic linear dimension (m)
- μ is the dynamic viscosity of the fluid (Pa·s or N·s/m^2 or kg/(m·s))
- ν is the kinematic viscosity of the fluid (m^2/s).

Step 2: Wall friction coefficient computation



$$f = \frac{64}{Re_D} \qquad (8.19)$$

For fully developed turbulent flow, the analysis is much more complicated, and we must ultimately rely on experimental results. Friction factors for a wide Reynolds number range are presented in the *Moody diagram* of Figure 8.3. In addition to depending on the Reynolds number, the friction factor is a function of the tube surface condition. It is a minimum for *smooth* surfaces and increases with increasing surface roughness, *e*. Correlations that reasonably approximate the smooth surface condition are of the form

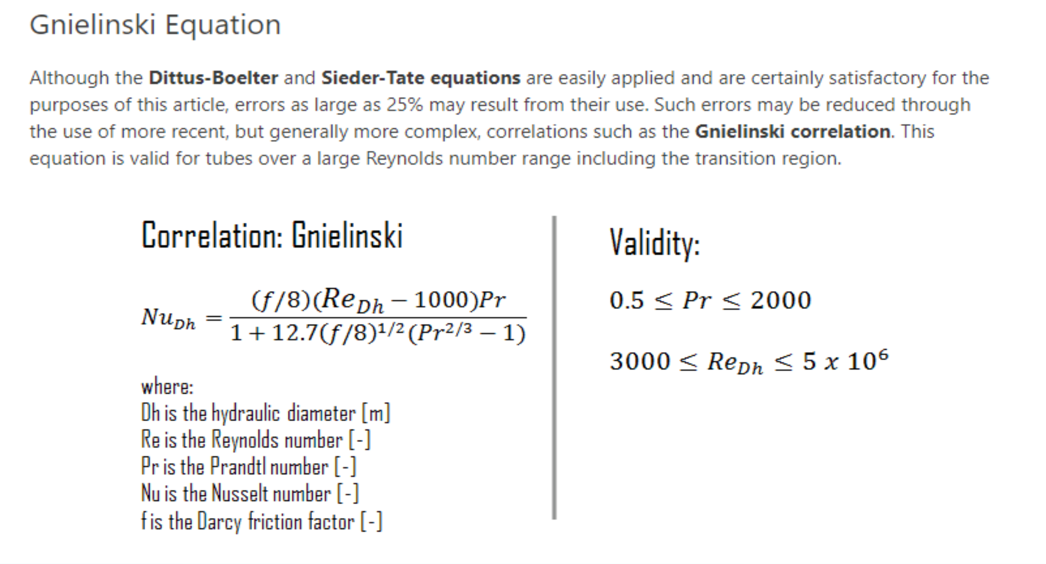
$$f = 0.316 Re_D^{-1/4} \qquad Re_D \lesssim 2 \times 10^4 \qquad (8.20a)$$

$$f = 0.184 Re_D^{-1/5} \qquad Re_D \gtrsim 2 \times 10^4 \qquad (8.20b)$$

Alternatively, a single correlation that encompasses a large Reynolds number range has been developed by Petukhov [4] and is of the form

$$f = (0.790 \ln Re_D - 1.64)^{-2} \qquad 3000 \lesssim Re_D \lesssim 5 \times 10^6 \qquad (8.21)$$

Step 3: Nusselt number computation



Gnielinski Equation

Although the **Dittus-Boelter** and **Sieder-Tate equations** are easily applied and are certainly satisfactory for the purposes of this article, errors as large as 25% may result from their use. Such errors may be reduced through the use of more recent, but generally more complex, correlations such as the **Gnielinski correlation**. This equation is valid for tubes over a large Reynolds number range including the transition region.

Correlation: Gnielinski

$$Nu_{Dh} = \frac{(f/8)(Re_{Dh} - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$$

where:
Dh is the hydraulic diameter [m]
Re is the Reynolds number [-]
Pr is the Prandtl number [-]
Nu is the Nusselt number [-]
f is the Darcy friction factor [-]

Validity:

$$0.5 \leq Pr \leq 2000$$

$$3000 \leq Re_{Dh} \leq 5 \, x \, 10^6$$

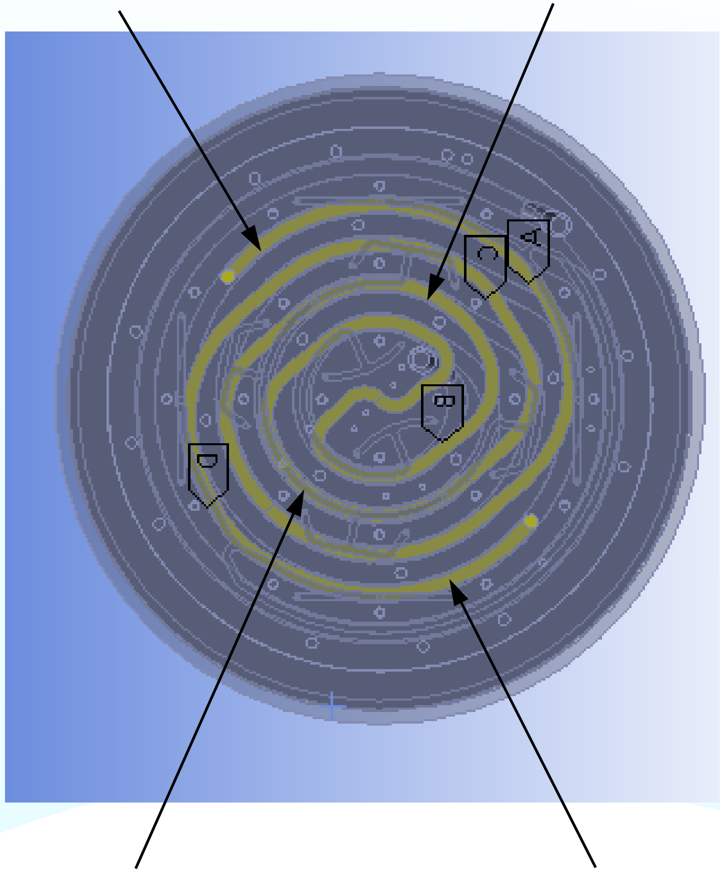
Step 4: Heat transfer coefficient computation

$$h = \frac{Nu_{Dh} \times k}{D}$$

Where,
h- heat transfer coefficient
Nu – Nusselt number
k – Thermal conductivity of the fluid
D- Hydraulic diameter of the tube/channel

HEAT TRANSFER COEFFICIENT COMPARISION

HTC by analytical method for FEA



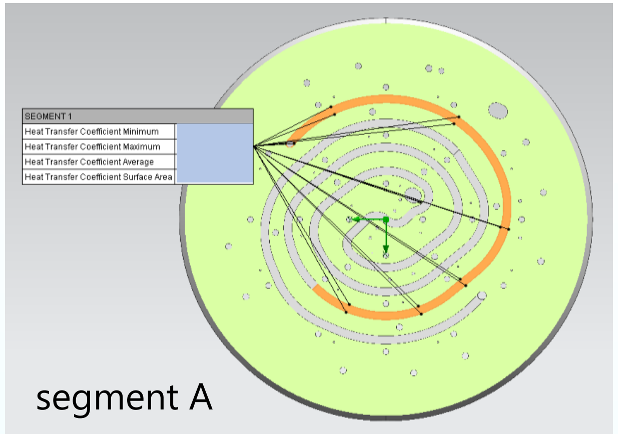
HTC for segment A (W/m^2-C)

HTC for segment B (W/m^2-C)

HTC for segment C (W/m^2-C)

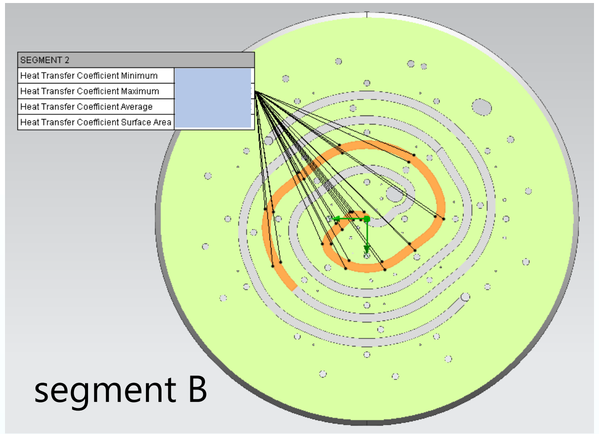
HTC for segment D (W/m^2-C)

HTC by CFD



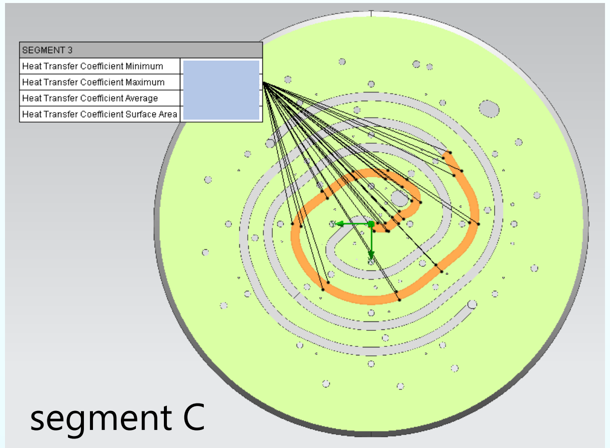
SEGMENT 1	
Heat Transfer Coefficient Minimum	[redacted]
Heat Transfer Coefficient Maximum	[redacted]
Heat Transfer Coefficient Average	[redacted]
Heat Transfer Coefficient Surface Area	[redacted]

segment A



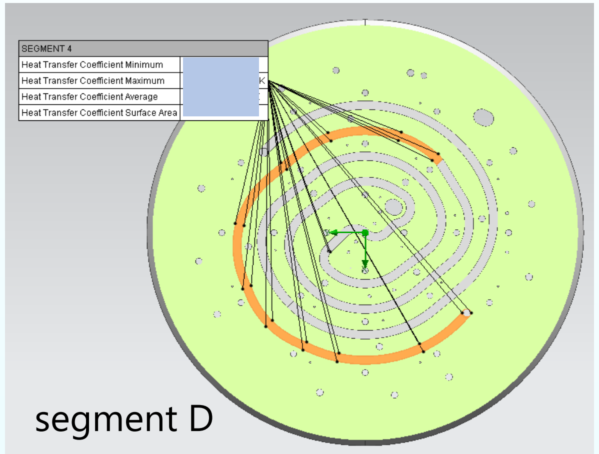
SEGMENT 2	
Heat Transfer Coefficient Minimum	[redacted]
Heat Transfer Coefficient Maximum	[redacted]
Heat Transfer Coefficient Average	[redacted]
Heat Transfer Coefficient Surface Area	[redacted]

segment B



SEGMENT 3	
Heat Transfer Coefficient Minimum	[redacted]
Heat Transfer Coefficient Maximum	[redacted]
Heat Transfer Coefficient Average	[redacted]
Heat Transfer Coefficient Surface Area	[redacted]

segment C

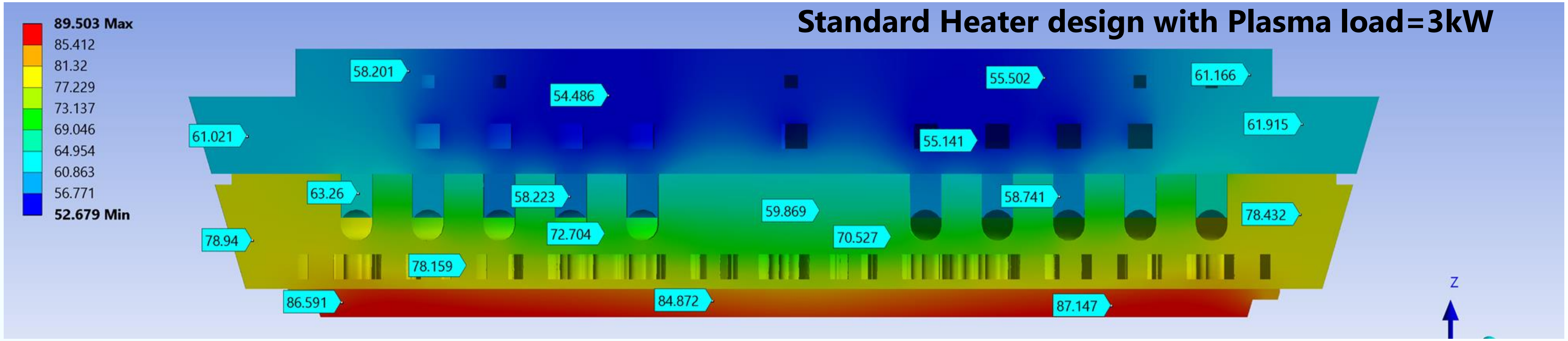


SEGMENT 4	
Heat Transfer Coefficient Minimum	[redacted]
Heat Transfer Coefficient Maximum	[redacted]
Heat Transfer Coefficient Average	[redacted]
Heat Transfer Coefficient Surface Area	[redacted]

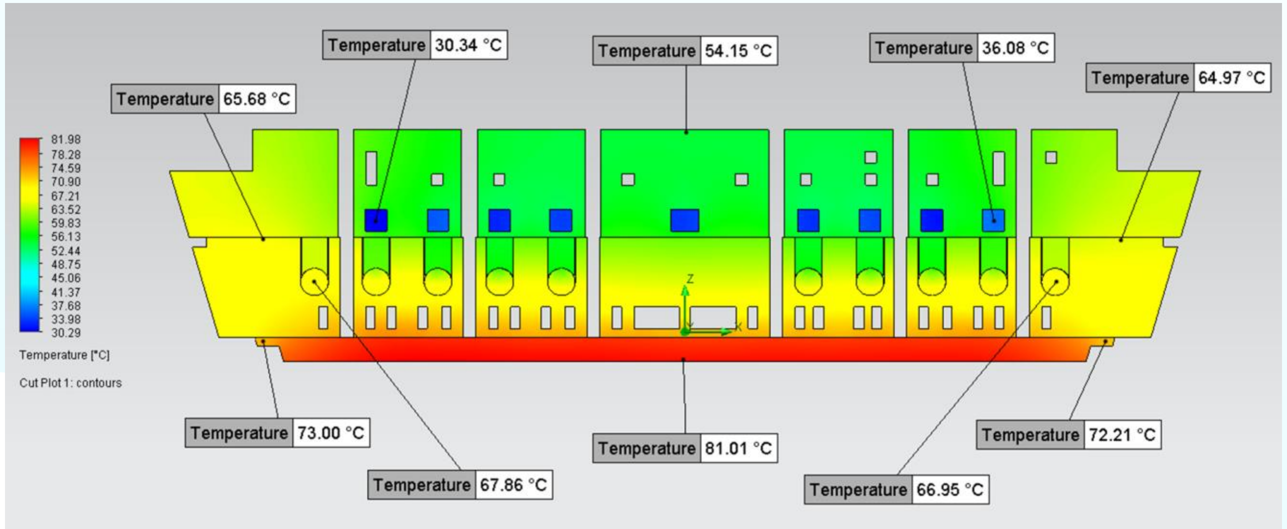
segment D

- ➢ Analytically calculated heat transfer coefficient values were in close correlation with computed values by CFD.

FEA AND CFD RESULTS COMPARISION


Standard Heater design with Plasma load=3kW

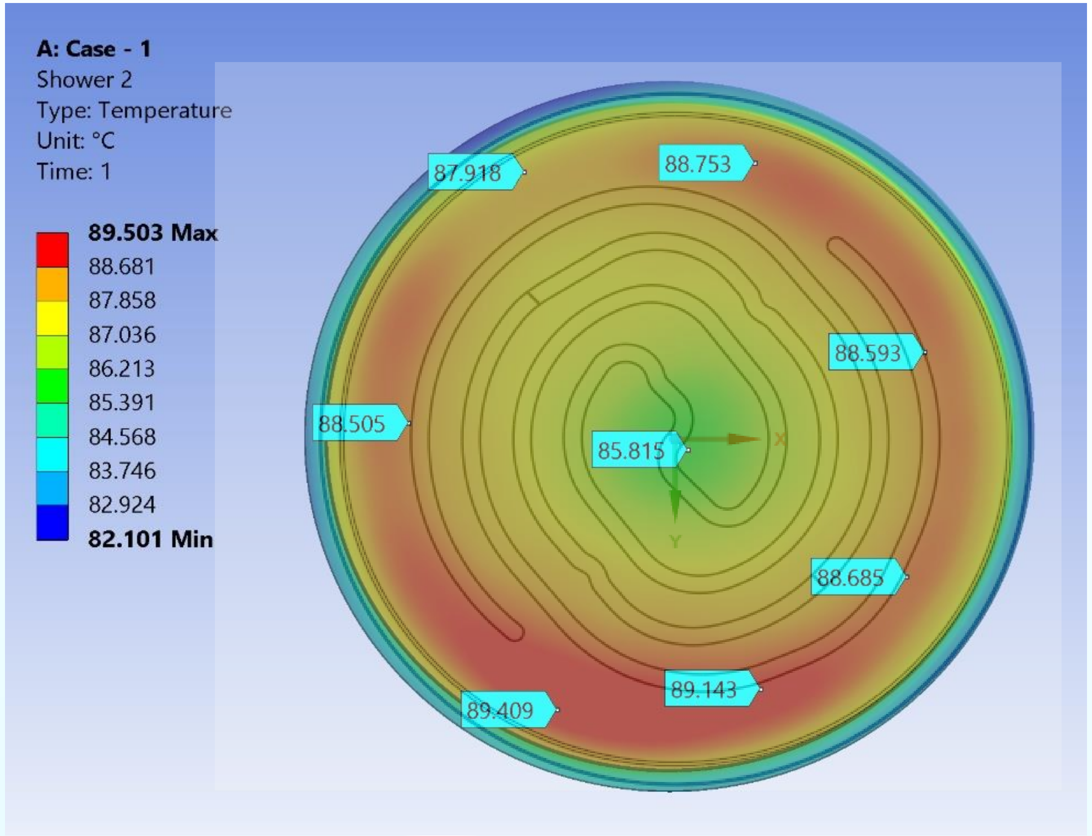
Overall Temperature – FEA approach



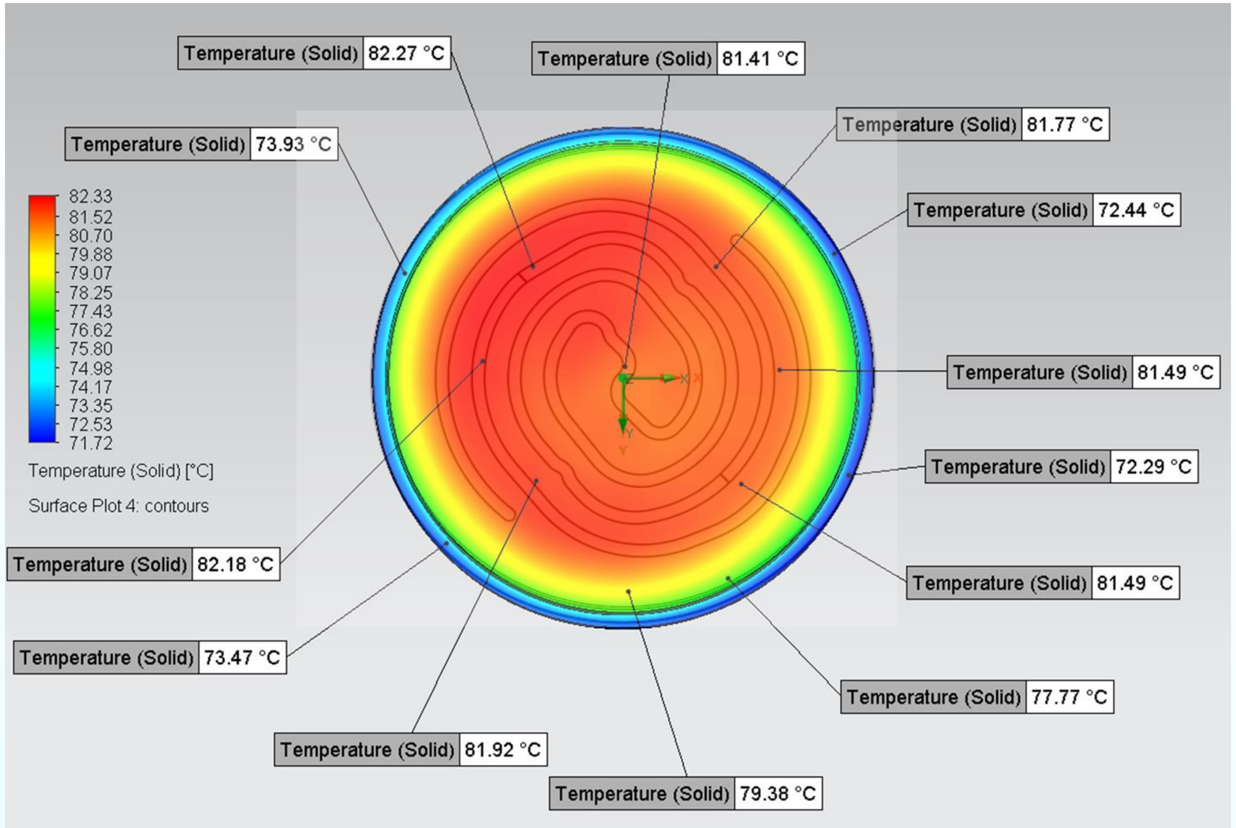
Overall Temperature – CFD approach

FEA AND CFD RESULTS COMPARISION

Standard Heater design with Plasma load=3kW



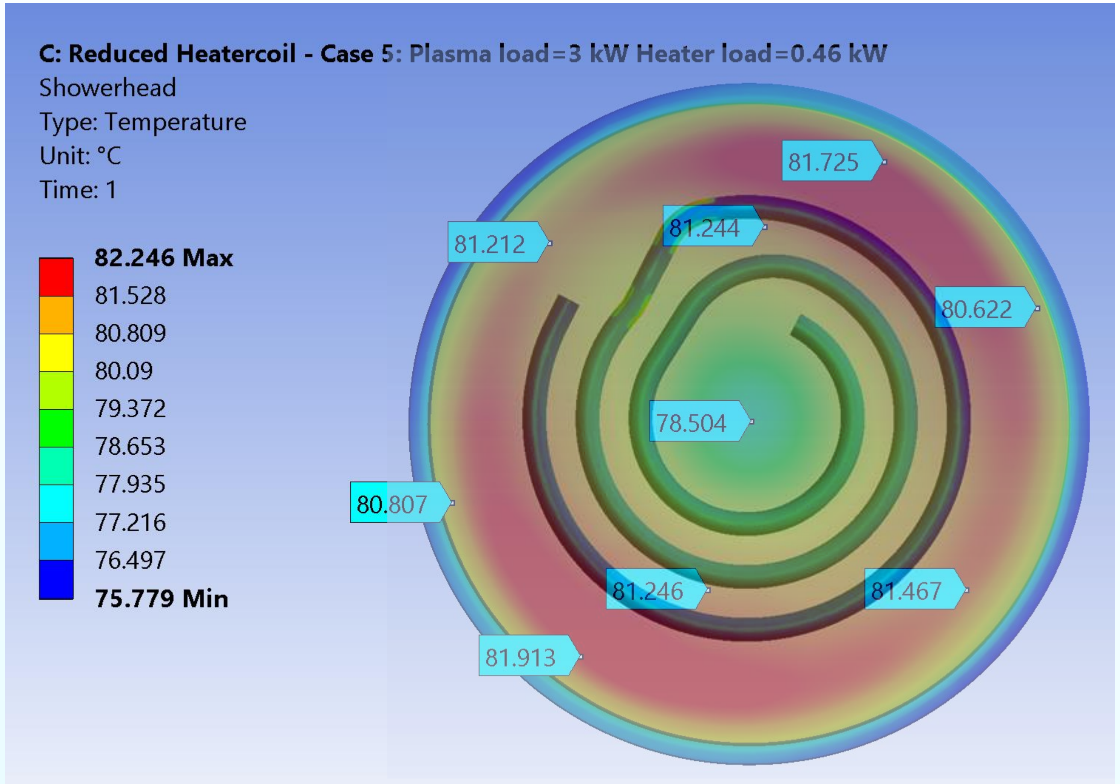
Shower head Temperature – FEA approach



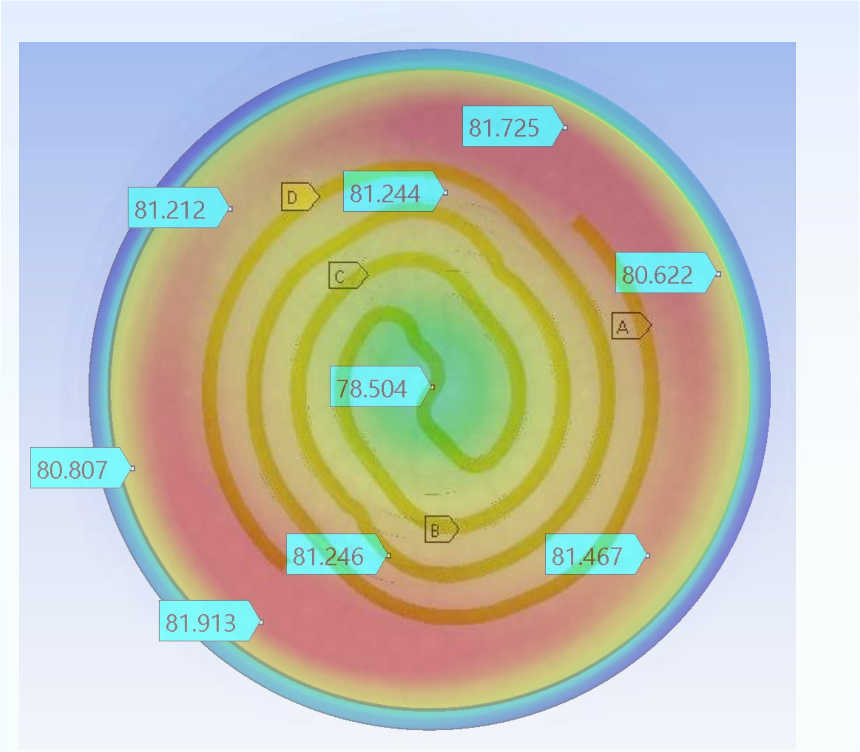
Shower head Temperature – CFD approach

REDUCED HEATER – SHOWERHEAD TEMPERATURE

Reduced heater design with Plasma load=3kW & Heater load=0.46kW



Shower head - Bottom Side Heater overlaid



Shower head, Heater & Cooling channel overlaid

- Maximum temperature of 82.2 ^{0}C and minimum temperature of 78.5 ^{0}C observed on bottom side of Showerhead. Hence a $\Delta T=3.7$ ^{0}C exists and is with in the desirable range. Average temperature over the bottom surface is 79.4 ^{0}C and is close to the requirement specifications.

Overall Comparison Table

Heater Path		Lower Thermal Gasket	Upper Thermal Gasket	Heater Power (kW)	Plasma Load (kW)	Coolant Flow (GPM)	Coolant Temperature (°C)	Showerhead Temperature (°C)		
								ΔT	Max. Temp.	Avg. Temp.
Standard Heater	CASE 1	OD = same as showerhead Thermal Impedance = .244 C-in^2/W	OD = 12.7” Thermal Impedance = .110 C-in^2/W	0	3	4	30	7.4	89.5	86.4
Reduced Heater	CASE 1	OD = same as showerhead Thermal Impedance = .244 C-in^2/W	OD = 12.7” Thermal Impedance = .110 C-in^2/W	0	3	4	30	4.3	85.1	82.6
	CASE 3	13.76 OD 0.273 C-in^2/W thermal impedance	8.53 OD 0.078 C-in^2/W thermal impedance	0.46	3	6	25	25.1	114.5	105.0
	CASE 5		10.5 OD 0.093 C-in^2/W thermal impedance	0.46	3	6	25	3.7	82.2	79.4

All temperature values are in °C

Overall Comparison Table

Heater Path		Lower Thermal Gasket	Upper Thermal Gasket	Heater Power (kW)	Plasma Load (kW)	Coolant Flow (GPM)	Coolant Temperature (^{o}C)	Showerhead Temperature (^{0}C)		
								ΔT	Max. Temp.	Avg. Temp.
Standard Heater	CASE 2A	OD = same as showerhead Thermal Impedance = .244 C-in^2/W	OD = 12.7” Thermal Impedance = .110 C-in^2/W	6	0	4	30	18.3	115.6	108.8
	CASE 2B			5	0	4	30	15.3	101.9	96.2
Reduced Heater	CASE 2A	OD = same as showerhead Thermal Impedance = .244 C-in^2/W	OD = 12.7” Thermal Impedance = .110 C-in^2/W	4.6	0	4	30	19.9	95.1	84.1
	CASE 2B			3.6	0	4	30	15.7	81.3	72.8
	CASE 4	13.76 OD 0.273 C-in^2/W thermal impedance	8.53 OD 0.078 C-in^2/W thermal impedance	4.14	0	6	65	10.6	120.7	116.9
	CASE 6		10.5 OD 0.093 C-in^2/W thermal impedance	4.14	0	6	65	14.2	107.4	99.5

All temperature values are in °C

CONCLUSION AND BENEFITS

Conclusion:

- For plasma load with standard heater, the temperature variation ΔT is 7.4 ^{0}C and a average temperature of 86.4 ^{0}C is observed while with a reduced heater the temperature variation ΔT is 4.3 ^{0}C and a average temperature of 82.6 deg C is observed. Hence showerhead with reduced heater gives design with less variation in temperature.
- For the reduced heater case with 3kW plasma load and 0.46kW heater load, maximum temperature of 82.2 ^{0}C and minimum temperature of 78.5 ^{0}C observed on bottom side of Showerhead. Hence a ΔT=3.7 ^{0}C exists and is with in the desirable range. Average temperature over the bottom surface is 79.4 ^{0}C and is close to the requirement specifications.

Benefits:

- Simulations were performed using the FEA approach to save computational cost and time.
- Customer saved both in prototyping and development costs.

CONTACT US

Advanced Engineering Services

https://aesgs.com

THANK YOU