

# AC-LGAD ToF : halfSTAVE prototype updates and heat transfer analysis for miniSTAVE and fullSTAVE

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- ◊ Updated glycol and water-cooling HTC values
  - ◊ Hand-calc estimation of HTC
  - ◊ Test Cases
    - ◊ -10 C 50% Glycol/Water
    - ◊ 0 C 50% Glycol/Water
    - ◊ 5 C Water
    - ◊ 18 C Water
- ◊ ASIC Direct to Hybrid Flex

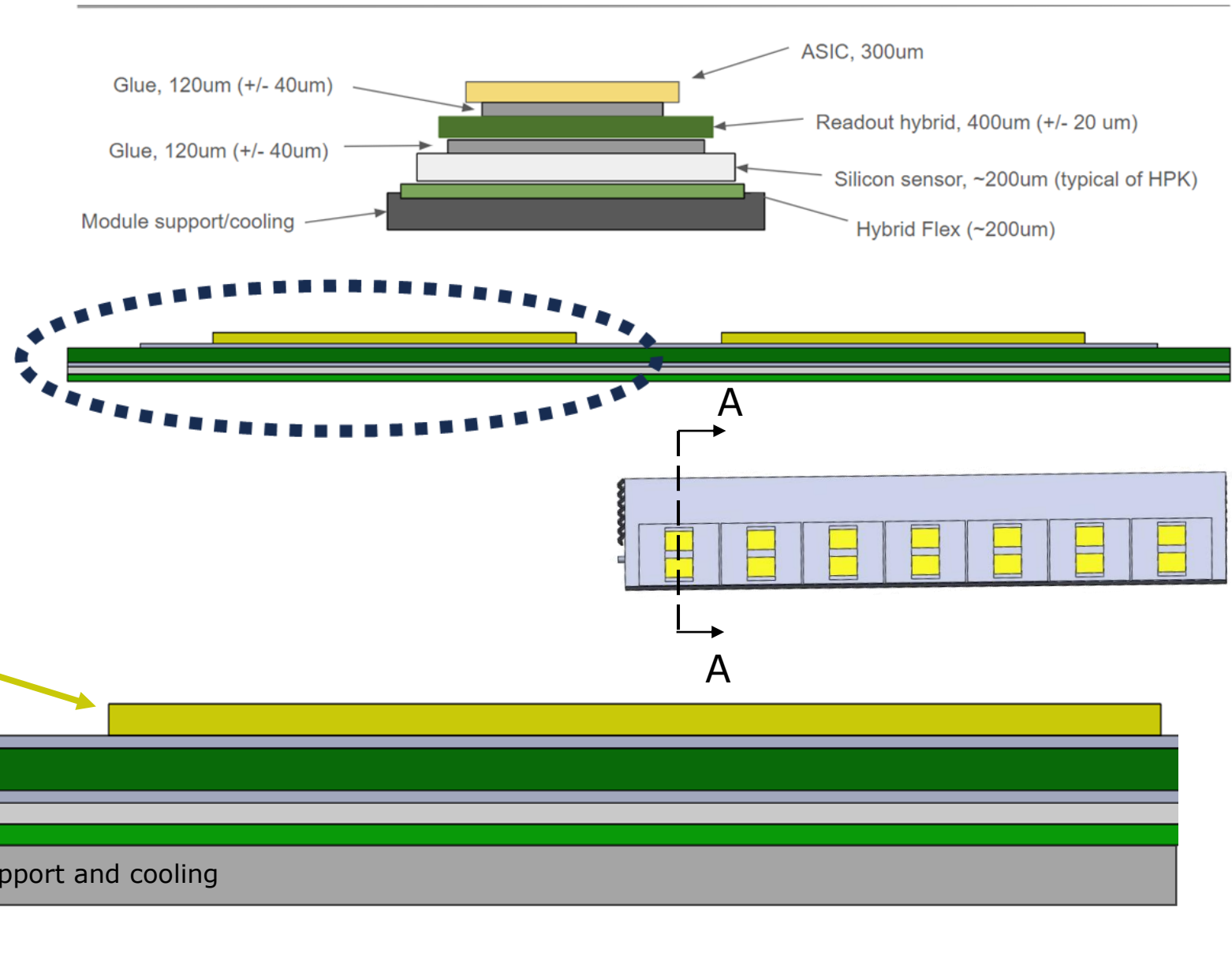
## Next Steps:

- ◊ Verify convective heat transfer coefficient values with Ansys Fluent heat pipe simulations

- ◊ The heat transfer coefficient in the pipe makes a huge difference on the temperature profiles
- ◊ We need a good way to do CFD coupled heat transfer simulations for getting performance metrics for the stove
- ◊ *Inputs that need refinement from Cooling group at EIC*
  - ◊ *What is the inlet pressure at AC-LGAD?*
  - ◊ *What is the flow rate available? – currently estimated at 1m/s.*
  - ◊ *Currently assumed as flow conditions – Reynold's number (Re) ~ 250,000 – laminar flow regime*

OLD SLIDE for reminder of work on-going

<u>Part Name</u>	<u>Thermal Conductivity (W/mK)</u>	<u>Thickness (µm)</u>
ROC and ASIC (PCB/Kapton properties)	0.97	400 and 300
Silicon Module	148	200
Carbon Face Sheet	Kxx - 180 Kyy - 150 Kzz - 1.36	200
Carbon Foam	25	6420
Loctite Epoxy (Glue)	1.28	120
Stainless Steel Pipe	16	716



Heat transfer coefficient estimated to be ( $h$ ) ~~1000 W/m<sup>2</sup>K~~ decaying down to ~~360 W/m<sup>2</sup>K~~ (at outlet) --

$$h = \frac{k \cdot Nu_L}{L}$$

$$\begin{aligned}
 & \text{Nusselt number } (Nu_L) \\
 &= \frac{\left(\frac{f}{8}\right) (Re - 1000) Pr}{1 + 12.7 \left(\frac{f}{8}\right)^{0.5} ((Pr)^{\frac{2}{3}} - 1)}
 \end{aligned}$$

- ⬠ Need better pressure inlet and pressure outlet understanding for refining the simulations further
- ⬠ Heat transfer coefficients in a pipe for water at room temp and pipe diameter of 5 mm used for the current estimation of  $h$ .
- ⬠ Please can Yi/NCKU help with this – cross check my numbers ?

For laminar flow in a pipe at room temp we found out the Prandtl ( $Pr$ ) and Reynolds ( $Re$ ) numbers using (Gnielinski, 1976) and first Petukhov eq. (1970) :

$$f = (0.790 \ln Re - 1.64)^{-2}$$

For flow conditions  $Re \sim 250,000$

$L$  = length (function of  $x$ );  $k$  = conductive heat transfer coefficient (W/mK)

- $Re = \frac{\rho * u * L}{\nu}$  (1)

- $\rho$  = density of fluid (kg/m<sup>3</sup>)
- $u$  = flow speed (m/s)
- $L$  = diameter(m)
- $\nu$  = dynamic viscosity of fluid(Pa\*s)

- Petukhov equation

- $f = (0.790 \cdot \ln Re - 1.64)^{-2}$  (2)

- Gnielinski equation

- $Nu = \left[ \frac{(f/8)(Re - 1000)Pr}{1 + 12.7 \left(\frac{f}{8}\right)^{0.5} (Pr^{\frac{2}{3}} - 1)} \right]$  (3)

- $Nusselt\ Number(Nu) = \frac{hL}{k}$  (4)

- $h$  = heat transfer coefficient (W/m<sup>2</sup>C)
- $L$  = characteristic length (m)
- $k$  = thermal conductivity (W/mC)

- $Q = hA(T_{Wall} - T_{Fluid})$  (5)

- $Q$  = heat transfer (W)
- $h$  = heat transfer coefficient (W/m<sup>2</sup>C)
- $A$  = Contact Area (m<sup>2</sup>)

- $Q = mc_p(T_f - T_i)$  (6)

- $m$  = mass (kg)
- $c_p$  = specific heat of fluid  $\left(\frac{J}{kgC}\right)$

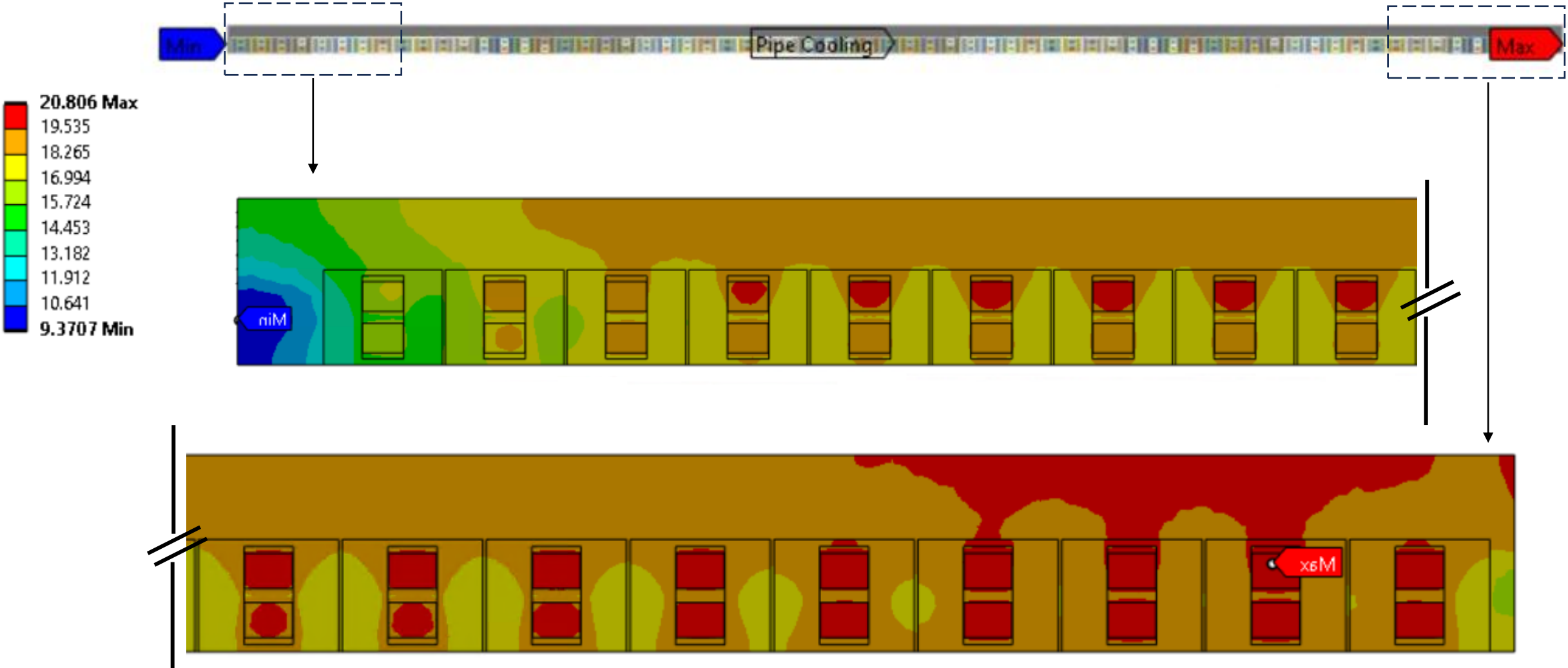
- Process

- $Re \rightarrow f \rightarrow Nu \rightarrow h \rightarrow Q \rightarrow T_f$
- Use iterative calculation in MATLAB to determine temperature increase of fluid using equations 5 and 6

**Note: potential for error in calculation, working to confirm values using heat pipe simulations in Ansys Fluent**

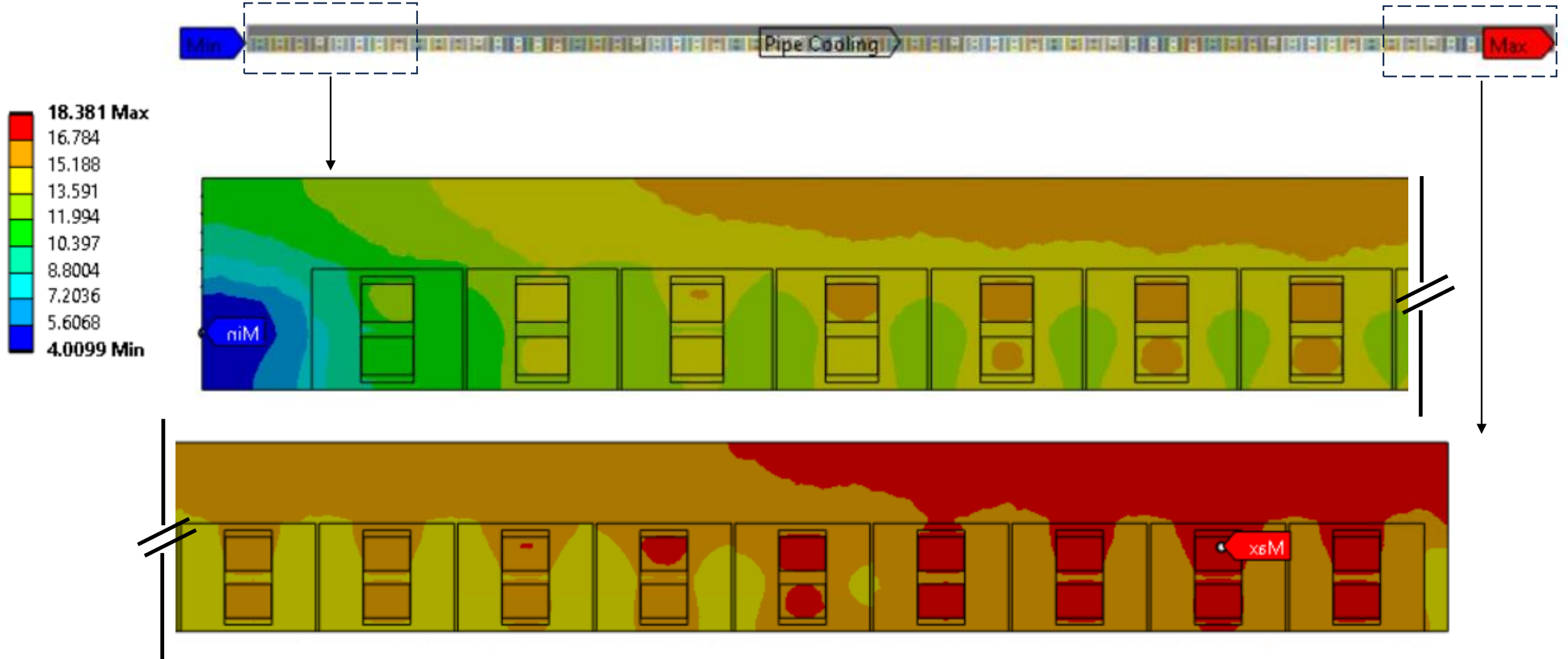
# 0 C Glycol Cooling (Full Stave)

HTC Range – 569 to 476 W/m<sup>2</sup>C



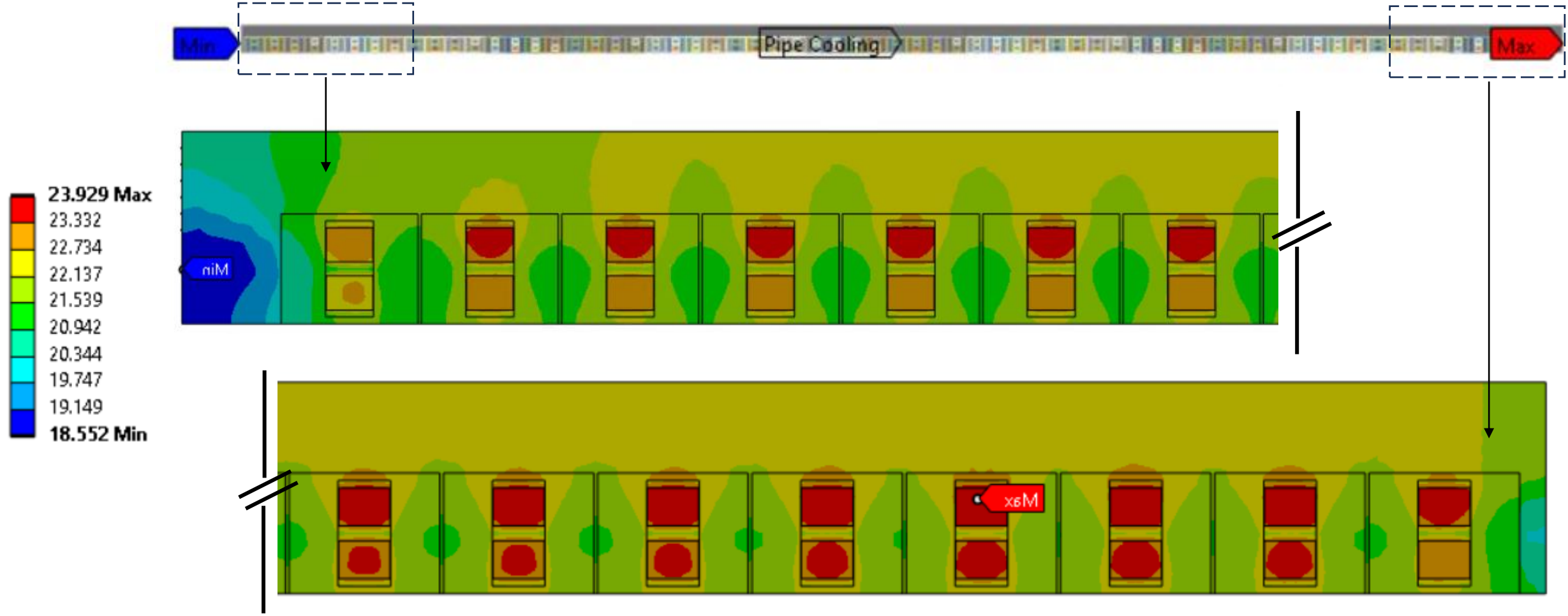
# -10 C Glycol Cooling (Full Stave)

HTC Range 447 to 335 W/m<sup>2</sup>C



# 18 C Water Cooling (Full Stave)

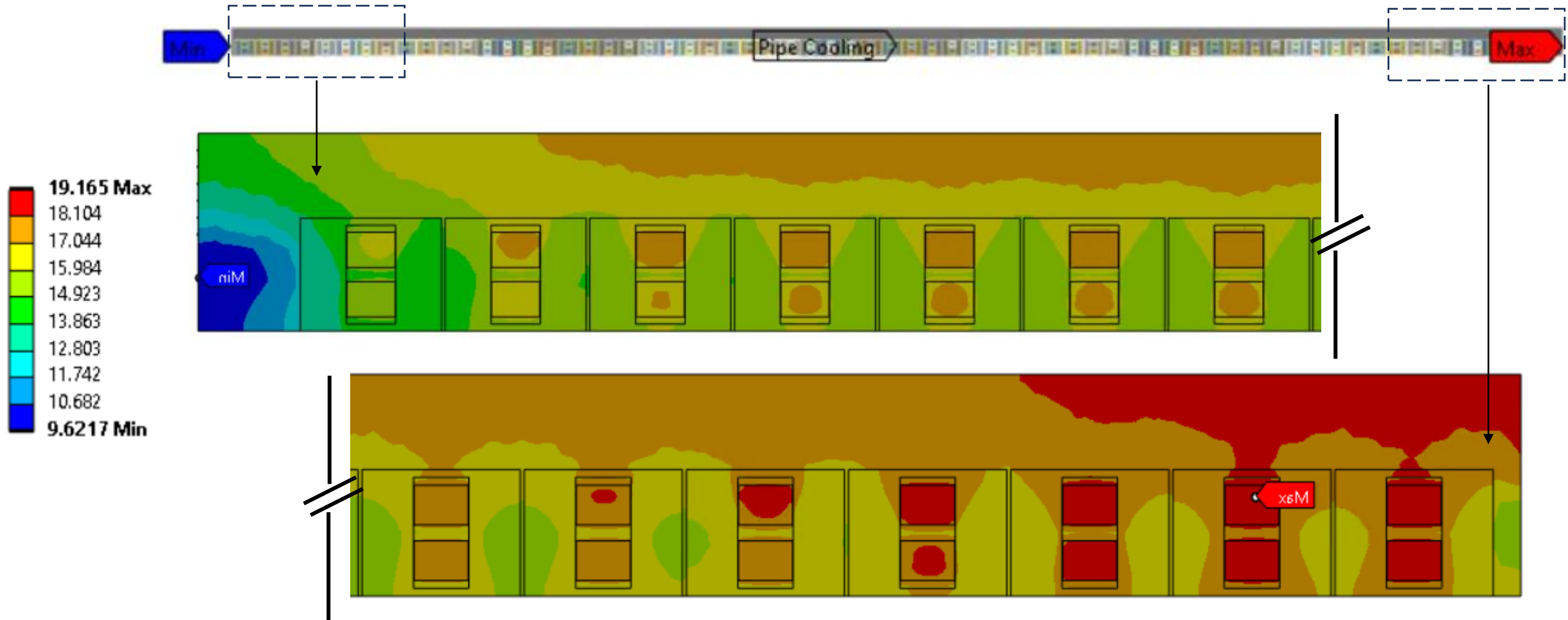
HTC Range 3364 to 2942 W/m<sup>2</sup>C





# 5 C Water Cooling (Full Stave)

HTC Range 1162 to 799 W/m<sup>2</sup>C



◊ We need experimental validations for all of these simulations.

◊ This is work in progress.

◊ Questions –

1. What is the maximum permissible  $\Delta T$  between coolant and sensor chip?
2. We need to make a critical interface plot for this stack up. What is the target maximum allowable temperature?
3. Updates on NCKU experimental work ?

Results from CMS work –  
<https://doi.org/10.48550/arXiv.2203.14347>

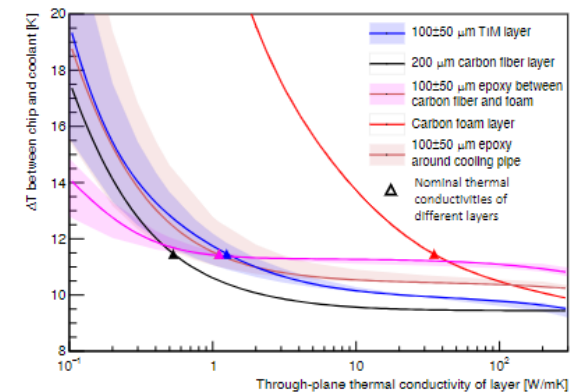
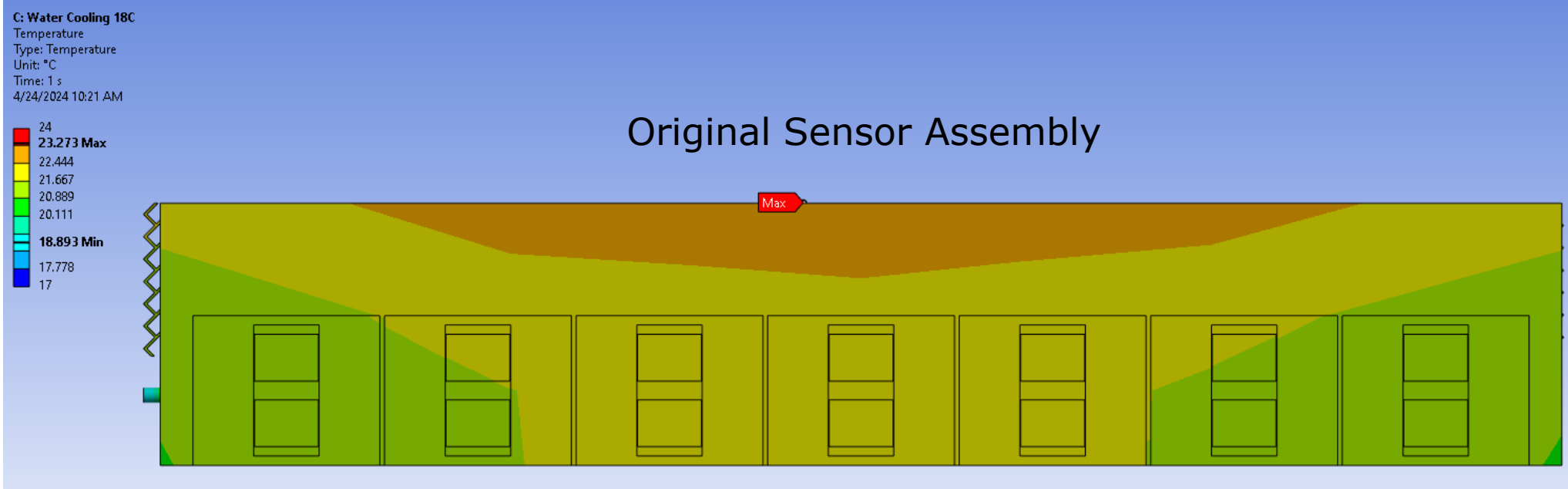
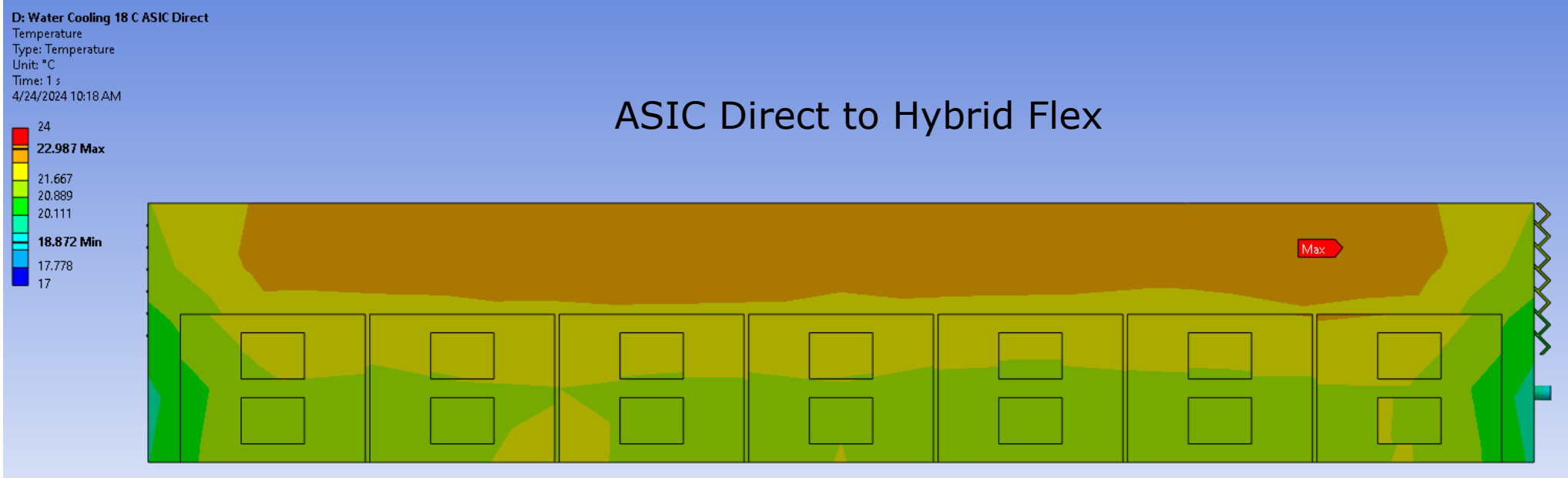
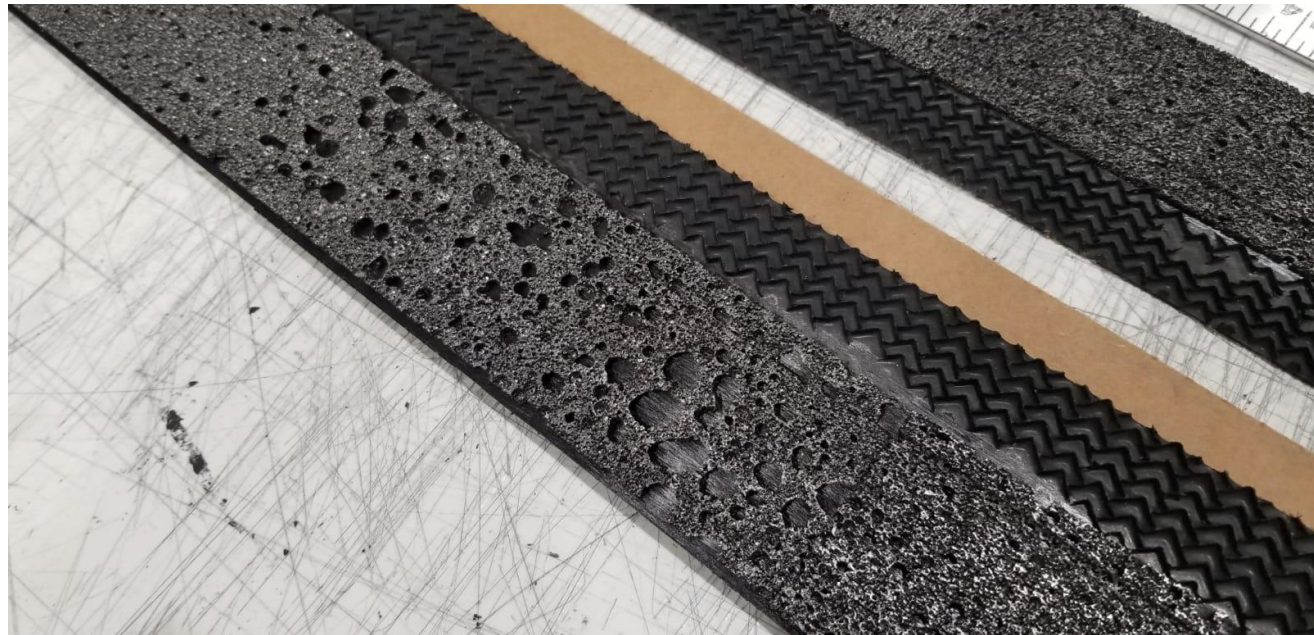
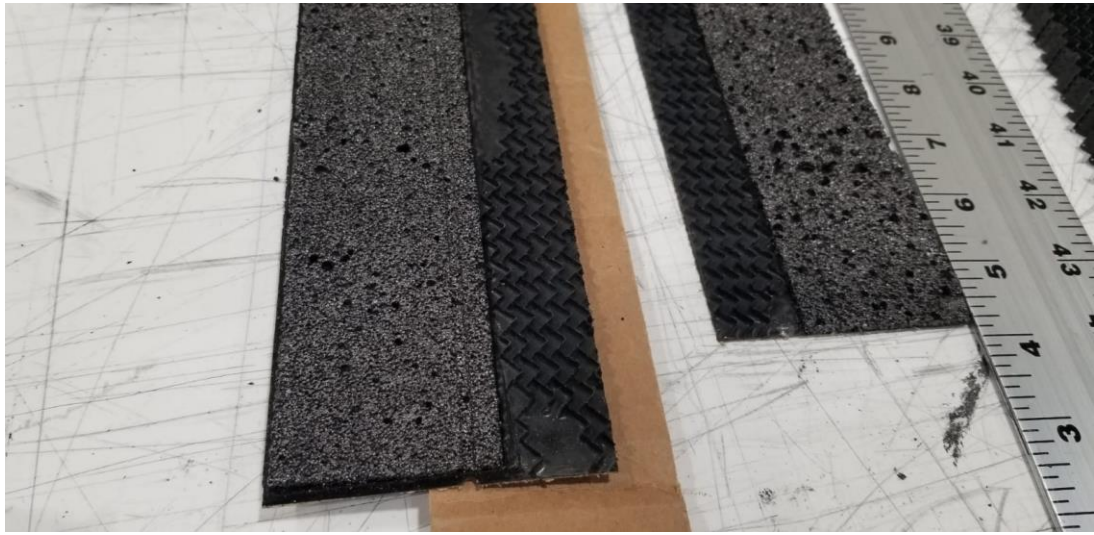
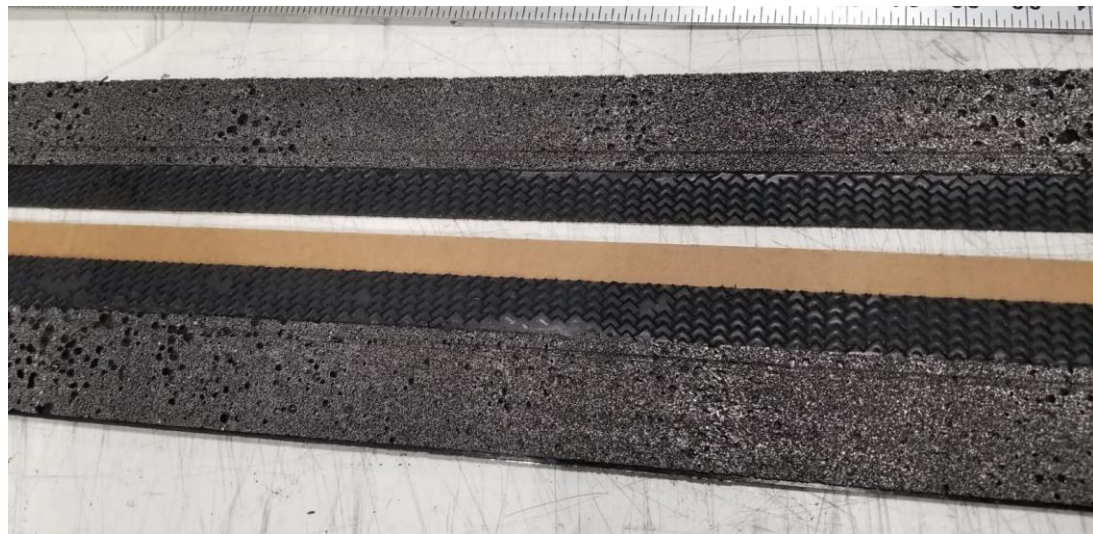


Figure 2: The temperature difference between the hottest point on the chip and the coolant as a function of the thermal conductivities of different layers of the hypothetical support structure shown in Figure 1. FEA simulation results are used to determine the functional form of the temperature dependence of the various materials. The shaded bands represent the impact of a  $\pm 50 \mu\text{m}$  thickness variation for the various epoxy interfaces. More details in the text.

When ASIC is directly on the hybrid flex there is better heat dispersion

**This was just a first attempt for miniSTAVE – same simulation for fullSTAVE in progress**





# halfSTAVE manufacturing – 1<sup>st</sup> prototype will be ready this week



- ⬢ The “cheaper” option of carbon foam we use has a lot of voids and inconsistencies – will this work for now? And / or for the AC-LGAD ToF ?
- ⬢ The gold standard foam used at ATLAS and CMS is – Lockheed Martin K9 foam. This is a super long lead time item and upto x5 to x8 more expensive than CFOAM HTC 35.
- ⬢ On-going work is more testing and design decision on what foam to use.

- ◈ The other detectors should not be affected with the choice of coolant and temperature that the AC-LGAD TOF runs at.
- ◈ Dry air ? Condensation when below dew point.
- ◈ Where is the space for this service for dry flushing? Can this go through structures? Do other detectors also need dry air flush?
- ◈ Figure out the path of the return pipe
- ◈ Implement that geometry and re-run the heat transfer simulations
- ◈ Can we do single stave or do we need to couple two staves to save on material budget ?