Forum on Tracking Detector Mechanics 2022

Challenges on the experimental validation of Finite Volume Model thermal simulation of Modules for the CMS Phase II Outer Tracker

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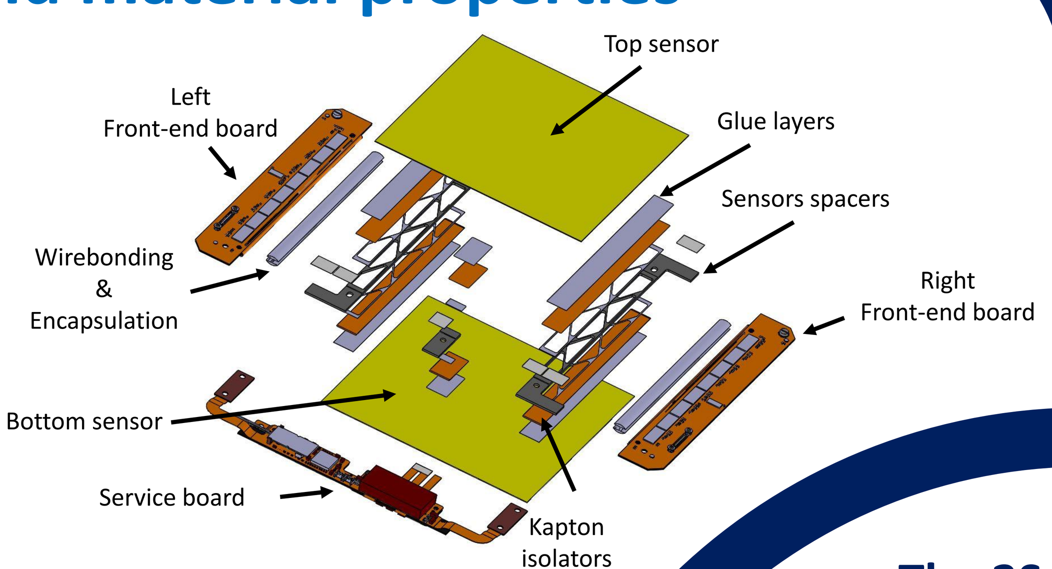
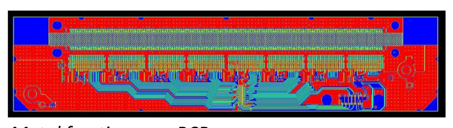
on behalf of CMS collaboration

Geometry and material properties

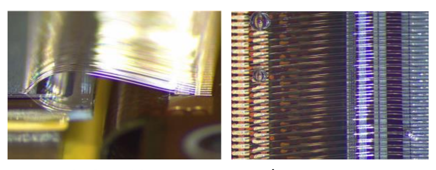
- **Geometry**
  - All the relevant components of the module are modelled.
  - A simplified shape is given to the electronic components and PCBs.
  - Non-structured grid is created for the meshing. Mean dimensions of cells in the model are 0.5 mm in-plane and 0.01 mm through plane.



- **Material allocation**
  - The majority of the material properties were taken from manufacturers' datasheets.
  - In some cases experimental measurements have been made on the thermal properties. Still some uncertainty remains for some materials.
  - PCBs and wirebonds require the introduction of equivalent properties given the impossibility of reproducing the copper traces and bonding wires by the adopted grid size.

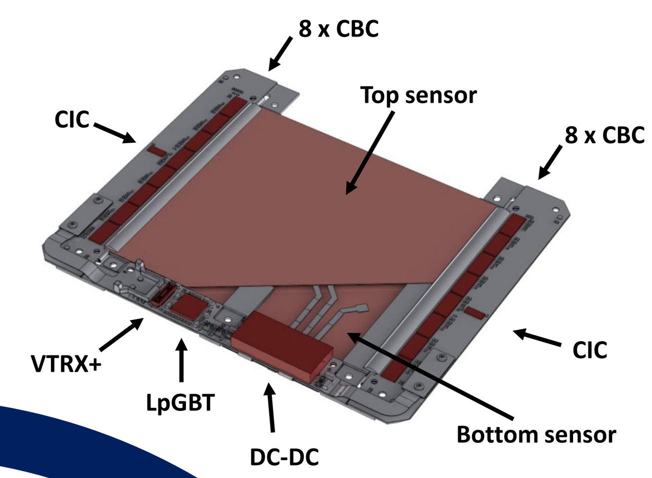


Metal fraction on a PCB



Detail of real bonding wires Ø=25μm

Heat loads



- **Low voltage (LV) dissipation:**
  - Inside active electronics components.
  - Maximum values expected are reported in the table.
  - LV loads are approximately constant with small temperature variations.
  - Loads vary with electronic activity, this has to be taken into account when permorming steady thermal measurements.

2S Module

Component	Power [W]
16 x CBC	2.160
2 x CIC	0.408
DCDC converter	2.028
lpGBT	0.358
VTRx+	0.206
Total	5.159

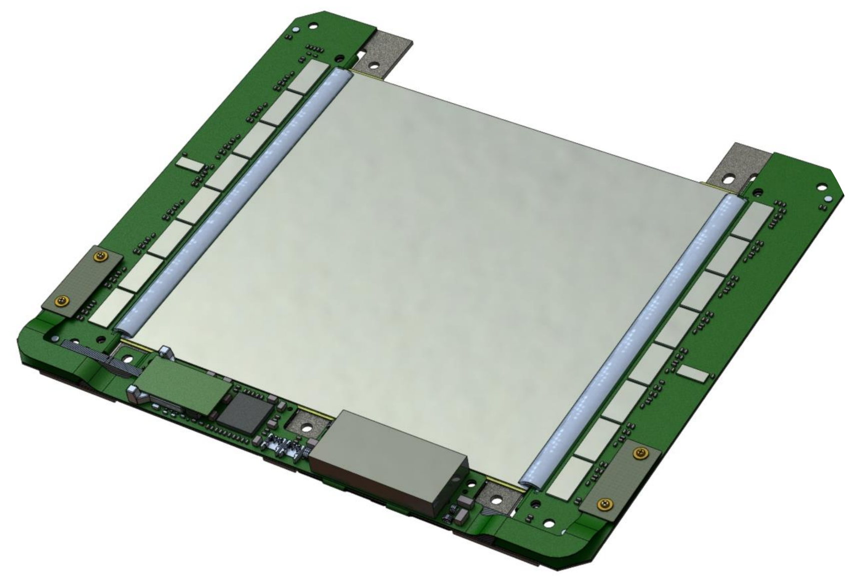
- **High voltage (HV) dissipation:**
  - Inside active volumes of sensors, due to leakage current.
  - HV loads are function of temperature and radiation damage, according to equation:

$$P_{sensor} = U_{bias}\,(\emptyset \cdot \alpha_0 \cdot V)\frac{T^2}{T_0^2}\,exp\left(-\frac{\Delta E}{2k_b}\left(\frac{1}{T}-\frac{1}{T_0}\right)\right)$$

  - HV loads also increase in presence of ambient lights. This needs to be monitored during test and eventually added in the model as additional load.

The 2S Module

A Finite Volume Model has been created to evaluate thermal performance of Modules.



Main features of the model can be collected in 4 groups: geometry, heat loads, active cooling and ambient conditions

Active cooling

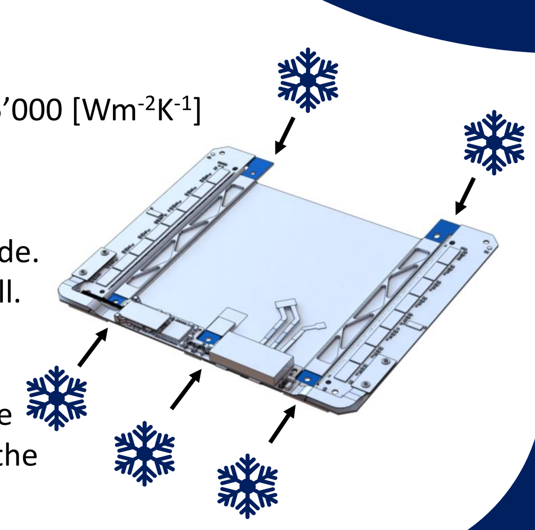
Modules are cooled through the contact with support structures corresponding to the points where there are screw joints. Support structures host the active cooling system. Different types of boundary conditions can be given depending on the properties of the cooling system.

- **In the Outer Tracker for phase-II Upgrade:**

Boiling in forced convection cooling system → Two-phase carbon dioxide flowing in 2mm pipes
This can be modeled as a convective boundary condition at the walls where the pipe is glued.
Main parameters of interest are:
Temperature of the cooling fluid (CO_2) → **T_{CO_2} [°C]**
Heat transfer coefficient resulting from the boiling activity → **HTC_{CO_2} [$Wm^{-2}K^{-1}$]**
They can be estimated through semi-empirical correlations
30% of uncertainties in calculations → worst case approach is T_{CO_2}=-33°C and HTC_{CO_2} = 5'000 [$Wm^{-2}K^{-1}$]

- **In some testing apparatuses (including the cooling box in Perugia)**

Thermoelectric cooling system → Peltier cells cooled by water-glycol fluid at the hot side.
This can be modeled as a fixed temperature condition at the cold face of the Peltier cell.
Main parameters of interest is just the temperature of the cold face of the Peltier → **$T_{PELTIER}$ [°C]**
If the Peltier is well sized and controlled by a PID system, the setpoint cold temperature can be controlled with ±0.1°C of variation, corresponding to a very low-uncertainty in the boundary conditions for the simulation.



Ambient conditions

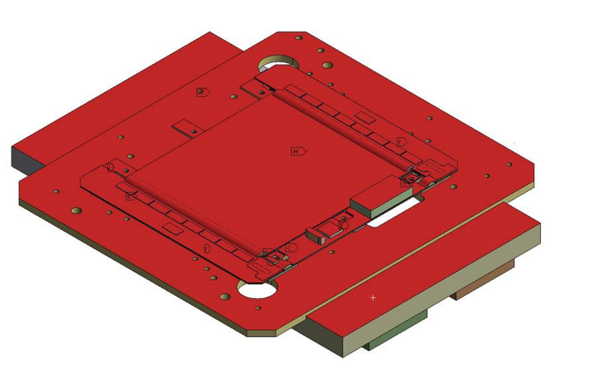
When modules are installed in the structures, dry air will be present in the surrounding space. This involves an exchange of heat with the exposed surfaces of the module (in red in the figure below). This heat exchange can be reproduced with a convective condition on the external surfaces, which takes the form of:

$$Q = h_{air} \cdot A \cdot \left(T_{air} - T_{wall}\right)$$

This requires that input values in the model should be given to the heat transfer coefficient for air (h_{air}) and the temperature of the air in the proximity of the module (T_{air}). These values depends on many factors, and can bring into the model results high uncertainties. For this reason, two approaches can be considered:

Adiabatic conditions → $h_{air} = 0$ $Wm^{-2}K^{-1}$

- Effect of convection neglected in the model
- Less uncertainties in the results related to air conditions
- Results comparable to a real case where the ambient air has the same temperature of the sensor i. e. ($T_{air} - T_{wall}$) = 0
- It gives more cautelative results until is $T_{air} < T_{sensor}$

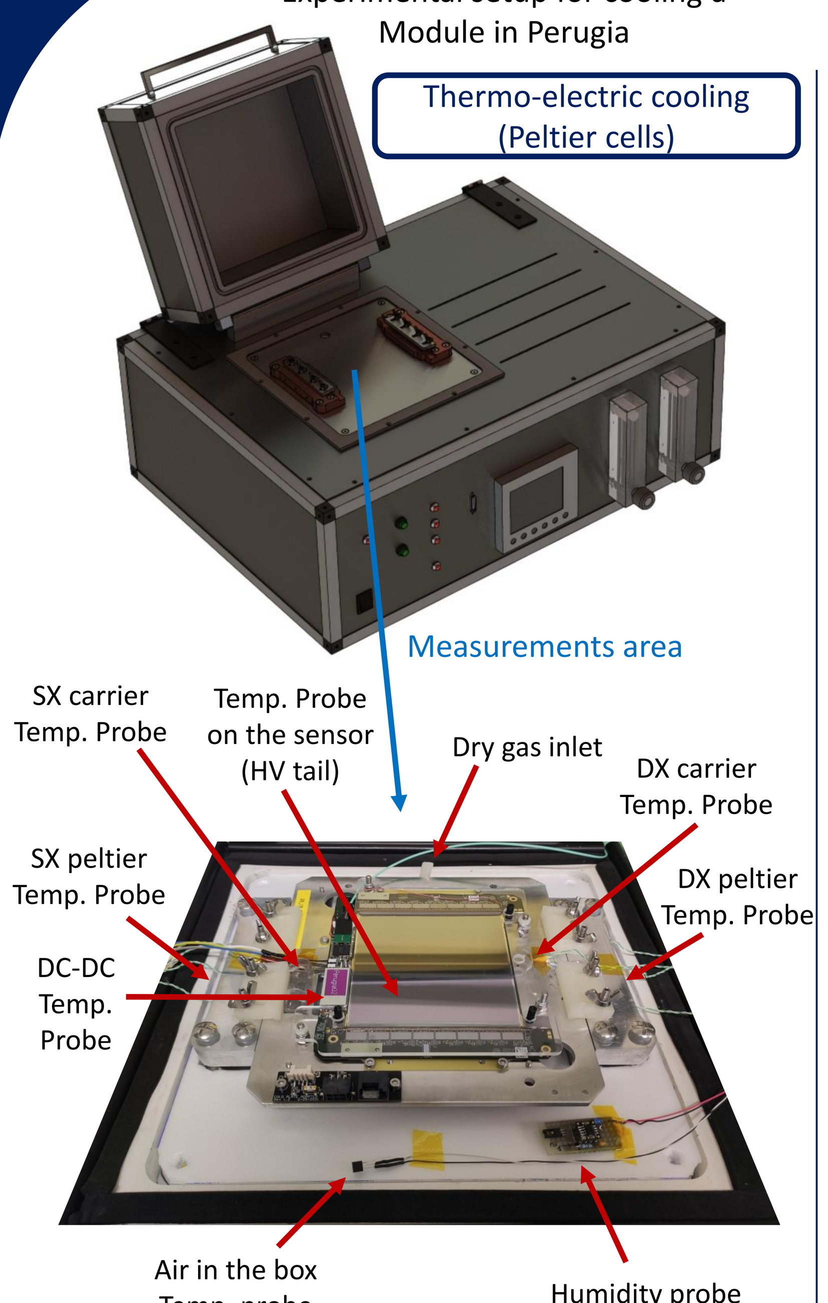


Convective conditions → $h_{air} > 0$ $Wm^{-2}K^{-1}$

- Effect of convection is considered, results closer to reality
- Difficulty on the evaluation of h_{air} and T_{air} because strogly dependent from the experimental conditions

Thermal test in the cooling box in Perugia

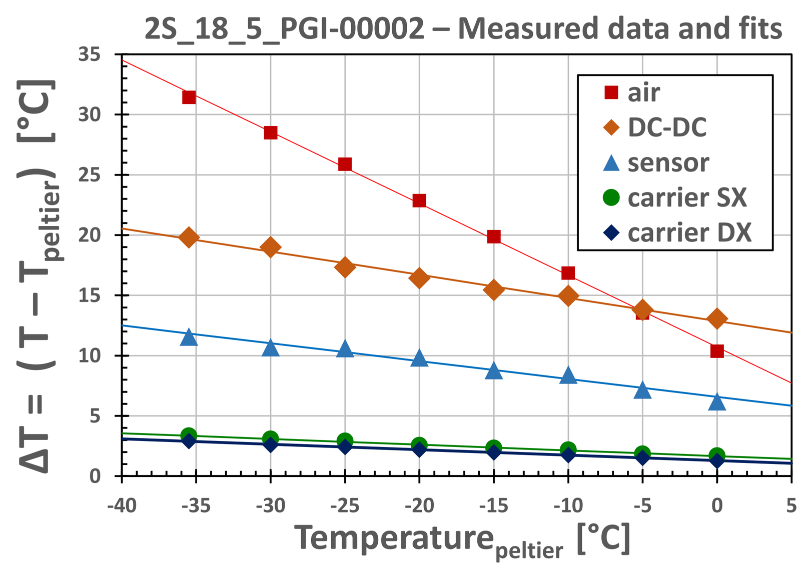
Experimental setup for cooling a Module in Perugia



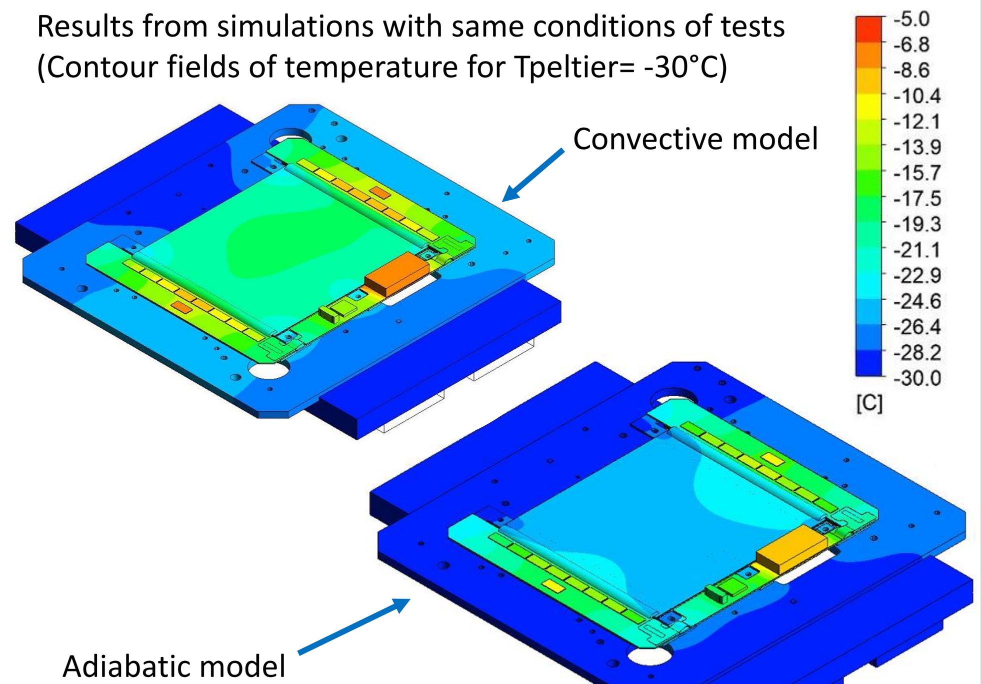
NOTE: air temperature in the box not controlled actively. Its temperature is the equilibrium between the cold surfaces and the warm dry gas at inlet (10 LPM)

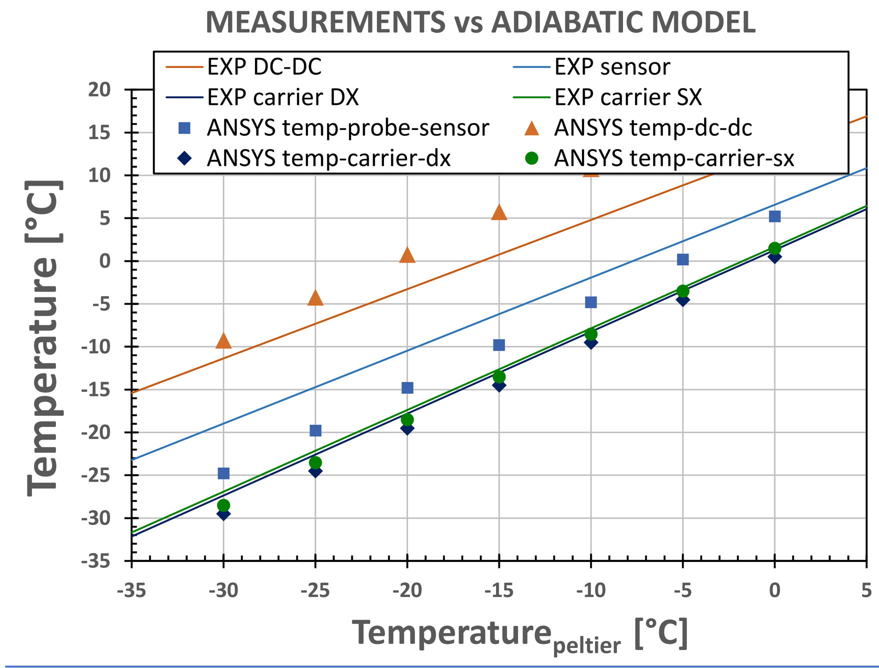
Results from measurements with following heat source:

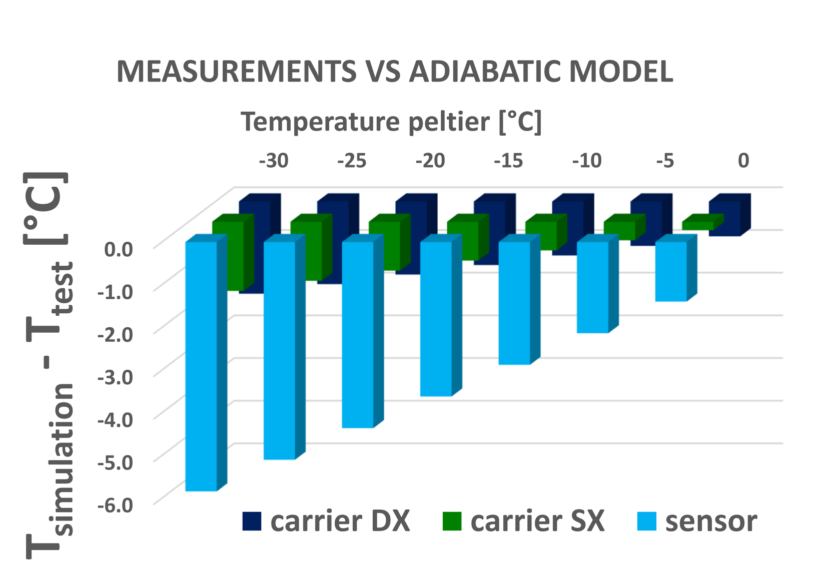
- Power HV = 0 W (un-irradiated sensors)
- Power LV = 4 W (Hybrids calibration)

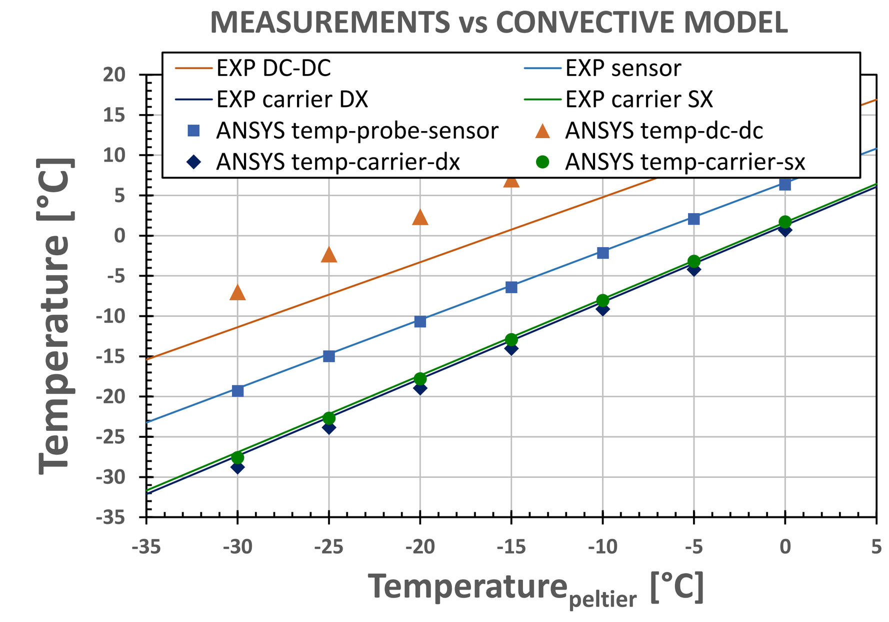


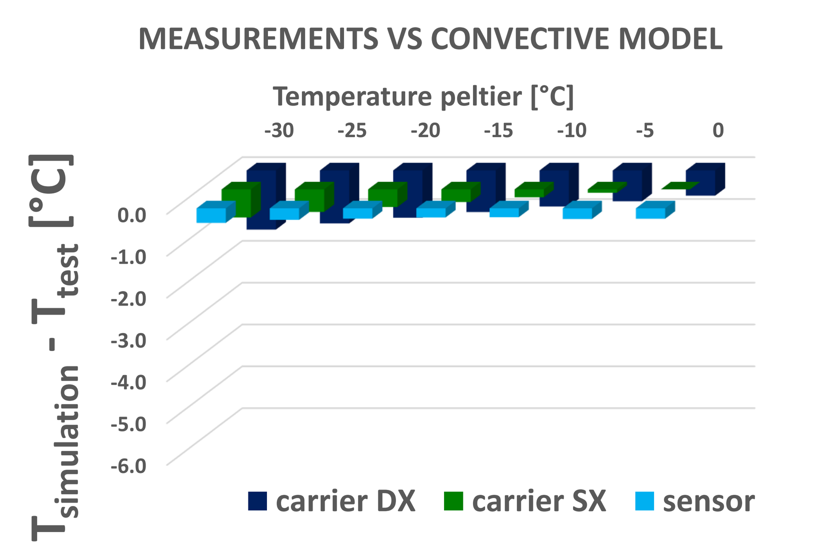
Results from simulations with same conditions of tests (Contour fields of temperature for Tpeltier= -30°C)











Temperature of the sensor well reproduced with the convective model and a value for $h_{air} = 6$ $Wm^{-2}K^{-1}$
Discrepancy between test and adiabatic model up to 6°C @ $T_{PELTIER}$ = -35°C