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Modelling CO₂ cooling of the ATLAS ITk Pixel Detector

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ABSTRACT

The Large Hadron Collider (LHC) physics program has been extended to the period 2026-2037 to deliver an order of magnitude more of proton-proton collisions compared to end 2023. To sustain the harsh conditions imposed to detectors in this period (radiation, high occupancy), the current ATLAS Inner Detector will be replaced by a new one using the most recent silicon sensor technologies. One of them is the generalization of the cooling with two-phase CO₂ flowing in titanium tubes located close to the silicon sensor and associated electronics. The choice of CO₂ relies on its most favourable thermo-physical properties in boiling state.

The radiation damage during the HL-LHC program will be quite harsh and the engineering of the local supports under such extreme conditions requires a deep understanding of the two-phase behaviour of the CO₂ flowing inside the titanium pipes, and a precise modelling of the heat transfer through the mechanical structure. However, the data available on CO₂ boiling in channels of small hydraulic diameter (say, below 3 mm) is limited and often affected by too large uncertainties. This enforces the detector designers to include large safety factors and long iterative phases of experimental measurements, which could be sensibly reduced by the availability of reliable models.

This thesis starts with a quick description of the ATLAS Upgrade Phase II project, a first outlook on the fundamentals of two-phase cooling technology, widely used to cool particle detectors in high-energy physics (HEP) experiments and discussion about the thermal management requirements of the future ATLAS Pixel Detector. Several custom tools were developed in python language to guide the design and optimisation of the CO₂ cooling system: simulation of heat transfer coefficient (HTC) and frictional pressure drop along evaporators, flow distribution studies in manifolds and calculation of thermal requirements.

In a second phase, the document presents the thermal test setup to measure thermal performances of thermal prototypes for the new detector (ITk) as well as the associated simulation based on Finite Element analysis and Heat Transfer Coefficient modelling. A novel fit method was implemented to extract from the measurements the different parameters of the ITk elements (in situ materials thermal conductivity, manufacturing variability, HTC...). The results are compared to the current CO₂ model predictions and the discrepancies discussed. A new set of parameters for the CO₂ model was defined to improve precision by factor two on the effective HTC values for the ITk working conditions and integrated in the python modelling tool. Finally, the impact of this work on the design of the Pixel detector is discussed.

RÉSUMÉ

Le programme scientifique du Grand Collisionneur de Hadrons (LHC) du CERN a été étendu à la période 2026-2037 pour augmenter le nombre total de collisions de paires de protons d'un ordre de grandeur. Pour que les détecteurs fonctionnent encore dans les conditions extrêmes imposées, le détecteur interne d'ATLAS va être remplacé par un nouveau (ITk) qui utilisera les dernières technologies de détecteur silicium. Une d'elles sera le refroidissement CO_2 diphasique circulant dans des tubes en titane liés aux détecteurs silicium par une structure porteuse. Le choix du CO_2 provient de ses très bonnes propriétés thermiques. Les effets des radiations sur les détecteurs seront importants. La conception de la structure porteuse dans ces conditions nécessite une bonne compréhension du comportement du CO_2 diphasique circulant dans des tubes titanes ainsi qu'une modélisation précise des échanges de chaleur à travers cette structure. Néanmoins, les données disponibles sur le comportement du CO_2 dans des tubes de diamètre inférieur à 3 mm restent limitées et sont souvent associées à de larges incertitudes. Cela impose aux concepteurs d'inclure de grands facteurs de sécurité et de valider les performances sur plusieurs versions de design. Des modèles du CO_2 plus précis éviteraient ces tâtonnements. Le document de thèse commence par une rapide description du programme d'upgrade du détecteur ATLAS et une revue des fondamentaux du refroidissement diphasique qui est largement utilisé dans les détecteurs de Physique des Particules puis une discussion sur les contraintes thermiques spécifiques au détecteur Pixel d'ATLAS. Plusieurs outils ont été développés en langage python pour guider la conception et l'optimisation d'un tel système de refroidissement : simulation du coefficient de transfert de chaleur (HTC) et perte de pression le long des évaporateurs et calcul des besoins thermiques. Dans la deuxième partie, le manuscrit présente le banc qui a servi à mesurer les performances thermiques de prototypes avec leurs simulations utilisant la méthode des Éléments Finis (FEA) et une modélisation des coefficients de transfert de chaleur (HTC). Une méthode originale d'ajustement a été utilisée pour transformer les résultats des mesures thermiques en mesure des paramètres de base des éléments du détecteur ITk (conductivité thermique des composants, variabilité de la production, HTC,...). Les résultats sont comparés aux modèles CO_2 actuels et les incompatibilités sont discutées. Un nouveau lot de paramètres pour le modèle CO_2 a été calculé permettant de réduire d'un facteur deux les incertitudes sur les valeurs de HTC dans les conditions de fonctionnement de l'ITk. Le document se conclut avec une présentation de l'impact de ces résultats sur la conception du détecteur Pixel.

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ATLAS PHASE-II UPGRADE

“Europe’s top priority should be the exploitation of the full potential of the LHC, including the high-luminosity upgrade of the machine and detectors with a view to collecting ten times more data than in the initial design.”

EUROPEAN STRATEGY FOR PARTICLE PHYSICS

MARCH 2013, CERN

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1.1 LHC programme at CERN

The European Center for Nuclear Research (CERN) located on the French/Swiss border, is the major research institution for Particle Physics in Europe. Since its creation in 1952, several accelerators and colliders have been built and continuously upgraded to study particles and their fundamental interactions. The circular Large Hadron Collider (LHC) is the current last element of a four-stage particle acceleration system and the largest and most energetic collider in operation worldwide. It accelerates two proton beams synchronously in opposite directions. They collide at four different locations (interaction points) with an energy in the center-of-mass up to 14 TeV. Four large particle detectors (ATLAS, CMS, ALICE and LHCb) are installed at each of the interaction points as illustrated in Figure 1.1.

Such high energy proton-proton collisions might produce fundamental heavy particles like the recently discovered Higgs boson. This discovery completed the list of particles predicted by the Standard Model. However, there are still many other fundamental questions not yet answered which LHC will try to address. The LHC baseline was designed for a maximum nominal center-of-mass energy of 14 TeV, an instantaneous luminosity peak of $1 \times 10^{34} \text{ cm}^2/\text{s}$

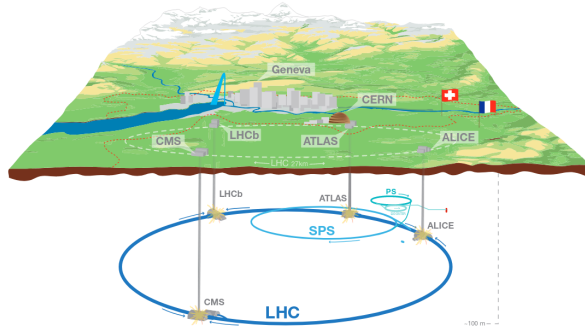


Figure 1.1: Location at the French/Swiss border of the LHC and its four experiments: ALICE, LHCb, ATLAS and CMS.

and total integrated luminosity of 40 fb^{-1} per year. As the baseline LHC design included a margin allowing it to reach two times the nominal design performances, the peak luminosity has been pushed to $2 \times 10^{34} \text{ cm}^2/\text{s}$. After 2026, with a nominal instantaneous peak luminosity of $2 \times 10^{34} \text{ cm}^2/\text{s}$, the amount of data acquired in a 10 year long period would only reduce the statistical error on the measurements by 40%. Therefore the LHC upgrade (HL-LHC project) plans to increase its nominal instantaneous peak luminosity by a factor of 2-4 to increase the luminosity by a factor 10. The programme shown in Figure 1.2, foresees a 2.5 year long shutdown (LS3) starting 2024, to upgrade the LHC for the High-Luminosity phase.



Figure 1.2: Schedule for the HL-LHC programme.

With a peak luminosity expected to reach from five to seven times the nominal instantaneous luminosity, the ATLAS and CMS detectors foresee also the upgrade of their subsystems (already started during the second long shutdown) to be prepared for a 10 year long operation (Phase II) with ten times more collisions than during the Run-3 campaign and integrated luminosity up to 3000 fb^{-1} . The price to pay for such high instantaneous luminosity is the increase of the number of pile-up events ¹.

¹Proton-proton collisions of no interest and thus considered as background. The likelihood of these processes occurring scales linearly with the luminosity.

1.2 ATLAS Detector

1.2.1 Overview

The current ATLAS Detector was officially proposed by the ATLAS Collaboration in 1992 [1] as a general purpose proton-proton experiment to be fully operational by the LHC startup in 2008.

One of the main goals of the experiment was to discover the Higgs boson that was only possible thanks to LHC's high energy proton-proton collisions. In 2012, the ATLAS and CMS Collaborations discovered the Higgs boson, with a measured mass of 126.0 ± 0.4 (stat) ± 0.4 (sys) GeV [2], for which there was already a strong sign of existence but no experimental evidence.

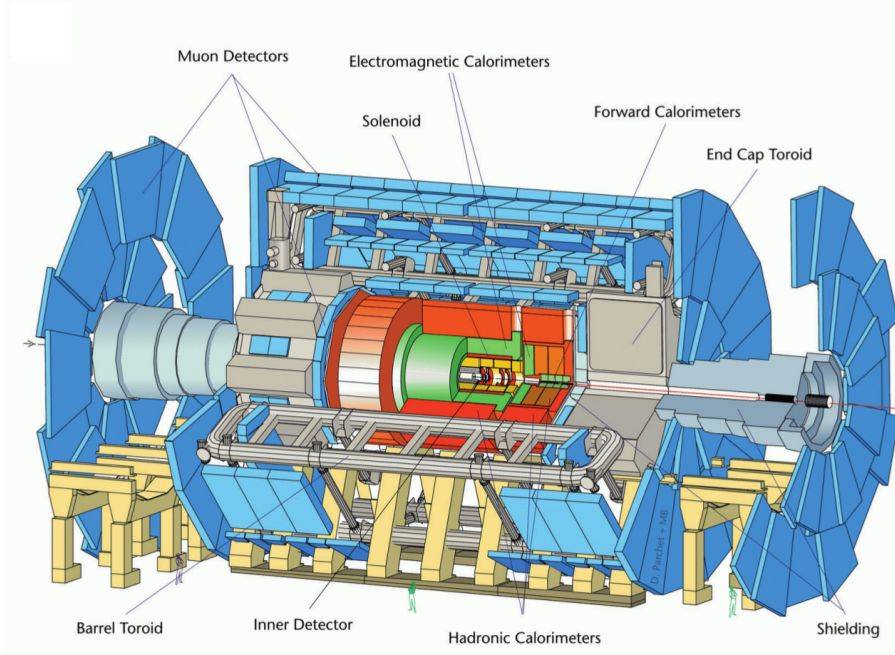


Figure 1.3: *ATLAS Detector and its subsystems.*

To observe the particles produced from the proton-proton collisions, the detector is divided in three main subsystems: muon spectrometer, calorimeters and the inner tracker as shown in Figure 1.3. Each subsystem is split into a barrel and two end end-caps to fully enclose other subsystems at its interior, as illustrated in Figure 1.4.

The muon subsystem is the outermost detector and comprises four kind of chambers: two (barrel and end-cap) working as trigger systems and other two to measure precisely the muon momenta. The calorimeters are used to identify particles and measure their direction and energy. The innermost calorimeter is the liquid argon electromagnetic calorimeter responsible for measuring the energy of electrons and photons. Surrounding it, is the hadronic calorimeter which complements the electromagnetic one to reconstruct the properties of particle jets. Finally, the innermost subsystem is the inner tracker (ID for Inner Detector) responsible for measuring the charged particle momenta and reconstruct their interaction vertices.

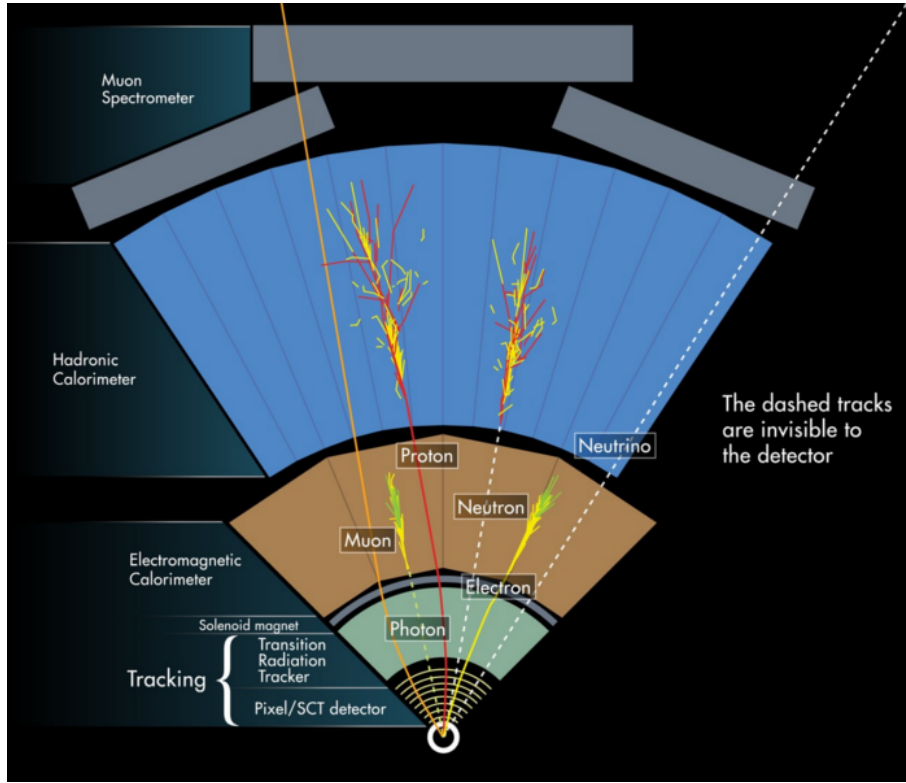


Figure 1.4: Scheme illustrating the sub-detectors arrangement in the ATLAS Detector with representation of different particle tracks and how they interact with each detector.

1.2.2 Inner Detector

The ATLAS Inner Detector (ID) is the closest sub-system to the beam pipe. At the LHC startup in 2008 it was composed of three independent sub-detectors: Pixel Detector, Semi Conductor Tracker (SCT) and the Transition Radiation Tracker (TRT). In 2014 the Insertable B-Layer (IBL) was installed. The IBL will be discussed in section 1.2.3. The four sub-detectors comprising the current Inner Detector are shown in Figure 1.5.

To locate different elements within its cylindrical geometry, ATLAS uses a right-handed coordinate system with origin at the nominal interaction point, at the center of the detector. The z axis runs along the beam. The x -axis points to the center of the LHC and the y -axis points upwards. Each element can be located with cylindrical coordinates (r, ϕ) for the x - y (transverse) plan, and the polar angle θ . The angle θ is often converted in pseudorapidity units η :

$$\eta = -\ln \tan \theta/2 \quad (1.1)$$

With a total length of 6.2 m and a diameter 2.1 m, the Inner Detector covers a range in $|\eta| < 2.5$, as illustrated in Figure 1.6.

The outermost detector is the TRT, a lightweight detector composed of gaseous proportional counters (straws filled with mix of Argon, CO_2 and Xenon) that provide a large number of tracking points with much less material. It provides around 30 hits per track, equivalent to a single point of $50 \mu\text{m}$ of precision, contributing to the pattern recognition and particle identification.

Both Pixel and SCT detectors rely on silicon sensors to detect charged particles with fine granularity. The Pixel Detector counts on pixels of $400 \mu\text{m} \times 50 \mu\text{m}$ and its proximity to the beam pipe to reconstruct precisely the position of the interaction vertex ² and thus, distinguish events of interest from events of pile-up and provide good flavour tagging.

²Position where it took place the collision between protons responsible for producing the particle identified.

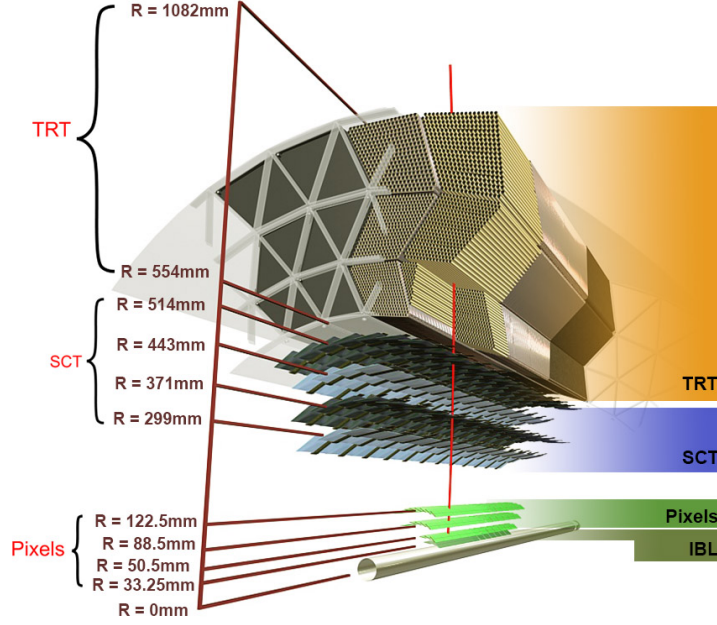


Figure 1.5: Sub-detectors comprising the ATLAS Inner Detector after installation of IBL.

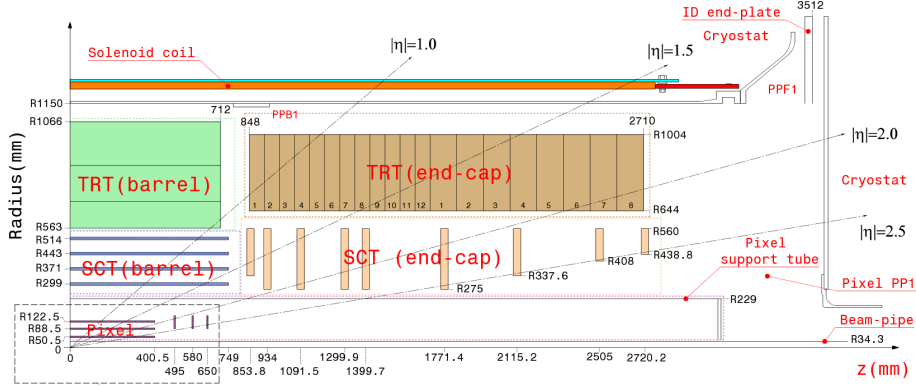


Figure 1.6: A cut view of the ATLAS inner detectors with the collision point is located in the left-bottom corner of the figure. Each detector is separated in barrel and the end-cap parts covering $|\eta|$ up to 2.5 [3].

To cope with the tracking performance requirements, the Inner Detector is designed to provide at least eight hits per track on the silicon and micro-strip layers. The number of layers are limited due to the high cost of silicon sensors and amount of material introduced. The original Pixel detector comprised three layers in the barrel and end-cap while the SCT has four layers in the barrel and ten layers in the end-caps. Both detectors are cooled using two-phase C_3F_8 cooling system. The innermost (and newest) layer in the Pixel detector is called B-layer because of its proximity to the beam pipe, determinant for good flavour tagging. Moreover, its mechanical structure had to be designed separately from the remaining two layers, in order to be replaceable if necessary due to possible radiation damage lifetime limitations.

1.2.3 Inner B-Layer

To improve the flavour tagging, a fourth layer called the Insertable B-Layer was installed during the first long shutdown (2013-2014, ATLAS Upgrade Phase 0), between the B-layer and the beam-pipe as shown in Figure 1.7. It is ≈ 66 cm long and composed of fourteen carbon composite staves around the beam pipe, providing a coverage up to $|\eta|$ of 3 [3].

It was built with the most recent pixel technologies. For example, thanks to the new pixel

sensor technologies (planar and 3D) with smaller pixel sizes of $250\,\mu\text{m} \times 50\,\mu\text{m}$ and faster read-out chips, the IBL has been able to sustain the high radiation dose and high particle occupancy induced by its proximity to the beam pipe. Also, a new cooling system based on two-phase CO_2 was introduced. These new technologies were already considered as the best candidates for the future ITk. The IBL served as well as a test bench to evaluate them in a smaller scale detector.

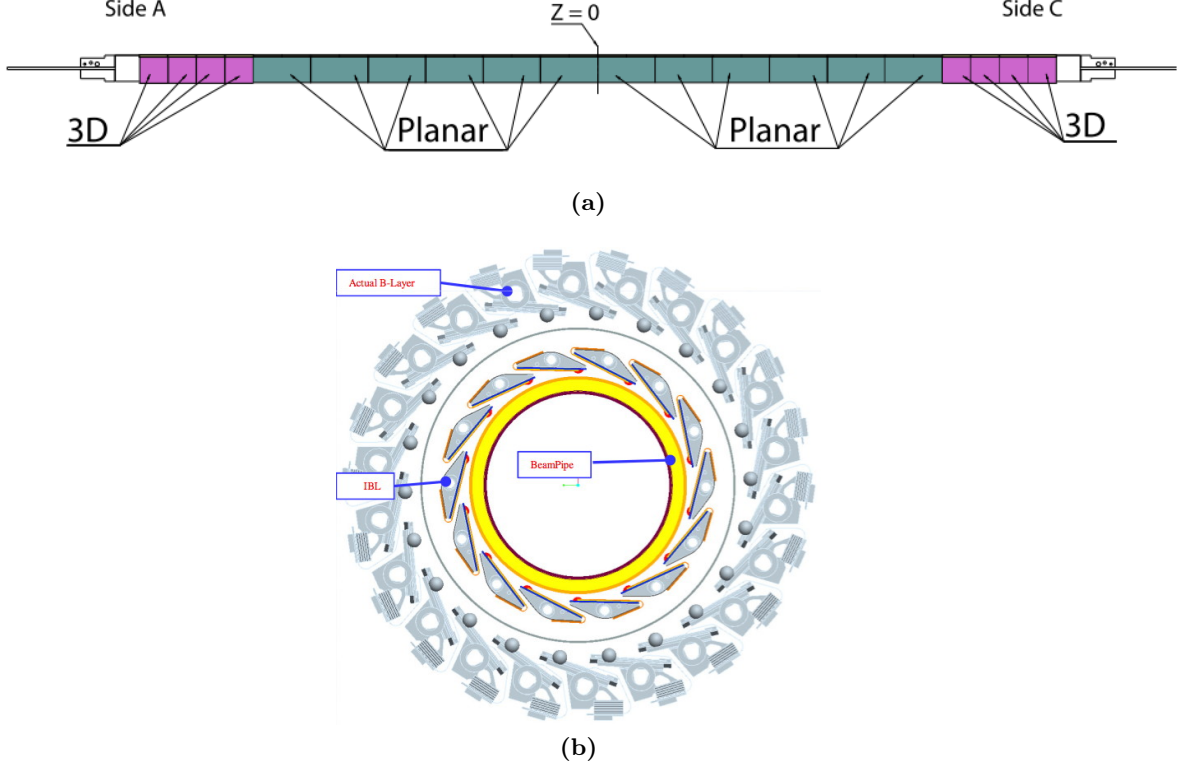


Figure 1.7: (a) Single stave of IBL with planar silicon pixel sensor at the central region and 3D sensors at the ends. (b) Location of the IBL between the B-Layer and beam pipe [3, 4].

1.3 ATLAS Upgrade Phase II: Inner Tracker

The LHC scientific programme will finish with the start of the High Luminosity campaign. Its aim is to start in 2026 and run for a decade, delivering an integrated luminosity of $3000\,\text{fb}^{-1}$ for proton-proton collisions at a center-of-mass energy of 14 TeV.

To reach $3000\,\text{fb}^{-1}$ in such a short period, the instantaneous luminosity will be increased up to $7.5 \times 10^{34}\,\text{cm}^2/\text{s}$. This will represent an average of 200 proton-proton collisions per bunch crossing, increasing the number of charged particles hitting the detector per unit area. In addition, the Inner Tracker will be exposed to ionizing radiation dose up to 10 MGy and fluence up to $10^{16}\,\text{n}_{\text{eq}}/\text{cm}^2$.

Finally, the tracker will extend to an $|\eta|$ value up to 4: one of the main targets, already detailed in the ATLAS Phase-II Upgrade Scoping Document [5], is to improve the significance of vector-boson fusion and vector-boson scattering channels by increasing the forward jet acceptance while minimising the pollution by jets originating from pile-up events. This will maximise the expected precision on the measurements of vector-boson fusion and vector-boson scattering production.

The expected radiation dose imposes the replacement of the current Inner Detector during the next LHC shutdown (2024-mid 2026). To fulfill the challenging running conditions, the new Inner Tracker (ITk), which will be a all-silicon tracking detector and will maximise the benefits from recently developed technologies.

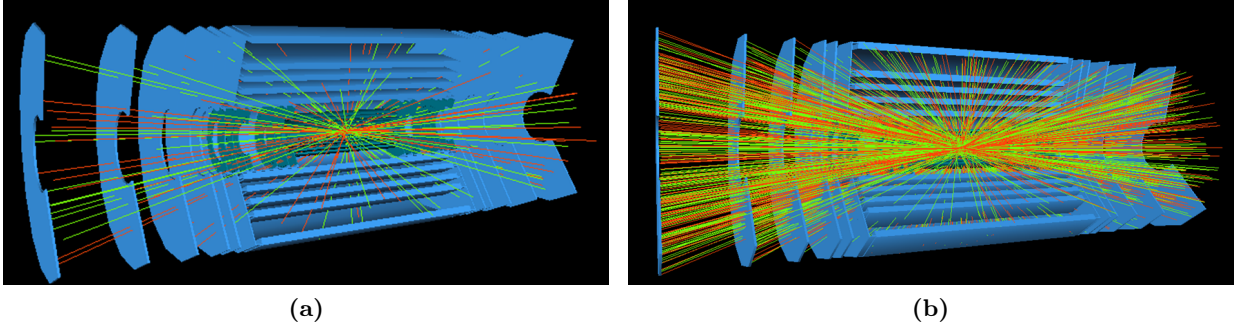


Figure 1.8: (a) Simulation of the ATLAS ITk traversed by a 23 pile-up events and (b) 230 pile-up events [4]

The future ITk will contain a silicon micro-Strip Detector surrounding a Pixel Detector. The main goal is to achieve similar or better tracking performance with respect to the current tracker but in much harsher conditions.

Given the high number of pile-up events, the density of tracks gets especially high close to the interaction point, where the particle tracks may even superpose - see Figure 1.8. Farther from the beam pipe, the high granularity of the sensors is less critical hence, strip sensor technologies with pitch sizes down to $75\text{ }\mu\text{m}$ are planned to be used in the Strip detector.

To distinguish tracks in this highly dense area, the Pixel Detector relies on small pixels with a pitch of $50\text{ }\mu\text{m} \times 50\text{ }\mu\text{m}$ (or $25\text{ }\mu\text{m} \times 100\text{ }\mu\text{m}$) with the silicon bulk thinned down to $100\text{ }\mu\text{m}$ (to minimise particle scattering) and fast read-out chips. It is designed with 3D pixel sensors on its innermost layer (layer 0) and planar pixel sensors in the remaining layers. The 3D pixel technology is used only in the first layer thanks to its radiation hardness. The high cost of the 3D sensors prevents their installation in the other layers. The radiation hardness and difference between planar and 3D sensors are discussed in Chapter 5.

1.3.1 Radiation environment studies

The simulations of radiation background for ITk are performed using the FLUKA particle transport code [6] and PYTHIA8 for event generation [7]. Figure 1.9a shows the results of FLUKA simulations of the 1 MeV neutron equivalent damage in silicon for the whole ITk and Pixel Detector in particular.

These estimates are particularly important to foresee the magnitude and type of radiation on the silicon sensors at each region of the tracker. The impact on the operation of the sensors and their thermal management is discussed in Chapter 5.

In the Pixel and Strip Detector Technical Design Reports [8, 9] the expected fluences are usually calculated from the average along z for each layer with a safety factor of 1.5. This safety factor is defined to cope not only with eventual mismatch between the simulation and measurements but also, the increase of the total radiation damage along z as shown in Figure 1.9b.

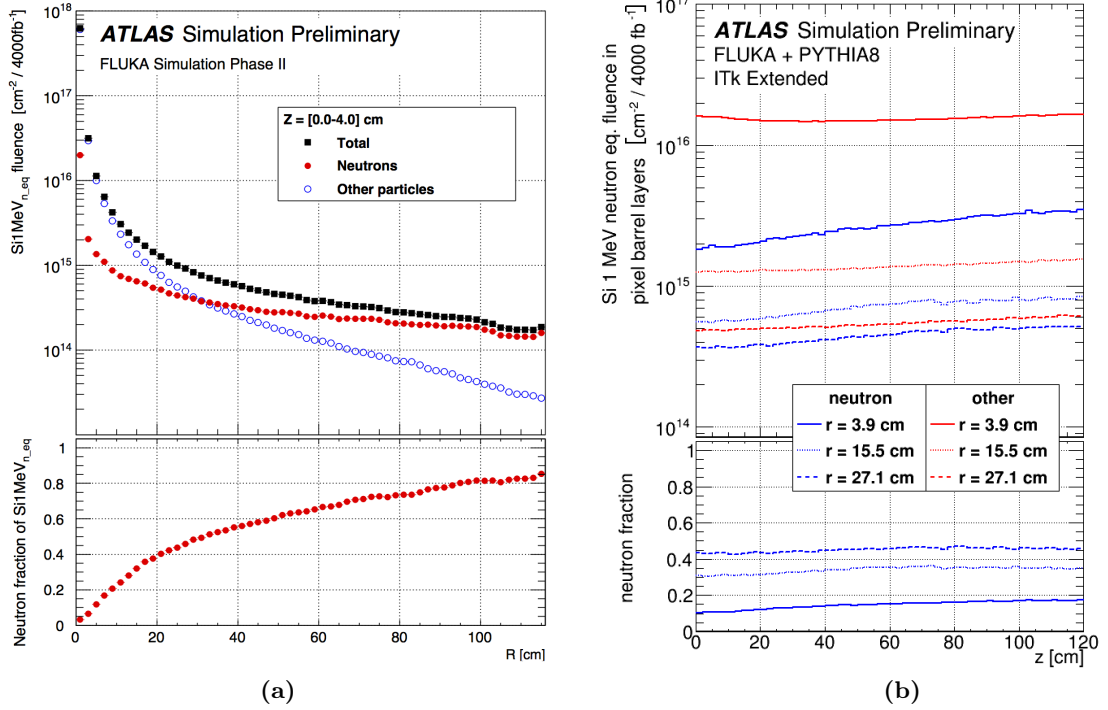


Figure 1.9: (a) Silicon 1 MeV equivalent fluence as a function of radius at the center of the ITk, subdivided into the component from neutrons and other particles, with values averaged for z between 0 cm and 4 cm. (b) Result from simulation for the Pixel detector as a function of z , in various layers.

1.3.2 Pixel Detector: inclined layout

After the removal of the TRT the Pixel Detector is extended from three to five layers. The main initial difficulty arising from a hit per layer (5 precision hits) requirement [8] up to $|\eta|$ of 4 is the cost. Covering a radius up to 350 mm from the beam pipe and a detector length up to 3 meters with a classical system of five cylindrical barrel layers and five end-cap disk layers would require to cover a prohibitive large surface of 20 m^2 with high precision silicon modules (pitches of $50 \mu\text{m} \times 50 \mu\text{m}$). To overcome this limitation, an innovative design was proposed in 2012 [10], where the pixel modules are progressively inclined along the z -axis to increase the solid angle covered by each module.

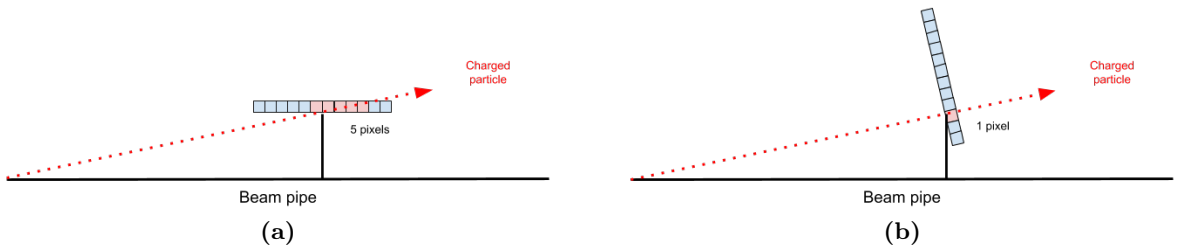


Figure 1.10: Description of particle crossing a flat and inclined sensor at large η .

For the end-caps, an arrangement of pixel modules inside rings instead of disks is considered. In the barrel, the pixel sensors are placed horizontally in the central part and inclined at the ends. Figure 1.10 shows that the inclined geometry reduces the amount of silicon crossed by the particle and the size of pixel clusters. Having fewer pixels needed to reconstruct a track not only reduces the amount of silicon but also the non instrumented material inside the detector

volume, needed to support, cool and read-out the pixel front-end electronics. The conclusion after comparing the performance of the inclined geometry to the baseline geometry showed that the total surface of silicon is reduced to 12 m^2 and that the tracking performance is improved [11]. Therefore, the ATLAS collaboration decided to consider the Inclined layout as the baseline for the future ITk Pixel Detector.

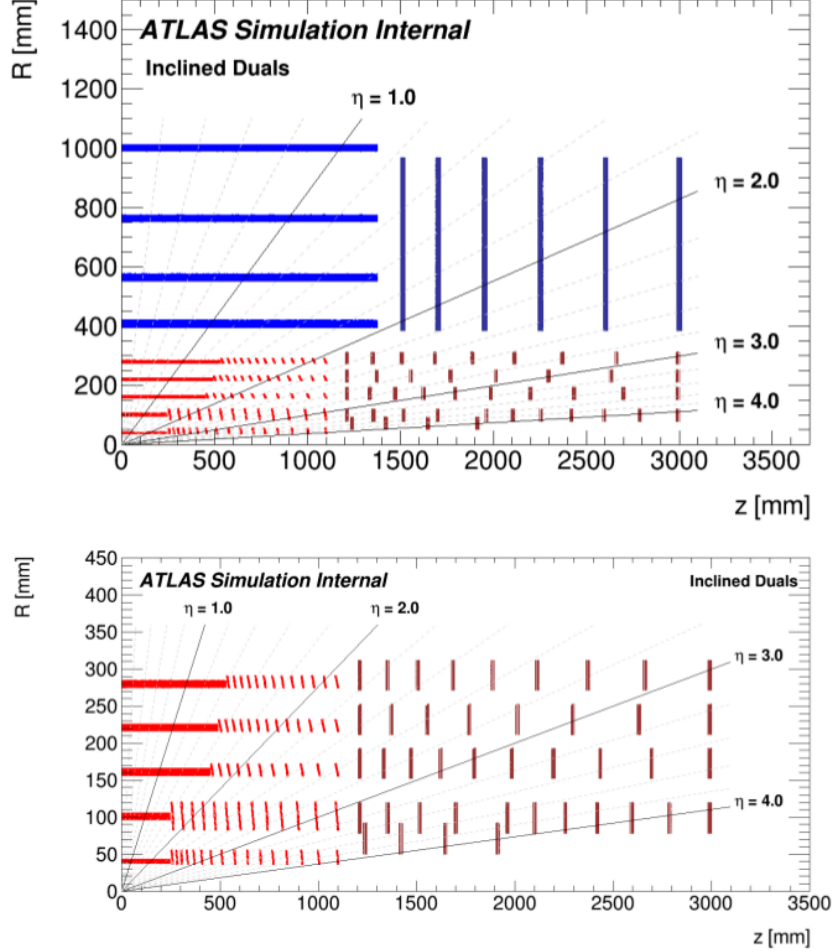


Figure 1.11: *Top: A schematic layout of the ITk Inclined Duals layout for the HL-LHC phase as presented in the Technical Design Report [8]. Bottom: The zoom onto the Pixel Detector. Only one quadrant of active detector elements are shown for both diagrams. The horizontal axis is the axis along the beam line with zero being the interaction point. The vertical axis is the radius measured from the interaction region.*

A schematic view of the Inclined Layout [8] is shown in Figure 1.11. In this baseline layout, the five pixel layers in the barrel are surrounded by the Strips detector, and the pixel detector itself is split into three subsystems: the two Innermost Barrel layers, designed to be replaceable after 2000 fb^{-1} of integrated luminosity, the three Outer Barrel layers (OB) and the end-caps rings (EC). In the following, we will focus on the Outer Barrel sub system, the most challenging one with respect to the thermal performance - see chapter 5.

1.3.3 Pixel Outer Barrel: local support

At the beginning of the ITk project, the Pixel Outer Barrel consisted of a phi-arrangement of long mechanical staves populated with two flavors of pixel modules³: 4 cm x 4 cm pixel modules with four readout chips (quad-modules) and 4 cm x 2 cm pixel modules with two readout chips (dual modules). Quad modules were used in the central part of the detector and dual inclined modules at larger pseudo-rapidities [8, p.7] The support staves are used for the mechanical stability of the detector, to carry the electrical services and to hold the titanium cooling pipes needed to evacuate the heat generated by the pixel modules.

Initially, two mechanical solutions were proposed as candidates for the inclined OB local support: the Alpine stave (Figure 1.12) and the SLIM longeron (Figure 1.13). Both concepts are based on rigid carbon structures (carbon fiber truss for SLIM, coated carbon foam for the Alpine stave), but use different thermal interface materials to transfer the modules heat load to the cooling pipes through the mechanical structure.

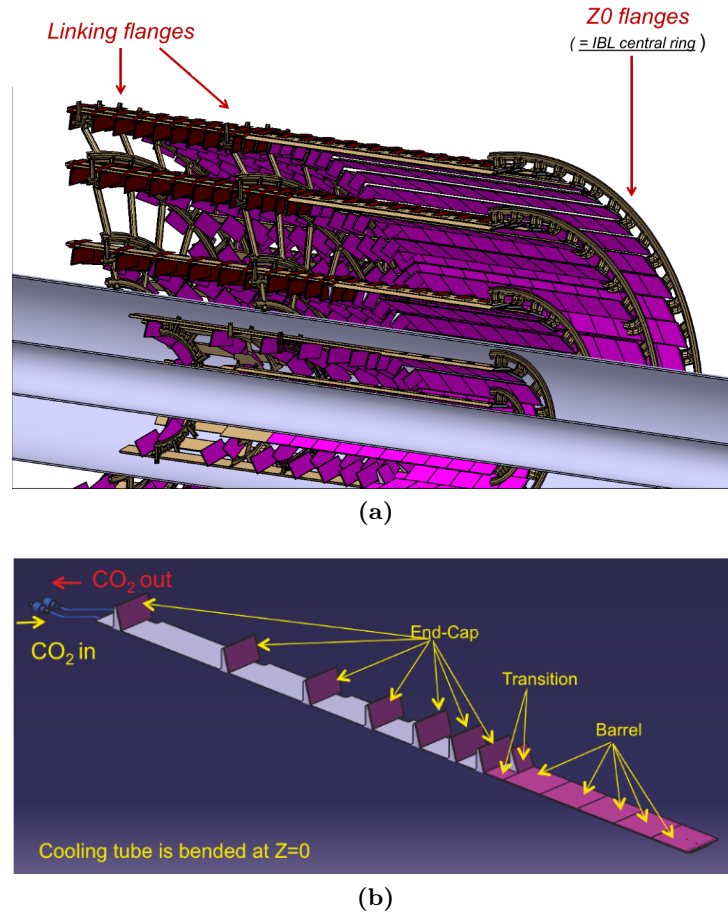


Figure 1.12: (a) Section of the Alpine local support showing five layers of the Pixel Detector with staves mounted on top of the linking and Z0 flanges. (b) Alpine stave with flat quad pixel sensor foreseen for the barrel section and duals for the inclined section with a single sensor in the transition from flat to inclined sections.

The Alpine support is similar to the IBL approach. It is composed of long staves built from a machined carbon foam structure wherein titanium cooling tubes are glued to establish the thermal contact between the mechanical structure and the coolant. The pixel modules are glued on top of the coated carbon foam flat section and onto the inclined machined "mountains"

³Pixel detector element composed of silicon sensor, readout chip and electrical services. Described in more detail in Chapter 5

(thus its Alpine name). These staves are screwed to a system of carbon composite flanges for mechanical stability as shown in Figure 1.12a.

On the other hand, the SLIM local support (stands for Stiff Longerons for ITk Modules) proposed a simplified mechanical structure composed only of composite structures ("longerons") hosting the titanium tubes and the pixel modules. Base blocks 20 mm wide are brazed to the titanium tube to host the flat quad pixel sensors while the inclined duals in the most forward region have 8 mm wide base blocks - see Figure 1.13b.

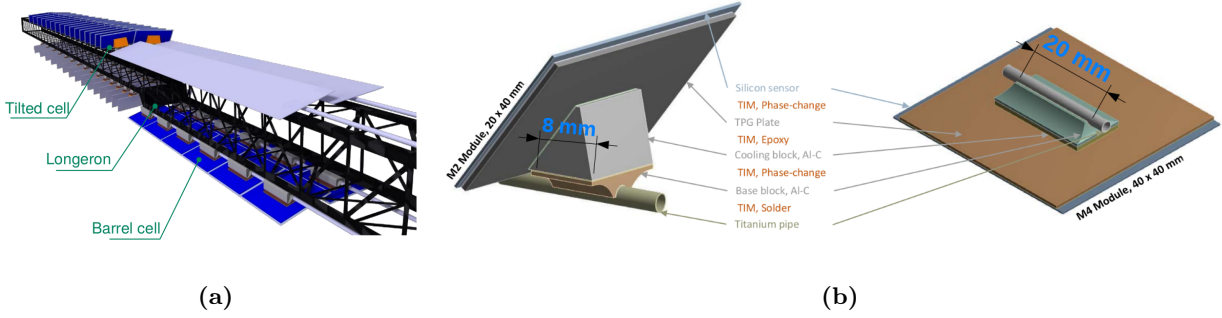


Figure 1.13: (a) SLIM longeron with flat barrel cells and tilted cell in the most forward region, mounted onto a composite structure. (b) Two types of base blocks brazed onto the titanium tubes, foreseen for the inclined and flat sections of the Pixel Outer Barrel.

The inclined supports of either layouts place the sensors farther away from the cooling lines compared to the previous generation of silicon detectors [12], triggering concerns about the ability to respect the thermal specifications. While the Alpine concept is close to the IBL design, with a cooling pipe fully enclosed inside the carbon foam to maximize the contact area between the pipe and the local support structure, the SLIM local support has a much smaller contact area between the cooling pipe and the module mechanical support.

Thanks to its modular design and better reworkability, the SLIM proposal was chosen as baseline for the future local support of the Pixel Outer Barrel.

In 2018, the LHCC imposed on both ATLAS and CMS the exclusive use of "quad modules" for both OB and EC sub systems, leading to major engineering changes. With wider sensors replacing the inclined duals, the design of the inclined cooling block (see Figure 1.13b), became a major challenge in terms of mechanical stability and thermal management.

After several design iterations, the flat and inclined sensors passed from a "longeron only" philosophy to a longeron in the central section of the barrel and inclined rings in the inclined part as shown in Figure 1.14a. The new design proposed, had to resolve new challenges forced by the different mechanical structures and services routing. However, as it was decided to use only one sensor type and the inclination of the modules is not anymore dictated by the geometry of the cooling block, the design of the assembly between the pixel module and titanium tube converged into a simpler and more modular design as shown in Figure 1.14b.

As referred to earlier, the thermal management of the pixel sensors has been one important constraint in the design of the local supports. The heat produced by the pixel sensors and front-end chips of the future Outer Barrel needs to be transported away, by two-phase CO_2 flowing inside the titanium tubes. The global thermal performance of the design will depend thus, on the thermal performance of the local support material interfaces between the sensor and the fluid and on the efficiency of the heat exchange between the tube and the coolant. In the following chapters the effectiveness of the heat exchange process in two-phase flows is discussed along with a comparison of the thermal performance of the local support with the thermal requirements.

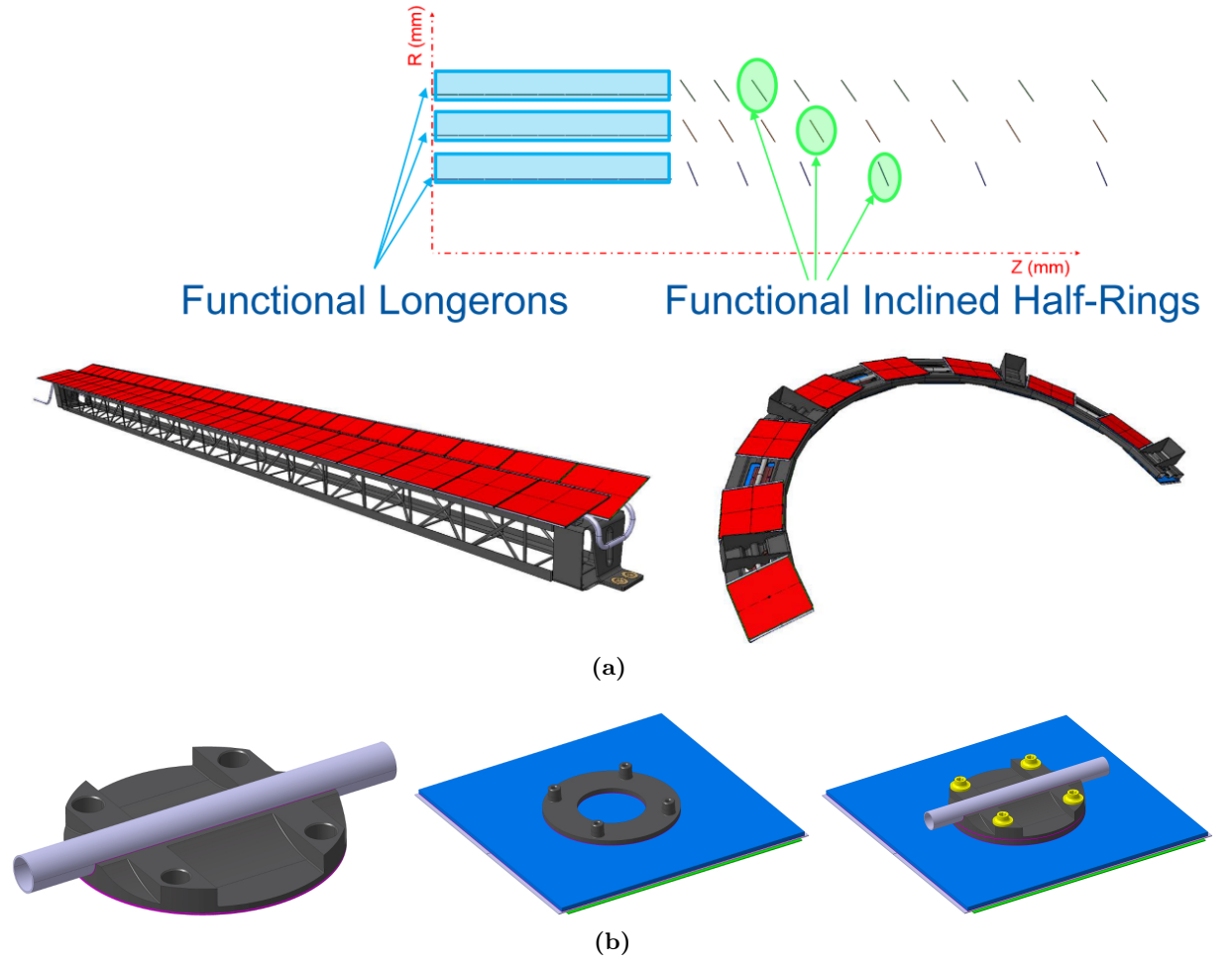


Figure 1.14: (a) Pixel Outer Barrel with quad sensors installed onto longerons in the central section and inclined half-rings in the forward inclined region. (b) Modular local support assembly of the pixel sensors used in the Outer. The cooling block is brazed to the titanium tube and the base block is glued to the pixel module. The pixel module is host in place by screwing the base block and cooling block (four yellow bolts at bottom left figure) [13].

SINGLE AND TWO-PHASE FLOWS IN MINI-CHANNELS

Fluids flowing inside channels are widely used as systems of heat transportation. Refrigeration systems use the fluid to absorb the heat dissipated along its way and transport the energy accumulated through the distribution channels, far from the heat sources, to avoid localised rise of temperature.

With single-phase refrigeration systems the flow of vapour/liquid see its pressure dropping and its temperature rising along the way from the refrigerant source to the end of the heat sink. On the other hand, two-phase refrigeration systems use saturated flows to extract heat from the heat source in a locally isothermal flow boiling process. In both cases, the channel's walls are responsible for establishing the thermal contact between the fluid and the heat source.

The effectiveness of exchange processes in single phase flows between the fluid and the channel's wall, is known to depend on the relationship between the heat exchange surface ($\propto D_h^1$) and the mass flow rate ($\propto D_h^2$). Hence, channels with smaller diameters with bigger surface area to volume ratio ($\propto 1/D_h$) [14] have typically better cooling performance (case of blood capillaries).

Particle detectors have been exploring cooling systems using two-phase flows where the heat exchange processes are more complicated to predict than with single-phase coolants. There is big interest in minimising the size of their cooling systems, hence it is important to well understand the heat exchange process taking place in two-phase flow streams inside small channels.

Regarding the channel classification scheme proposed by Kandlikar and Grande [15], the silicon detectors have been using mini-channels for their refrigeration systems with inner diameters (D_h) between 0.5 mm (for capillaries) and 3 mm (for evaporators).

Chapter outlook and personal contributions:

The present chapter gives an overview of several concepts used in the literature to describe the various types of single and two-phase flows. It presents the heat transfer properties and the predictions on the flow thermo-physical properties, considered crucial for the design of a cooling system. A section is dedicated to comment on the origin of super-heating, its relationship with the nucleate and convective boiling and the theoretical limits given by the Van der Waals theory of fluids. Even if it is not crucial to the subject of this work, it has been a hot topic in the CO₂ cooling community for particle detectors. The semi-empirical predictive methods to calculate the pressure drop and heat transfer coefficient used to design the future evaporative cooling system for the ATLAS ITk are presented in detail and their validity discussed.

Chapter contents

¹The term hydraulic diameter $D_h = \sqrt{\frac{4A}{\pi}}$, where A stands for the cross sectional area of the channel, is used to normalise the calculation of the equivalent diameter of non-cylindrical channels.

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2.1 Dimensionless numbers and thermodynamic properties

There are several dimensionless numbers and thermodynamic parameters that will be often referred throughout the description of single-phase and two-phase flows. They are commonly used in Fluid Mechanics to describe the transport properties of a given fluid.

Each number is derived from the relationship between different thermodynamic properties and kinetics (momentum, velocity, acceleration) of a fluid stream with complementary units.

The most common thermodynamic properties and kinetic parameters used are the following:

Volumetric mass density (ρ): Calculated dividing the total mass by the total volume and commonly called density. This property is deeply related to the fluid state and is determinant on the kinetic behaviour of the fluid (buoyancy, gravity, static pressure, diffusion, etc) [kg/m³].

Dynamic viscosity (μ): Usually referred as absolute viscosity or only viscosity, translates the shear stress in force units between two moving adjacent fluid layers. The bigger the viscosity, the more energy [kg/(m s)].

Kinetic viscosity (ν): Calculated dividing the dynamic viscosity by the density and represents the resistance of a fluid to flow when no external force is applied [m/s].

Surface tension (σ): Represents the force needed to deform the surface of a liquid. This property is particularly relevant in flow boiling where the resistance to surface deformation will be major in the bubble growth and bubble dynamics in flow boiling processes [N/m].

Specific heat capacity (c_p): Translates the raise of temperature of a unit of mass from heat absorption. Fluids with bigger heat capacity are consequently better refrigerants as they will reach a lower equilibrium temperature if compared with other less performing coolants [J/K].

Thermal conductivity (k): Ability to transmit heat through the fluid, which has direct impact on the heat exchange between the fluid and the tube walls, thus in the heat transfer coefficient of the fluid [W/(m K)].

Reduced pressure (p_r): Dimensionless parameter calculated from the ratio between the absolute pressure of the fluid and is critical pressure.

Mass flow rate (MF): The velocity of flows will be often described by its mass flow rate [kg/s]. Other kinetic parameters are derived from this.

Volumetric flow rate ($\dot{Q}$): Differently from the mass flow rate, the volumetric flow rate represents the volume crossing a control cross section in a duct and can be calculated by dividing the MF by the density [m³/s].

Velocity (u): The velocities are in m/s and can be calculated dividing $\dot{Q}$ by the cross sectional area of the pipe. Liquid and gas velocities can be calculated by considering the respective cross sectional areas $[m/s]$.

Mass velocity (G): Calculated by dividing the mass flow rate by the cross sectional area of the tube and used in several dimensionless numbers $[kg/(m s^2)]$.

Wall shear stress (τ_w): Has force units and represents the force parallel to a given surface. It is often used to quantify the slip boundary of a fluid at the pipe inner wall. Big shear stress lead to different fluid velocity profiles and higher pressure drops due to energy loss from friction between liquid and tube walls $[N/m^2]$.

Reynolds number

The Reynolds number is often used to describe the level of turbulence in a fluid calculated from the ratio between the inertial and viscous forces. The inertial forces are calculated from the flow velocity, density and hydraulic diameter while the viscous force is represented by the dynamic viscosity:

$$Re = \frac{\rho u D_h}{\mu} \quad (2.1)$$

Nusselt number

The Nusselt number (Nu) is widely used for the calculation of heat transfer efficiency in a fluid and is calculated from the ratio of convective to conductive heat transfer.

Its evaluation may be not trivial in turbulent flows where the convective and conduction heat flows are difficult to describe.

For the simplest case scenario, where the heat flows are parallel to each other and perpendicular to the flow stream (case of a laminar flow) the Nusselt number can be described by the following:

$$Nu = \frac{h D_h}{k} \quad (2.2)$$

where h stands for the heat transfer coefficient (sometimes also referred by HTC), D_h the hydraulic diameter and k the fluid's thermal conductivity.

The Nusselt number in a fully developed laminar flow is expected to be constant and depends on the tube geometry and wall heat transfer boundary condition [14, p.131]. For a circular tube under a constant heat flux boundary condition, the Nusselt number (Nu_H) is estimated at 4.36 while the number under constant wall temperature (Nu_T) is estimated at 3.66 [14, p.108].

Other correlations were derived to adapt the Nusselt number to turbulent flows. This will be explored afterwards for the calculation of the heat transfer in single and two-phase flows.

Prandtl number

The Prandtl (Pr) number is calculated from the ratio between momentum and thermal diffusivity.

$$Pr = \frac{\mu c_p}{k} \quad (2.3)$$

Small numbers mean that the heat diffusivity dominates the heat transfer behaviour while higher numbers mean that the heat transfer is dominated by momentum diffusivity. The Prandtl number is therefore of extreme importance when one wants to calculate the heat transfer coefficient in forced convection flows.

Froude number

The Froude number (Fr) is defined as the ratio of the flow inertia to the gravitational forces.

$$Fr = \frac{u^2}{gD_h} \quad (2.4)$$

As the set of correlations used to estimate the flow pattern map use often the mass velocity (G) instead of fluid velocity (u), an equivalent expression is used to calculate the Froude number [16]:

$$Fr = \frac{G^2}{\rho^2 g D_h} \quad (2.5)$$

where g stands for the gravitational acceleration.

This number has a strong dependency on the type of flow regimen present in the flow stream, thus it is often coupled with empirical observations to define transition boundaries between different types of flow patterns.

Weber number

The Weber number (We) is calculated from the ratio between inertia and surface tension (σ) forces.

$$We = \frac{\rho u^2 D_h}{\sigma} \quad (2.6)$$

similarly to the Froude number, the expression is often used in correlations defined to estimate the boundaries between different flow patterns and defined in terms of G instead of u :

$$We = \frac{G^2 D_h}{\rho \sigma} \quad (2.7)$$

This dimensionless number is especially important in two-phase flows where the balance between inertial and surface tension forces are determinant in the dynamics of the formation of bubbles and droplets .

2.2 Single-phase liquid flow in mini-channels

A single-phase flow can present either a laminar or a turbulent flow. The following sections, discuss the transition from laminar to turbulent flow in single-phase streams and how the level of turbulence is taken in consideration in the prediction of the frictional pressure drop along the tube and the transfer of heat between heat tube walls and the fluid.

2.2.1 Laminar and turbulent flow

The cross sectional velocity profile of the fluid indicates its level of turbulence. In a fully developed laminar flow, the velocity profile can be described by a parabolic Hagen-Poiseuille velocity profile, with its axis parallel to the duct walls. On the other hand, in a turbulent flow, the velocity profile is not well defined and rather chaotic.

The transition from laminar to a turbulent flow is usually observed for Re numbers above ≈ 2300 .

$$Re = \frac{\rho u D_h}{\mu} \quad (2.8)$$

where ρ stands for the density of the fluid, u its velocity, D_h the hydraulic diameter of the duct and μ the dynamic viscosity.

The velocity profile in a laminar flow will depend mainly on the shear stress between the fluid and the channel's wall τ_w , the dynamic viscosity of the fluid μ and will have direct impact on the pressure drop.

$$\tau_w = \mu \left. \frac{du}{dy} \right|_w \quad (2.9)$$

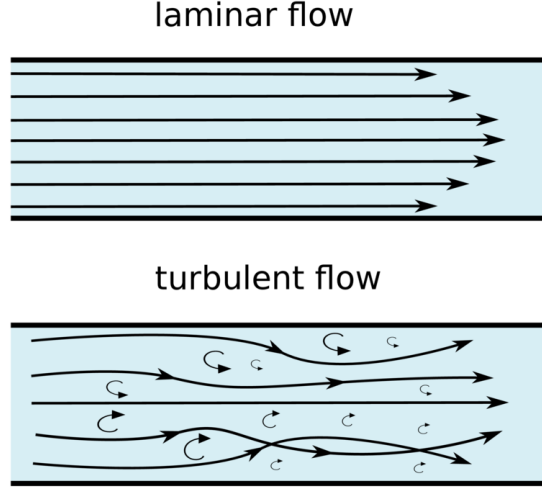


Figure 2.1: Representation at the top figure of the Hagen-Poiseuille velocity profile of a laminar, and a chaotic stream velocity present in turbulent flows.

2.2.2 Pressure drop

The pressure drop along a total length L in an horizontal single-phase liquid flow with an average velocity u_m , can be described by the following equation

$$\Delta p = \frac{2f\rho u_m^2 L}{D_h} \quad (2.10)$$

where the Fanning friction factor f is obtained by:

$$f = \frac{\tau_w}{(1/2)\rho u_m^2} \quad (2.11)$$

In a fully developed laminar flow respecting the Hagen-Poiseuille velocity profile, the friction factor f in a circular duct is defined by a Poiseuille number Po , equals to 16.

$$Po = fRe \quad (2.12)$$

2.2.3 Roughness effects on the pressure drop

The investigation of the effects of the surface roughness in turbulent flows started with the works of Darcy (1857) and was followed by others like Fanning (1886), Mises(1914) and Nikuradse (1937). Part of their work was dedicated to develop the concept of "relative roughness" in a pipe and on systematic studies of its impact on the effective friction factor.

Moody (1944) took the available data from previous works and defined the well known Moody diagram relating the Darcy friction factor f_{Darcy} as a function of the relative roughness present in a pipe ϵ/D , where ϵ represents the roughness and D the hydraulic parameter [14]. The Darcy

friction factor was named after Darcy's contributions to roughness studies and is defined as a fourth of the Fanning friction factor introduced earlier.

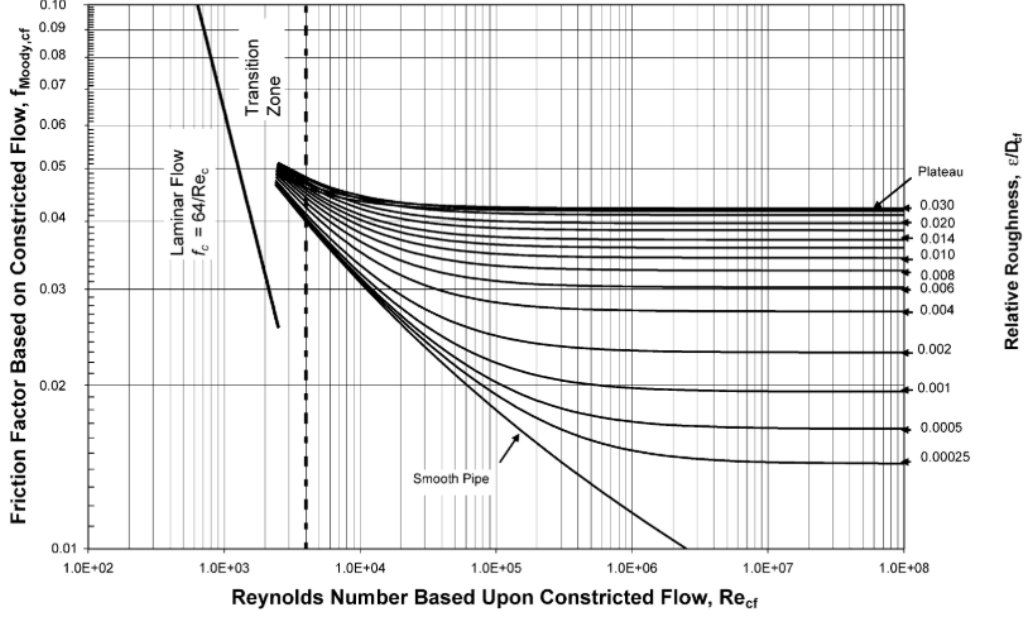


Figure 2.2: *Moody diagram based on the constricted flow diameter [17]*

The constricted flow model was proposed by Taylor, Carrano, and Kandlikar where the Reynolds number and the flow area are redefined from the constricted diameter D_{cf} :

$$D_{cf} = D_h - 2\epsilon \quad (2.13)$$

The $f_{Moody,cf}$ represented in the moody diagram at Figure 2.2, represents the effective friction factor in the constricted flow model for different surface roughness. In a laminar flow, the surface roughness has no effect on the friction factor, which still respects the equality referred earlier, $fR_e = 16$ set by the Hagen-Poiseuille velocity profile:

$$f_{Moody,cf} = f_{Darcy} \left[\frac{D_{cf}}{D_h} \right] \quad (2.14)$$

$$f_{Moody,cf} = 4Po/Re_{cf} \Leftrightarrow f = 64/Re_{cf} \quad (2.15)$$

In the constricted flow, the considered flow parameters are the following:

$$\Delta p = \frac{2f_{cf}\rho u_{m,cf}^2 L}{D_{h,cf}} \quad (2.16)$$

$$u_{m,cf} = \frac{\dot{m}}{A_{cf}} \quad (2.17)$$

$$Re_{cf} = \frac{\rho u_{m,cf} D_{h,cf}}{\mu} \quad (2.18)$$

In the fully developed laminar region respecting $0 \leq \epsilon/D_{h,cf} \leq 0.15$, the friction factor follows the standard definition

$$f_{cf} = \frac{Po}{Re_{cf}} \quad (2.19)$$

In the fully developed turbulent region, also for $0 \leq \epsilon/D_{h,cf} \leq 0.03$, one can use the equation proposed by Haaland [14]:

$$f_{cf} = \frac{f_{Darcy}}{4} = \frac{1}{4} \left\{ -18 \log_{10} \left[\left(\frac{1}{3.7(D_{cf}/\epsilon + 2)} \right)^{1.11} + \frac{6.9}{Re_{cf}(\frac{D_{cf}}{D_{cf}+2\epsilon})} \right] \right\}^{-2} \left[\frac{1}{1 + \frac{2\epsilon}{D_{cf}}} \right]^5 \quad (2.20)$$

In the fully developed turbulent region with $0.03 \leq \epsilon/D_{h,cf} \leq 0.05$ and $Re_{cf} < 10^8$, the friction factor is set at:

$$f_{cf} = 0.0105 \quad (2.21)$$

2.2.4 Heat transfer in sub-cooled fluid

The heat transfer coefficient (h_l) between a heated tube wall heated (heat flux $\dot{q}$) and the sub-cooled liquid is mainly driven by the thermal conductivity of the liquid (k_l) and tube diameter. The Nusselt number is constant and will depend on the tube wall heat transfer boundary condition and duct geometry as stated in section 2.1.

$$h_l = k_l \frac{Nu}{D_h} \quad (2.22)$$

$$T_w - T_f = \frac{\dot{q}}{h_l} \quad (2.23)$$

Where the $T_w - T_f$ is the difference of temperature between tube wall and fluid. In a turbulent flow the Nusselt number Nu for the liquid phase can be calculated from the equation proposed by Dittus and Boelter [18]:

$$Nu = 0.023 Re_l^{0.8} Pr_l^{0.4} \quad (2.24)$$

where k_l , and Pr_l stand for the liquid thermal conductivity and Prandtl number.

2.3 Two-phase flow in mini-channels

Several applications like steam generators, cryogenic systems and the evaporators used in silicon detectors, control the heated surface thanks to the cooling effectiveness of a two-phase mixtures. In a two-phase flow, the volume fraction of the vapour in the vapour-fluid mixture is referred as the void fraction ϵ , while the corresponding mass fraction is referred as vapour quality x . A developed two-phase flow inside an evaporator may present different flow patterns, depending on the flow conditions and the physical properties of the fluid. The overall thermal performance of the two-phase mixture highly depends on the heat transfer between the tube wall and vapour/liquid phases, between the two phases and convection in the fluid. Hence, it is important to investigate the transition between patterns and check how the heat transfer mechanisms are affected.

2.3.1 Concepts used in two-phase flows

While describing the fluid dynamics in a two-phase flow, several recurrent concepts are often used:

Heat flux (q)

The thermal performance in a cooling pipe depends on the heat transfer coefficient, h or htc , between the pipe wall and the fluid:

$$h = \frac{q}{T_w - T} \quad (2.25)$$

where q stands for the local heat flux in W/m^2 , T_w the temperature of the innerwall of the pipe in contact with the fluid and T the temperature of the fluid.

The local heat flux is an important parameter as the heat transfer coefficient is often dependent on its magnitude.

Void fraction (ϵ)

The void fraction stands for the ratio of the cross sectional area that is occupied by the vapour phase. This parameter is fundamental to calculate the liquid and vapour velocities of liquid and vapour phases, thus, to estimate either the pressure drop along the pipe and heat transfer coefficient.

Vapour quality (x)

Similarly to the void fraction, the vapour quality translates one other relative quantity between liquid and vapour phases. It translates the ratio of mass of the vapour phase to the total mass of the flow:

$$x = \frac{M_v}{M_v + M_l} \quad (2.26)$$

Along a two-phase cooling pipe with total length L , the continuous absorption of heat will evaporate the liquid, increasing the amount of vapour and changing its enthalpy. Assuming that both phases are in thermodynamic equilibrium (same pressure and temperature) along the pipe and that there is saturated liquid at the inlet of the pipe section, the vapour quality at the position z along be calculated from:

$$x = \frac{h(z) - h_l}{h_{LV}} \quad (2.27)$$

where h_l is the enthalpy at the saturated liquid, h_{LV} is the latent heat the given pressure and $h(z)$ the estimated enthalpy at the position z . Considering an in-compressible fluid, the vapour quality at the end of the pipe (x) can be calculated from the equation:

$$h(L) = h_l + \frac{\pi D_h}{M} \int_{z=0}^L q(z) dz \quad (2.28)$$

where $q(z)$ in W/m^2 is the local heat flux at position z .

2.3.2 Sub-cooled and saturated flow boiling

A sub-cooled fluid entering the evaporator will be fully liquid and exchange heat with the tube wall through direct contact conduction. The overall heat transfer will depend on the thermal contact between the fluid and wall (wettability), the liquid thermal conductivity and the free convection created by internal temperature gradients, that will depend on the level of turbulence.

If the wall temperature is above the corresponding saturation temperature, a nucleation process may be triggered with the creation of bubbles on the heated surface. Depending on the level of sub-cooling, the produced vapour may be cooled rapidly by the surrounding liquid and

collapse(condense) immediately. This nucleation process takes place in small cavities present on the tube wall, usually determined by its roughness.

If the temperature of the wall is not above the saturation temperature, the fluid will be continuously heated and thus, see its pressure and temperature increasing until reaching the saturated state.

When the liquid is in the saturated (non-subcooled) state, the tube wall will consequently be at a higher temperature when heated, leading to bubble nucleation. The local wall temperature in contact with the bubble will see its temperature increasing, caused by the lower conductivity of the vapour, compared to the liquid. The continuous evaporation will contribute to the increase of the bubble size until it eventually boils off the surface, triggering the nucleate boiling process (on-set boiling).

2.3.3 Onset boiling

The temperature profile in the liquid from the tube wall to the fluid bulk will depend on the amount of liquid, its thermal conductivity and on the free convection in the fluid.

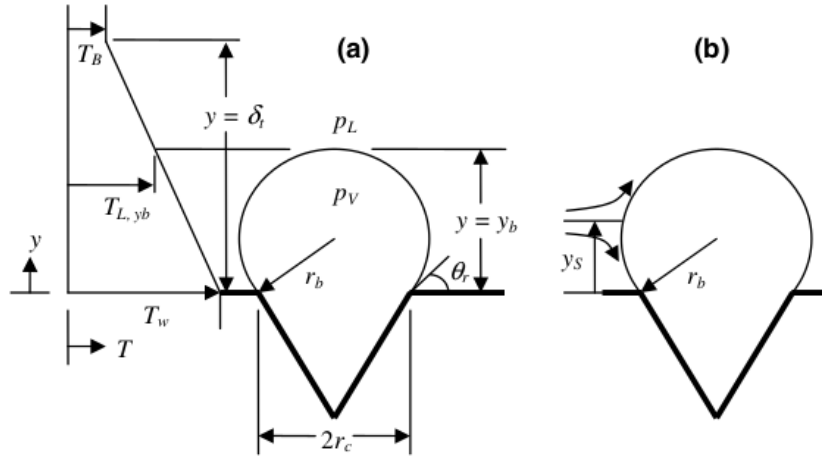


Figure 2.3: Temperature profile (a) and stagnation streamline (b) around a bubble [19]

As soon the onset boiling (ONB) takes place, the ensemble of forces dictating whether the bubble stays growing in contact with the surface or whether it boils off. This depends on: surface tension between liquid and vapour σ , the size (radius) of the bubble r_b , the bubble buoyancy (gravitational forces), the liquid temperature and velocity field around the bubble. These parameters are determined by the liquid film thickness, thermal conductivity, heat transfer coefficient and vapour quality.

Kandlikar worked on analytical formalism to determine the liquid super-heating level (ΔT_{sat}) needed to experience nucleate boiling in small cavities with a given size. Figure 2.4 illustrates a bubble created in a small cavity characterized by its "mouth" size $2r_c$. The excess of pressure inside the bubble ($p_v - p_l$), with a radius r_b , and the liquid film thickness δ_t is given by:

$$\Delta T_{sat} = T_w - T_{sat} \quad (2.29)$$

$$p_v - p_l = 2\sigma/r_b \quad (2.30)$$

$$\delta_t = k_l/h_l \quad (2.31)$$

where k_l is the thermal conductivity of the liquid and h_l the heat transfer coefficient with liquid flow before nucleation.

From the analysis of the thermal equilibrium between the bubble and the temperature of liquid stream flowing around it, its is estimated the range of critical cavity radius $\{r_{c,min}, r_{c,max}\}$ needed to trigger the bubble nucleation.

$$\begin{aligned} \{r_{c,min}, r_{c,max}\} &= \frac{\delta_t \sin \theta_r}{2.2} \left(\frac{\Delta T_{sat}}{\Delta T_{sat} + \Delta T_{sub}} \right) \\ &= \times \left[1 \mp \sqrt{1 - \frac{8.8\sigma T_{sat}(\Delta T_{sat} + \Delta T_{sub})}{\rho_v h_{LV} \delta_t \Delta T_{sat}^2}} \right] \end{aligned} \quad (2.32)$$

Where the h_{LV} , ΔT_{sat} and ΔT_{sub} stand for latent heat of vaporization, wall local super-heat and bulk sub-cooling.

The cavity radius nucleating first under this conditions is:

$$r_{c,crit} = \frac{\delta_t \sin \theta_r}{2.2} \left(\frac{\Delta T_{sat}}{\Delta T_{sat} + \Delta T_{sub}} \right) \quad (2.33)$$

And by solving equation 2.32 using equation 2.33, the corresponding wall super-heat can be calculated from the following:

$$\Delta T_{sat,ONB} = \sqrt{\frac{8.8\sigma T_{sat} \dot{q}}{\rho_v h_{LV} k_l}} \quad (2.34)$$

where $\dot{q}$ is the local wall heat flux.

if the cavity size r_c is not known, then the nucleation will occur first in the cavity that leads to the minimum wall super-heat to trigger nucleation. The wall super-heat required to nucleate in a cavity with a given size r_c is given by [19]:

$$\Delta T_{sat} |_{ONB \text{ at } r_c} = \frac{1.1 r_c \dot{q}}{k_l \sin \theta_r} + \frac{2\sigma \sin \theta_r}{r_c} + \frac{T_{sat}}{\rho_v h_{LV}} \quad (2.35)$$

The previous analysis neglects the dynamic effects that the fluid stream velocity and level of turbulence may have on the bubble pressure balance. In fast laminar flows in small pipes where the velocity profile is more abrupt, it is hard to determine whether this promotes boiling or restrains the bubble in place.

In either case, the bubble will keep increasing in size and pressure and the liquid temperature will raise until reaching the condition for nucleation. This effect will translate into a wall super heat and a corresponding pressure spike as shown in Figure 2.4 [14]. The big rise of pressure will lead to flow instabilities, namely due to back flow, that will eventually trigger the nucleate boiling and propagate along the pipe.

This super-heat effect has been observed with a much lower slope in ATLAS IBL detector [3, p.81] and CMS thermal management studies, with a developed super heated liquid in titanium tubes that eventually reached the evaporators. Heating the fluid smoothly along the pipe, and the titanium low roughness were found to be the causes for this unwanted effect.

Solving equation 2.35 for a two-phase CO₂ flow, one obtains the ΔT_{sat} shown in Figure 2.5 considering different heat fluxes and saturation temperatures. The estimations show that pipes with smoother surfaces are expected to need a relatively high wall (up to 7 °C) super-heat to trigger boiling and that this value does not depend much on the heat-flux. This particularity may be the reason for the observed smooth super-heating of the liquid along small pipes (capillaries) in titanium tubes, only subjected to the radiative heat pick-up from the environment.

The solution found to trigger boiling was to apply a localised big heat flux, to trigger locally the big spike shown in Figure 2.4 before the evaporator, guaranteeing a saturated flow boiling along the detector section. Other solutions like introduction of pressure drop elements promoting reversed flow or rougher surfaces are proposed [19].

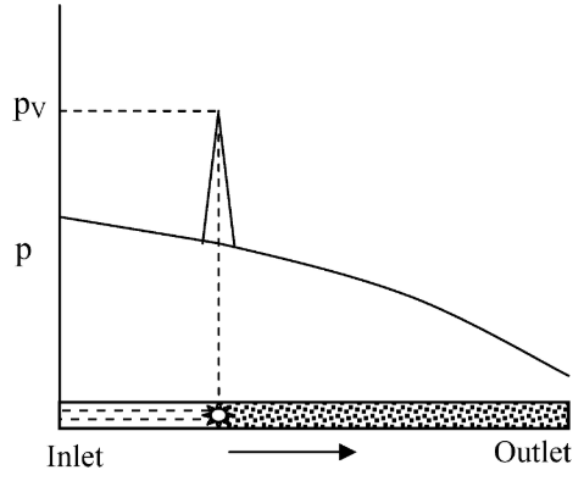


Figure 2.4: Schematic representation of pressure variation following nucleation during flow boiling in a micro-channel [19]

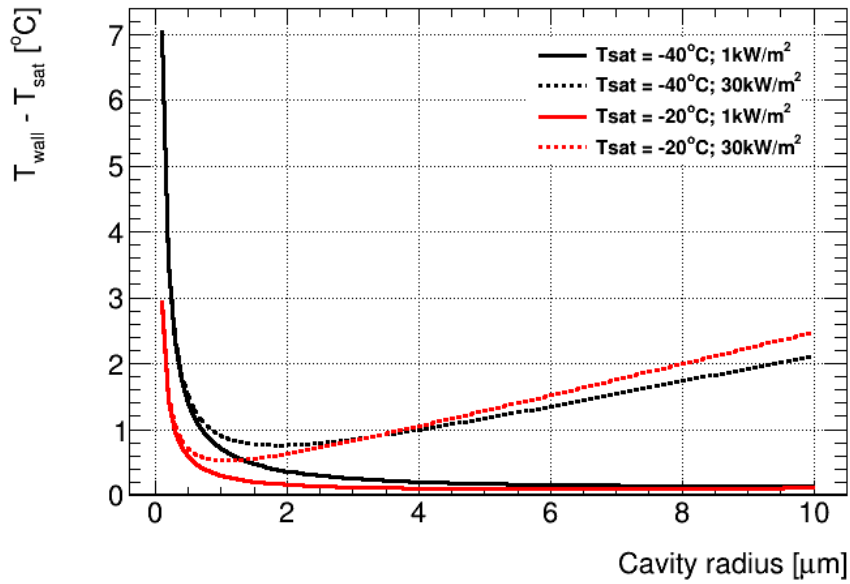


Figure 2.5: Estimated ΔT_{sat} needed to trigger bubble nucleation as a function of cavity radius for different cooling conditions using two-phase CO_2 .

2.3.4 Flow patterns

A flow with sub-cooled fluid and continuously heated along the way, will eventually develop into a two-phase flow and pass through several flow patterns on its two-phase mixture, as the liquid evaporates - as illustrated in Figure 2.6 for a vertical tube.

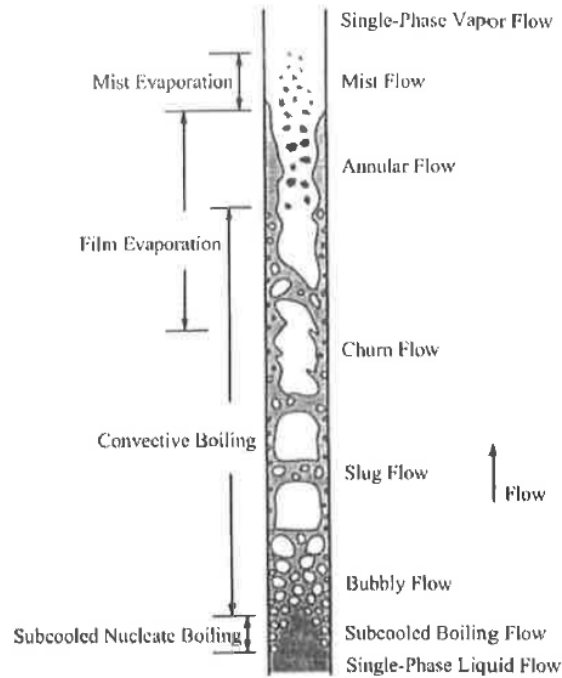


Figure 2.6: Illustration of how different flow patterns are developed during the boiling process in low (at the right side of the vertical tube) and corresponding process of heat exchange (left side of the tube) [20].

When the fluid reaches saturation, first bubbles start appearing on the inner tube wall surface, in what is called on-set boiling. As the fluid continues to be heated, these bubbles will grow and eventually detach under buoyancy (for bubbles under the liquid stream) and forced convection effects in a nucleate boiling process. The agitation created by the bubbles flowing around will perturb the flow velocity profile, creating more turbulence thus enhancing the heat transfer. Depending on the mass flow, this may lead to a Bubbly flow at higher mass flows with small vapour bubbles flowing around, or to "Dispersed Bubbly", with bigger bubbles, for lower mass flows, as they take longer to detach from the surface.

At higher mass flows, the bubbles will eventually accumulate in the central region of the tube leaving a thin film of liquid flowing in contact with the tube wall in a Annular flow. This process will continue until the thin liquid film is populated with so many bubbles that they will merge together, and remain in contact with the tube wall. The beginning of this process is called the dry-out incipience and takes place first at the top part of the tube wall due to the vapour buoyancy, and continues until the dry-out completion, when there is not more liquid in contact with the walls. The remaining liquid in the duct is now dispersed in the form of droplets in a Mist flow, similarly to the vapour bubbles in the Bubbly flow.

At lower mass flows, the bigger bubbles in the dispersed flow will continuously accumulate forming different types of intermittent bubble arrangements such as Churn flow, Plug/Slug, or will accumulate at the top the tube forming a Stratified flow, with a liquid layer at the bottom, until it dries out. After reaching the dry-out, the flow passes also through dry-out and reaches the Mist flow regimen, similarly to what happens for higher mass flows.

With intermediate mass flows, the Stratified flow may see the liquid layer presenting a wavy profile (Stratified Wavy) due to the liquid-vapour slip interface with little turbulence. Getting

to higher mass flows, the wave front will eventually touch the top tube wall and develop into an Annular flow.

Several works were dedicated to develop correlations able to predict the flow pattern of a fluid, given the vapour quality and the mass flow. The semi-empirical method used in this thesis to predict the heat transfer coefficient on CO₂ saturated flows is based on the Cheng–Ribatski–Wojtan–Thome CO₂ pattern map [21]. This is discussed further in section 2.5.

2.3.5 Heat transfer: Nucleate and convective boiling

As soon as the nucleate boiling is triggered, the turbulence created by the bubbles flowing around, enhances the bubble detachment rate hence increasing the heat transfer. This enhancement is usually represented by the boiling dimensionless parameter Bo :

$$Bo = \frac{\dot{q}}{k_l G} \quad (2.36)$$

where G stands for the mass flow.

With the continuous production of bubbles, the vapour quality increases, leading to higher fluid velocities. Higher void fraction and higher velocities will create a thinner liquid film in contact with the tube wall, which will reduce the boiling heat transfer but improve the forced convection. This effect that depends on the vapour quality x , the liquid density ratio ρ_v/ρ_l and relative difference of viscosities μ_l/μ_v , is usually represented in terms of the Lockhart-Martinelli parameter [22] X_{tt} :

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_v}{\rho_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_v} \right)^{0.1} \quad (2.37)$$

The rate of bubble production soon stabilizes and an Annular flow takes shape. In an Annular flow, the liquid thin film will get thinner and the homogenization of the fluid temperature will be get faster through forced convection created by the vapour phase. At some point, the heat transfer through forced convection dominates over the nucleate boiling.

The proposed correlations used to estimate the heat transfer coefficient in flow boiling are usually restricted to one fluid and to either one of the two situations: sub-cooling or saturated boiling. Most of the recent correlations follow the proposal of Chen [23] to treat the nucleate boiling and non-boiling forced convection² as separate processes. The overall two-phase heat transfer coefficient h_{tp} is calculated as a linear relationship between both.

The forced convection is usually based on the heat transfer of the liquid phase, proposed by the Dittus-Boelter equation 2.24, multiplied by an enhancement factor E , based on the Martinelli parameter X_{tt} , which varies with the vapour quality to account for the increase of the heat transfer due to higher flow velocities and thinning of the liquid film. The nucleate boiling in its turn, can be multiplied by a suppression factor S to account for the lower effective super-heat available to trigger boiling, due to thinner film boundary [24]. Figure 2.7 illustrates how these two components vary in a developed flow, from fully liquid to only vapour.

2.3.6 Frictional pressure drop of two-phase flows in mini-channels

The pressure drop in a mini-channel represents the energy (potential and kinetic) lost by the fluid due to friction on the walls and turbulence, and can be calculated as the sum of the following components:

$$\Delta p = \Delta p_c + \Delta p_{f,1-ph} + \Delta p_{f,tp} + \Delta p_a + \Delta p_g + \Delta p_e \quad (2.38)$$

²In his work, the nucleating boiling is referred as micro-convective process while the forced convection is stated as a macro-convective process.

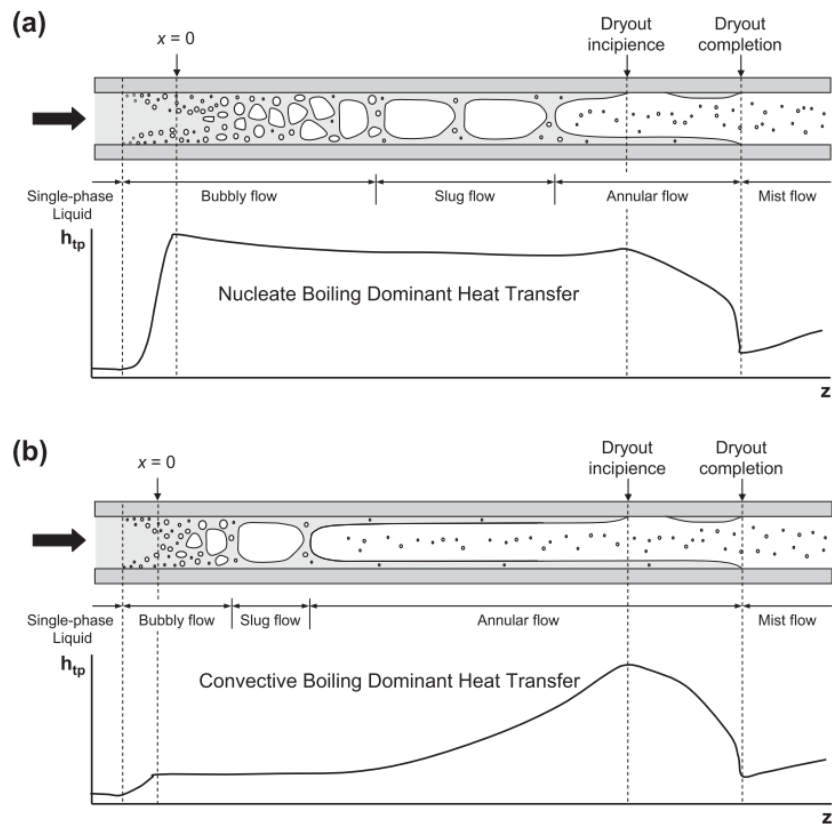


Figure 2.7: Different flow regimes, wall dry-out and variation of heat transfer coefficient along uniformly heated channel for (a) nucleate boiling dominant heat transfer and (b) convective boiling dominant heat transfer [25].

where the subscripts p_c stand for pressure drop at the contraction entrance; $f, 1 - ph$ the single-phase pressure drop due to friction; f, tp is the two phase pressure frictional pressure drop; a is the acceleration caused by the liquid evaporation; g is the gravitational pressure drop; and e the expansion at the outlet [14, p.262]

The pressure drop due to sudden contraction of the channel is calculated with the following equation:

$$\Delta p_c = \frac{G^2}{2\rho_l} \left[\left(\frac{1}{C_o} - 1 \right)^2 + 1 - \frac{1}{\sigma_c^2} \right] \psi_h \quad (2.39)$$

where G is the mass flux, σ_c the contraction area ration and C_o the contraction coefficient given by:

$$C_o = \frac{1}{0.639(1 - \frac{1}{\sigma_c})^{0.5} + 1} \quad (2.40)$$

ϕ_h is the two-phase homogeneous flow multiplier given by:

$$\phi_h = [1 + x(\frac{\rho_l}{\rho_v} - 1)] \quad (2.41)$$

The exit pressure loss is calculated from:

$$\Delta p_e = G^2 \sigma_e (1 - \sigma_e) \phi_s \quad (2.42)$$

where σ_e is the area expanion ratio and ϕ_s is the separated flow multiplier:

$$\phi_s = 1 + \left(\frac{\rho_l}{\rho_g} - 1 \right) [0.25x(1 - x) + x^2] \quad (2.43)$$

The acceleration pressure drop, assuming homogeneous flow is calculated from:

$$\Delta p_a = G^2 \nu_{lv} x_e \quad (2.44)$$

where ν_{lv} is the difference between the specific volumes of vapour and liquid phases, and x_e is the change in vapour quality between the inlet and outlet.

The gravitational pressure drop can be calculated from the following equation:

$$\Delta p_g = \frac{g(\sin \beta)L}{\nu_{lv} x_e} \ln \left(1 + x_e \frac{\nu_{lv}}{\nu_l} \right) \quad (2.45)$$

where β is the angle made between the flow direction and the horizontal plane.

The frictional pressure drop for the single phase region is usually calculated from equation 2.10 referred to earlier.

There are in the literature several methods used to predict the pressure drop in two-phase flows. Each is more suited to a particular fluid, channel dimensions or type of mixture (with more gas or liquid). It mostly depends on the experimental data it was compared against. This is one of the main drawbacks of the several existent empirical methods. This thesis bases its pressure drop estimates in a more phenomenological correlation proposed by Cheng et al., that will be discussed further in section 2.5. Differently from mere empirical models, it tries to broaden its application range by taking into account the interfacial flow structure like the gas liquid partitioning in the tube (pattern maps). The model becomes more complex but less dependent on the experimental database against which it was compared.

Many of the methods present however some similarities in their approaches. The pressure drop of two phase mixtures are usually derived from the expected pressure drop in a single phase flow, by using equation 2.10, multiplied by a two-phase multiplier ϕ_{tp} . One of the first

proposing this approach was Lockhart-Martinelli. In phenomenological approaches these two-phase multipliers may take into account the flow pattern like the one used in this thesis, discussed in section 2.5.

2.4 Thermodynamic properties of Van der Waals fluids

The Van der Waals (VdW) theory of fluids treats two-phase fluids as a system of interacting particles subject to the same thermodynamic properties. In the formalism describing the repulsive and attractive forces between molecules, the thermodynamic properties of a VdW fluid are presented in a universal manner, normalised with critical thermodynamic properties: p_c , t_c , critical enthalpy U_c , critical density n_c and critical volume V_c [26].

Even if they may not perfectly describe real fluids, some generic thermodynamic properties are relevant to understand how the thermal performance of any two-phase cooling system depends on its thermodynamic state.

2.4.1 Critical state and reduced parameters

During the phase change, the liquid and gas phases share the same pressure and temperature. The energy absorbed/dissipated by the system is responsible for changing the system internal energy in a phase change process (either liquid evaporation or gas condensation).

The critical point expressed in terms of its critical pressure p_c and critical temperature t_c , defines the end of the coexistence curves (or saturation line). Above the critical point, the fluid is said to be in super-critical state, with diffuse thermodynamic properties that can be more liquid or gas like.

The pressure-volume (P-V) diagram with isotherm lines, situates the liquid-gas coexistence thermodynamic states for a given fluid, in terms of its volume, pressure and temperature - see the P-V diagram in terms of the critical exponents $\hat{p} = p/p_c$, $\hat{V} = V/V_c$ and $\hat{T} = T/T_c$ at Figure 2.8.

A fluid with a thermodynamic state situated above the critical point is a super-critical fluid, below and at the left of the coexistence curves (saturation line) is said to be sub-cooled, while that at its right, it is said the fluid is in a super-heated state.

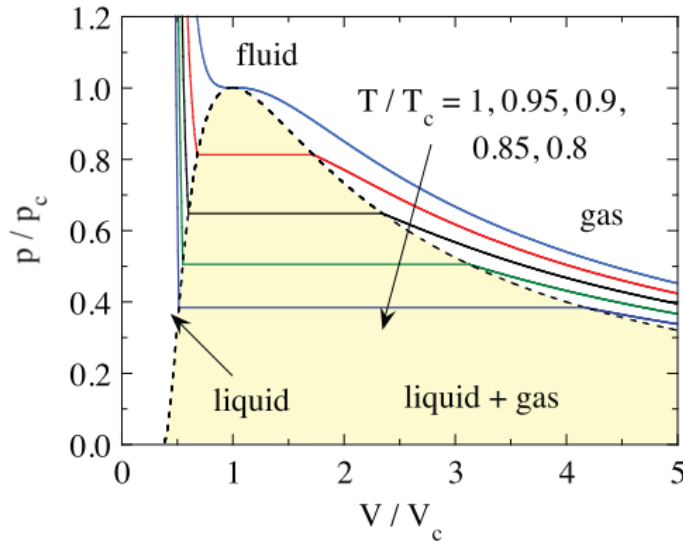


Figure 2.8: Illustration of how the critical pressure and volume ratios vary in a P-V diagram. The colored curves represent the iso-thermal lines. The blue curve passing by the critical point corresponds to the critical temperature ($\hat{T} = 1$). The remaining curves correspond to lower temperatures from a $\hat{T}$ of 0.95 to 0.8 [26].

2.4.2 Coexisting liquid-gas mass densities

The relative difference between the reduced coexisting gas and liquid densities (n_{liquid} and n_{gas} parameters) change, as we get farther from the critical point, as shown in Figure 2.9a. One can observe how the relative difference of volumes explodes at reduced temperatures below the curve's inversion point (close to $\hat{T}$ of 0.6) in Figure 2.9b.

Bigger relative volume occupied by the gas phase at lower reduced pressures $\hat{p}$, lead to bigger difference in the relative velocities of liquid/gas phases in a two-phase flow. As shown previously, this relationship has direct impact on the pressure via the Martinelli parameters and two-phase multipliers presented in equations 2.37 and 2.41, meaning that two-phase mixtures at lower reduced pressures, will consequently present higher frictional pressure drops.

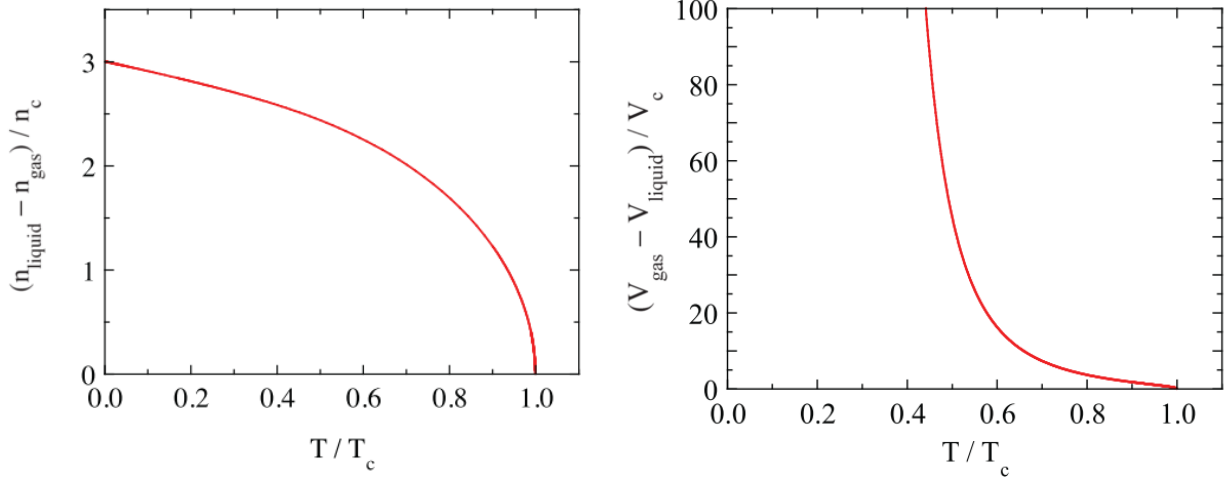


Figure 2.9: At (a) is shown the relative difference between the mass densities of liquid and vapour phases as function of the temperature and at (b) the corresponding difference between vapour and liquid specific volumes [26].

2.4.3 Specific heat capacity in sub-cooled liquids and super-heated gases

The Prandtl number for the liquid phase, defined in equation 2.3, has direct impact on the heat transfer between the tube wall and the liquid phase. Hence, the specific heat capacity at constant pressure c_p will also have a direct influence on the heat transfer coefficient.

Figure 2.10 shows how the c_p (normalised with the number of particles Nk_b) of sub-cooled liquid increases as it approaches to the coexistence state (saturation line in the P-H diagram). The colored curves represent isobar lines, like the ones shown in Figure 2.8. The black curve corresponds to the isobar line at the critical pressure ($\hat{P} = 1$). For $\hat{T} < 1$ the fluid is sub-cooled and sees its specific heat increasing until crossing the saturation line at $\hat{T} < 1$ where its specific hits its maximum. At lower reduced pressures the isobar lines cross the saturation line at lower temperatures and the maximum specific heat gets way lower. This is why two-phase mixtures show an improvement of their cooling capabilities at higher reduced pressures and why sub-cooled fluids see their cooling performance increasing as they saturate.

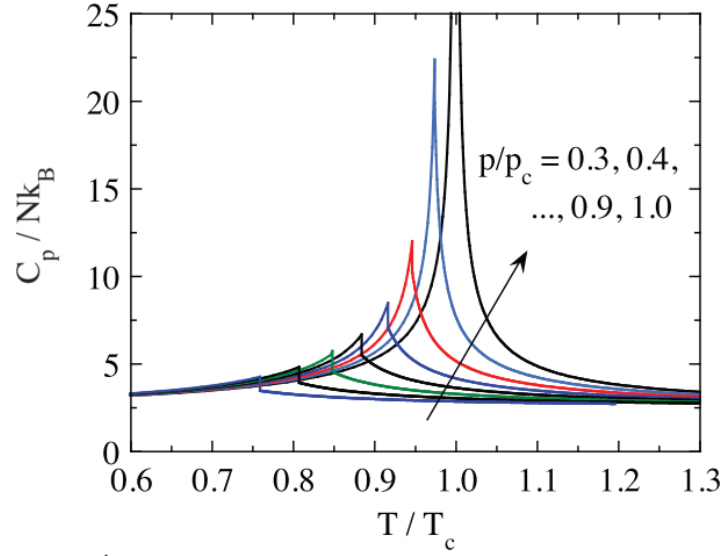


Figure 2.10: Specific heat of a fluid for critical pressures (p/p_c) from 0.3 to 1.0 at different critical temperatures. Each colored line represents a isobar line crossing horizontally the P - V diagram like the one shown in Figure 2.8 [26].

2.4.4 Latent heat

The latent heat of evaporation is important for cooling purposes as it defines the amount of heat possible to be absorbed by unit of mass of fluid, before turning completely into vapour. Figure 2.11 shows how the latent heat increases at lower reduced temperatures.

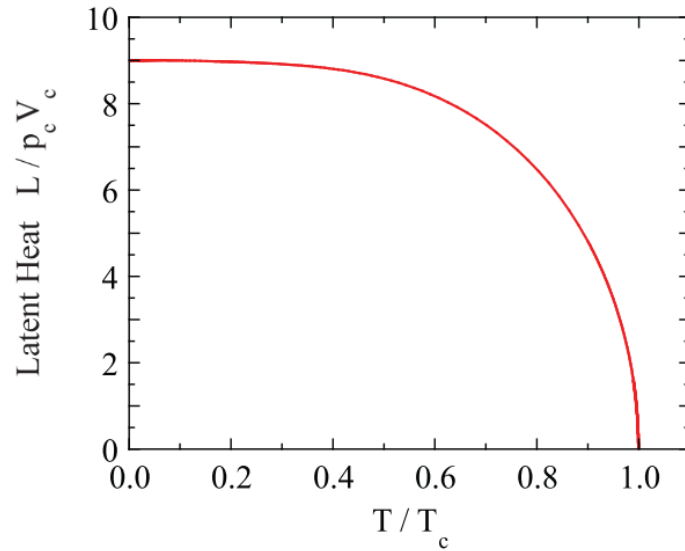


Figure 2.11: Latent heat in a two-phase mixture with the decrease of its critical temperature [26].

2.4.5 Super-cooling and super-heating

Super-cooling and super-heating are predicted by the VdW theory as first-order hysteresis in phase transitions upon cooling and warming processes [26]. Figure 2.12a shows the allowable processes in a fluid with $\hat{p} = 0.3$.

When the gas cools, it nucleates in small droplets in a process that may take some time to overcome potential energy barriers due to surface tension between liquid and gas phases. In

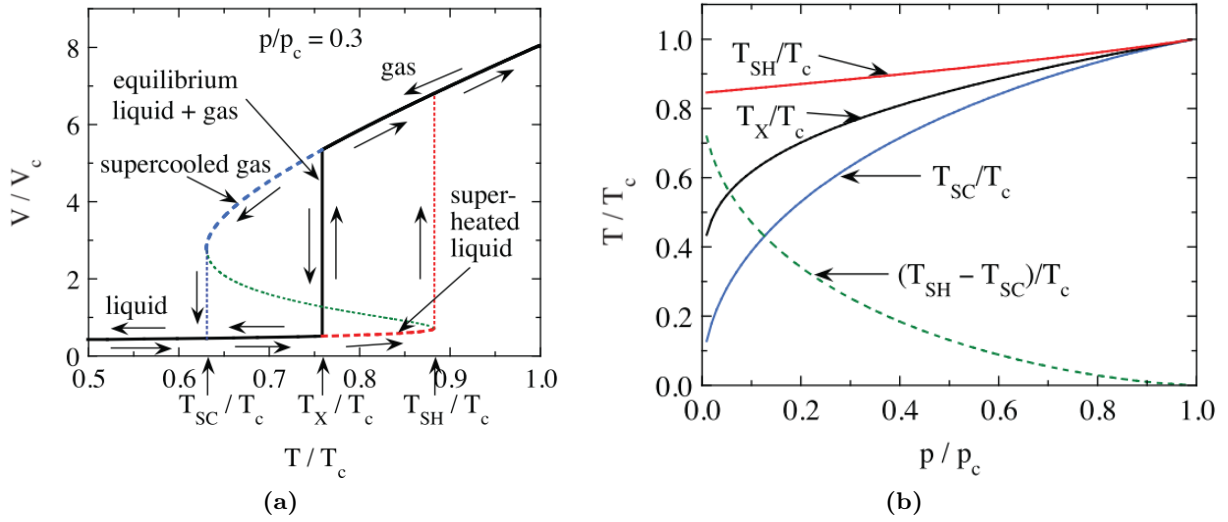


Figure 2.12: Diagrams at (a) with allowed super-cooling (blue dotted lines) and super-heating processes in a two-phase mixture with a reduced pressure of 0.3, predicted by the Van der Waals theory of fluids. Figure (b) shows the theoretical limits of super-heating and super-cooling temperatures at different reduced pressures [26].

rapid cooling, there may be no time to nucleate the vapour, thus it keeps being cooled below the coexistence temperature in a super-cooling process. The VdW theory predicts a minimum super-cooling temperature below which, the vapour is expected to condense rapidly in a irreversible transition, represented by the vertical blue dotted line.

In a similar process, there may be liquid super-heating if there is a potential barrier avoiding bubble nucleation. The bubble nucleation depends on the velocity and temperature of the liquid surrounding the bubble being formed. This gets more difficult to be triggered in smooth surfaces. The irreversible process represented by the vertical dotted red line lead to an abrupt evaporation of the liquid. This sudden fluid expansion is accentuated at lower reduced pressures as the $\Delta\hat{V}_X = \hat{V}_{gas} - \hat{V}_{liquid}$ is bigger - see Figure 2.8.

The super-heating has already been presented as the cause for the observed retardation of the onset boiling in IBL [3, p.81] when cooled at temperature close to -20°C .

Figure 2.12b shows how the super-heating margin $\Delta T_{SH} = T_{SH} - T_X$ gets higher at lower temperatures, giving more room to super-heated liquid. At temperatures closer to the critical point, the ΔT_{SH} gets much lower. The abrupt evaporation burst happens before experiencing considerable super-heating effects, triggering immediately the nucleate boiling due to the turbulence created by the sudden vaporization of the liquid. As consequence, the super-heating effects are more likely to happen at lower temperatures.

2.5 Semi-empirical and flow pattern based predictive methods

This thesis bases most of its estimates for the frictional pressure drop and heat transfer coefficients of CO_2 saturated flows on semi-empirical correlations [16, 27]. Rather than a universal set of correlations, the proposed prediction methods, take into account the phenomenological properties of CO_2 saturated flows. The current section is mostly dedicated to describe how the correlations are formulated.

It is known that the effectiveness of the heat transfer in a two-phase flow highly depends on its flow regime. Hence, the methods described are based on CO_2 pattern maps to properly predict the flow pattern and calculate several thermo-fluidic properties accordingly (frictional pressure drop, heat transfer coefficient and critical heat flux, etc).

2.5.1 Model specific parameters

Several dimensionless numbers are used to develop the methods. Some of them are quite common to many correlations, as the already referred Weber number (We), Nusselt number (Nu) and Prandtl number (Pr_l), while others are specific to the model's approach.

Circular and non-circular channels are treated equally, taking into account their hydraulic diameter D_h :

$$D_h = \sqrt{\frac{4A}{\pi}} \quad (2.46)$$

where A is the total cross sectional area of the pipe and A_l and A_v the corresponding vapour and liquid parts:

$$\begin{aligned} A_v &= A\epsilon \\ A_l &= A(1 - \epsilon) \end{aligned} \quad (2.47)$$

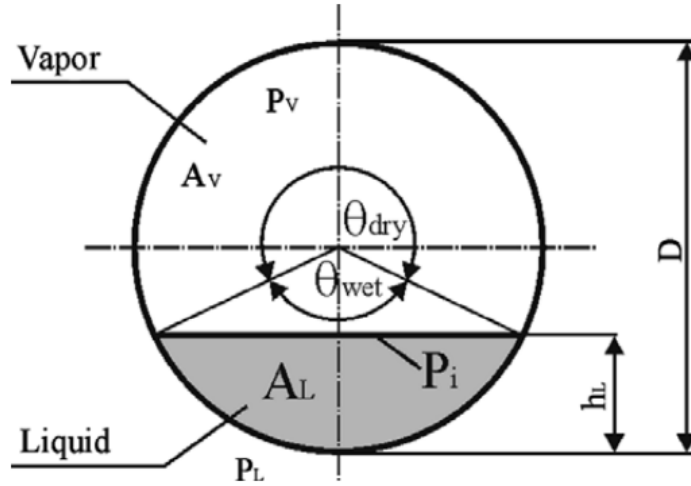


Figure 2.13: Sectional view of a cylindrical tube illustrating some parameters used in the semi-empirical predictive methods [16, 27].

The void fraction ϵ represents the vapour mass fraction, and is determined by Thome and Hajal [28], taking into account the Rouhani-Axelsson drift flux model [29]:

$$\epsilon = \frac{x}{\rho_v} \left[(1 + 0.12(1 - x)) \left(\frac{x}{\rho_v} + \frac{1 - x}{\rho_l} \right) + \frac{1.18(1 - x)[g\sigma(\rho_l - \rho_v)]^{1/4}}{G\rho_l^{1/2}} \right]^{-1} \quad (2.48)$$

The dry angle θ_{dry} defines the ratio of the tube perimeter in contact either with liquid or vapour. The θ_{strat} representing the θ_{dry} in a stratified flow (like the one shown in Figure 2.13) is calculated with the equation proposed by Biberg [30]:

$$\begin{aligned} \theta_{strat} = & 2\pi - 2 \left\{ \pi(1 - \epsilon) + \left(\frac{3\pi}{2} \right)^{1/3} [1 - 2(1 - \epsilon) + (1 - \epsilon)^{1/3} - \epsilon^{1/3}] \right. \\ & \left. - \frac{1}{200}(1 - \epsilon)\epsilon[1 - 2(1 - \epsilon)][1 + 4(1 - \epsilon)^2 + 4\epsilon^2] \right\} \end{aligned} \quad (2.49)$$

Figure 2.13 illustrates a sectional view of a tube with the liquid standing at the bottom of the tube while the tube walls on top remain dry.

Six geometrical dimensionless parameters are defined from ϵ and θ_{strat} to characterize the liquid-gas partitioning inside the channel:

$$h_{ld} = \frac{h_l}{D_h} = 0.5 \left(1 - \cos \left(\frac{2\pi - \theta_{strat}}{2} \right) \right) \quad (2.50)$$

$$P_{ld} = \frac{P_l}{D_h} \quad (2.51)$$

$$P_{gd} = \frac{P_g}{D_h} \quad (2.52)$$

$$P_{id} = \frac{P_i}{D_h} = \sin \left(\frac{2\pi - \theta_{strat}}{2} \right) \quad (2.53)$$

$$A_{ld} = \frac{A_l}{D_h^2} = \frac{A(1 - \epsilon)}{D_h^2} \quad (2.54)$$

$$A_{gd} = \frac{A_v}{D_h^2} = \frac{A\epsilon}{D_h^2} \quad (2.55)$$

$$(2.56)$$

2.5.2 Pattern maps

The pattern maps are usually represented by 2D maps with different regions, depending on the mass flow and vapour quality of the saturated flow. Figure 2.14 illustrates different pattern maps retrieved for a CO₂ flow inside a mini-channel with a hydraulic diameter of 2.3 mm.

Each region in the pattern map correspond to a different types of regimens and are defined from the following $G(x)$ curves:

x_{IA} : Intermittent (I) to Annular (A) flow transition boundary

x_{di} : Dry-out incipience vapour quality defining the transition from annular/stratified wavy (A/SW) to dry-out (D)

x_{de} : Dry-out completion corresponding to the transition from dry-out to mistflow (M)

G_{SW} : Mass flow separating the stratified wavy (SW) from intermittent and annular flow

G_{strat} : Mass flow below which the liquid be accumulated at the bottom of the duct presents a stratified (S) regimen

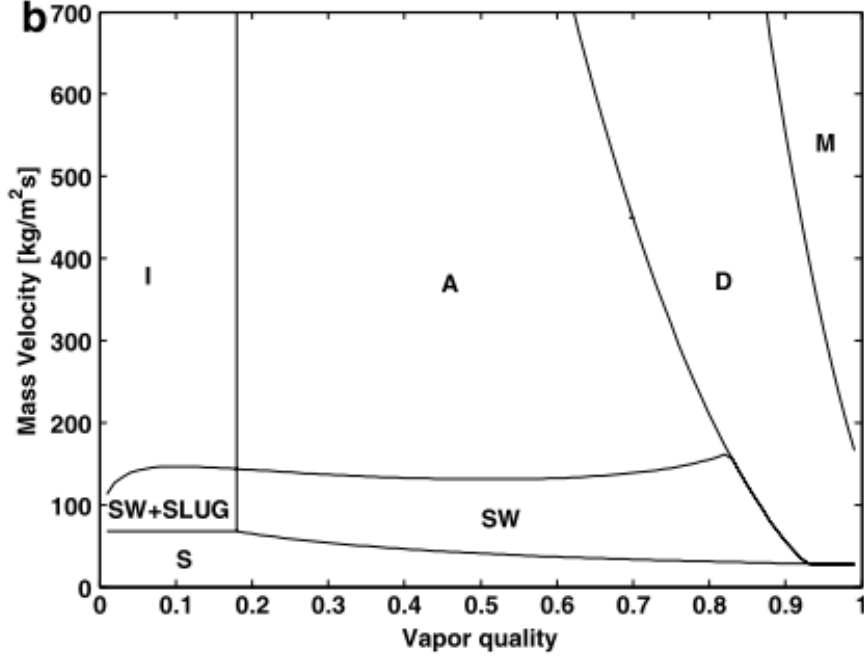


Figure 2.14: Improved pattern map for CO_2 ($D_h = 2.3 \text{ mm}$, $T_{\text{sat}} = 10^\circ\text{C}$, $q = 8 \text{ kW/m}^2$) based on the pattern map of Cheng et al. [31]

2.5.3 Prediction of frictional pressure drop

The frictional pressure drop model proposed by Cheng et al. comes from the adaptation of the Moreno Quiben and Thome [31–33] models (focused on R-22, R-410a and R-134a refrigerants) to a CO_2 pressure drop database.

The total pressure drop is assumed to be the sum of gravitational Δp_g , accelerational Δp_a and frictional pressure drops Δp_f , introduced earlier in the calculation of frictional pressure drop in two-phase flows.

In horizontal tubes, Δp_a is calculated as the following:

$$\Delta p_a = G^2 \left\{ \left[\frac{(1-x)^2}{\rho_l(1-\epsilon)} + \frac{x^2}{\rho_v \epsilon} \right]_{\text{out}} - \left[\frac{(1-x)^2}{\rho_l(1-\epsilon)} + \frac{x^2}{\rho_v \epsilon} \right]_{\text{in}} \right\} \quad (2.57)$$

The gravitational pressure drop Δp_g is calculated applying the equation 2.45.

$$\Delta p_g = \frac{g(\sin \beta)L}{\nu_{lv}x_e} \ln \left(1 + x_e \frac{\nu_{lv}}{\nu_l} \right) \quad (2.45)$$

with β as the angle made between the flow direction and the horizontal plane.

The frictional pressure drop will be inherently dependent on the flow pattern i.e. the pattern map.

Slug and intermittent flows pressure drops are calculated from the interpolation between the pressure drop expected in liquid only (LO) flows and the pressure drop calculated for annular flows at the Intermittent/Annular transition boundary (x_{IA}). The pressure drop calculation for liquid only flow is equivalent to what was proposed earlier for single-phase flows, in equation 2.10:

$$\Delta p_{LO} = \frac{2f\rho_m^2 L}{D_h} \quad (2.58)$$

$$\Leftrightarrow 4f_{LO} \frac{L}{D_h} \frac{G^2}{2\rho_l} \quad (2.59)$$

where the f_{LO} is the equivalent to the Fanning friction factor, thus $4f_{LO}$ will correspond to the Darcy friction factor in constricted flow presented earlier in section 2.2.3. However, the roughness effects are ignored by simply considering:

$$f_{LO} = \frac{0.079}{Re_{LO}^0.25} \quad (2.60)$$

$$Re_{LO} = \frac{GD_h}{\mu l}$$

The pressure drop interpolation for a Slug/Intermittent and Bubbly flow, from full liquid until a void fraction of ϵ_{IA} is calculated as follows:

$$\Delta p_{SLUG+I} = \Delta p_{LO} \left(1 - \frac{\epsilon}{\epsilon_{IA}}\right) + \Delta p_A \left(\frac{\epsilon}{\epsilon_{IA}}\right) \quad (2.61)$$

$$\Delta p_B = \Delta p_{LO} \left(1 - \frac{\epsilon}{\epsilon_{IA}}\right) + \Delta p_A \left(\frac{\epsilon}{\epsilon_{IA}}\right) \quad (2.62)$$

where the pressure drop for an annular flow Δp_A proposed is the following:

$$\Delta p_A = 4f_A \frac{L}{D_h} \frac{\rho_v u_v^2}{2} \quad (2.63)$$

$$f_A = 3.128 Re_v^{-0.454} We_l^{-0.0308} \quad (2.64)$$

Similarly to the annular flow, the pressure drop for a Stratified Wavy (SW) flow depends mainly on the vapour phase thermodynamic properties:

$$\Delta p_{SW} = 4f_{SW} \frac{L}{D_h} \frac{\rho_v u_v^2}{2} \quad (2.65)$$

with f_{SW} being somewhere in between the friction factor of single phase vapour f_v and the friction factor for an annular flow f_A , depending on the wet perimeter θ_{dry} (in annular, intermittent and bubbly flow, the perimeter is considered to be completely wet, thus a $\theta_{dry} = 0$):

$$f_{SW} = \left(\frac{\theta_{dry}}{2\pi}\right) f_v + \left(1 - \frac{\theta_{dry}}{2\pi}\right) f_A \quad (2.66)$$

The dry angle θ_{dry} in a stratified wavy flow is retrieved from the following equation, true for vapour qualities above x_{IA} :

$$\theta_{dry_{x>x_{IA}}} = \theta_{strat} \left(\frac{G_{wavy} - G}{G_{wavy} - G_{strat}}\right)^{0.61} \quad (2.67)$$

The dry angle for vapour qualities below x_{IA} is estimated from the following interpolation:

$$\theta_{dry_{x\leq x_{IA}}} = \theta_{strat} \frac{x}{x_{IA}} \left(\frac{G_{wavy} - G}{G_{wavy} - G_{strat}}\right)^{0.61} \quad (2.68)$$

The transition from Slug flow to SW follows the same logic as Δp_{SLUG+I} :

$$\Delta p_{SLUG+SW} = \Delta p_{LO} \left(1 - \frac{\epsilon}{\epsilon_{IA}}\right) + \Delta p_{SW} \left(\frac{\epsilon}{\epsilon_{IA}}\right) \quad (2.69)$$

The Mistflow is treated as a homogeneous flow that behaves like a single phase with thermodynamic properties derived from the homogeneous void fraction:

$$\epsilon_H = \left(1 + \frac{(1-x)\rho_v}{x\rho_l}\right) \quad (2.70)$$

$$\rho_H = \rho_l(1 - \epsilon_H) + \rho_v \epsilon_H \quad (2.71)$$

$$\mu_h = \rho_l(1 - \epsilon_H) + \rho_v \epsilon_H \quad (2.72)$$

$$Re_h = \frac{GD_h}{\mu_H} \quad (2.73)$$

the friction factor of mist flow was retrieved from the experimental data:

$$f_M = \frac{91.2}{Re_H^{0.832}} \quad (2.74)$$

The pressure drop for stratified flows was defined similarly to the stratified wavy flow. It is calculated as a linear interpolation from the pressure drop of single phase flow at low vapour quality to the pressure drop at the boundary x_{IA} :

$$\Delta p_{strat}(x < x_{IA}) = \Delta p_{LO} \left(1 - \frac{\epsilon}{\epsilon_{IA}}\right) + \Delta p_{strat}(x \geq x_{IA}) \left(\frac{\epsilon}{\epsilon_{IA}}\right) \quad (2.75)$$

where the $\Delta p_{strat}(x \geq x_{IA})$ is calculated similarly to Δp_{SW} , but considering θ_{strat} instead:

$$f_{SW} = \left(\frac{\theta_{dry}}{2\pi}\right) f_v + \left(1 - \frac{\theta_{strat}}{2\pi}\right) f_A \quad (2.76)$$

Finally, the pressure drop estimated for the dry-out is retrieved from the interpolation between the pressure drop right before (Δp_{tp}) reaching dry-out incipience (x_{di}) until the dry-out completion (x_{de}). The dry-out incipience may take place in an annular flow or stratified wavy and it will complete into a mistflow.

$$\Delta p_{dryout} = \Delta p_{tp}(x_{di}) - \frac{x - x_{di}}{x_{de} - x_{di}} [\Delta p_{tp}(x_{di}) - \Delta p_M(x_{de})] \quad (2.77)$$

The dry-out incipience, completion and Intermittent/Annular boundary vapour qualities, are calculated as follows:

$$x_{di} = 0.58 \exp \left[0.52 - 0.236 W e_v^{0.17} F r_{v,Mori}^{0.17} (\rho_v/\rho_l)^{0.25} (q/q_{crit})^{0.27}\right] \quad (2.78)$$

$$x_{de} = 0.61 \exp \left[0.57 - 0.502 W e_v^{0.16} F r_{v,Mori}^{0.15} (\rho_v/\rho_l)^{-0.09} (q/q_{crit})^{0.72}\right] \quad (2.79)$$

$$x_{IA} = \left[1.8^{1/0.875} \left(\frac{\rho_v}{\rho_l}\right)^{-1/1.75} \left(\frac{\mu_l}{\mu_v}\right)^{-1/7} + 1\right]^{-1} \quad (2.80)$$

with the vapour Weber and Froude numbers defined in section 2.1. The critical heat-flux is calculated with the Kutateladze correlation [34].

$$q_{crit} = 0.131 \rho_v^{0.5} h_{lv} [g\sigma(\rho_l - \rho_v)]^{0.25} \quad (2.81)$$

2.5.4 Prediction of heat transfer coefficient

The phenomenological approach of Thome [27] to predict the two-phase heat transfer coefficient (h_{tp}) is based on the Kattan, Thome, and Favrat [35] general equation:

$$h_{tp} = \frac{\theta_{dry} h_v + (2\pi - \theta_{dry}) h_{wet}}{2\pi} \quad (2.82)$$

where h_v is the heat transfer coefficient of the vapour phase on the dry perimeter, calculated from the often used Dittus-Boelter for single phase flows:

$$h_v = 0.023 Re_v^{0.8} Pr_v^{0.4} \frac{k_v}{D_h} \quad (2.83)$$

The heat transfer coefficient on the wet perimeter is retrieved from the following combination of nucleate (h_{nb}) and convective (h_{cb}) boiling heat transfer processes, for every pattern map but mistflow and dry-out:

$$h_{wet} = \left[(Sh_{nb})^3 + h_{cb}^3 \right]^{1/3} \quad (2.84)$$

where S corresponds to the heat suppression factor applied to the nucleate boiling heat transfer that takes places at vapour qualities after the Intermittent/Annular transition boundary, due to the thinning of annular liquid film:

$$\text{If } x < x_{IA}, S = 1 \quad (2.85)$$

$$\text{If } x \geq x_{IA}, S = 1 - 1.14 \left(\frac{D_h}{0.00753} \right)^2 \left(1 - \frac{\delta}{\delta_{IA}} \right)^{2.2} \quad (2.86)$$

$$(2.87)$$

here, δ represents the liquid film thickness in contact with the tube wall, which is calculated in the model using the expression proposed by El Hajal et al [28].

$$\delta = \frac{D_h}{2} \sqrt{\left(\frac{D_h}{2} \right)^2 - \frac{2A_l}{2\pi - \theta_{dry}}} \quad (2.88)$$

with A_l as the cross sectional area occupied in the liquid defined by the equation 2.47.

The nucleate boiling heat transfer coefficient h_{nb} is calculated with the Cheng-Ribatski-Wojtan-Thome proposed correlation [21]:

$$h_{nb} = 131 p_r^{-0.0063} (-\log_{10} p_r)^{-0.55} M^{-0.5} q^{0.58} \quad (2.89)$$

The convective boiling heat coefficient is calculated for every pattern map but mistflow and dry-out, using the following expression:

$$h_{cb} = 0.0133 Re_\delta^{0.69} Pr_l^{0.4} \frac{k_l}{\delta} \quad (2.90)$$

$$Re_\delta = \frac{4G(1-x)\delta}{\mu_l(1-\epsilon)} \quad (2.91)$$

with a liquid film δ that can go up to half the duct diameter $D_h/2$.

The heat transfer coefficient for the mistflow (h_M) steps out from the generic approach proposed in equation 2.82 for a more single-phase approach with homogeneous parameters, similarly to what was performed for the frictional pressure drop methods.

The heat transfer coefficient is calculated, considering the following expressions

$$h_M = 2 \times 10^{-8} Re_H^{1.97} Pr_v^{1.06} Y^{-1.83} \frac{k_v}{D_{eq}} \quad (2.92)$$

$$Y = 1 - 0.1 \left[\left(\frac{\rho_l}{\rho_v} - 1 \right) (1-x) \right]^{0.4} \quad (2.93)$$

$$Re_H = \frac{GD_h}{\mu_v} \left[x + \frac{\rho_l}{\rho_v} (1-x) \right] \quad (2.94)$$

the dry-out heat transfer coefficient is calculated from the interpolation between the heat transfer at the dry-out incipience ($h_{tp}(x_{di})$) to the dry-out completion ($h_M(x_{de})$):

$$h_{dryout} = h_{tp}(x_{di}) - \frac{x - x_{di}}{x_{de} - x_{di}}[h_{tp}(x_{di}) - h_M(x_{de})] \quad (2.95)$$

2.5.5 Heat transfer model confidence region

The semi-empirical method described in the previous sections, has been used throughout the thesis to predict the heat transfer coefficient of the CO₂ and is often referred by "Thome model". Similarly to other methods present in the literature, it relies on several correlations whose exponents are derived from fitting the predictions to the experimental database available. Hence, its precision is deeply dependent on the considered phase space.

The authors of the model concluded that the set of correlations were able to describe 80% their experimental database with a precision of $\pm 30\%$. The experimental database covered saturation temperatures from -28°C to 25°C , heat fluxes from 2 kW/m^2 to 46 kW/m^2 , tube diameters from 0.6 mm to 10 mm and mass velocities from $50\text{ kg/m}^2\text{s}$ to $1500\text{ kg/m}^2\text{s}$.

Figure 2.15 shows the database used to fit the model is well suited for the IBL detector operating conditions. However, in the barrel part of the ATLAS ITk, the heat flux values flowing from the pixel modules to the cooling fluid through the brazed local support can surpass 100 kW/m^2 .

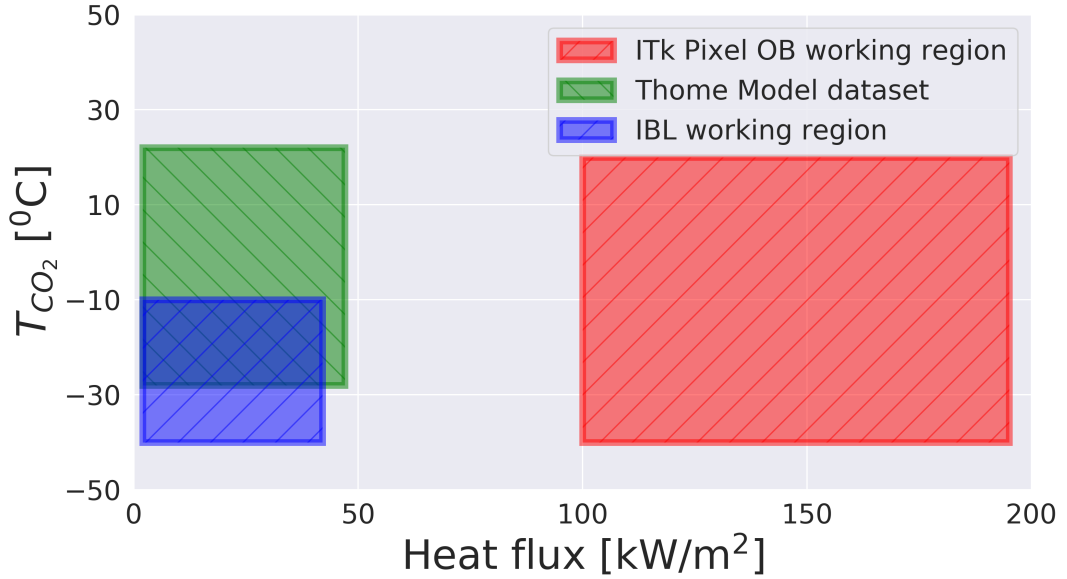


Figure 2.15: $(T_{CO_2}, \text{heat flux})$ phase-spaces for IBL and ITk Outer Barrel. The ITK Pixel Outer Barrel will work at much higher heat fluxes.

The model currently used to estimate the heat transfer coefficient (HTC) has never been validated for heat fluxes of this magnitude. This is particularly sensitive, given that the corresponding temperature gradient between the coolant and the tube wall (ΔT_{conv}) increases with the heat flux shown in Figure 2.16.

To cope with the lack of experimental data, several testing campaigns were conducted, to check if the model was still valid in the working region of interest for the future ATLAS ITk Pixel Detector.

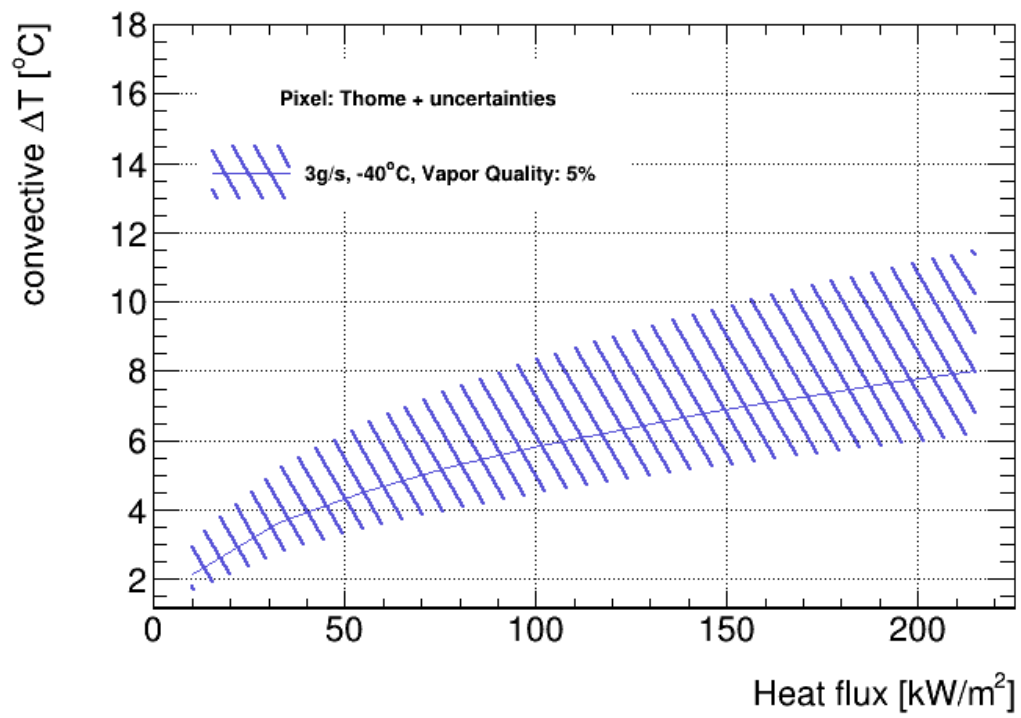


Figure 2.16: Zone of validation of the convective temperature gradient between CO_2 and tube wall predicted by the convective model varies with the heat flux at the tube wall.

CO₂ COOLING FOR TRACKING DETECTORS

Silicon detectors need to be cooled to keep their temperature stable, maximise the signal to noise ratio and to prevent against radiation damage dependent thermal runaway. In the previous generation of silicon detectors for Particle Physics built in the 1990s, they were cooled with water in single-phase mode in "leakless" sub-atmospheric pressure systems. The main drawback was the water's high freezing point (0°C) and low efficiency which makes difficult the task of cooling down detectors below the water's freezing point [36] and electrical conductivity, even with antifreeze added. One of the advantages of two-phase cooling is that the heat extraction does not change the temperature other than due to pressure drop along the evaporator. As the detector signal to noise ratio depends on the sensor temperature, the temperature uniformity across all the sensors will consequently benefit the uniformity of sensor behaviour.

To cool its silicon detectors around -7°C , the ATLAS ID considered initially an aqueous solution with a freezing point depressant (methanol) that used the latent heat of fusion of suspended microscopic ice crystals to improve the heat capacity and operated in a near isothermal process [36]. This idea was afterwards dropped in favor of an evaporative cooling system based on perfluorocarbons (C_3F_8 or R218) thanks to their more favourable thermo-physical properties, non-toxicity, non-flammability and radiation resistance. This solution was implemented for the SCT and the Pixel layers of the ATLAS detectors installed in the 2000s.

In the meantime, a new evaporative cooling solution based in CO_2 (R744 refrigerant) has been gaining popularity in the high energy physics community. This new technology has already been used to cool the ATLAS Inner B-Layer (IBL) detector installed in 2014 and has been chosen as the baseline to cool the entire ATLAS ITk to be installed in 2024.

Chapter outlook and personal contributions:

The present chapter describes in detail, the current and future cooling systems used to cool the silicon sensors, putting into context the state points in the P-H diagram with the actual thermodynamic processes taking place in the ATLAS cooling systems (current and future). Whilst the presentation of the arguments justifying the transition from C_3F_8 to a CO_2 based cooling system, several thermodynamic properties are compared. Some aspects of the flow distribution systems, especially important for the future ATLAS ITk Pixel Detector are introduced.

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3.1 ATLAS cooling system

The current components of the ATLAS ID (Pixel and SCT) use a fluorocarbon based evaporative cooling system [37]. The temperature of the sensors has to be kept permanently below the required -10°C for the SCT and 0°C for the Pixel detector.

Amongst the various candidates, octafluoropropane (C_3F_8 or R218) has been selected for its adapted thermo-physical properties [38]: good chemical stability under high radiation environments, non-flammability, non-corrosivity, non-toxicity and non-ozone depleting behaviour. Compared to a water mono-phase based system, the evaporative cooling system is able to operate at much lower temperature (down to -25°C) and with 10 to 20 times less mass flow to evacuate ≈ 60 kW of heat dissipated in the module front-end electronics. This has been possible thanks to its much higher phase change heat capacity than the monophasic heat capacity of water.

3.2 Thermosiphon recirculator

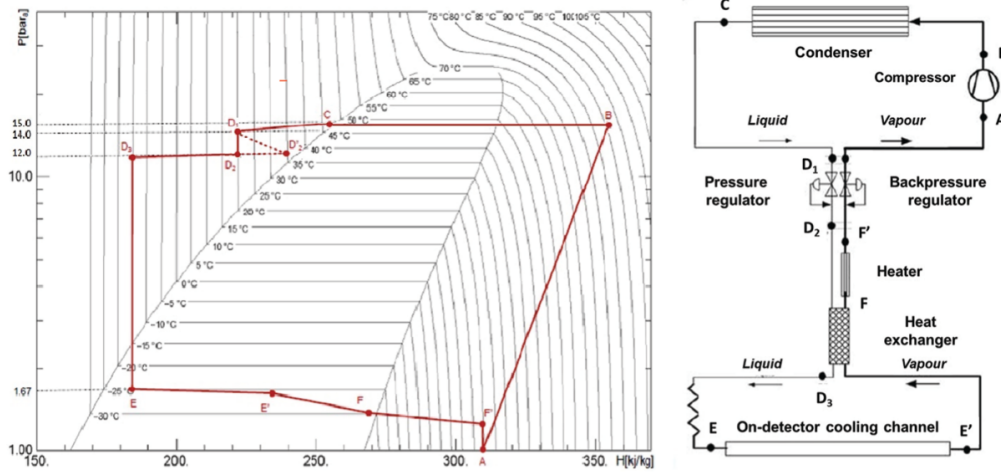


Figure 3.1: Thermodynamic cycle of the present compressor-driven C_3F_8 evaporative cooling system and the corresponding components in the cooling circuits. [38].

The ID cooling system was initially based on a typical compression-condensation cycle - see Figure 3.1. The compressors used in-line with the main cooling loop had to be oil-free to avoid contamination of the coolant. Oil-free compressors are known to be more fragile and require considerably more maintenance than oil-compressors, especially when used in extreme temperatures and for long periods of time. Considering it takes such an important role on the cooling cycle, relying on the reliability of the compressor was considered too risky and the lowest attainable temperature was higher than expected due to an underestimation of the pressure drop on the return line (from point F to A on the schematics). The change to a gravity-driven thermosiphon recirculator (Figure 3.2) simplified the system, reduced the amount of seals needed

to avoid leaking and the pollution in the system from particles produced by active elements in the primary loop like the piston rings of the compressor [38].

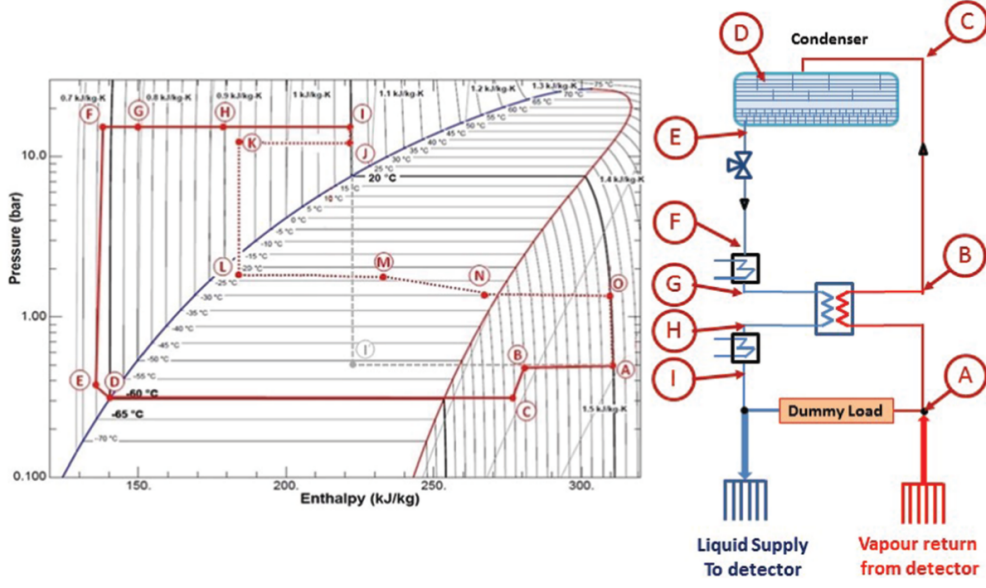


Figure 3.2: Thermodynamic cycle of the full scale thermosiphon system [38].

With the thermosiphon, the refrigerant is condensed and stored at the surface (point E), in a sub-cooled state (around -65°C and 0.4 bar). To reach the highest pressure point of 16.1 bar [39] in the diagram (from point E to F), the system takes advantage of the hydrostatic pressure difference along the ≈ 92 m depth between the ATLAS detector and the condenser storage unit. The point I represents the fluid state being supplied to the evaporator volume. The grey dashed lines represent the commissioning thermodynamic cycle with the fluid being bypassed through a separate circuit with a dummy load, responsible for the pressure drop and enthalpy change from point I to A. The fluid passes then through a heat exchanger coupled to the G-H section, so the fluid at point B reaches a temperature close to point H. Finally, it flows to the condenser, responsible to evacuate all the heat, sub-cool and restore the fluid to the liquid state.

When not passing through the dummy load, the system feeds the individual detector circuits through sub-cooling heat exchangers (as in the compressor system) between detector inlet and the return line (symmetric enthalpy change of J-K and A-B sections). The pressure drop along the capillaries are represented by the section K-L and the heat exchange between inlet and outlet sets the point M at same temperature as point K. Thanks to the use of local sub-cooling heat exchangers, the inlet state K is sub-cooled, lowering the vapour quality at the entrance of the evaporator section at point L. This is an important feature, needed to be taken into account during the design of a cooling system, that allows to reduce the chances of reaching dry-out in the evaporator section.

3.3 Migration from C_3F_8 to CO_2

The design of an evaporative cooling system for silicon trackers goes beyond the typical engineering constraints of most of the systems of refrigeration: available space, materials used, cost, reliability, amount of heat to evacuate, maximum allowable temperature on the heat source, design of heat sink, etc. On top of this, it is particularly important to fulfill the following specifications, when the goal is to cool silicon sensors placed inside the tracker volume:

- Minimize material in radiation length unit X/X_0 (mass of tubes and refrigerant);

- Keep sensors' temperature stable;
- Temperature uniformity across all sensors;
- Ability to route pipes in confined spaces;
- Reliable operation under harsh radiation environments;
- Remote monitoring and control of the system, far from the detector;
- Temperature stability and well controlled.

The C_3F_8 (or $\text{C}_3\text{F}_8/\text{C}_2\text{F}_6$ blend) system with a thermosiphon recirculator would have been able to successfully fulfil these constraints. The main reason for the future ATLAS ITk to operate with CO_2 is its better cooling performances with smaller tubes at low temperatures ($\approx -40^\circ\text{C}$).

The many favourable thermo-physical properties of CO_2 are going to be crucial for the future ATLAS ITk which will be entirely composed of silicon, with a total power dissipation estimated around 250 kW [9]. This is much higher than the current 60 kW [38] produced by the entire ID, raising the need for a more efficient refrigerant instead of scaling up the current system by a factor more than four.

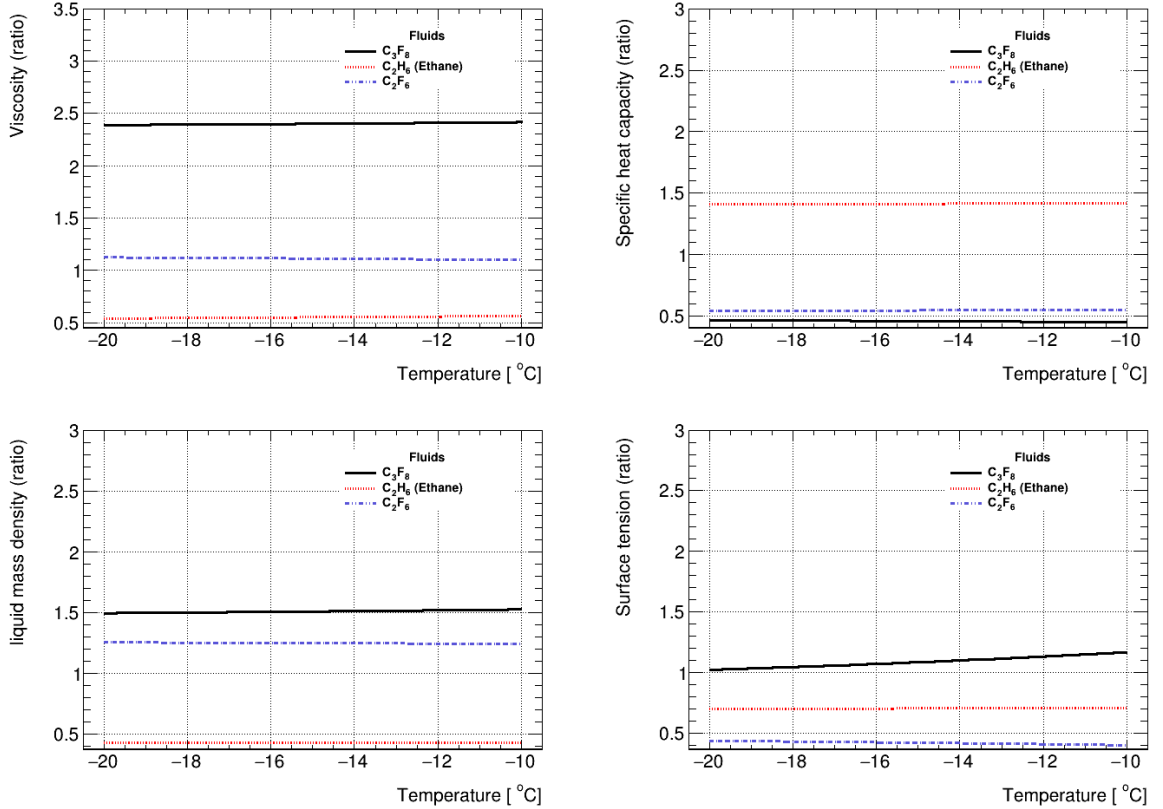


Figure 3.3: Comparison of thermodynamic properties as ratio between the each of the fluids and CO_2 . The properties were calculated using the CoolProp python package [40].

Relevant thermo-physical properties of several refrigerants are compared with CO_2 in Figure 3.3 and Figure 3.4. The liquid CO_2 presents higher specific heat capacity, lower viscosity and higher surface tension than most of the other refrigerants (the good properties of the *Ethane* are commented further). The specific heat is responsible for the overall thermal performance of the fluid as it impacts the performance of the liquid in contact with the inner wall of the pipe.

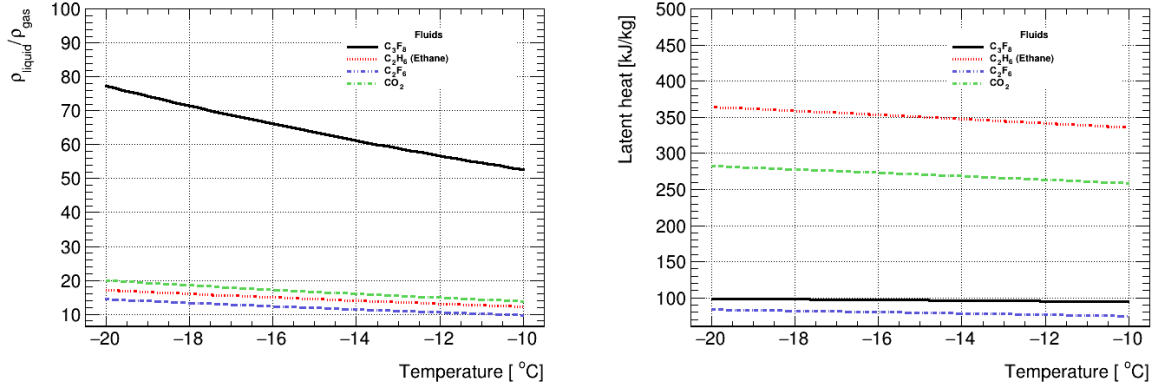


Figure 3.4: Comparison of liquid/vapour density ratio and latent heat between refrigerants. The properties were calculated using the CoolProp python package [40].

Finally, the low viscosity and low surface tension facilitates the boiling process, responsible for enhancing the heat exchange between the gas and liquid phase.

Figure 3.4 shows the density ratio between liquid and vapour phase and latent heat for the same refrigerants. A high density ratio and low latent heat of the C_3F_8 compared to other fluids is particularly problematic. In evaporators where a significant amount of heat is produced, the lower the latent heat, the faster the vapour quality increases. The system must thus operate with at a higher mass flow to avoid reaching dryout. On top of this, the large density ratio between liquid and vapour leads to larger pressure drop along the cooling lines, causing a non homogeneous temperature profile along the evaporators.

The temperature profile is nonetheless, not the only parameter dictating the thermal performance in an evaporator. Figure 3.5 shows that, ultimately, it will be the maximum temperature gradient ($\Delta T_{\Delta P+HTC}$) the variable of interest.

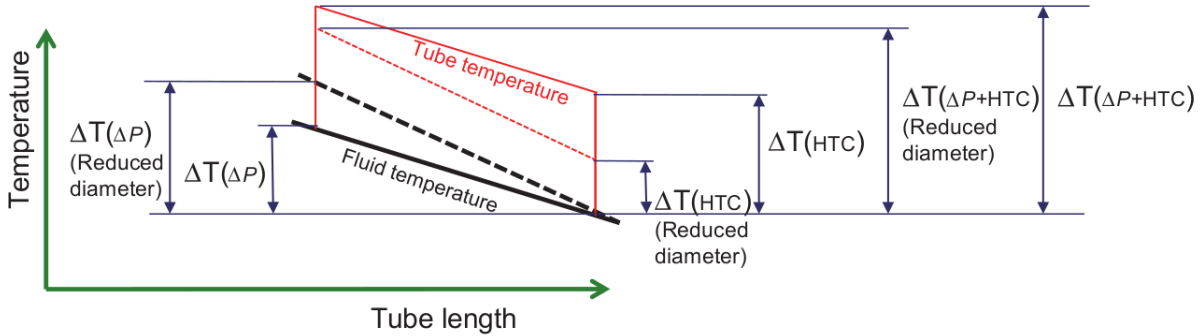


Figure 3.5: Temperature profile estimated along tube length of two pipes cooled with two-phase fluid with different diameters. The black solid and dashed lines represent the decrease of fluid's temperature along the tube due to frictional pressure drop. The temperature variation in the fluid is represented by $\Delta T(\Delta P)$. The red lines illustrate the tube temperature. The temperature difference between tube wall and fluid is represented by $\Delta T(HTC)$. The temperature difference between the lowest temperature of the fluid and warmest temperature in the tube is represented by $\Delta T(\Delta P + HTC)$ [41]

A way to compare the overall performance of different fluids is to plot an alternative combined heat transfer coefficient that takes into account the warmest spot along a given setup [42] as a function of the tube diameter.

$$HTC_{combined} = \frac{q}{\Delta T_{total}} \quad (3.1)$$

This exercise illustrated in Figure 3.6 shows that, not only the optimal tube diameter is smaller for CO₂ compared to the other options, but it also presents a relatively better thermal performance. A similar approach can be performed to estimate the optimal pipe size to be user in a given setup. This will be explored further in section 4.5.

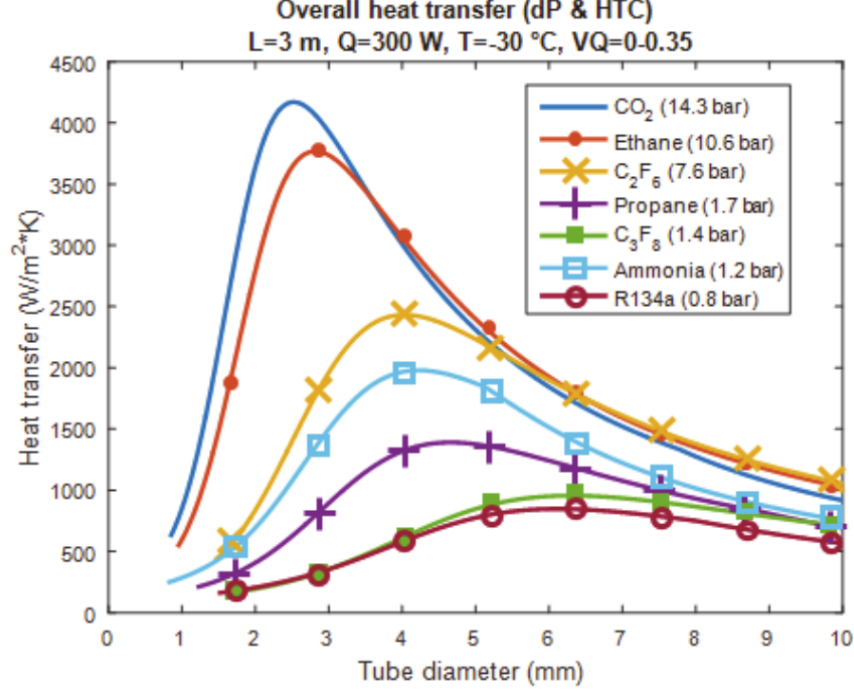


Figure 3.6: Combined heat transfer coefficient ($HTC_{combined}$) calculated for different fluids for a specific setup. The pressure drop is calculated with the Friedel correlation [43] and the heat transfer coefficient with the Kandlikar correlation [44].

The specific heat at constant pressure of the liquid and its mass density impact directly the performance of the fluid. With a similar specific heat, a fluid more dense, will present lower temperature gradient in a given volume of fluid. Also, the vapour stays more compressed, with a lower ρ_l/ρ_g ratio, leading to lower volume flows and smaller relative speed between gas and liquid phase as the the vapour quality increases.

The liquid specific heat and viscosity depend mostly on the fluid's reduced pressure (see Chapter 2) and not necessarily on its pressure. *Ethane*, for example, presents lower reduced pressure than CO₂. Hence, its higher specific heat capacity and lower viscosity. It is the higher density of CO₂ that leverages its performance when compared to the *Ethane*. Its performance is eventually surpassed at bigger tube diameters, because the lower viscosity and ratio ρ_l/ρ_g of *Ethane* start to dominate over the density. Regardless its good cooling performance, *Ethane* would never be used in a secondary application like cooling underground detectors due to its flammability.

CO₂ is therefore the current best choice, not only because it shows better thermal performances but also because it is cheap, radiation hard, not toxic and has a low greenhouse potential [42]. The main challenge to use CO₂ is its high working pressure between 10 bar and 20 bar (from -40°C to -20°C) which is way higher than the low operating pressures of 400 mbar to 3 bar with C₃F₈. One must be more careful to not overhead the pump heads and the seals must be able to withstand the high pressure to avoid leaks.

3.4 Radiation length: CO₂ vs C₃F₈

When a particle crosses material inside the detector volume, it will loose energy and deviate from its original direction due to multiple scattering. The loss of energy and multiple scattering degrades the precision of track reconstruction. Thus, the mass crossed by the particle inside the tracker volume must be minimised.

The reduction of the energy of an electron crossing a material is described by its radiation length. It has length units (typically in *cm*) and represents the mean length of material that one electron needs to traverse to reduce its energy by the factor $1/e$. Therefore, one should prioritize the use materials with a long radiation length inside the tracker volume.

The formula used to calculate the radiation length (in *cm* units) for an element is the following [45, 46]:

$$X_0 = \frac{716.4 \times A}{Z(Z+1) \ln \frac{287}{\sqrt{Z}} \times \rho} \quad (3.2)$$

where Z is the material's atomic number, A the mass number and ρ the materials mass density in g/cm^3 units.

In the case of CO₂ and C₃F₈, the radiation length can be calculated from the equation:

$$\frac{A_0 N_0}{X_0} = \sum \frac{A_i N_i}{X_i} \quad (3.3)$$

where A_0 and A_i are correspondingly the average atomic mass and the atomic mass of each constituent, both in *g/mol*. The N_0 and N_i are correspondingly the total number of moles and the number of moles of each constituent.

Considering the liquid densities of both coolants at -40°C we obtain a radiation length of 32.4 cm for CO₂ and 21.2 cm for C₃F₈.

On top of the longer radiation length, the optimal tube diameter found for CO₂ is two times smaller than the optimal diameter for C₃F₈. Hence, a cooling system optimised for CO₂ will have around a third of relative X/X_0 ¹ compared with the C₃F₈ (considering tubes equally thick) which is a major argument favouring the use of CO₂.

3.5 CO₂ 2PACL cooling system

Two possible approaches for designing cooling systems are shown in Figure 3.7. The traditional method is the vapour compression system used in the early days of the ATLAS ID as described before. The evaporator is placed inline with the refrigeration cycle, relying on controlled heaters to vaporize the remaining liquid before exhausting to the compressor. One of the advantages of this method, is that warm transport lines do not demand proper insulation releasing space constraints. The main disadvantages are the need of active elements heaters inside the detector volume and an oil-free compressor to avoid polluting the circuit [47]. To solve these limitations, the system migrated for a termosiphon recirculator as described in section 3.2.

The second possibility denominated by 2PACL (two-phase accumulator controlled loop) is to feed directly the evaporator using a liquid pumped system, from an external sub-cooled source. The main limitation is that the cold transfer lines need to be well insulated while the main advantage is that the system does not rely on any active element inside the detector area². Nonetheless, the possibility to place all the hardware away from the detector while still controlling the cooling system in the detector was the major argument to consider this approach to cool IBL.

¹The material thickness normalized to its radiation length, X/X_0 , is commonly taken as parameter of reference to compare the effective radiation length of two different materials.

²By principle, compressor and termosiphon systems do not need active elements inside detector volume although ATLAS uses electrical heaters.

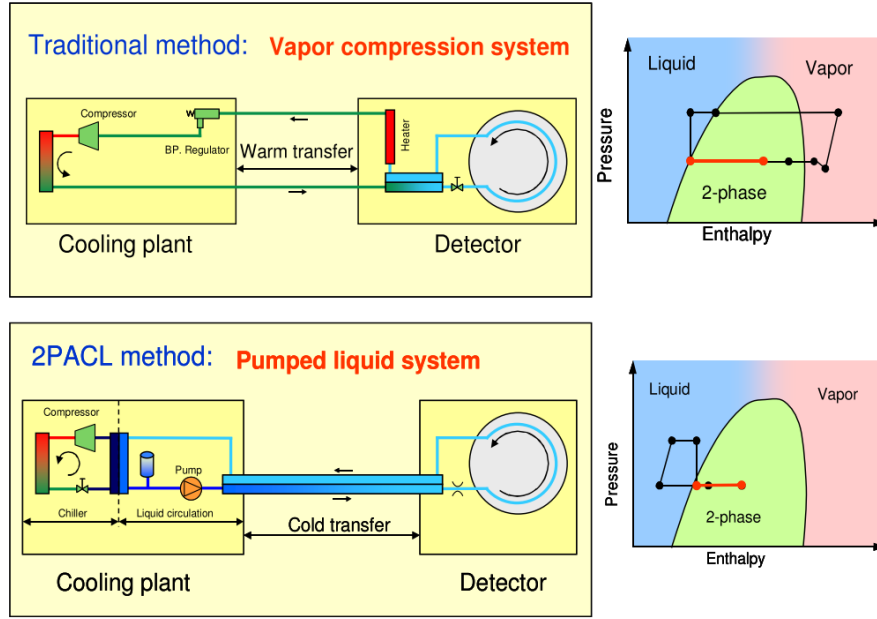


Figure 3.7: Principle of operation of cooling systems based on vapour compression and liquid pumping [47].

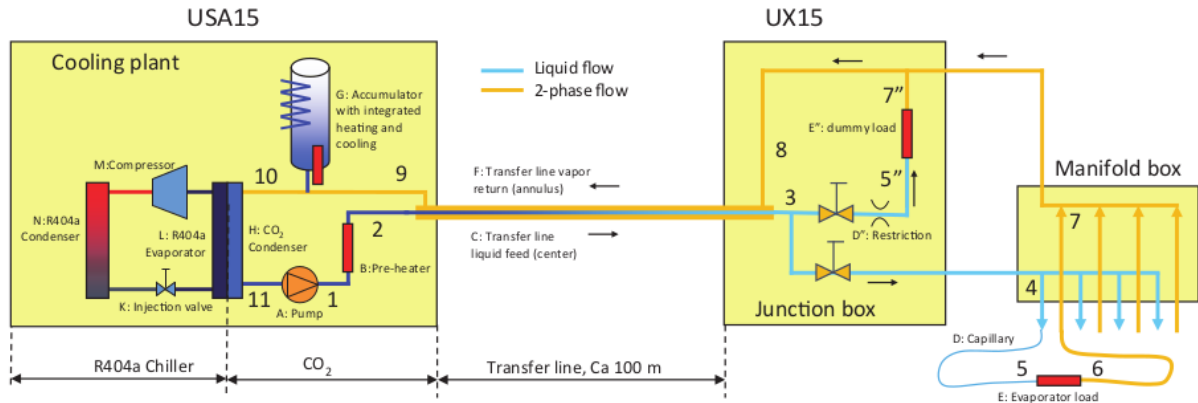
Another important feature is the ability to separate the chiller circuit from the main cooling loop. This novel method was invented during the design of the AMS02 Tracker Temperature Control System (TTCS), launched to the International Space Station in 2011 [48] and LHCb-VELO Temperature Control System (TCS) [47]. However, it was during the operation of LHCb-VELO that the method has been validated in real conditions for the first time.

Some years later this new cooling concept was successfully implemented in the IBL detector [49] and in CMS pixel "phase-I" cooling system [50]. The systems have been proven very reliable as they experienced very little downtime since their installation. The schematic shown in Figure 3.8 illustrates how the cooling plant used to cool IBL, placed in the services cavern (USA15), is connected via cold transfer lines to the experimental cavern (UX15), closer to the detector.

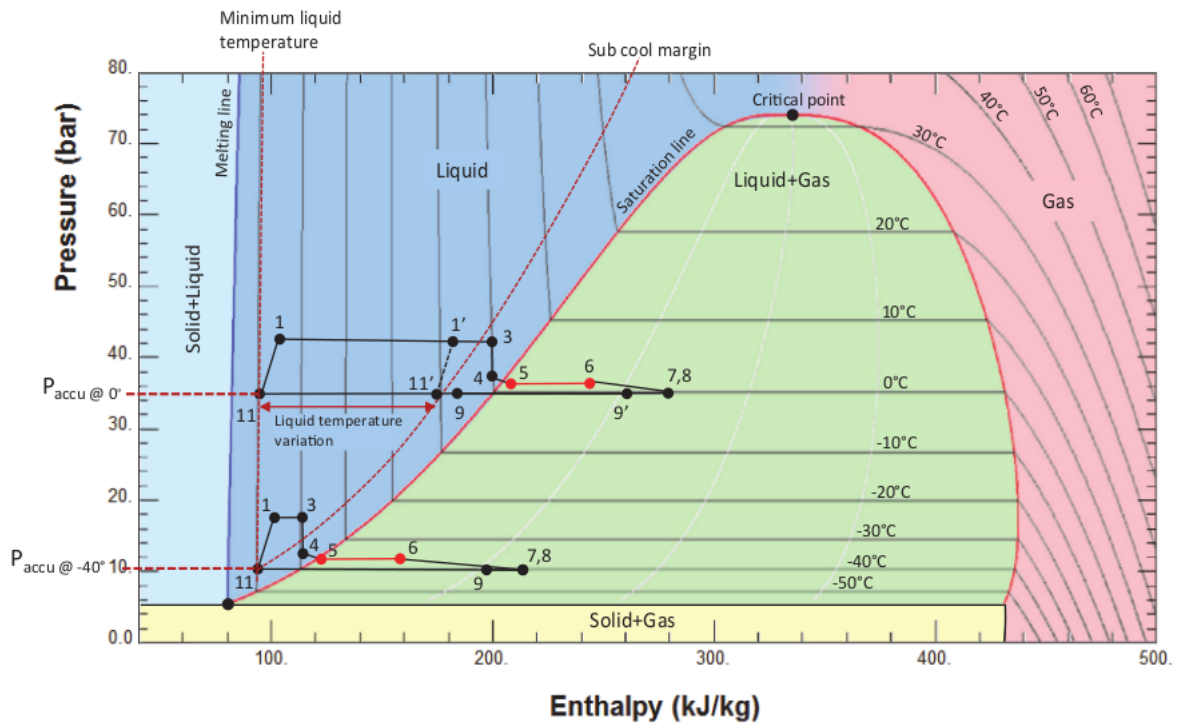
To understand the full refrigeration cycle shown in Figure 3.8, one can start at the pumped liquid (P1), sub-cooled by the condenser. It flows through concentric transfer lines which act like a heat exchanger with the liquid feed lines surrounded by the vapour return line. The vapour quality on the return line ranges typically from 30% to 50%, which still guarantees a good heat exchange in the transfer lines. As the temperature on the return line is higher than the sub-cooled liquid, the pumped liquid is going to get warmer (P1-3) until it thermalises close to the temperature of the return line (depends on the thermal exchange efficiency).

Now at the junction box (P3), the manifold is fed with liquid CO_2 to facilitate the distribution across the parallel cooling lines. The pressure drop and enthalpy change until a two-phase flow enters the evaporator at (P3-5). For the ATLAS ITk this will be regulated by a warm nose system explained in the next section.

The fluid returning from the evaporator to the accumulator (P7-10), flows through big pipes, to minimise the pressure drop. The pressure drop is expected to be minimal so the pressure set in the accumulator will dictate the temperature on the return line. As long as the return line of the evaporator respects the maximum vapour quality requirements, the evaporator loops are somehow independent of the main cooling loop and the cooling conditions at the evaporator level can be controlled by active elements (pumps, accumulator, pre-heater and warm nose heaters) far from the detector volume.



(a)



(b)

Figure 3.8: At (a) is shown scheme locating the various parts of the cooling system for ATLAS [49] and at (b) the P-H diagram with the corresponding state points.

In the meantime, all the critical active elements like pumps, compressor, chiller and accumulator are kept easily accessible for intervention. In UX15 a few active elements remain to control CO₂ feeding the manifold box which distributes the fluid among the detector cooling lines, as well as a parallel circuit that can bypass the experiment cooling loop and use a dummy load and a flow restrictor for system commissioning purposes.

3.6 Two-phase accumulator

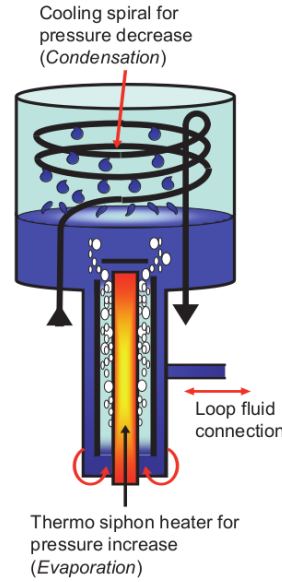


Figure 3.9: *Accumulator concept with integrated evaporator and condenser used in the 2PACL cooling systems [42].*

The accumulator in a 2PACL configuration, is a versatile CO₂ storage unit, capable of condensing gas and boiling the liquid in its interior. Before the system start-up, there is only liquid in the condenser and accumulator, where CO₂ is permanently cooled by the chiller. Before turning on the circulation, the accumulator boils the liquid with a heater, accumulating gas at the top of the vessel from evaporation. This will provoke an increase of pressure (due to the liquid-mass difference of mass densities), pushing the liquid out of the accumulator to the cooling lines. The liquid will gradually fill the system, pressurizing and heating the remaining gas present in the lines. Eventually, the gas will reach the two-phase region and condense at room temperature.

The accumulator is designed to maintain a minimum level of liquid CO₂ otherwise the heater will start heating the gas instead. When the system is filled with liquid, the pumping system is turned on and the accumulator can stop heating and rely on its integrated condenser (spiral wherein flows cool CO₂ from the condenser) to cool the gas and tune its pressure.

As explained before, the pressure in the accumulator is close to the pressure of the evaporator outlet. When the pressure is low enough, so the evaporator is fed with saturated liquid (cycle C at Figure 3.10), the accumulator pressure set point will dictate the evaporator outlet temperature.

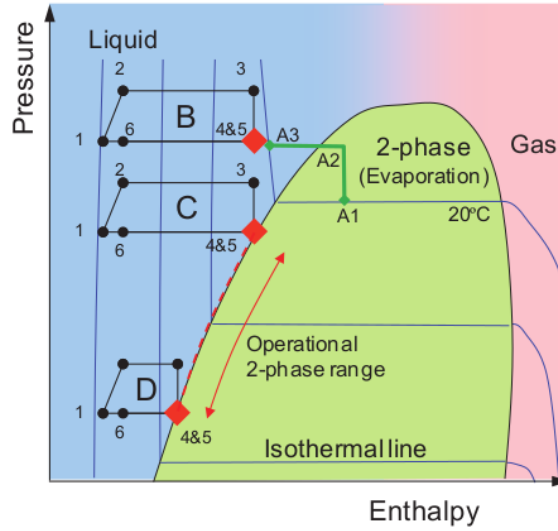


Figure 3.10: 2PACL cooling system refrigeration cycles from startup (the main loop is fully liquid) with continuous decreasing of accumulator pressure until reaching the saturation line (operational 2-phase range) [42].

3.7 Future ATLAS ITk CO₂ cooling system

A single cooling plant was enough to cool 1 kW [49] produced by the Inner B-Layer Detector. This cannot be scaled to cool the 250 kW expected to be dissipated by close to 1000 evaporators in the ITk [9].

There will be seven individual cooling plants (one of them is a spare unit for redundancy purposes) with two of them dedicated to cool the Pixel Detector: one for the Barrel and the other for the End Cap.

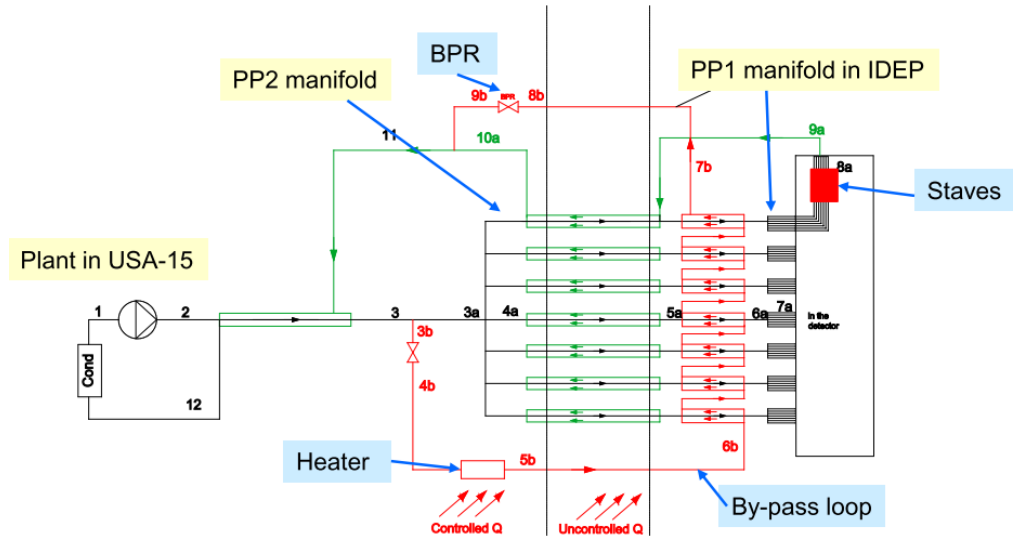


Figure 3.11: Scheme representing the warm nose concept from one side of the detector, where a heated bypass loop (in red) is used to thermalise the inlet capillaries before the manifold at PP1. [9]

The transport of the refrigerant from the cooling plants located in the USA-15 room illustrated in Figure 3.11.

For the Barrel, each of the two of available plants has 69 kW of refrigeration power. The

main transfer line will be divided at PP2 (second patch panel) into sixteen concentric flex lines³ (eight per side) with an average cooling power of 4.3 kW each.

Closer to the detector, at PP1, the flex lines are connected to splitter boxes placed at each of the ID end plates (IDEP). The splitter boxes convert the concentric flex lines in two separated lines (liquid feed and two-phase return lines), which are then routed to manifolds responsible to distribute over several parallel detector loops.

It is important that all the distribution lines share the same pressure drop. The ITk cooling system has been designed taking in consideration a total pressure drop of 10 bar per cooling loop, with a maximum budget of 2 bar in the evaporator section. The total pressure drop will be ultimately dictated by the sizing (length and diameter) chosen for the pipes and the power dissipated in the evaporator loops (explored in section 3.9). For this reason, each cooling plant is typically dedicated to one of the subsystems, where the cooling lines are expected to have similar lengths and the evaporators the same power dissipated.

3.8 Warm nose scheme

To guarantee a two-phase state at the inlet of the evaporators, the cooling system foresees a warm nosing system that regulates indirectly the inlet vapour quality, installed at PP2, thus outside the detector volume. This is represented at Figure 3.11 by the red boxes surrounding the inlet tubes at PP1.

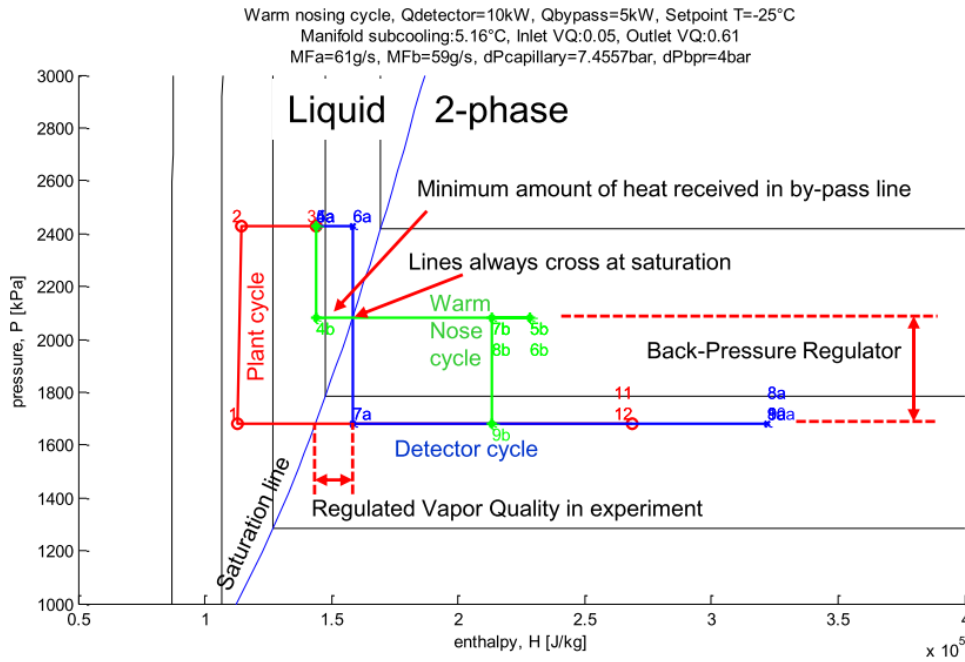


Figure 3.12: *P-H diagram describing how the warm-nose bypass loop regulates the inlet vapour quality in the experiment [9].*

The principle of operation, illustrated in the complementary Figures 3.11 and 3.12, is based on a back pressure regulator to set the temperature of the liquid lines at the point where they

³Flexible tube comprising an inlet tube with the refrigerant used to cool the experiment, placed inside a larger tube with the fluid returning from the experiment. The tubes are thermally in contact, acting like a heat exchanger and the outermost tube may be insulated from the exterior if needed. The ability to couple inlet, return line and a heat exchanger in a single component, highly simplifies the system.

enter the PP1 manifolds. In this way, one can control the temperature of the liquid before the manifolds and hence, regulate the vapour quality at the experiment entrance [9].

3.9 Flow distribution

Particle detectors usually contain several parallel cooling branches responsible to cool different evaporators manifolded together. The manifolds, placed far from the detector volume, distribute the refrigerant (still liquid) to the evaporators via long tubes.

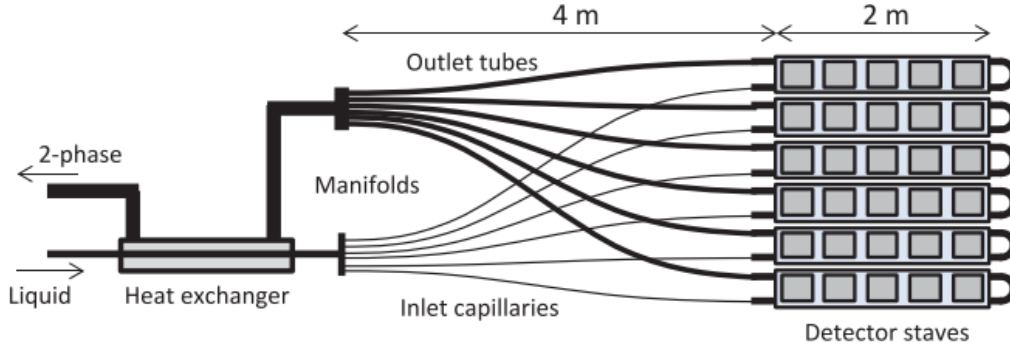


Figure 3.13: Typical layout of a cooling system of a particle detector with six identical parallel cooling lines composed of an evaporator section preceded by an inlet capillary and followed by an outlet tube. The coolant is fed in liquid phase and sub-cooled by the return line via an internal heat exchanger setting the liquid at the inlet a two-phase flow in the return line at the same temperature [42].

A group of manifolded evaporators, like the ones illustrated in Figure 3.13, share the same pressure drop. If all the evaporators are designed with the same pipe dimensions and dissipate the same power, the pressure drop and mass flow is expected to be the same for all. However, in case there is one evaporator absorbs less heat, the amount of vapour inside its pipe will reduce, as its pressure drop. The pressure unbalance is compensated by an increase of the mass flow, reducing the mass flow in the remaining loops.

The risk in the case of particle detector cooling, is that sharing the fluid from the operating branches may cause dry-out in some of them. The solution found to overcome this problem, is to manifold all the evaporators in series with individual inlet capillaries sharing similar design.

With an inlet pressure drop at the capillaries higher than the pressure drop expected at the evaporators, the overall pressure drop at the manifold becomes less sensitive changes at the evaporator. They have also a second role of provoking a pressure drop so that we enter the two-phase region inside the evaporator section.

To design manifolds, the chosen size of capillaries takes into account the flow unbalance but also the pressure drop needed to enter the evaporator in two-phase regime. In section 4.6 the tool and the method used to calculate the flow distribution in manifolds with multiple branches will be presented.

3.10 Outer Barrel manifolding scheme

In the Outer Barrel there will be two main types of structures holding the pixel modules: longerons and half inclined rings. The total power of ≈ 52 kW in the outer barrel considers an average of 0.7 W/cm² power dissipated in each of the 2236 quad pixel modules.

The longerons hosting 36 quad modules will dissipate around 404 W while the half inclined rings will dissipate from 180 W to 325 W (units at higher radius host more modules). Each one of the sixteen cooling lines is expected to be shared between either longerons or half inclined

rings to avoid fluid distribution issues. The current manifolding scheme considers a grouping of seven longerons per manifold for each side. Thus, a total six of the sixteen lines dedicated to feed the thirty four longerons loops. The remaining ten cooling lines will be shared between the ninety-two half inclined rings with manifolds grouping nine loops each (two of the manifolds will need to group ten instead to cope all the inclined ring units).

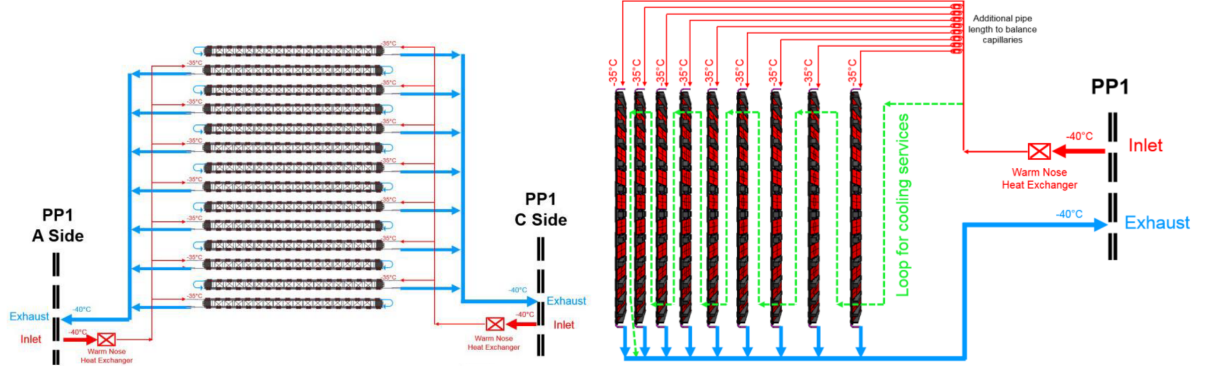


Figure 3.14: *Manifolding scheme for the longerons and inclined half rings for the ATLAS ITK Pixel Outer Barrel.*

Figure 3.14 illustrates how the half inclined rings and longerons will be manifolded, and Figure 3.15 shows the length for each pipe section (inlet capillary, connector, evaporator and outlet tube) for the longerons. The lengths are expected to be close to the ones stated in the Figure but may change slightly as it will depend on the access to the longeron. Nevertheless, each group of eight should share the same design.

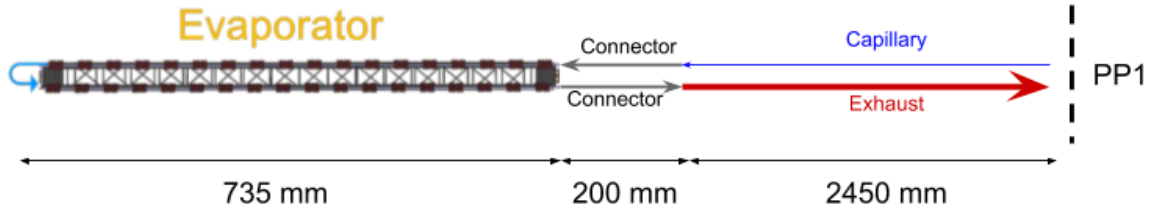


Figure 3.15: *Pipe lengths for the Outer Barrel longerons cooling loop.*

CO₂ COOLING SIMULATION TOOL

To design CO₂ cooling systems, the CERN cooling group have been performing simulations predicting the coolant behaviour with a tool developed in MATLAB called CoBra (stands for CO₂ branch calculator) [51].

Among other applications, this tool has been determinant for the optimization of IBL tube dimensions [52] and to properly predict the tube wall temperature profile along its cooling lines.

Nevertheless, there was a recurrent need to compute the heat transfer coefficient, to compare with the experimental data retrieved from several testing campaigns conducted at LAPP. Hence, a custom made tool, inspired by CoBra, was developed in the python programming language, implementing the semi-empirical methods described in section 2.5 to estimate the heat-transfer coefficient and pressure drop.

The implementation in python was versatile and easily coupled with the measurements which was determinant to compare huge datasets with the estimates and retrieve conclusions about the ability to explain the data with the currently used model.

Chapter outlook and personal contributions:

This chapter is focused on the description of the developed custom tool, with some examples of how it can be used to simulate real case scenarios. The tool is validated with experimental data, used to optimise the tube design of the pipe diameter of the future Pixel Detector and simulates the flow distribution in manifolds. The web interface developed as a front-end for simple computations and made available to the ATLAS collaboration is also presented.

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4.1 Steady state iterative algorithm

The program is able to simulate the heat along a cooling line composed of several cylindrical pipes connected in series, which one can call elements. Each element is unique and owns the following attributes as input:

- Name
- Inner Diameter [m]
- Tube wall roughness [m]
- Length [m]
- Angle with the vertical plane [°]
- Bending angle [°]
- Local heat flux [W/m²]
- Effective heated surface (heat flux may be applied to only one part of the tube) [m²]

Each element may be divided in several other child sub-elements, sharing the parent properties. This is usually integrated in the calculation for a more refined iterative extrapolation on the enthalpy and pressure change along the flow stream.

In a cooling line, the fluid properties inside each element are changed in an iterative process due to the frictional pressure drop along the tube and heat absorption. The fluid properties are dictated by boundary thermodynamic conditions, defined at the first element in the series: pressure, enthalpy and mass velocity. The remaining elements are created as extensions from the inlet element, without need for more information about the coolant's state.

A cooling line is considered a sequence of N individual elements as illustrated in Figure 4.1 where the pressure and enthalpy of element k , is calculated from the pressure and enthalpy of the first element (set by the boundary conditions) and the integrated change along the next k elements.

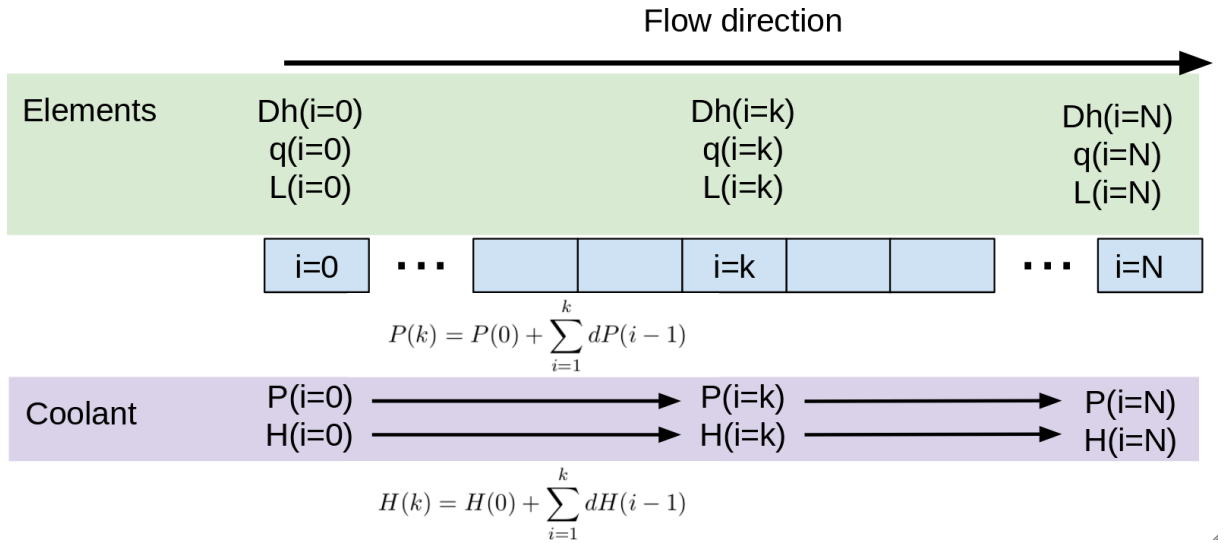


Figure 4.1: Illustration of the iterative process used to calculate the change of the thermal properties of the refrigerant along a cooling line.

At the beginning of the iterative process, only the first two sub-elements have information about the flow thermodynamic properties. At each iterative step, two subsequent sub-elements are evaluated ¹, to calculate the enthalpy and pressure of the next one. The process jumps then to the next pair of sub-elements in the series and redo the calculation. The fluid is considered in-compressible, thus the sectional mass flow is considered constant along the pipe.

¹To compute the accelerational pressure drop one needs the information about two consecutive elements (inlet and outlet) as shown in equation 2.57 on page 34

4.2 Program description UML diagram

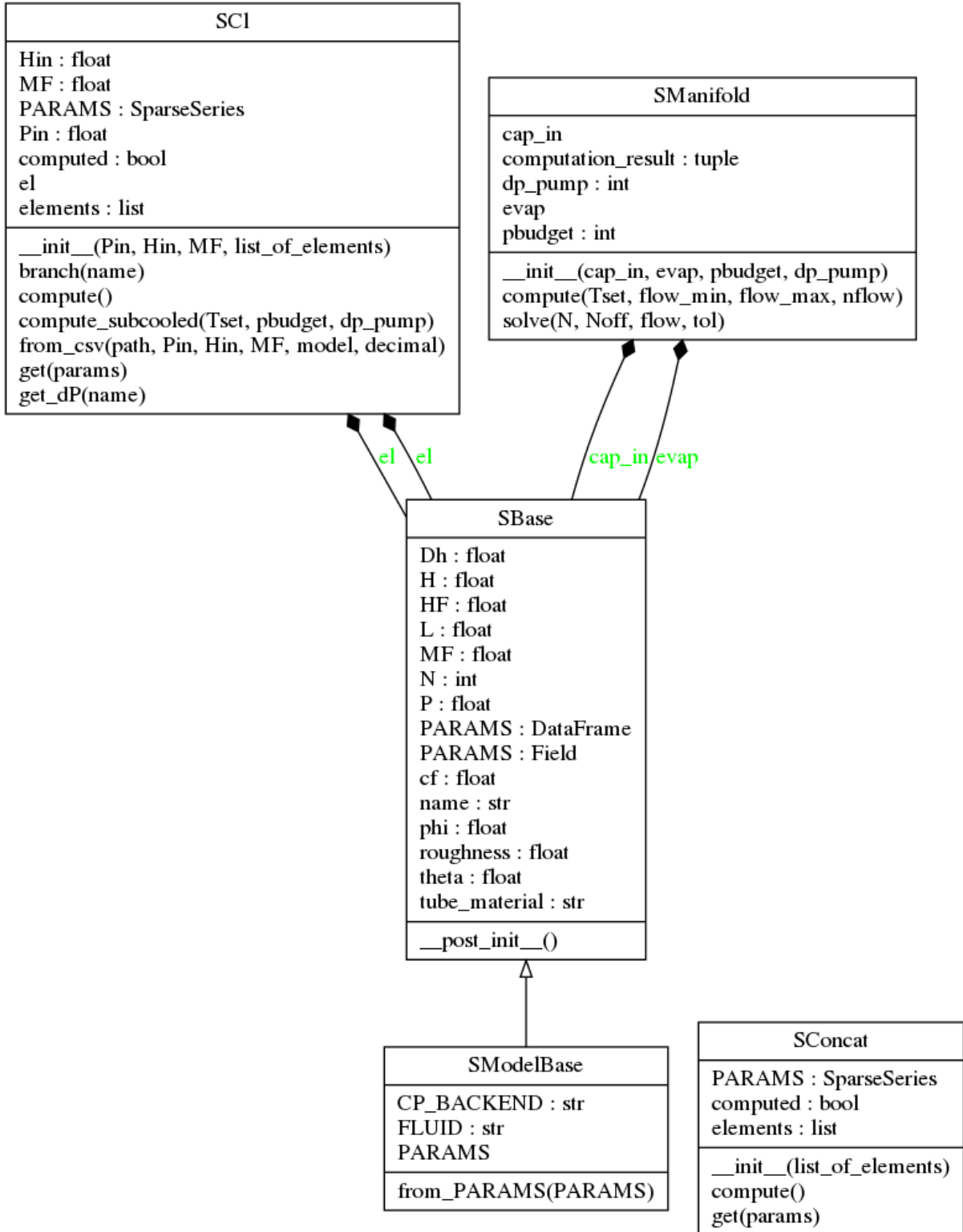


Figure 4.2: UML diagram of the simulation tool showing the different classes inheritance and corresponding methods and attributes.

The program is class-based and the Figure 4.2 illustrates the respective Unified Modelling Language (UML) which gives an outlook on the class inheritance, attributes and methods.

Each element is instantiated from the *SBase* class. A set of attributes like the inner diameter, wall roughness, bending angle, the angle of the tube with the vertical plan and local heat flux are needed in order to construct the element class object. To cope with the cases that just part of the tube may be heated (top half or bottom half for example), a new attribute defining a coefficient applied to the total heated surface was added to calculate the right amount of heat absorbed in cases the tube is not uniformly heated.

Any model plugged in the tool must inherit from the *SModel* class, and have defined methods to calculate: pressure drop, heat transfer coefficient and flow pattern. Although the tool currently only includes the phenomenological methods described in section 2.5, any other set of correlations could be defined and easily integrated.

Several tools are provided in the *utils* module. It is possible to concatenate a sequence of elements with the *SConcat* class to compute several parameters provided by CoolProp and the model. This is especially useful to compare the results (heat transfer coefficient, pressure drop, incipience vapour quality, pattern map) between different types of elements.

To simulate the flow along a sequence of elements, one can use the *SCl* class that takes as input the inlet pressure, enthalpy and mass flow and the sequence of elements that compose the whole cooling line. The iterative calculation is provided by the *compute* method.

As in the case of the CO₂ cooling system for the future ATLAS ITk, the system is conditioned by the temperature set at the accumulator and the temperature and pressure gradient driven at the pump level. A second method in the *compute_subcooled* class (*compute_subcooled*) was included to take into account these parameters to iteratively retrieve the right inlet pressure and enthalpy to respect the system conditioning.

4.3 About Coolprop

To calculate the thermodynamic properties of the fluid, the simulation tool uses the open source C++ library Coolprop [40], able to compute equations of state for several fluid based on Helmholtz energy formulations. With this construction, all the thermodynamic properties of interest can be obtained from partial derivatives of the Helmotz energy [53–57]. The equations of state are typically valid for sub-cooled, super-heated and super-critical fluids.

This library has several bindings with other programming languages, python included. The abstract interface implemented in Coolprop can be easily coupled with the widely used proprietary library REFPROP, developed by researchers at the National Institute of Standards (NIST), which is used in CoBra.

4.4 Comparison with experimental data

The tool was able to replicate several simulations computed with CoBra [52] where it was extensively used to optimise the tube dimensions of the IBL.

Colijn and Verlaet in their work [58] cooled 800 mm long stainless steel pipes with inner diameters of 1.2 mm and 1.6 mm to study the pressure drop and dry-out incipience. A 715 mm section of the tube was heated uniformly by running a DC current through the stainless steel. Along the heated section, several thermocouples were bonded to the pipe’s surface. To provoke dry-out, the power applied was increased continuously until a temperature spike was observed at the end of the heated section.

Figure 4.3 shows side by side the experimental data (Figure 4.3a) and simulation (Figure 4.3b). With a pipe of 1.2 mm inner diameter, the dry-out was observed when the tube was heated with 62 kW/m². The dry-out was observed 650 mm from the inlet and the simulation tool has been able to predict the dryout close to this same location.

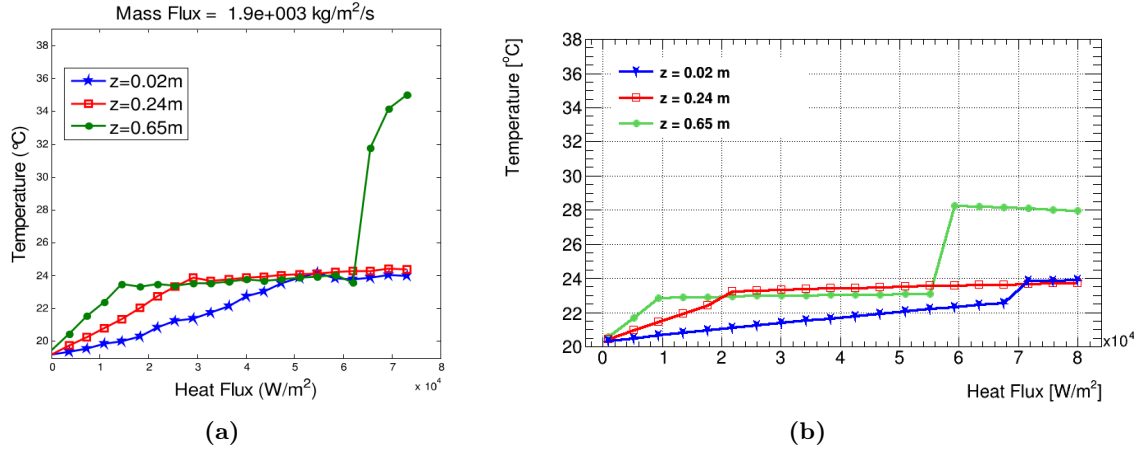


Figure 4.3: Comparison between measurements at the *left* with the dry-out observed at 0.65 m from the beginning of the heated section of the pipe when applied 62 kW/m². At the *right* is the corresponding prediction.

Other measurements were conducted to estimate the pressure drop in the 800 mm prototype with sub-cooled liquid flowing inside pipes with inner diameters of 1.6 mm and 1.2 mm. As in Figure 4.3 the experimental data is showed next to the prediction of the tool.

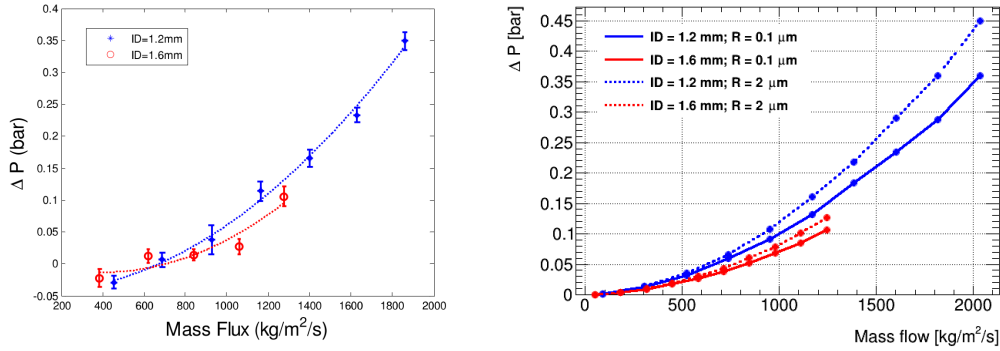


Figure 4.4: Comparison between measurements at the *left* and the predictions using the simulation tool at the *right*.

The good agreement between simulation and the measurements shows that, not only the model implemented predicts well the pressure drop and vapour quality for the dry-out incipience, but also that it is able to successfully estimate the change of thermodynamic properties of the fluid along heated pipes.

4.5 Optimisation of design

Once the coolant is chosen, the decision on the optimal tube diameter must be taken. The tube diameter can be optimised by maximizing the objective function (HTC_{vol}) [52]:

$$HTC_{vol} = \frac{q}{\Delta T_{total}} \times \frac{1}{V_{evap}} \quad (4.1)$$

The maximisation of HTC_{vol} described in equation 4.1 derives from the $HTC_{combined}$ formulated in section 3.3. It is used to minimise the ΔT_{total} and total volume of the evaporator V_{evap} for a given total heat load. This approach has the merit of minimising the impact of several parameters however, it does not consider any parameters weight. If the X/X_0 of the coolant is

not a major concern, there is no way to favour the minimisation of the ΔT_{total} . In the same way, one may want to favour the minimisation of X/X_0 in case of large safety margins in terms of temperature gradient. The algorithm must also evaluate how the change in the volume of the evaporator affects the heat flux and calculate ΔT_{total} accordingly. The integrated heat absorbed is taken as a constant and the magnitude of the heat flux has, a priori, no interest in being maximised.

To cope with these considerations another way of optimising the tube dimensions is proposed. Considering that the evaporator volume of each element V_{evap} is proportional to D_h^2 , the cooling line may be composed of elements with different lengths L_i and diameters D_{h_i} , and the total heat load is constant, an equivalent objective function (f_{obj}) to be optimized is given by the following:

$$f_{obj} = \frac{1}{w_1 \Delta T_{total}} + \frac{1}{w_2 \sum_{i=0}^N D_{h_i}^2 L_i} \quad (4.2)$$

where w_1 and w_2 are the optimisation weights applied correspondingly to the total temperature gradient and evaporator volume. Depending on the relative difference between the weights, one may prioritize the minimisation of X/X_0 ($w_2 > w_1$) or temperature gradient ($w_1 > w_2$).

Example with simulation tool

The tool includes in its *utils* module the *optimise* method that will take a cooling line and a range of tube diameters as arguments to retrieve the result of the objective function and obtain the optimized value.

As an example, we will take an evaporator with the same dimensions of the longerons of the ATLAS ITk Outer Barrel: a cooling pipe 1.47 m long (to cool two 0.735 m longerons). To simplify, we will consider a total power of 400 W (≈ 10 W/quad cell) to be dissipated by the 36 quad-cells distributed homogeneously along the tube walls (in reality the heat loads will be rather localised) and CO₂ at -35°C with a mass flow of 4 g/s at the inlet.

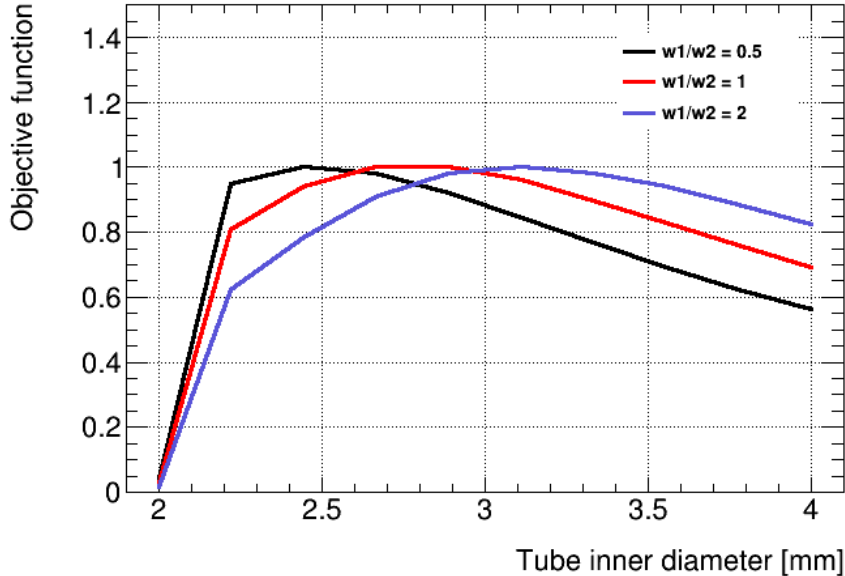


Figure 4.5: Optimisation of tube diameter for future ATLAS ITk outer barrel when cooled with -35°C CO₂ with 4 g/s and 400 W of power dissipated.

The results in Figure 4.5 shows that the chosen 2.5 mm inner diameter is within the recommended sizes. The optimization was performed considering different weights and scaled so

the result is dimensionless and its maximum is one for comparison purposes. The results show that increasing the diameter from 2.5 mm to 3 mm would favour the minimisation the effect of ΔT_{total} .

4.6 Modelling flow distribution in parallel lines

As referred earlier in section 3.9, the detectors are usually composed of several parallel branches manifolded together. The inlet capillaries create a big pressure drop before the evaporators, so the flow distribution is less sensitive to different power loads at the evaporators.

To calculate the flow distribution, the tool has the possibility to instantiate a manifold for the *SManifold* class, and could calculate the expected flow loss in one evaporator if one or all the remaining evaporators in the manifold are turned off. One can see how important it is, to correctly dimension the inlet capillaries before the evaporators, from the simulations shown in Figure 4.6, which took as an example the manifold presented in section 3.10 to be used for the ATLAS ITk outer barrel longerons.

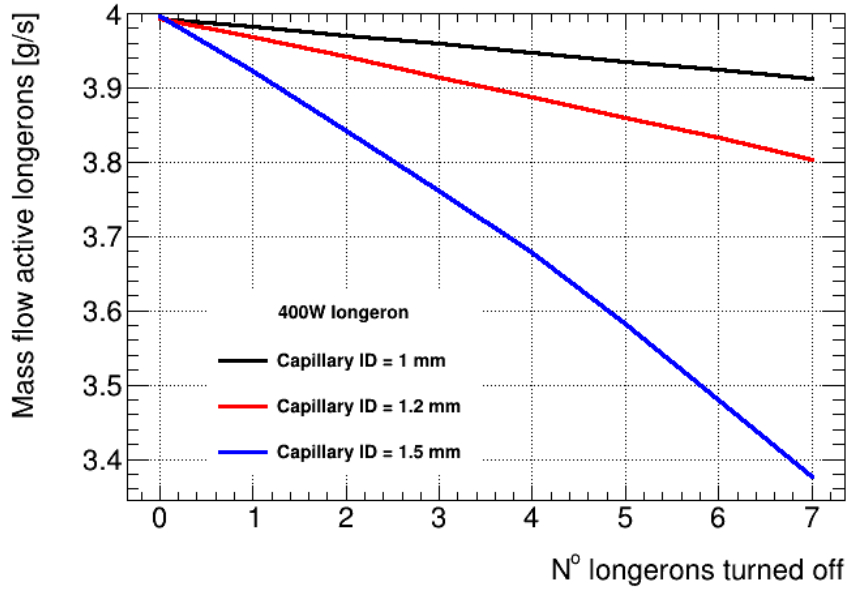


Figure 4.6: Flow unbalance studies for the ATLAS ITk outer barrel baseline manifold with eight evaporators dissipating 400W each fed with a constant mass flow of 32 g/s of CO_2 (4 g/s per evaporator). The results show the drop of mass flow expected in an active branch if no power is dissipated in one or more of the remaining branches in the manifold. The simulation was performed considering different diameters for the inlet capillary.

4.7 Web user interface

The design process of local supports is dynamic, with frequent changes. Therefore, there is often the need to recompute the expected heat transfer coefficient and pressure drop. Also, these predictions depend not only on the design but also on the coolant thermodynamic state.

To ease the access of any member of the ATLAS collaboration to the prediction tool for calculating the frictional pressure drops and heat transfer coefficient of the CO_2 a web interface was built using the Dash framework from Plotly [59], (deployable in the local computer) where these estimates can be easily retrieved.

The interface counts currently with four different tabs. Each tab aims to have one specific usage. Figure 4.7 shows the web page with the first tab selected used to compute heat transfer

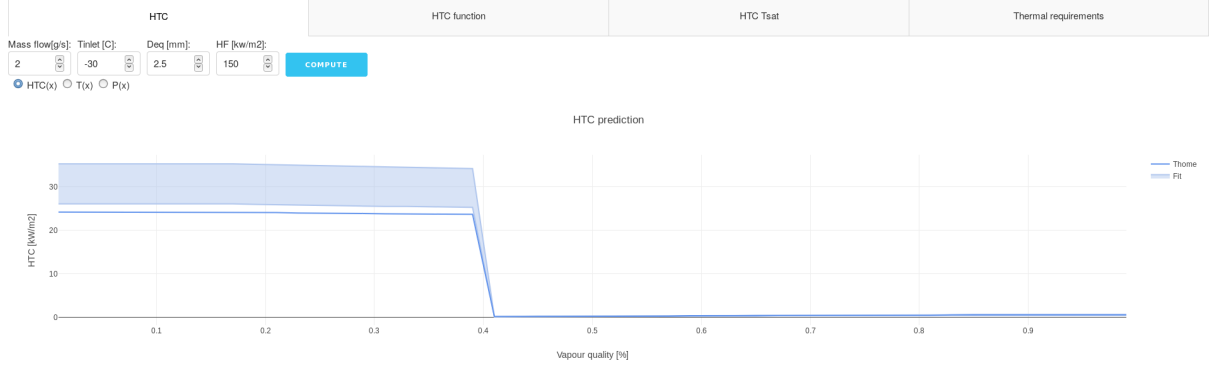


Figure 4.7: Front page of the developed web application with the first tab selected in CERN web-servers [60]. The HTC is calculated as a function of the vapour quality given the coolant mass flow, saturation temperature, inner diameter of the tube and local heat flux. The nominal HTC calculated from the Thome model can be compared with the fit re-parameterization proposed in chapter 7 with an uncertainty of $\pm 15\%$.

coefficient estimates for the CO_2 given the heat flux, CO_2 temperature and vapour quality. It is calculated using the semi-empirical model describe in section 2.5 and can be compared with the re-parameterisation proposed in section 7.6.1. Still in the first tab, one can compute the expected frictional pressure drop.

Figure 4.8 shows the second tab used to compute the heat transfer coefficient as a function of the heat flux. This curves were especially useful for people responsible for the Ansys FEA analysis of local supports at CERN who could now easily retrieve the estimates and plug in their models.

Figure 4.9 shows the third tab used to calculate the heat transfer coefficient as a function of the saturation of the temperature. This was implemented so people in CERN EP-DT group could compare their latest heat transfer coefficient experimental databases with re-parameterisation further proposed.

Finally, the fourth tab provides the thermal requirements for the ATLAS ITk outer barrel, for each one of the layers and is provided by another tool developed in python that will be presented in section 5.10.

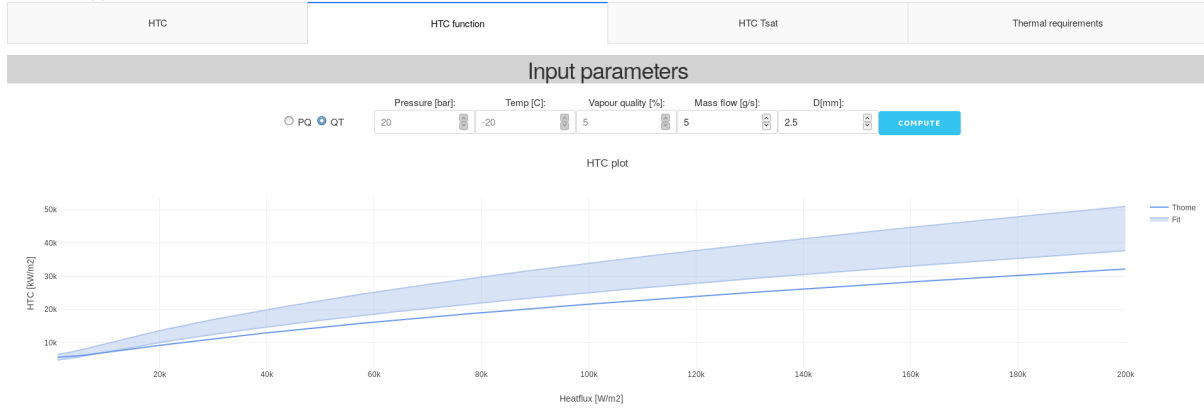


Figure 4.8: Front page of the developed web application with the second tab selected deployed in CERN web servers [60]. The HTC is calculated as a function of the heat flux given the coolant mass flow and inner diameter of the tube. The CO_2 thermodynamic properties can be set either from the input pairs vapour quality and saturation pressure (QT) or from vapour quality and saturation temperature (PQ). The nominal HTC calculated from the Thome model can be compared with the fit re-parameterization proposed in chapter 7 with an uncertainty of $\pm 15\%$.

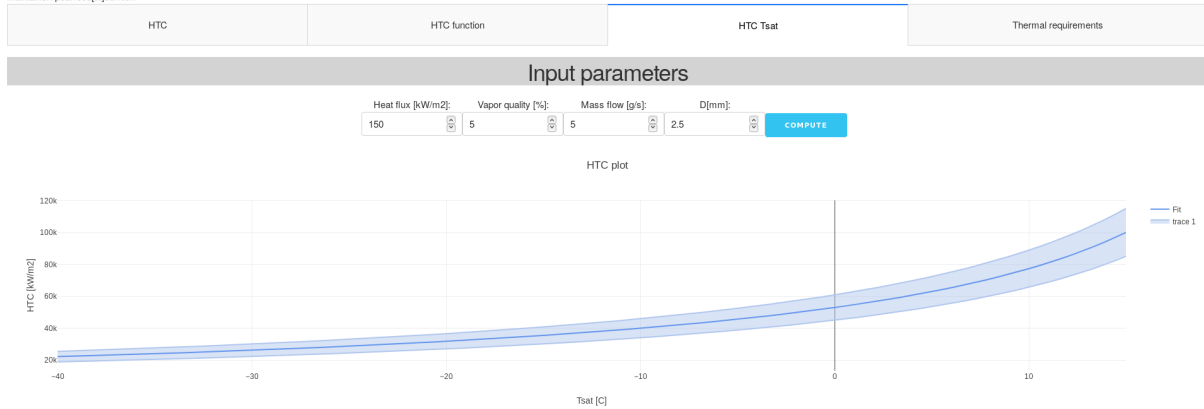


Figure 4.9: Front page of the developed web application with the third tab selected in CERN web servers [60]. The HTC is calculated as a function of the saturation temperature given the coolant mass flow, inner diameter of the tube, local heat flux and vapour quality. The nominal HTC calculated from the Thome model can be compared with the fit re-parameterization proposed in chapter 7 with an uncertainty of $\pm 15\%$.

4.8 Summary and outlook

The application backend is quite versatile and one could easily include other implementations, useful to design evaporative cooling systems.

The documentation and source code are available for the collaboration. It includes description of the tool, installation steps, usage examples and advanced configurations.

As future features, one could foresee the inclusion of other heat transfer coefficients and frictional pressure drop predictive methods that may be better suited for other coolants. Also, in the current version, there is still no consideration on the pressure drop due to the pipe bending and inlet constrictions.

THERMO-FLUIDIC MANAGEMENT OF THE ATLAS ITk PIXEL DETECTOR

Due to the expected harsh radiation conditions, the pixel sensors will need to operate at low temperatures (below -10°C). The maximum operation temperature will depend mostly on the associated radiation damage and the sensor technology.

The mechanical structure supporting the sensors and the read-out chips must thus, not only be able to keep them in place, but also to be coupled with a cooling system, capable of evacuating all the heat produced, so the pixel sensors are kept below their maximum temperature. The right assessment of the thermo-fluidic management of the Pixel detector is therefore of extreme importance, to guide the design of the mechanical structures and cooling services.

The current chapter will discuss the importance of the thermal management of planar and 3D silicon sensors, the limitations of the CO_2 cooling and the impact on the design of the local support for the outer barrel layers of the Pixel Detector.

Chapter outlook and personal contributions:

In order to understand what is the total power produced in the Pixel Detector to be taken into account to design the cooling system, it is explained in detail how one estimates the power dissipated in all the components in a Pixel module throughout the life of the detector. A custom tool developed to compute the set of thermal design requirements and estimate the critical limits (power and cooling temperature) of the system is described. The results are complemented by the tool described in the previous Chapter for a complete thermo-fluidic characterisation of the CO_2 cooling system for the Outer Barrel of the future Pixel Detector.

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5.1 Silicon in semiconductor trackers

Particle detectors rely on the signal generation from the interaction of charged particles with matter. In solids, the electrons energy levels are in bands. The electrons bounded to the atoms are in the valence band while at higher energies, the electrons in the conduction band can move freely.

Insulator materials have a big band gap, thus it is more unlikely to have electrons in the conduction band. In good conductors like metals, the valence and conduction bands overlap, so valence electrons can move freely. If the band gap is small enough, the material is considered a semiconductor.

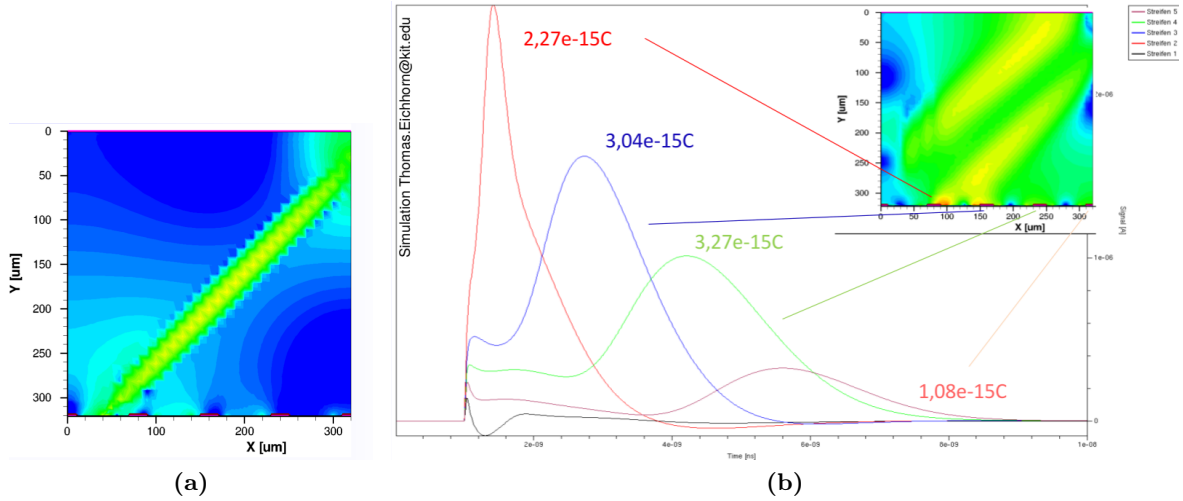


Figure 5.1: (a) Charge density in a strip sensor (300 μm thick and 25 μm wide pixels) 1ns after a mip started crossing the depleted region with a 45° angle. (b) Signal induced at each of the electrodes connected to individual pixels.

An electron can become free only if it absorbs sufficient energy to overcome the energy band gap to become available in the conduction band. Pure silicon is an intrinsic semiconductor and due to its small band-gap ≈ 1.12 eV [61] there is an average of $1.45 \times 10^{10} / \text{cm}^3$ (at room temperature) electrons in the valence band migrating to the conduction band (free charges), due to thermal excitation in an intrinsic carrier generation process (electron-hole pair production).

A charged particle crossing the silicon can release enough energy to promote electrons from the valence to the conduction band producing an electron-hole (e-h) pair (≈ 3.6 eV needed). However, a minimum ionizing particle (mip) would only create $7 \times 10^6 / \text{cm}^3$ e-h pairs, which is significantly smaller than the number of free charge carriers. Hence, bare silicon is not suitable to detect signals from crossing charged particles.

However, the silicon crystal structure can be changed with a doping process¹, introducing new energy levels in the forbidden energy gap. Adding elements from the fifth group like phosphorus or arsenic, creates an in-balance of more electrons than holes (type n silicon) while doping with elements from the third group like boron, adds more holes than free electrons (type p). By combining positively and negatively doped silicon, a pn-junction is created with a central region where free electrons and free holes migrate freely (diffusion current) until thermal equilibrium is reached creating a region depleted of mobile charge carriers.

By applying a reversed bias voltage, this depleted region is increased until the free electrons of n-type silicon populate all the free holes of the p-junction, creating an insulating layer (fully depleted region). The voltage needed to fully deplete the diode is called depletion voltage.

In a p-n junction, if an ionizing particle crossing the silicon exchanges enough energy with electrons in the depleted region occupying the holes, they migrate to the free holes in a e-h pair production process, which will then drift in the electric field created by the reverse bias and induce an electrical signal on the metal electrodes.

In a matrix of several p-n junctions (pixels), the drift of electrons and holes from a single particle will induce a signal pulse shared by several pixels - see Figure 5.1. By gathering the pulse shape and integrated measured charge from every pixel it is possible to get a precise crossing location of the particle, thus suitable for particle tracking.

5.2 Radiation damage in silicon sensors

Ionizing radiation (charged particles) crossing the detector will produce charges in the silicon oxide of silicon detectors, while neutrons will induce defects in the bulk region [61, p. 25]. Both surface and bulk generation currents compose the total induced leakage current which will increase with the radiation damage. In the bulk, charge trapping will occur, reducing the signal and higher depletion voltage will need to be applied for full depletion.

In the fully depleted mode, the signal is proportional to the overall generated charge and a crossing particle can be detected. Not operating in fully depleted mode lowers the signal-noise ratio as the charges created in the undepleted bulk region do not contribute to the signal. Also, the operation of an undepleted device leads to additional noise because of the increase in its capacitance. A way to compensate this effect is to raise the operation voltage to operate in fully depleted mode which will also drive a raise in the bulk generated current as $I_{bulk} \propto \sqrt{V}$ for $V \leq V_{dep}$ [61, p. 25].

$$I_{bulk}(T_S) \propto T_S^2 \exp^{-T_S/T_A} \quad (5.1)$$

The bulk defects add charges closer to the conduction band, increasing the chances of charge production due to thermal excitation. The dependence of the bulk generated current with temperature is formalised by equation 5.1 with T_A being the activation temperature defined as $E_g/2k$. E_g is an effective energy gap for carrier generation, empirically ≈ 1.2 eV for radiation damaged silicon and k the boltzmann constant.

While the temperature of the sensor is not in equilibrium, the power dissipated by the sensor due to its leakage current will raise, as well as its leakage current. This process is called *thermal runaway* of the sensor and will lead to a positive feedback loop that will at some point diverge reaching the destruction of the sensor. To avoid this rise of noise and risk of damaging the sensor it is important to cool the sensors and keep their temperature stable.

¹Dopants are usually introduced by ion implantation into the silicon crystal structure to replace silicon atoms.

5.3 3D and planar hybrid pixel detectors

The ATLAS ITk Pixel detector will be fully equipped with hybrid pixel detectors composed of a single passive silicon sensor with arrays of $50\ \mu\text{m} \times 50\ \mu\text{m}$ or $25\ \mu\text{m} \times 100\ \mu\text{m}$ sized pixels connected to front-end (FE) readout chips. The high-density connectivity between the two components is obtained through flip-chip bonding, where bump-bonds will establish the electric contact between front-end and silicon pixels, as shown in Figure 5.2.

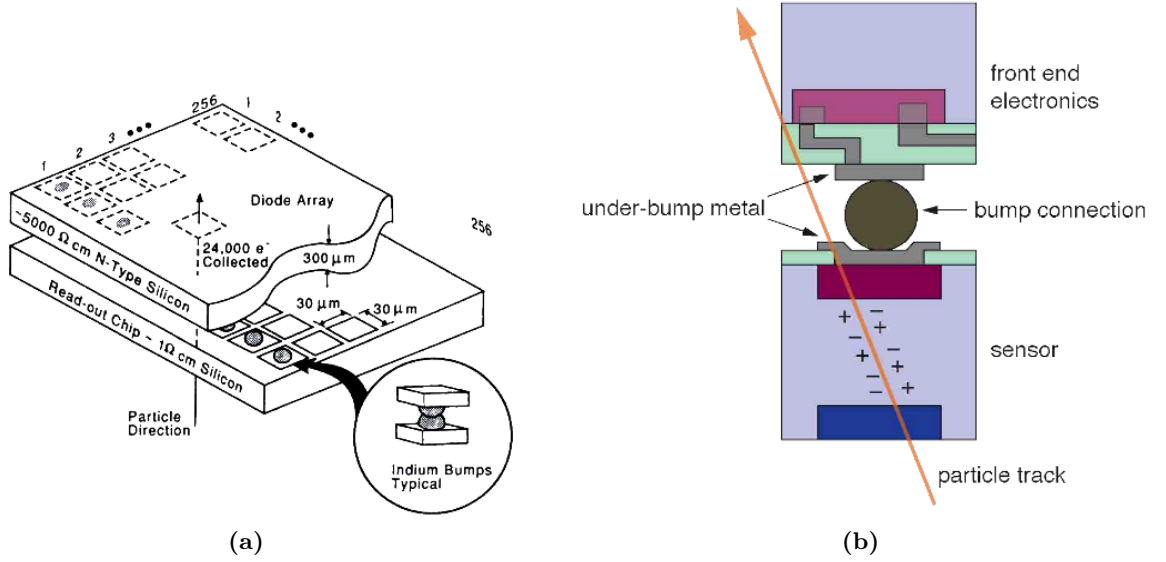


Figure 5.2: (a) Illustration of an hybrid pixel detector and (b) sketch illustrating its cross section. The silicon sensor is connected to the electronics via the bump bond to capture the charge created by the particle track traversing the sensor volume [62, 63]

Two types of silicon sensor technologies will be used: 3D and planar sensors. In 3D silicon detectors the distance between the cathode and anode is not coupled to the device thickness as in the case of planar sensors - see Figure 5.3. Therefore, the distance can be much smaller lowering the operational voltage for full depletion and reducing the chances of charge trapping in the bulk region. This translates into lower leakage currents, hence less power dissipation after irradiation.

3D sensors have been chosen for the innermost layer, thanks to their radiation hardness and low power dissipation. As planar sensors still cost less to produce and have higher production yield, they were chosen for the other layers.

Nevertheless, as big efforts are being made to increase the yield of future 3D sensors production and lower costs, they could soon replace planar sensors as the baseline technology used in HEP experiments [8, p. 149].

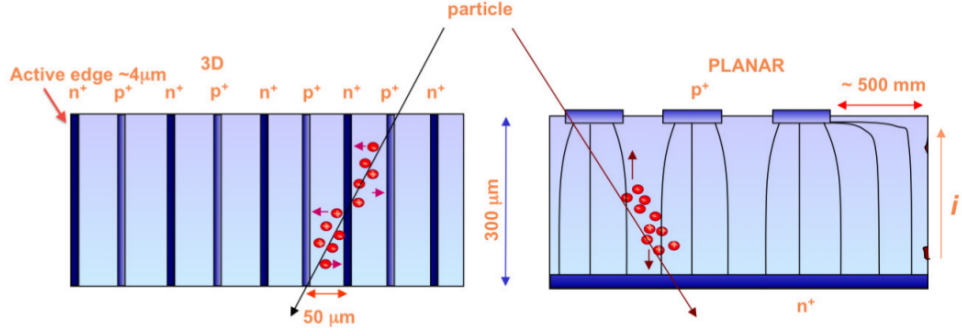


Figure 5.3: Schematic view of electrode arrangement and charge collection in 3D and planar pixel sensors [8]

5.4 ATLAS Pixel module

The hybrid pixel module is the basic unit of the Pixel Detector. The innermost layer of the inner system will use single chip modules (2 cm x 2 cm) while the remaining barrel layers will use quad modules composed of four front-end read-out chips (4 cm x 4 cm) [64].

A flexible PCB (module flex), bonded to the back part of the silicon sensor, connects the front-end chips to read out and power distribution systems via wire-bonding² (Figure 5.4). All the pixel modules will share the same readout chips but different sensor technologies are used depending on the expected radiation damage.

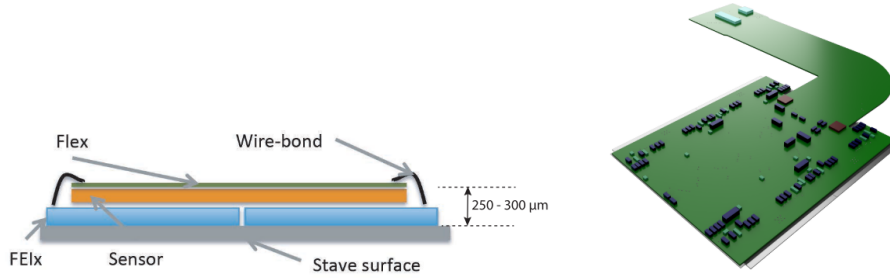


Figure 5.4: Stack up view of pixel module elements (left) and top view of a quad module flex (right)

5.4.1 Outer Barrel local support

The local support is responsible for holding the pixel modules in a fixed position over time and to establish the thermal connection with the cooling circuit, to evacuate the power dissipated by the pixel module and its services. It is composed of materials with low mass, high rigidity and high thermal conductivity to minimise the amount of material inside the detector while keeping the modules aligned and cooled.

A module cell is composed of four front-end chips bump-bonded to the silicon sensor and glued to a thermal pyrolytic graphite (TPG) plate, as shown in Figure 1.14b at section 1.3.3. The ensemble is then bonded to a cooling block. The module cell is integrated in the longeron by screwing the cooling block to a base block with a thermal interface material (TIM) in between. A titanium tube is soldered to the base block and glued to the truss composite structure.

²Wire bonding is a semi-automatic method often used to electrically connect the integrated circuits to printed circuit boards via micro-metric wires.

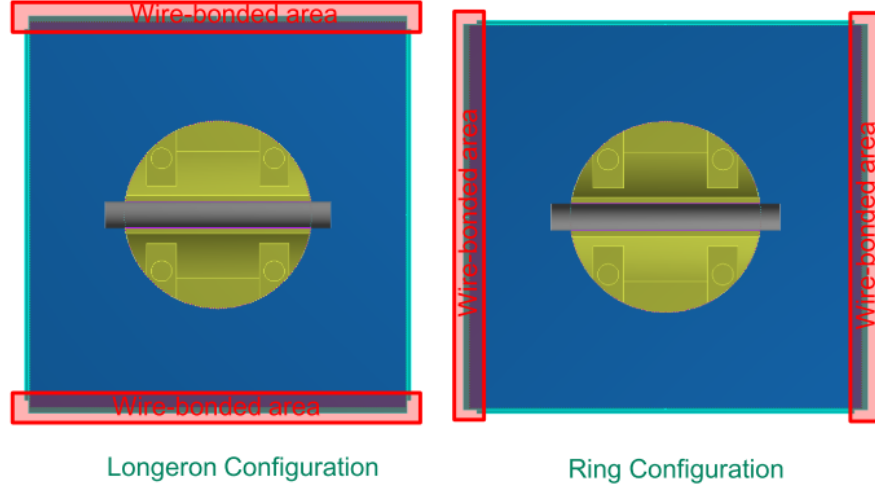


Figure 5.5: Pixel module cell and the relative orientation of the titanium cooling tube to the FE wire bonded area of the pixel modules mounted on the longeron and inclined rings.

The base block presents a symmetric design so it can be used to host the module cells in the longerons and inclined rings, with different cooling tube orientation relative to the FE wire-bonds (see Figure 5.5) using the same mechanical parts.

5.4.2 ITk Pixel serial powering

Powering thousands of modules individually would result in a excessive number of cables impossible to afford, especially inside the tracker volume.

Therefore, for the ATLAS ITk it was decided to connect the modules with a single serial power line (SP Chain), driven by a constant serial power current source (I_{SP}) with a maximum of 64 front-end chips³ (i.e 16 pixel quad-modules).

As shown in Figure 5.6, there will be several types of power sources located in the USA15 cavern, responsible for powering the front-ends (LV), sensors (HV), opto-boxes (V_{gbl} , V_{optoT} and V_{optoR}), DCS chips (V_{dcs} and V_{can}).

The pixel modules in a SP-Chain are connected in series with a common LV line to power all the front-ends. Depending on the length of a SP-Chain there may be two or more HV lines connected in parallel to the sensors for the bias voltage. These powering lines are connected from the module flex to the sensor and front ends via wire bonds. The connection between the the pixel module power lines and the power sources are made via Type-0 services routed along the stave or ring (stave flex) connecting the module flex to a "End of stave" board (EoS board) in a services gathering location called patch panel 0 (PP0). PP0 is connected to a second patch pannel (PP1) via Type-II services, followed by Type-III cables to connect to the power sources - see Figure 5.6.

The power provided by the low voltage power supplies must be enough to power the front-end chips while taking into account the power loss in the cables. The front-end chips will be responsible for most of the power consumption. The sensors will have smaller contributions while the cables (type I, II and III) together are expected to account a maximum of 30% of the power budget⁴.

³The limitation of 16 quad modules in a SP Chain is limited by the single communication bus with the DCS controller [8, p. 260]

⁴Most of the power loss will be in Type-III cables, far away from the detector volume.

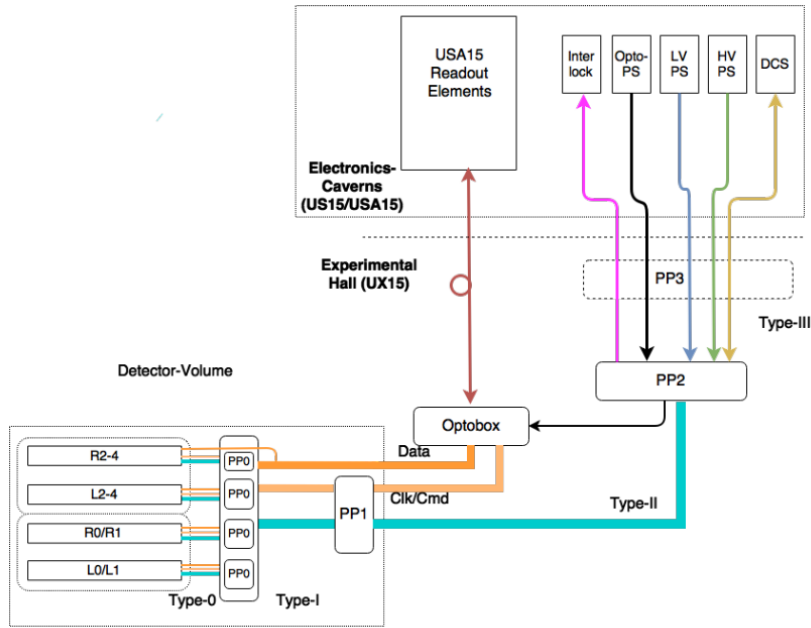


Figure 5.6: *Routing of electrical services from the USA15 cavern to the detector volume [8].*

5.4.3 Module flex

The module flex is a flexible PCB responsible for interfacing the silicon sensor and front end chips with the local support Type-0 electrical services, required to operate the pixel module (Figure 5.7). It is glued to the rear of the sensor and connected to the front-end chips via wire bonding as shown in Figure 5.4.

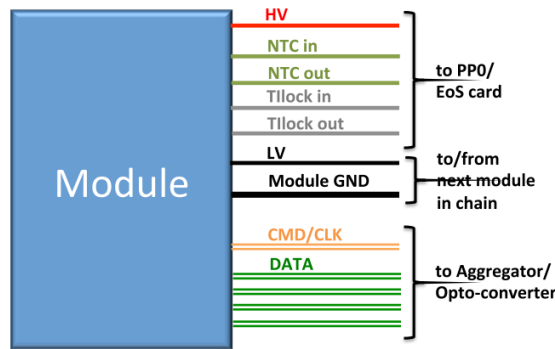


Figure 5.7: Schematic with electrical connections between the Pixel module and electrical services [8]

The HV supply lines provide the needed sensor bias voltage and the LV lines power the front-end chips. All the needed passive and active elements to operate and readout the front-end chips [8, p. 199] will be integrated in the flex.

Two temperature sensors (NTC) placed on the module flex will inform the Detector Control System (DCS) about the module temperature via four dedicated lines. One of the sensors is monitored by the VDS chip, responsible for bypassing a serial power chain (SP-Chain) in case there is a module malfunction. The second sensor is connected to the interlock system to monitor the performance of the cooling system.

The module flex will also host the aggregator chip responsible for multiplexing together the up-link data streams from the front-end chips connected to the front end chip via wire-bonds.

Only 10% of the power budget for Type-0 services is allowed to be dissipated on the module flex, corresponding to a maximum of 2% of the FE power.

5.4.4 RD53A front-end chip

The FE chip is based on the RD53A/B integrated circuit developed by the RD53 collaboration, created as joint effort between ATLAS and CMS to obtain a common chip design able to comply with the HL-LHC challenging conditions [64]. It is the active element responsible for recording the charge collected at each pixel and holding the information until the trigger decision. The information must be readout with no loss, avoiding fakes and fast enough to comply with the triggering frequency needed, that will be increased during the HL-LHC operation.

It will be produced with 65 nm CMOS technology assuming 50 μm sized pixels with a pixel matrix of 400 pixels (20 mm) wide by 384 pixels (18.2 mm) tall [64]. The FE and silicon sensor pixels will be connected via bump-bonds in a extremely precise flip-chip process.

For leakage currents above 10 nA per pixel, the front end chip uses a continuous feedback circuit to prevent the input signal from saturating [64, p. 21]. As this compensation reduces the signal to noise ratio, a maximum of 10 nA per pixel is required. This limitation will have an impact on the thermal management of the detector as the pixel leakage current depends on its temperature - shown further in section 5.10.2.

5.4.4.1 RD53A Load Model

The current supplies feeding the LV lines will be required to deliver from 1.4 V and 5.6 A to a maximum of 1.53 V and 8 A [8] for each group of four front-end chips. This specification assumes the FE will dissipate on the quad module from 0.49 W/cm² to 0.77 W/cm² (for a 4 cm $\times$ 4 cm sized sensor).

Each module in the SP-Chain will contribute with V_{module} to the total voltage drop along the power line. The voltage and current set in the LV power supplies must be able to handle the total voltage drop along the modules in series and the power loss in the cables. A current overhead from 10% to 20% is foreseen (amount of current above the nominal values) to ensure sufficient power to the front ends in case of current unbalance at the module level. This will consequently increase the total power dissipated in the front ends, which should be taken into account as a safety factor in the thermal management specifications.

The red slope illustrated in Figure 5.8 represents the load model for the module with an effective resistance ($R_{effective}$) of 230 m Ω and a V_{offset} of 0.8 V (V_{offset} can be set for each chip). Each one of the four chips will be equipped with two shunt low-drop (ShuntLDO) regulators to set a constant operating voltage on its analog and digital circuit. The remaining current not drawn in the chip, is shunted by the regulator to the local module ground [8, p. 260].

Regardless of any malfunctioning leading to an open-circuit front-end chip (analog, digital or both) the supplied current is kept constant. The remaining three front end chips will thus share the total current. This will raise the voltage drop across the chip and consequently increase the total power dissipated in the module. The module voltage drop and current per chip must be below the maximum values of 2 V and 4 A [64].

Table 5.1 shows the expected $R_{effective}$ and V_{offset} to be set in the barrel modules and the corresponding V_{module} for different serial currents (I_{SP}).

Table 5.1: Approximate values of $R_{effective}$ and V_{offset} to be set in the Barrel pixel modules [65].

Layer	V_{offset}	$R_{effective}$	V_{module} @ $I_{SP} = 5A$	V_{module} @ $I_{SP} = 6A$	V_{module} @ $I_{SP} = 7A$
Layer 0	1.2 V	75 m Ω	—	1.650 V	1.725 V
Layer 1-4	1.06 V	90 m Ω	1.510 V	1.600 V	—

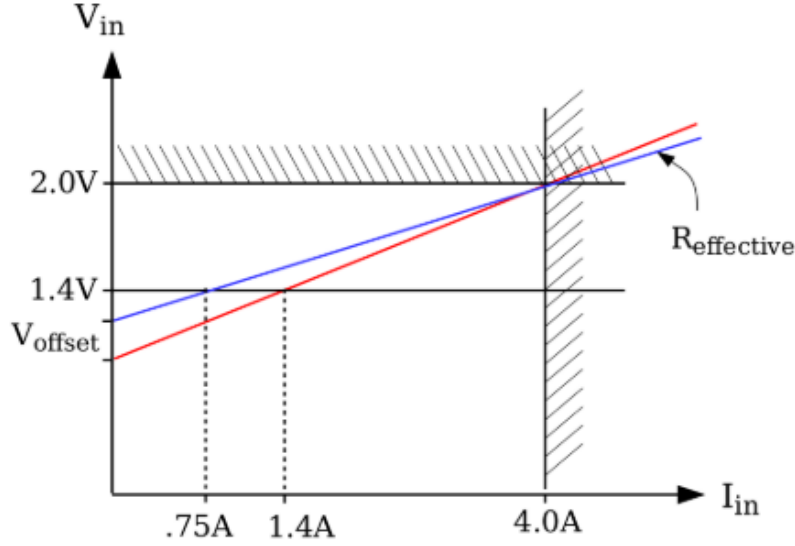


Figure 5.8: Load model RD53A with the V_{module} estimated as a function of serial power current I_{in} . The blue and red curve represent two different operating combinations of V_{offset} and $R_{effective}$ on the pixel module, respecting the required maximum voltage of 2 V and maximum current of 4 A [64].

5.5 Pixel Detector thermo-fluidic requirements

Experimentally, the thermal performance of a local support is measured from the temperature difference ΔT_{global} between the coolant and the hottest point on the silicon sensor surface.

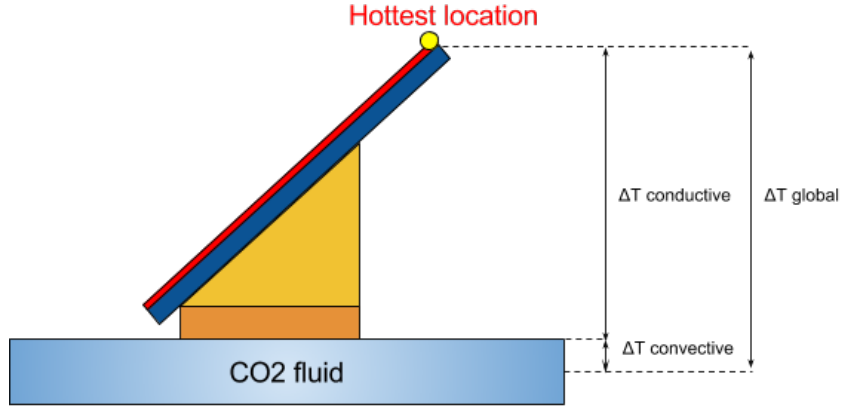


Figure 5.9: Scheme illustrating the contributions to the temperature difference measured between the coolant (two-phase CO_2) and the sensor's hottest point (ΔT_{global}). The conductive (ΔT_{conv}) and convective (ΔT_{cond}) contributions are simulated separately and characterize the CO_2 cooling performance and the material thermal conductivities respectively.

The convective contribution (ΔT_{conv}) is defined as the difference between the two-phase CO_2 temperature and the inner titanium tube wall temperature. ΔT_{conv} is driven by the CO_2 thermal performance. The conductive contribution (ΔT_{cond}) corresponds to the maximum temperature gradient between the inner tube wall and the hottest location on the sensor surface. ΔT_{cond} depends solely on the local support thermal conductivity.

The maximum temperature measured on the sensor surface must comply with the thermal requirements defined according the thermal management design constraints.

The requirements are defined in terms of the local support thermal figure of merit TFM_{TOT}

with Kcm^2/W units, calculated with equation 5.2. The ΔT_{global} is used to calculate TFM_{TOT} . ΔT_{conv} and ΔT_{cond} are used to calculate the corresponding convective and conductive components TFM_{HTC} and TFM_K . The power dissipated at the sensor is expressed in power per unit of module's area, ϕ_{mod} :

$$\begin{aligned} TFM_{TOT} &= TFM_{HTC} + TFM_K \\ &= \frac{\Delta T_{\text{conv}}}{\phi_{mod}} + \frac{\Delta T_{\text{cond}}}{\phi_{mod}} \end{aligned} \quad (5.2)$$

$$\phi_{mod} = \phi_{sensor} + \phi_{FE} + \phi_{services}$$

Considering the nominal operating conditions to calculate ϕ_{mod} , one can calculate $TFM_{nominal}$. However, to calculate the thermo-fluidic requirements of the final design TFM_{design} , several safety factors are taken into account:

Total fluence: The upper limit of the total fluence for each layer is given by the nominal value (see Table 5.3) multiplied by 1.5⁵.

Front-end power dissipation: In Table 5.2, the nominal SP chain current expected for every layer in the barrel is listed. The design value considers the possibility of having 20% current overhead, on top of the nominal current. For design purposes, it is usually assumed a maximum ϕ_{FE} of 0.8 W/cm^2 for the front ends at the inner system and 0.7 W/cm^2 for the outer barrel [8].

Module flex services power dissipation: The $\phi_{services}$ at the module flex is limited by design to a maximum of 2% of the front-end power dissipation.

Production variability: A safety factor is considered on top of the thermal figure of merit of the local support (TFM_K) to consider production variability of the local supports. After production of IBL, it was measured global TFMs of $19.5 \pm 4.5 \text{ Kcm}^2/\text{W}$ [3, p.50] on one stave with 16 modules i.e $\pm 25 \%$ of variability. For the ITk Pixel Detector, which will have many more modules than IBL, the estimated variability was set at $\pm 50 \%$. Assuming a gaussian distribution $3\sigma \approx 1.5\mu$ (only 3% of produced cells would not respect $\pm 50 \%$ variability), a conductive TFM (TFM_K) equal to 2/3 of the maximum allowable value and a design acceptance of $\sigma = 16.7\%$ were considered as the final TFM design requirement - see Figure 5.10.

⁵Table 5.3 lists the average total fluence along the z-axis for a given radial position

Table 5.2: Estimate for nominal and design dissipated power on the front-end chip and module flex normalized to the sensor area. To calculate ϕ_{FE} it is considered the V_{offset} and $R_{effective}$ listed in Table 5.1.

Layer	Nominal		Design	
	SP Current [66] [A]	ϕ_{FE} [W/cm ²]	SP current [A]	ϕ_{FE} [W/cm ²]
Layer 0 (flat)	6.80	0.68	8.16	0.86
Layer 0 (rings)	5.10	0.47	6.12	0.59
Layer 1 (flat)	5.86	0.58	7.03	0.76
Layer 1 (rings)	5.95	0.59	7.14	0.72
Layer 2 (flat)	5.53	0.54	6.64	0.69
Layer 2 (rings)	5.57	0.54	6.68	0.69
Layer 3 (flat)	5.10	0.48	6.12	0.61
Layer 3 (rings)	5.06	0.48	6.07	0.61
Layer 4 (flat)	4.94	0.47	5.93	0.59
Layer 4 (rings)	4.85	0.45	5.82	0.57

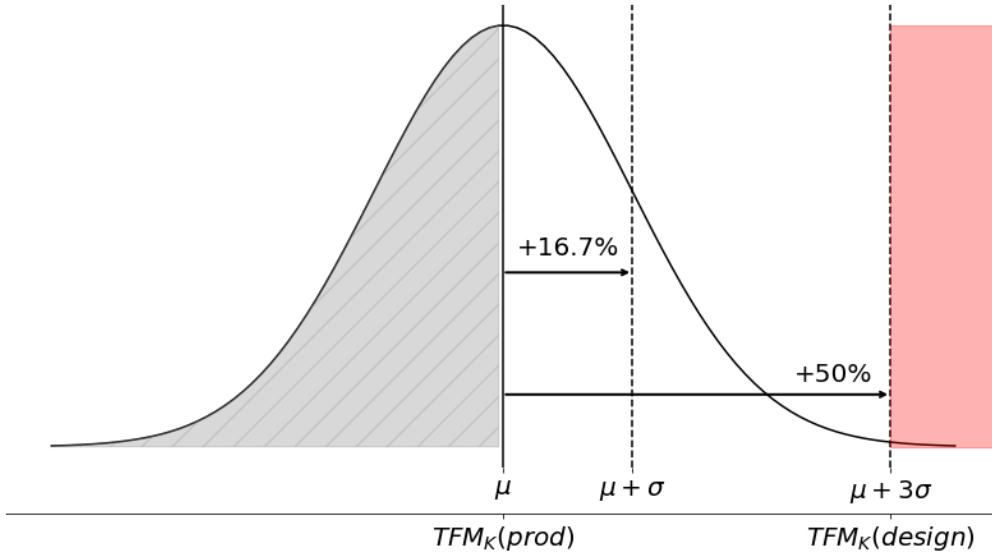


Figure 5.10: The required thermal performance for the local supports before production is calculated as the mean from a Gaussian distribution with $\mu + 3\sigma$ equals the maximum allowable TFM by design.

5.6 Radiation background for Phase II

Table 5.3: Average 1 MeV neutron equivalent fluences expected for the different layers of the Pixel Detector.

Layer	Sensor	R (mm)	Luminosity(fb ⁻¹)	Total fluence (10 ¹⁴ n _{eq} /cm ²)
Layer 0	3D	39	2000	89.5
Layer 1	planar 100 μm	99	2000	23.3
Layer 2	planar 150 μm	160	4000	23.3
Layer 3	planar 150 μm	228	4000	14.0
Layer 4	planar 150 μm	291	4000	9.80

The radiation damage from ionizing particles for the ATLAS ITk during Phase II can be expressed in 1 MeV neutron equivalent damage units (n_{eq}/cm^2). Both neutron and charged particle damages were retrieved separately from FLUKA simulations [67] at different radial positions and summed up in terms of effective 1 MeV neutron flux per cm^2 , giving the expected total fluence on the detector - see section 1.3.1. The nominal fluences listed in Table 5.3 are usually multiplied by a safety factor of 1.5 for a topmost fluence estimate.

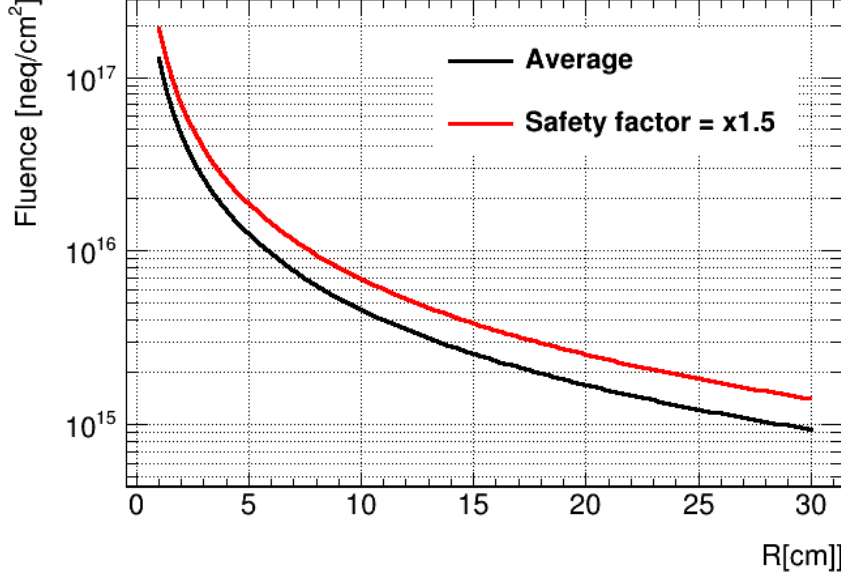


Figure 5.11: Average fluence in n_{eq}/cm^2 expected in the ATLAS Pixel barrel region as a function of distance from the beam pipe for integrated luminosity of 4000 fb^{-1} (black curve) and 6000 fb^{-1} (red curve considers a safety factor of 1.5).

5.7 Total leakage current and power dissipation

From equation 5.3 one can estimate the leakage current of the sensor by knowing its leakage current (I_{ref}) at a given temperature of reference (T_{ref}).

$$I_{bulk}(T) = I_{ref} \left(\frac{T}{T_{ref}} \right)^2 \exp \left(-\frac{E_{eff}}{2k_B} \left[\frac{1}{T} - \frac{1}{T_{ref}} \right] \right) \quad (5.3)$$

The leakage current at -25°C (T_{ref}), has been measured for 3D and planar sensors after being irradiated with 23 MeV and 25 MeV protons at fluences from 10^{15} to $10^{16} n_{eq}/cm^2$ [9].

The total power dissipated by the sensor (ϕ_{sensor}) depends also on the applied bias voltage which will continuously be increased to compensate irradiation defects on the bulk. The 3D sensors will see their bias voltage raised from 50 V at a fluence of $15 n_{eq}/cm^2$ to 200 V at $2 \times 10^{16} n_{eq}/cm^2$. The planar sensors will operate at 400 V, 600 V and 1000 V for fluences of 2×10^{15} , 5×10^{15} and $10 \times 10^{15} n_{eq}/cm^2$ [68]. The expected change of the bias voltage for all the sensors, considered for simulation purposes in this thesis is presented in Figure 5.12.

Considering the expected leakage currents and the bias voltage ramp we obtain the expected power dissipation curves shown in Figure 5.13.

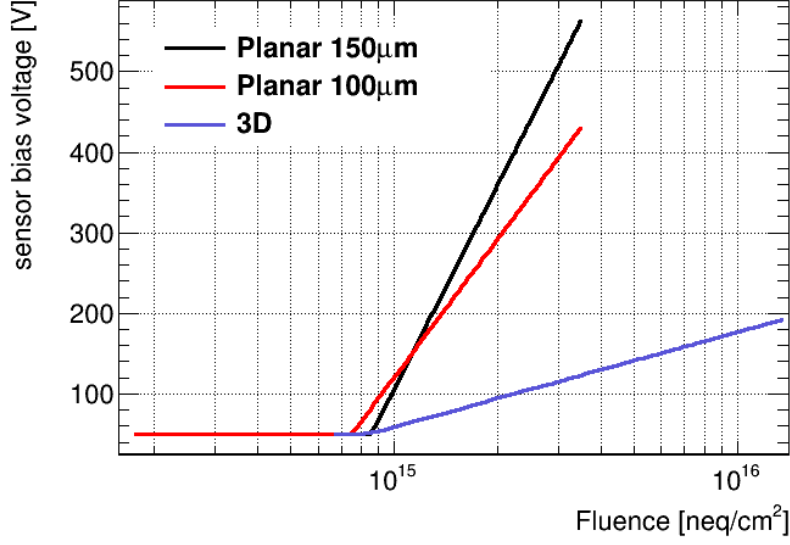


Figure 5.12: *Estimated bias voltage as a function of radiation damage (fluence) for all sensor types to be used in the Pixel Detector. The minimum voltage needed for full depleting the sensor is assumed to start at 50 V for every sensor.*

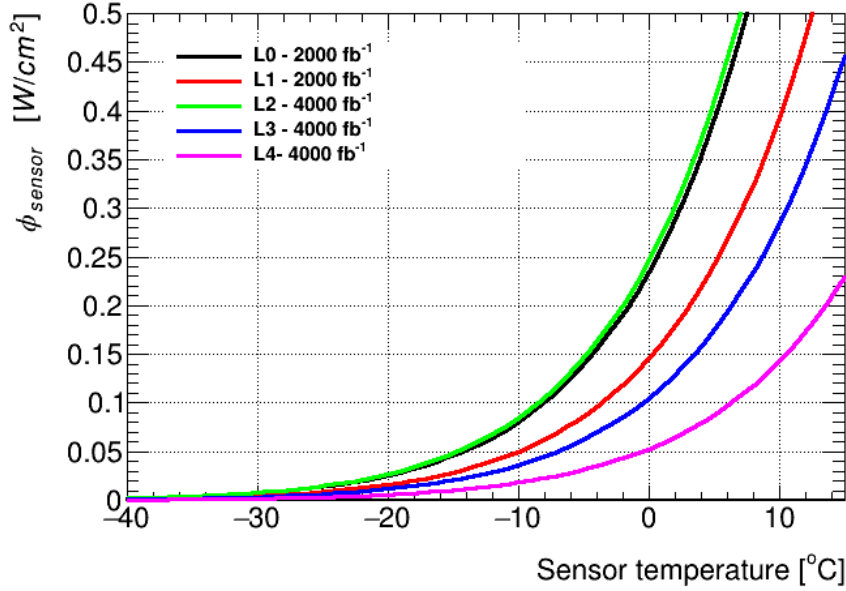


Figure 5.13: *Power dissipation of pixel sensors after integrated luminosity of 2000 fb⁻¹ for the innermost two layers and 4000 fb⁻¹ for the outer layers, taking a safety factor of 1.5 into account.*

5.8 Thermal performance and non homogeneous heating

In normal operation, the heat will not be evenly distributed over its surface but rather concentrated at its periphery, close to the wire bonds and in the longeron (Figure 5.5) furthest from the tube. Finite Element Analysis (FEA) results from ANSYS⁶, show how the non homogeneous FE power dissipation will affect the highest temperature of the sensor and the temperature gradient over its surface. The convective effect was included in the simulation integrating the CO₂ HTC dependency with the temperature and local heat flux. The curves were retrieved from the web

⁶Simulations conducted by Diego Feito from CERN EP-DT group using a detailed description of the local support materials and heat transfer coefficient estimates retrieved from the web application presented in Chapter 4

application presented in chapter 3.

The possibility of having one of the quad module front-ends not working was also investigated. As explained before, due to the serial powering scheme, this case scenario would lead to an increase of the power dissipated at the periphery of the remaining front-ends, increasing the sensors maximum temperature. The impact is shown in Figure 5.14 for the flat quad-module cells on Layer 2 of the outer barrel cooled with CO_2 at -35°C . The simulations considered a 1.5 safety factor on top of the fluence estimation, 0.7 W/cm^2 dissipated by the front-end read-out chips and 0.07 W/cm^2 dissipated by the module flex services which is in accordance to the safety margins to be considered.

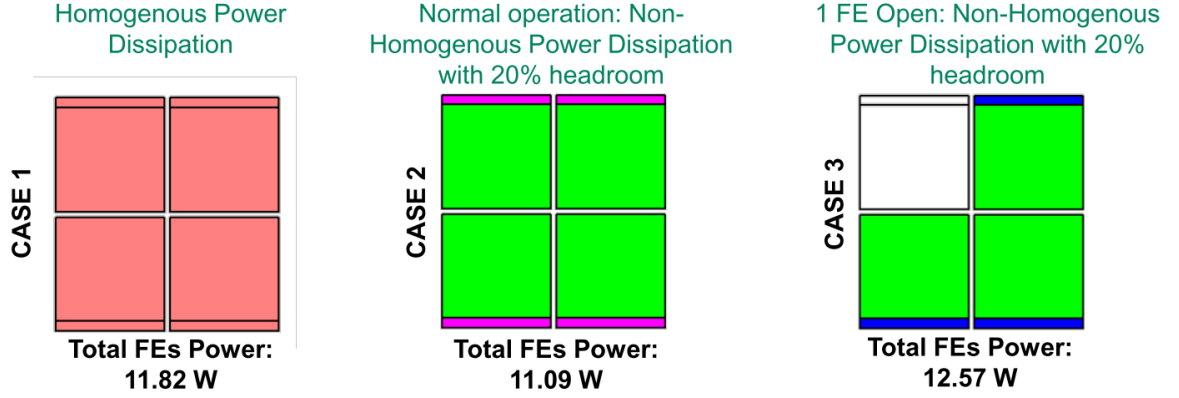


Figure 5.14: Schemes illustrating how the power is distributed over the sensor's surface with homogeneous FE power dissipation (CASE 1), normal operation with most of the power dissipated at the FEs border (CASE 2) and if there is a FE not working (CASE 3) [13].

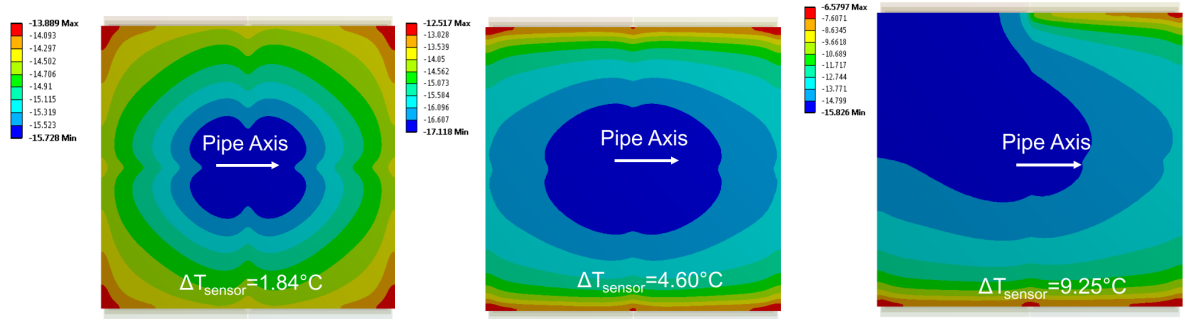


Figure 5.15: FEA results for flat quad-modules on Layer 2, cooled with -35°C CO_2 , considering homogeneous power dissipation on the four chips (left); power dissipation concentrated at FEs periphery (center); and one FE open-circuit (right). The homogeneous power dissipation with 20% headroom corresponds to a total 0.84 W/cm^2 dissipated by the module. [13]

The most problematic scenario is indeed the possibility of having a malfunction in one of the front-ends. The total power increases as it will be distributed over a smaller surface. In Figure 5.15 is shown the consequent increase of 6°C on the topmost temperature and this doubles the temperature gradient across the sensor from 4.6°C to 9.2°C .

The homogeneous dissipation illustrated in CASE 1 at Figure 5.14 is known to be unrealistic. Therefore, one should rely on the CASE 2 for an estimate for thermal performance of the local support. With an expected ΔT_{global} of 22.5°C and a power dissipation at the module of 0.79 W/cm^2 , the global TFM is estimated at $28.6\text{ Kcm}^2/\text{W}$. The conductive part corresponding the local support itself measured a TFM_K around $19.8\text{ Kcm}^2/\text{W}$.

5.9 Minimal model and calculation of thermal specifications

Two main specifications will drive the maximum allowable temperature on the sensor (T_{max}): the maximum leakage current per pixel of 10 nA (T_{10nA}) and the maximum sensor temperature before thermal runaway ($T_{S,crit}$). Both depend on the total power dissipated by the sensor, front-end chip and services.

A simplified model was proposed by Beck and Viehhauser [69] where the silicon sensor is modeled as an ideal source of heat $Q(T_S)$ with negligible thermal resistance (sensor is 150 μm thick) connected to a heat sink at constant temperature T_C (CO_2) by a thermally resistive path ($R_C + R_l$) independent of temperature and heat flux - see the Thevenin equivalent in Figure 5.16. In this model, the source of heat represents the total power dissipation of the pixel module, the R_l the TFM_K and R_l the TFM_{HTC} .

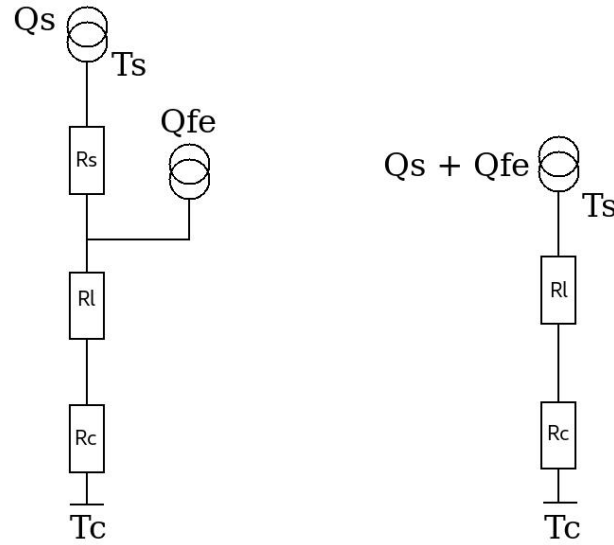


Figure 5.16: Equivalent Thevenin circuit from the minimal model assumptions with Q_{fe} and Q_s as the power dissipation at the front-end and silicon sensor; T_s and T_c being respectively the sensor and the CO_2 temperatures; and thermal impedances of the sensor, local support and CO_2 represented by R_s , R_l and R_c .

Figure 5.17 illustrates graphically a way to verify if the thermal performance of the local support at layer of the Pixel Outer Barrel is able to comply with the sensor thermal requirements when cooled at -35°C . The black and red lines represent respectively the dissipated heat (front end, sensor and services) and the heat conducted by the local support with a given thermal impedance. The interception of the conduction line with the power dissipation line gives the operating point. If the curves do not intercept, the local support is not able to evacuate the power dissipated by the pixel module hence, the sensor is thermally unstable. As T_{10nA} is also a upper temperature limit to be taken into account, T_{max} must be the lowest temperature between T_{10nA} and $T_{S,crit}$.

The slope of the load line is inversely proportional to the thermal figure of merit (TFM_{TOT}) of the local support. The maximum TFM before triggering the thermal runaway or the maximum 10nA per pixel will set the critical operating point at $T_{S,crit}$ and $\phi_{mod,crit}$. From Figure 5.17 one can conclude that the local support must have a TFM_K low enough to avoid thermal runaway.

The TFM_{TOT} corresponds to the sum of the thermal impedance of the local support (TFM_K) and the convective thermal impedance (TFM_{HTC}). A constant TFM_{TOT} assumes both convective and conductive contributions are constant (no dependence on the module heat flux neither the CO_2 temperature).

As the heat transfer coefficient (HTC) of the coolant strongly depends on the heat flux and

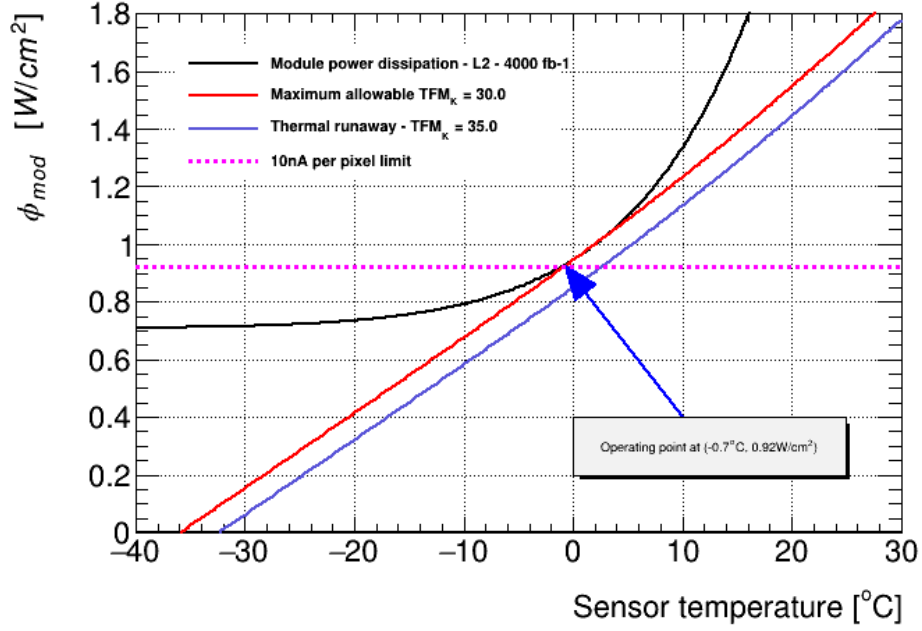


Figure 5.17: Illustration on how the maximum allowable sensor temperature and corresponding module power dissipation is estimated for pixel modules on Layer 2, after an integrated luminosity of 4000 fb^{-1} multiplied by a safety factor of 1.5 and 20% current overhead on the FE. The operating point of the sensor is derived from the interception of its load line (red line) and the module power dissipation curve (black line). To respect the requirements, the operating point must correspond to a leakage current below the limit of 10 nA per pixel. For layer two (L2) of the outer barrel, the local support is required to have a TFM_K below $30 \text{ Kcm}^2/\text{W}$.

CO_2 temperature, it would not be realistic to consider a constant TFM_{HTC} regardless of these parameters. Changes in the HTC, lead to a parabolic rather than a linear load line as shown in Figure 5.17.

5.10 ATLAS thermal management utility tool

A tool was developed in the python programming language to compute the thermal management specifications for a local support responsible for cooling an irradiated pixel sensor with two-phase CO_2 . To estimate the heat transfer coefficient of the CO_2 , it integrates the tool described in Chapter 4. It is able to calculate the expected power dissipated by the module, the CO_2 and sensor critical temperatures ($T_{C,crit}$ ($T_{S,crit}$)) and the critical local support thermal impedance (TFM_K).

The tool was implemented to estimate the thermal requirements for all layers of the ATLAS Pixel Detector presented in this thesis. Thanks to its modular structure, it can be easily adapted to other detectors/local supports.

The estimates retrieved with the app were validated on several occasions, by being systematically in agreement with independent studies conducted internally in the ATLAS collaboration (prediction of CO_2 critical temperature during the life of the detector, power dissipation of the module before runaway, thermal requirements, ...).

Program UML

Figure 5.18 shows the program inheritance diagram for the Module class, which represents a pixel module.

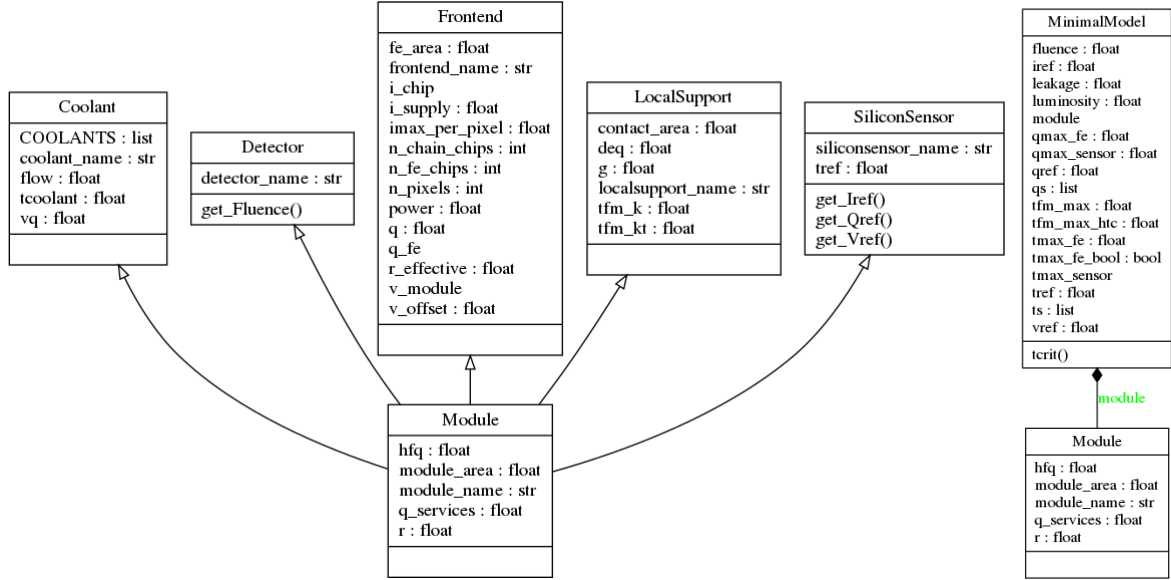


Figure 5.18: UML diagram

To create a pixel module for Layer 2 of the outer barrel, one must declare a CO₂ instance for the coolant, a Detector instance for the pixel detector, a Front-End instance for the quad front end, a SLIM local support and a 150 μm thick planar sensor. Each module object is characterized by its properties and methods listed in the UML diagram - Figure 5.18. The difference between each type of module, is the set of instances used in the assembly and the declared attributes like its radial position from the beam pipe (ex: $r = 39 \text{ mm}$ for Layer 0).

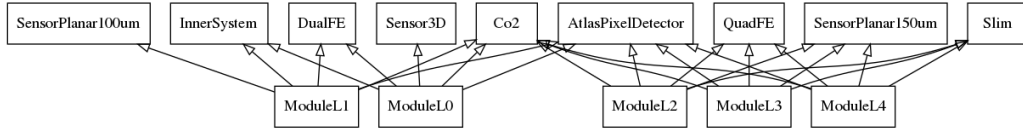


Figure 5.19: UML diagram of utility tool used to calculate the thermal specifications for the ATLAS ITk Pixel Detector.

The minimal model class shown in Figure 5.18 gathers the set of methods needed to calculate the thermal specifications and critical values for a given module.

The critical T_{FM} before thermal runaway, for a local support considering a linear conduction line (as proposed in the original minimal model [69]) is given by the tfm_max property with tfm_fe_bool set to *False*

The thermal requirements presented in this thesis were calculated using this tool. The limitation for the maximum 10 nA current per pixel limit was chosen (by enabling the tfm_fe_bool property) and the model described in section 2.5 for the HTC dependence on the heat flux and CO₂ temperature (tfm_max_htc).

5.10.1 Thermal requirements

Using the tool described and considering the requirements listed in section 5.5 the nominal, design and production thermal requirements were calculated and displayed in Table 5.4.

Before reaching the final design, engineers and physicists worked continuously to find the best compromise. At each iteration there are changes that will impact the thermal specifications of the local support. This is hard to estimate if there is not a tool integrating the various parameters that need to be taken into account. To ease the task of recomputing the estimated

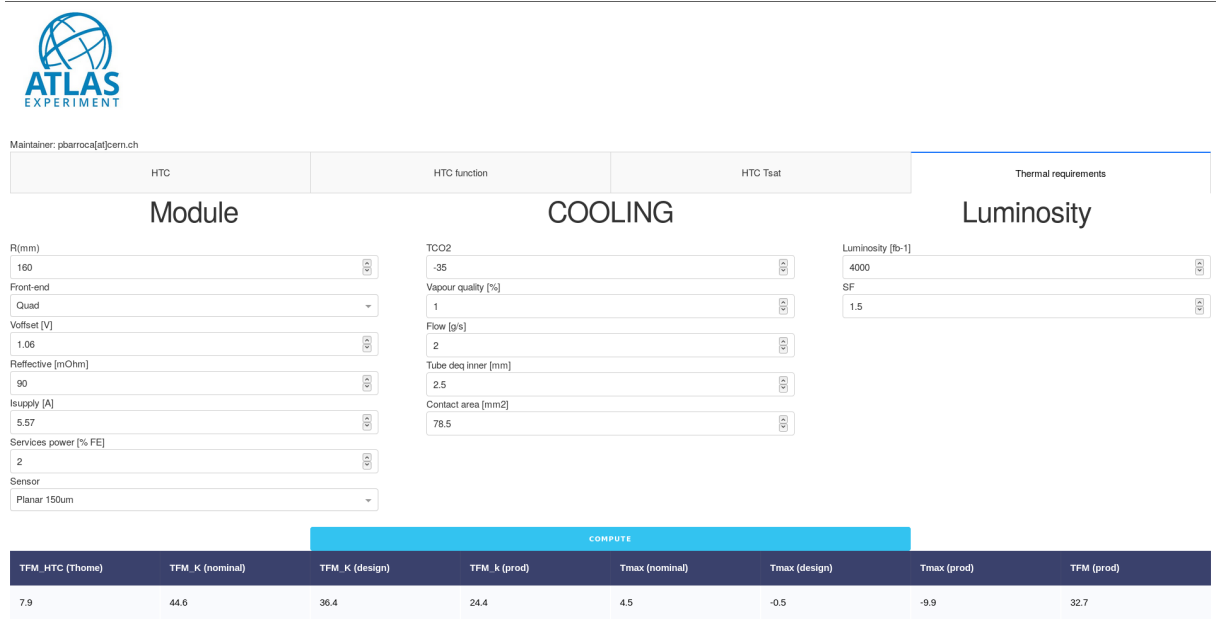
Table 5.4: Specifications for the Thermal Figure of Merit (TFM_{TOT}) in Kcm^2/W units. The final prototypes must respect the requirement set for production which considers the total fluence multiplied by a safety factor of 1.5, 20% current overhead on the front end chip and the module production variability.

Layer	Nominal		Design		Production			
	T_{max}	TFM_{TOT}	T_{max}	TFM_{TOT}	T_{max}	TFM_{HTC}	TFM_K	TFM_{TOT}
Layer 0	5.5 °C	41.2	3.5 °C	30.4	-11.5 °C	2.9	20.4	23.2
Layer 1	6.7 °C	56.7	5.3 °C	42.7	-9.7 °C	4.7	28.6	33.3
Layer 2	4.5 °C	52.9	1.3 °C	37.2	-8.7 °C	7.8	24.9	32.7
Layer 3	12.9 °C	66.6	8.1 °C	46.1	-5.1 °C	8.0	30.9	38.9
Layer 4	22.8 °C	83.1	13.9 °C	53.9	-1.9 °C	8.0	36.1	44.1

These results can be easily retrieved from the web app [60].

thermal specifications after every change on the design, this python package was developed to work seamlessly with the CO₂ simulation tool presented in chapter 3 to compute the heat transfer coefficient.

In the web interface presented in section 4.7, a fourth tab was integrated to rapidly calculate the thermal specifications of the local support for a given set of inputs - see Figure 5.20. With an easy to use web application, engineering teams and physicists could rapidly evaluate how changes in the design would affect the thermal management of the local support.



TFM_HTC (Thome)	TFM_K (nominal)	TFM_K (design)	TFM_k (prod)	Tmax (nominal)	Tmax (design)	Tmax (prod)	TFM (prod)
7.9	44.6	36.4	24.4	4.5	-0.5	-9.9	32.7

Figure 5.20: Thermal management tool integrated in a web interface to calculate the thermal specifications of the local support for the ATLAS Pixel Detector.

5.10.2 Critical CO₂ temperature

The cooling system must ensure that the sensors in the detector are always operating below their critical temperature to avoid thermal runaway.

At the beginning of the detector operation, the power dissipation of the module is less critical so there is more headroom for the coolant temperature to be set in the CO₂ cooling system. This

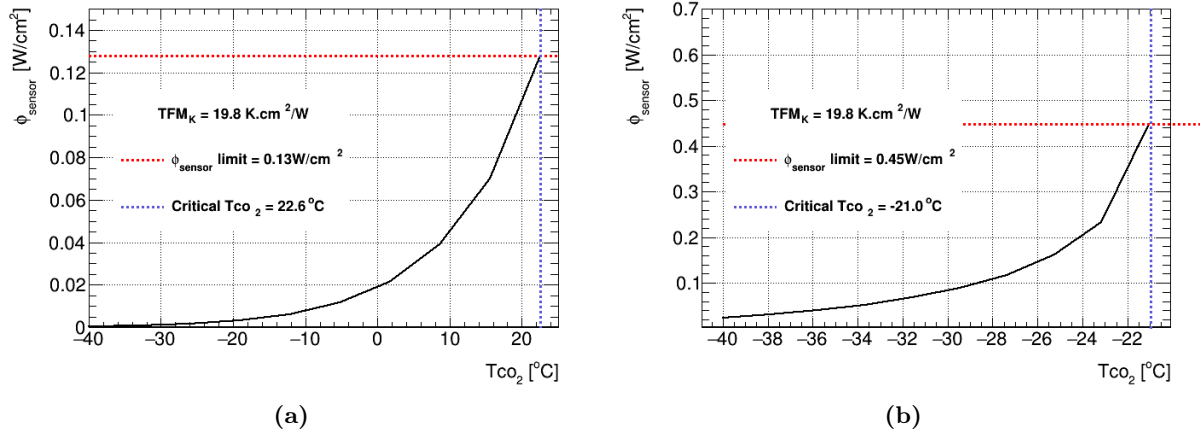


Figure 5.21: Calculation of critical CO_2 temperature for the layer 2 of the Outer barrel with a total fluence of 100 fb^{-1} in (a) and 4000 fb^{-1} in (b) (safety factors considered).

is visible in the estimates of the CO_2 critical temperature shown in Figure 5.21 for a local support with a TFM of $19.8 \text{ K.cm}^2/\text{W}$.

It is thus important to estimate the critical coolant temperature profile throughout the lifetime of the detector, so cooling system controllers are well aware of the safety margins.

Figure 5.22 shows the CO_2 critical temperature profile for a local support in Layer 2, with a TFM_K of $30 \text{ K.cm}^2/\text{W}$ (maximum allowable by design) and considering all the safety factors referred to previously. Two critical profiles are traced: one with the 10 nA current per pixel limitation and a second one, only considering the thermal runaway at the sensor. A third consideration, with a TFM_K of $19.8 \text{ K.cm}^2/\text{W}$, was added to show how the thermal impedance of the local support impacts the profile.

One can conclude from Figure 5.22 that the 10 nA limitation is the most constraining requirement. It is important to remember that this limitation is specific to the technology used by the pixel readout chip pumping heat into the silicon substrate and that this result corresponds to the most pessimistic case scenario as it considers the highest allowable TFM_K with conservative safety margins.

5.10.3 Dry-out

The results presented above assumed a priori the presence of two-phase CO_2 with an annular or bubbly pattern flowing inside the cooling pipes. It is crucial to work under this flow patterns to guarantee good wettability of the inner tube walls and for a good cooling performance. In case there is dryout, vapour will be in contact with the tube walls instead of liquid, degrading the heat exchange effectiveness between the coolant and tube wall. This will lead to an increase of the tube wall temperature and thus, the sensor temperature, risking the destruction of the pixel module.

The CO_2 flow pattern can be predicted using the set of parameterized functions proposed by the Moreno Quibén and Thome model discussed before in section 2.5. The mass flow and vapour quality are the main parameters driving the flow pattern. The vapour quality, by its turn, will depend on the heat exchange between the evaporator (detector) and the coolant, changing its enthalpy. As the detector power dissipation is known, one can estimate the minimum mass flow needed to guarantee an annular or bubbly flow in the detector section.

The vapour quality above which the flow transits to dry-out, is called dry-out incipience (x_{di}). Hopefully, the change from annular to dry-out does not happen suddenly. The estimated vapour quality for dryout incipience must be taken with a safety margin. A vapour quality of 30% – 40%

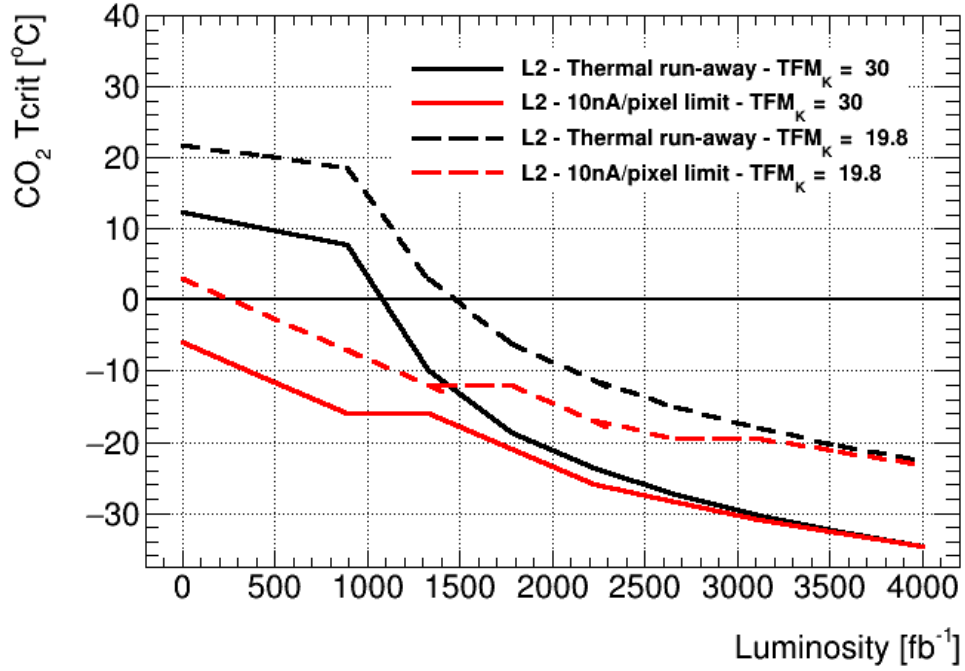


Figure 5.22: Critical CO_2 temperature above which the thermal management requirements will not be respected for Pixel Layer 2 with safety factors taken into account. Black and red solid lines show the critical profile for a local support with a TFM of $30 \text{ Kcm}^2/\text{W}$ only considering thermal run-away and considering the 10 nA/pixel front end limitation. The red dashed line considers a local support with lower thermal impedance (TFM of $19.8 \text{ Kcm}^2/\text{W}$).

is typically what is set as maximum allowable vapour quality at the end of an evaporator because there is yet no experimental data in the literature observing dryout below this range.

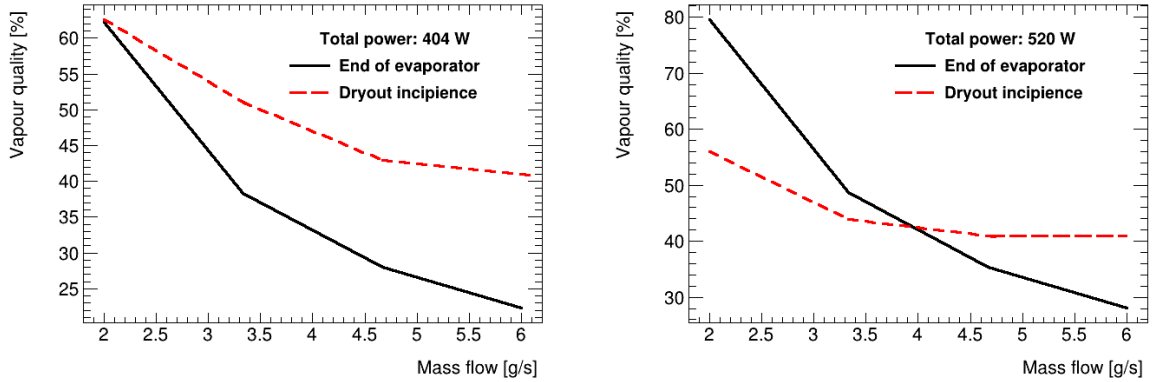


Figure 5.23: Estimate of dry-out limit using a CO_2 temperature of -35°C from the interception of dry-incipience estimate (**red line**) and the vapour quality expected at the end of the evaporator (**black line**).

The cooling lines that will see most of the power to be dissipated are the ones responsible for cooling the flat cells mounted on the longerons. As shown in section 3.9, a single cooling line is expected to cool down the 36 quad modules installed onto a longeron. The second layer will be the layer in the outer barrel with more power dissipated by the sensors (due to higher radiation damage) and front-ends (higher currents). Thus, layer 2 will be once more, be taken as reference for the whole pixel outer barrel.

The topmost estimate for the power dissipated in the module is given by the sensor operating

point, as shown in Figure 5.17. With 0.92 W/cm^2 , the 36 modules dissipate up to 520 W.

Using the simulation tool we can retrieve the vapour quality at the beginning and end of the longeron as a function of the inlet mass flow and check how far we are from the dry-out incipience.

The result shown in Figure 5.23 will assume the ideal: a CO_2 temperature of -40°C at PP1, a pressure difference between the PP1 outlet and the entry of cooling loop equal to the 2 bar budget set by the cooling system (leading to an inlet temperature around -35°C) and a pressure drop at the capillary, so we enter the evaporator with 100% of saturated liquid CO_2 .

From the results it is possible to conclude that, in order to ensure that the detector will not reach the dry-out at the end of life of the detector, the cooling system must be designed to feed each cooling line with at least 4 g/s of CO_2 . The results also show that, in the beginning of the detector operation, as the power dissipated will be lower (the results show for the case of an average power dissipation of 0.7 W/cm^2 per module) the minimum mass flow requirement will be way less constraining ($\approx 2 \text{ g/s}$).

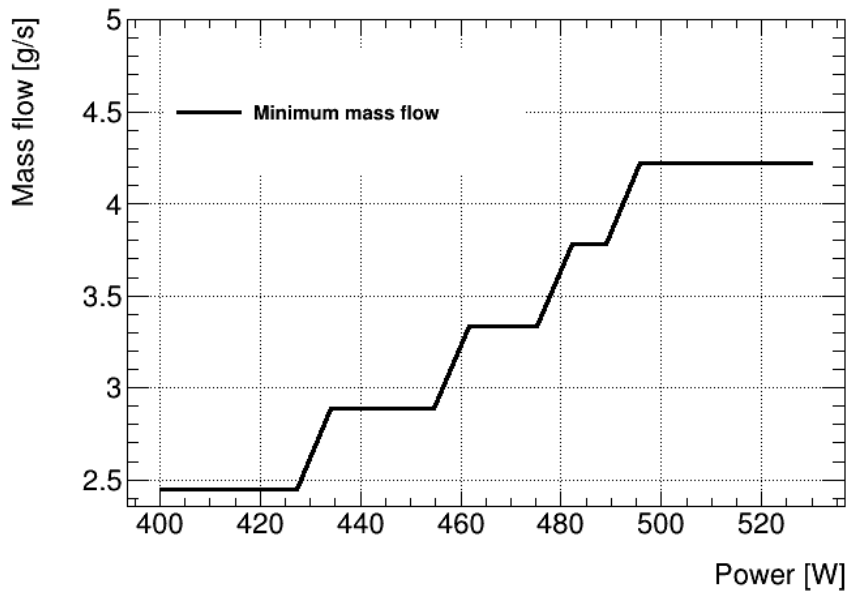


Figure 5.24: Results for the minimum flow needed in the longeron to guarantee an annular flow along the evaporator section.

Another way to present the results is shown in Figure 5.24, where the required mass flow range needed to avoid dry-out is presented as a function of power dissipated in a longeron.

These results do highlight how important is that the CO_2 cooling control system keeps track of the power being dissipated in the evaporators while keeping an eye on these estimations to guarantee there will not be dry-out in the evaporator section.

5.10.4 Sizing of capillaries

The CO_2 cooling system is designed for a maximum pressure drop of 10 bar in evaporators cooling loop.

Setting the CO_2 temperature at the outlet of PP1 at -40°C ($\approx 10 \text{ bar}$) we will need a pressure drop in the inlet capillary above 8 bar to enter the evaporator with a two-phase CO_2 mixture. The pressure drop in the capillary will depend on its diameter, material roughness and the mass flow (its length is fixed a priori).

One must avoid a total pressure drop above 10 bar to not compromise the CO_2 pumping system. Hence, the way to design a capillary is to consider the maximum mass flow that the

detector will eventually need to be cooled down and evaluate the estimated total pressure drop. As commented earlier, the minimum mass flow required to avoid dry-out is way higher at the end of life of the detector, than at the beginning (4 g/s instead of 2 g/s).

To investigate what is the best suited diameter for the inlet capillaries for the outer barrel, it is calculated the expected total pressure drop for a flow of 5 g/s with an inlet capillary 2.45 m long (see Figure 3.15) but different diameters and wall roughness - see Figure 5.25.

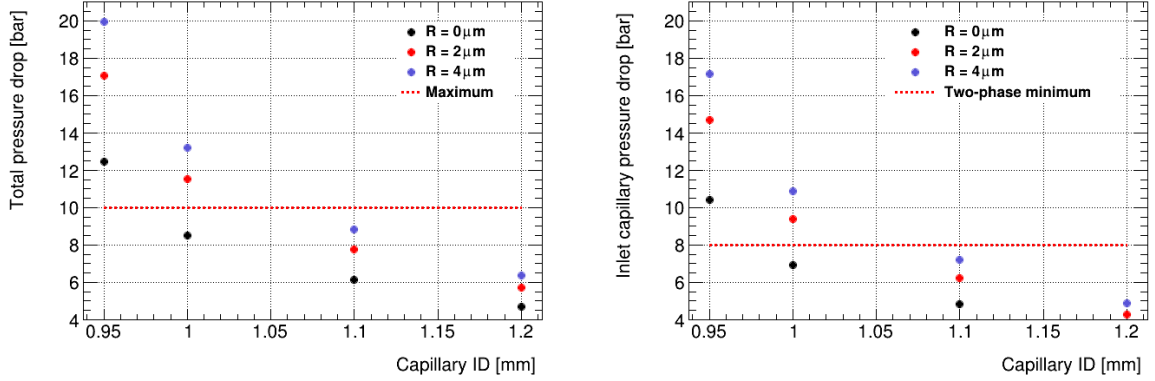


Figure 5.25: Total pressure drop expected from in a cooling loop (from PP1 to PP1) for an outer barrel longeron, at the end of life of the detector (520 W) with 5 g/s of CO₂ for different types of capillaries. *R* stands for inner tube wall roughness.

From the predictions, one can conclude that the capillaries will most probably need an inner diameter between 1 mm and 1.1 mm, depending on the tube wall roughness. The fact that the pressure drop highly depends on the pipe roughness, stresses the need for a prior fluidic qualification tests before choosing between the capillary candidates.

5.10.5 Warm nose operation

At the beginning of the detector operation, the power dissipated in the longerons will not be as high as at the end of life of the detector. Taking into consideration Figure 5.24, there is indeed a big difference between the mass flow needed depending on the power being dissipated.

Although there is a margin for the mass flow, one should keep in mind that if the capillary is dimensioned to work with 5 g/s, lowering the mass flow may compromise the two-phase flow at the evaporator inlet.

Hopefully, there is the warm nose bypass system (Figure 3.11), which is an useful tool in case the pressure drop at the capillary is not enough to enter the two-phase region at the evaporator inlet. The setback for using the warm nose is the increase of the temperature at the evaporator inlet. From Figure 5.26, is possible to evaluate the impact of the operation of the warm nose depending on the pressure drop seen at the inlet capillary for different temperatures set in the accumulator. The temperature difference might be optimistic as it is assumed at the inlet of evaporator sits on the saturation line, without a vapour quality security margin usually considered (as illustrated Figure 3.12 in section 3.8).

These results may be especially useful if during the starting phase of the detector, when the cooling system may work at warmer CO₂ temperatures, the power dissipated will be minimal, and the cooling system needs for any reason to work with with mass flows below the nominal value for which the inlet capillaries were designed.

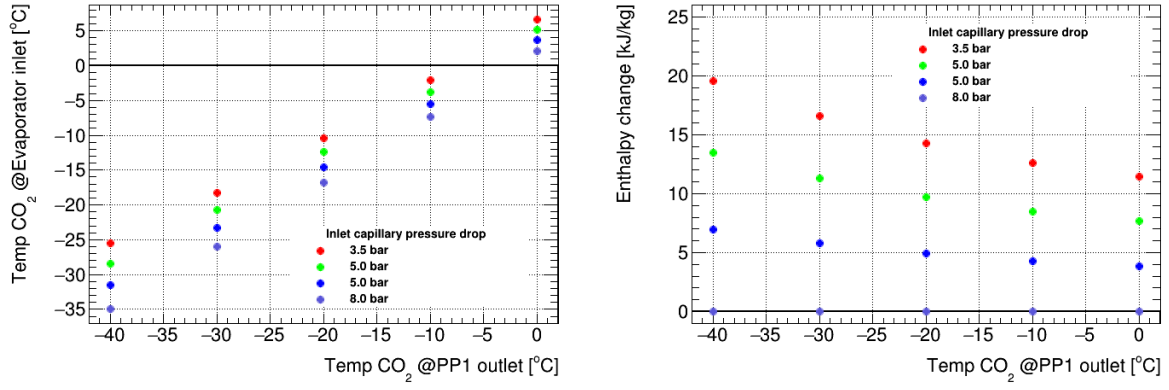


Figure 5.26: The figure on the **left** shows the expected temperature difference between the PP1 outlet (temperature set by the CO₂ cooling system accumulator) and the inlet of evaporator. The result is calculated considering different pressure drops at the inlet capillaries (from 8 bar to 3.5 bar). A pressure drop below the 8 bar needs to be compensated by the warm nose, responsible to change the enthalpy of the subcooled CO₂ until it reaches the two-phase region. The corresponding enthalpy change needed in each scenario is shown in the figure at the **right**.

5.11 Temperature profile of the ATLAS Pixel Detector

The expected temperature profile for the Layer 2 of the Outer Barrel is shown in Figure 5.27. It considers a $TFM_K = 19.8 \text{ Kcm}^2/\text{W}$, for which is expected a total module power dissipation of 0.67 W/m^2 at the end of life of the detector, thus a total power dissipation around 386 W across the 36 quad-modules in the longeron.

Assuming the maximum allowable conductive TFM of $30 \text{ Kcm}^2/\text{W}$ is respected, the sensor is expected to have temperatures between -20°C and -5°C - see Figure 5.27.

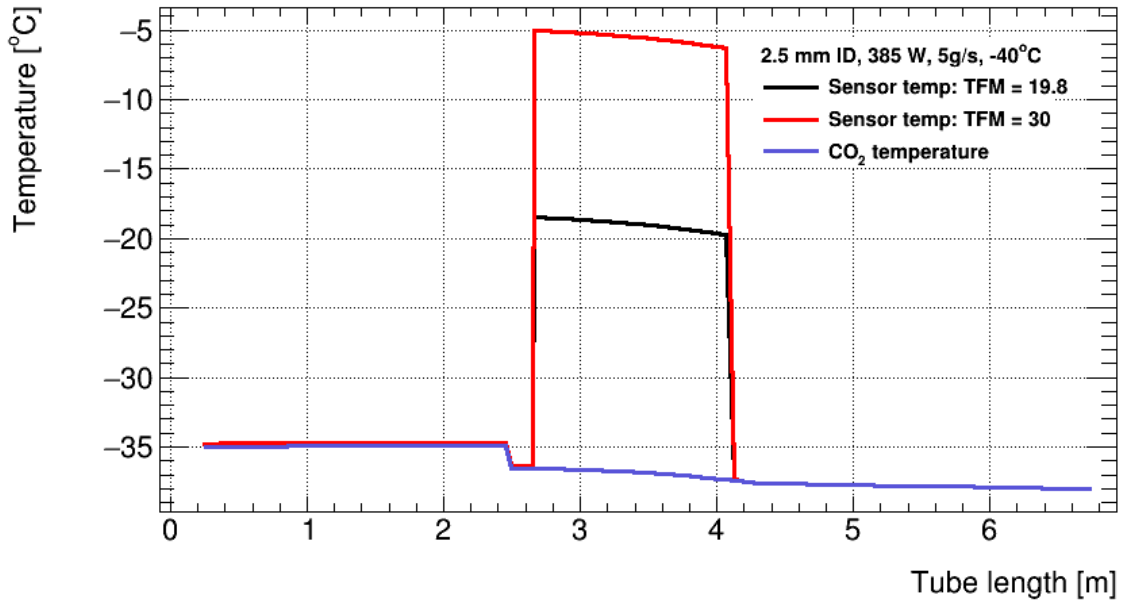


Figure 5.27: Module temperature profile along the 36 modules installed onto the longeron, cooled with 5 g/s of CO₂ with a temperature at PP1 (outlet) of -40°C . Simulation considers a local support with a conductive TFM of $19.8 \text{ Kcm}^2/\text{W}$ (estimated from simulations) and $30 \text{ Kcm}^2/\text{W}$ (maximum allowable thermal figure of merit).

5.12 Manifolding and flow distribution

The cooling services will be manifolded in groups of seven longerons and nine inclined rings. As explained in section 3.9, it is important to investigate how the CO₂ will be shared between parallel cooling lines in all possible scenarios.

Each longeron and inclined ring is equipped with an individual inlet capillary, responsible for the pressure drop before the evaporator section. The manifolding groups are going to be organized so the inlet capillaries will be the same for every single evaporator in the manifold.

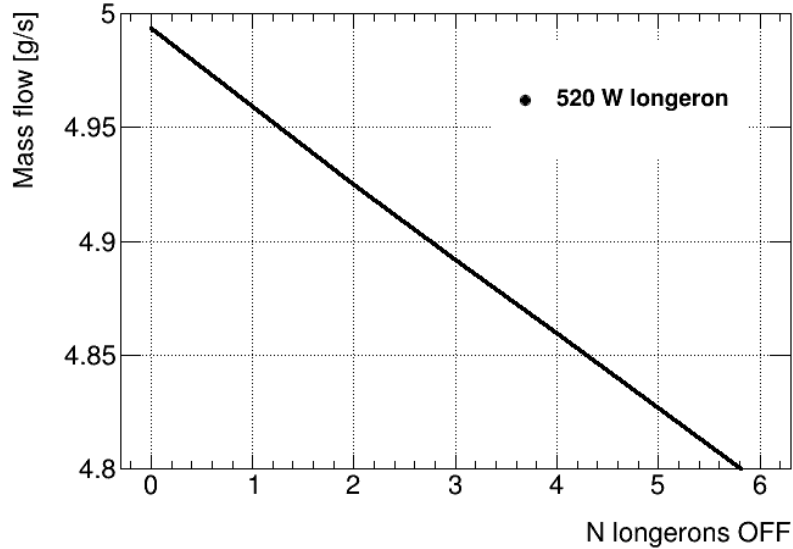


Figure 5.28: Change of mass flow in a longeron when the power on other longerons sharing the same manifold is turned off. Capillaries 2.45 m long with 1.1 mm inner diameter and the cooling system feeding 35 g/s of CO₂ into the manifold i.e. 5 g/s per longeron.

Capillaries are used both to enter in the two-phase region the closest possible to the evaporator section and to reduce the sensitivity of the mass flow to changes on the pressure drop experienced along the evaporator section.

There may be flow distribution issues in parallel lines feeding similar evaporators (same cooling pipe and power dissipation) if there are big differences on the power dissipated between evaporators.

In a group of seven longerons, if one longeron is turned off, it is expected a slight decrease of the mass flow on the remaining six longerons. However, the nominal mass flow at each cooling line should always respect the minimum mass flow condition referred in the previous section.

The CO₂ simulation tool provides one utility to estimate the flow distribution in groups of several evaporators. The highest flow unbalance one may experience, will be at the highest power dissipated in one longeron while all others manifolded in parallel are turned off. Thus, was calculated what is the expected change of mass flow in one evaporator turned on (longerons and inclined rings) when one or more evaporators connected in parallel is/are turned off. This was conducted considering 520 W dissipated per longeron (worst case scenario) in a seven longeron manifold, with a 1.1 mm ID capillary with 0.1 μ m roughness.

Thanks to the high pressure drop in the capillaries when compared to the pressure drop in the evaporator, the impact is expected to be small. For the worst case scenario, with just one longeron operating, the mass flow would drop from 5 g/s to 4.8 g/s. In such conditions, if we want to keep providing the flow needed, the cooling system would need to compensate with roughly 0.2 g/s per cooling line i.e 1.5 g/s at the PP1 inlet.

The same exercise can be performed for every manifold intended to be used in the detector.

For more reliable estimates, one needs, in the first place, an experimental fluidic characterisation of the capillaries and titanium pipes to be used, to cross check if the experimental pressure drops measured are in agreement with the estimates.

THERMO-FLUIDIC QUALIFICATION OF FIRST LOCAL SUPPORTS

During the ITk Pixel detector design phase, two mechanical solutions were being considered to host the inclined modules: the "Alpine mountains", and the SLIM base blocks. The former had the advantage of very high thermal performance, based on the same monoblock carbon foam conception as the IBL detector, while the second concept, quite innovative, had the significant advantage of being "reworkable", to ease the replacement of faulty modules during the detector assembly.

To compare simulations with the first testing on real prototypes, an experimental test bench has been built at LAPP, capable to replicate the CO₂ cooling conditions and minimise systematic sources of error from air convection, by including a vacuum vessel.

The goal of the measurement campaigns was to verify the compatibility of both local support concepts with the thermal requirements. It was also the opportunity to verify the validity of the thermal simulation used to optimise the thermal materials and interfaces to get the best performance while minimising the amount of passive material.

Chapter outlook and personal contributions:

The expected thermal performance of each prototype is estimated from a finite elements analysis performed with ABAQUS (*Dassault Systemes*) [70]. The simulations are presented and the modelling discussed. Special care is given to the uncertainties committed in the simulation of the thermal performance of the prototypes. This new approach is at the basis of the next test campaigns where every sample was preceded by an extensive assessment of uncertainties in the conductive and convective model used to estimate the global thermal performance. The test setup was conceived in an iterative process throughout several conducted testing campaigns. The continuous effort to improve the setup reliability by minimising the experimental sources of errors were decisive in the chosen instruments, sensors and used methods for thermal probe attachment. The use of a vacuum vessel was determinant to understand why the radiative heat pick-up is important to be taken into account whilst the qualification of samples cooled with evaporative systems. These first tests were important to validate the reliability and robustness of the test setup and tools used to treat the experimental data.

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6.1 Alpine prototypes

The Alpine local support is an IBL-like machined carbon foam stave. While in the IBL all the modules are placed along the cooling lines, the alpine stave also supports inclined modules as discussed in section 1.3.3.

Originally, the Alpine design used two titanium tubes with 150 μm thick walls and an inner diameter of 2.4 mm for a maximal thermal efficiency, as shown in Figure 6.1b. In order to minimise the passive material, a second prototype, with just one cooling tube only, was also built - see Figure 6.1a. The thermal testing campaigns were conducted on two prototypes with two tubes (M016 and M017) and three prototypes with a single tube (M019, M020, M021).

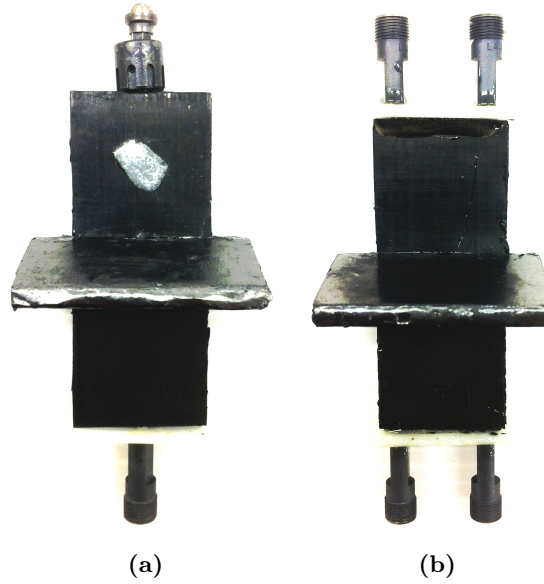


Figure 6.1: *Alpine local support prototypes with one and two titanium tubes.*

Materials

With an inclined foam structure, glued with Stycast epoxy to titanium tubes, new challenges arose from the thermal management and physics standpoint. The final design and materials were decided upon following an extensive study [52] to evaluate the best compromise between minimisation of material and thermal performance.

A carbon foam with a density of 0.6 g/cm³ was chosen as the best candidate for its good thermal conductivity to mass density ratio. It is coated with a flexible graphite sheet (FGS), to prevent carbon dust deposition on the sensors, and to improve the thermal conductivity of the local support. The composite K13 is used to provide mechanical rigidity and the thermal

pyrolytic graphite (TPG) glued onto the back of the front end chip is fixed onto the Alpine "mountain" with a temperature interface material (MPC 315).

To be as close as possible to the thermal behaviour of the local support in real detector conditions, the prototypes were loaded with 700 μm thick silicon heaters (36.2 mm x 21.3 mm), to mimic a dual silicon module. The prototypes were then equipped with two temperature probes, one on each top corner.

Figure 6.2 shows a schematic view of an alpine local support, and the materials used for the coating of the carbon foam. The thermal conductivities of the prototype materials and respective uncertainties are discussed in the next paragraph.

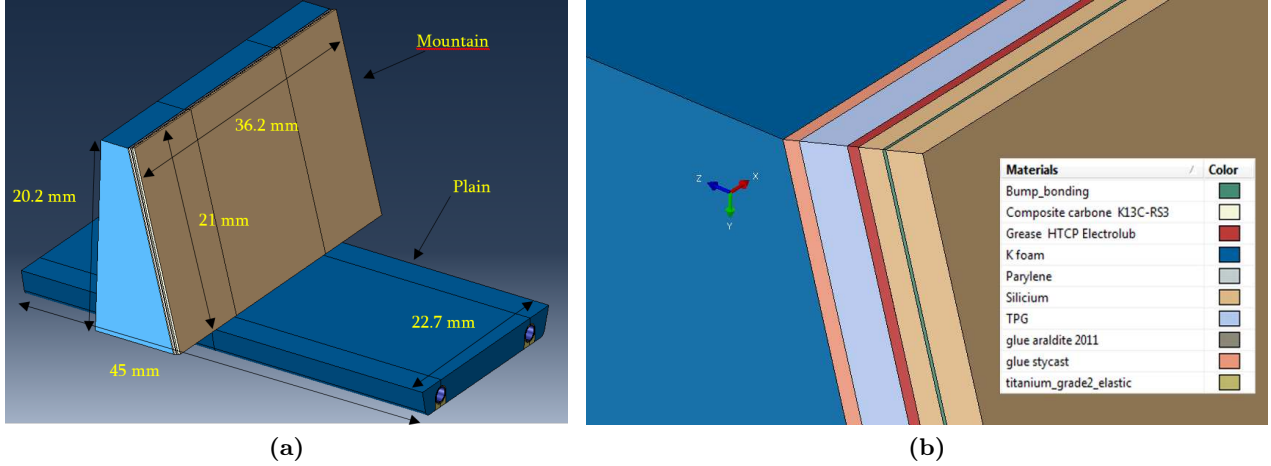


Figure 6.2: Schematic view of an alpine "mountain" (a) and stack-up of the assembly materials (b).

FEA uncertainties

The gradient of temperature from heat conduction through the local support thermal interfaces is retrieved from the heat conduction solution of a numerical finite element analysis (FEA) on a 3D model of the local support. Models used to describe the design of real structures and their materials are always limited to some degree of uncertainty. Either in terms of its dimensions or the properties of the materials relevant for the type of study.

In stochastic FEA analysis, independent studies are computed, considering that each property of the material can be retrieved from a probability function, representing its level of uncertainty. For the Alpine model, only the thermal conductivity of the materials was considered unknown. The models are relatively simple with a global thermal performance inversely proportional to the change of thermal conductivity of each material. Therefore, in order to estimate the uncertainty on the FEA Heat Transfer analysis, the lower and upper bounds of the uncertainty on the thermal conductivity for every material were considered (see Table 6.1), to calculate the range for the gradient of temperature expected from thermal conduction at the local support.

The sensitivity on each of N parameters (σ_i) is considered to be retrieved from a normal probability distribution. Thus, the total uncertainty on the FEA results (σ_{FEA}) is calculated with the root sum squared method:

$$\sigma_{FEA} = \sqrt{\sum_{i=0}^k \sigma_i^2} \quad (6.1)$$

The final result for the conductive contributions (ΔT_{cond}) of the alpine prototypes with one and two tubes are $1.43 \pm 0.19^\circ\text{C/W}$ (TFM_K $11.5 \pm 1.2 \text{ Kcm}^2/\text{W}$) and $1.07 \pm 0.06^\circ\text{C/W}$ (TFM_K $8.6 \pm 1 \text{ Kcm}^2/\text{W}$) respectively - see Figure 6.3.

Material	Thermal conductivity [W/(m K)]	Uncertainty [W/(m K)]	Sensitivity [°C/W]	Uncertainty @6W [°C]
Kfoam 150	150	$\pm 10\%$	[-0.05, +0.06]	[-0.3, +0.4]
MPC 315	4	± 2	—	—
K13 MTM49-3	(96, 0.5, 0.5)	$\pm 10\%$	—	—
FGS-010	(750, 750, 10)	$\pm 10\%$	[-0.01, +0.01]	[-0.06, +0.06]
TPG	(1500, 1500, 15)	$\pm 10\%$	—	—
Silicon	190	± 40	[0, +0.01]	[0, +0.06]
Stycast epoxy	1.1 [71]	$\pm 30\%$	[-0.2, +0.3]	[-1.2, +1.8]
Titanium	23	± 2	—	—

Table 6.1: List of materials and corresponding thermal conductivities considered in the FEA model for the Alpine prototype with one tube at -25°C . For the materials with lack of information on their thermal conductivity, an uncertainty of $\pm 10\%$ is considered. Sensitivity below 0.01°C/W is ignored.

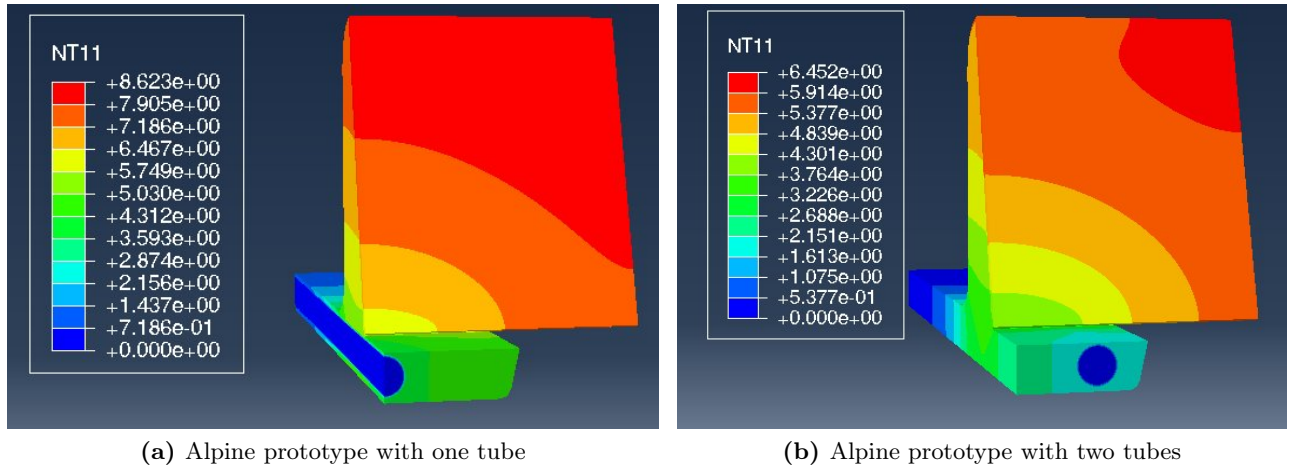


Figure 6.3: Expected ΔT_{cond} from ABAQUS FEA results, for Alpine prototypes with one and two cooling tubes (only half of the module was modelled to speed up simulation), when applying a total heat load of 6 W. The simulation considered the central values from Table 6.1

Establishing a good thermal contact between porous carbon foam and the pipe (contact resistance) is quite difficult to simulate. Moreover, the Stycast epoxy interface is the main source of uncertainty as it is the material performing the worst thermally and its contact thickness is hard to control and model. The error due to different curing types was reported to $\pm 20\%$ [71]. A total uncertainty of $\pm 30\%$ was considered to cope with unknown contact resistances. The results for the expected ΔT_{cond} contribution is shown in Figure 6.4.

These limitations were assumed to be the main cause for the big variability in terms of thermal performance experienced in IBL - discussed in chapter 1. While for IBL this could be afforded thanks to big safety margins, this might be not the case for the future ATLAS ITk. The lack of control on the thermal interfaces were one of the causes for dropping the Alpine prototype design in favor of the more robust SLIM solution for the local support.

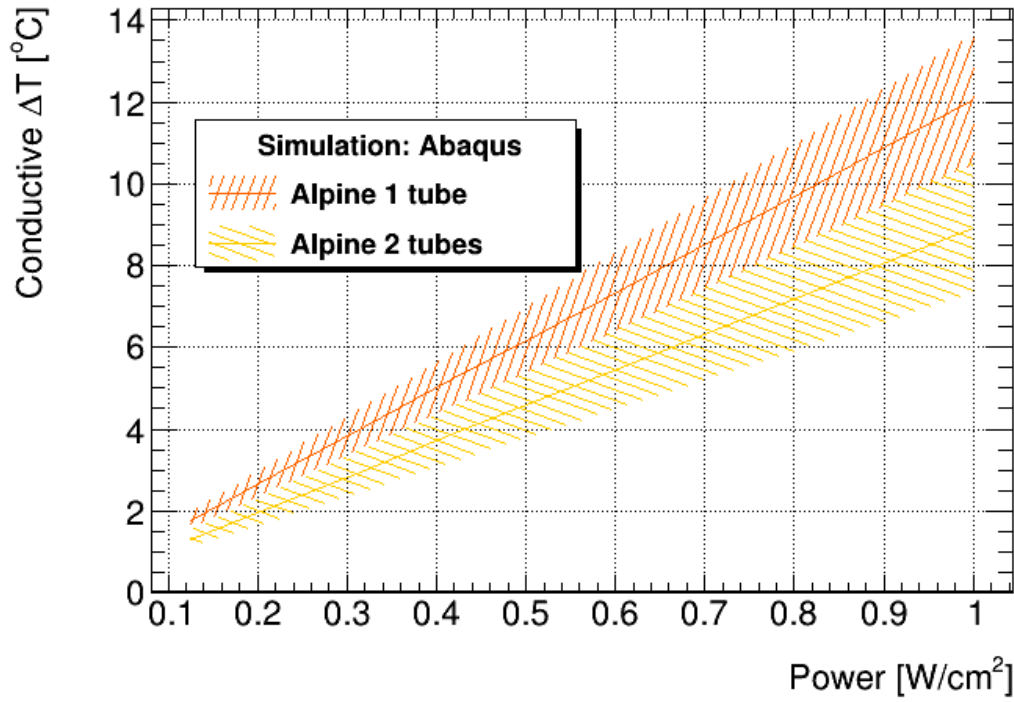


Figure 6.4: *Expected performance from heat conduction of alpine prototypes with one and two cooling tubes.*

Modelling the heat flux on cooling pipe

In the Alpine prototypes, the titanium tubes are surrounded by carbon foam, homogenizing the heat flux all around the tube. Still, it is the top part of the tube where the heat flux is expected to be higher.

The simulations were conducted with a simplistic approach, where the whole inner pipe wall is assumed to have the same temperature regardless of the local heat flux. This is a too optimistic assumption because the inner wall temperature will depend on the local heat transfer coefficient of CO₂ which, in turn, depends on the local heat flux.

Given that lower heat fluxes lead in fact to lower temperatures at the tube wall (for example Figure 2.16 in section 2.5.5), the lower heat flux at the bottom part of the tube will contribute to a more even distribution of the heat flux around the tube wall. Therefore, the average heat flux retrieved at the outer tube walls was the value considered to estimate the heat transfer coefficient for the Alpine prototypes.

Figure 6.5 shows the heat flux distribution around the Alpine prototype with two pipes. The mean heat flux and corresponding uncertainty (one sigma standard deviation) were calculated considering the heat flux at each of the FEA model nodes. The same exercise was performed for the prototype with one tube - see Figure 6.6.

The expected heat flux profiles of $30.5 \pm 2.5 \text{ kW/m}^2$ and $18.5 \pm 2.2 \text{ kW/m}^2$ (6 W on the heater) for the prototypes with one tube and two tubes respectively, and are taken as input to compute the expected ΔT_{cond} contribution shown at Figure 6.4.

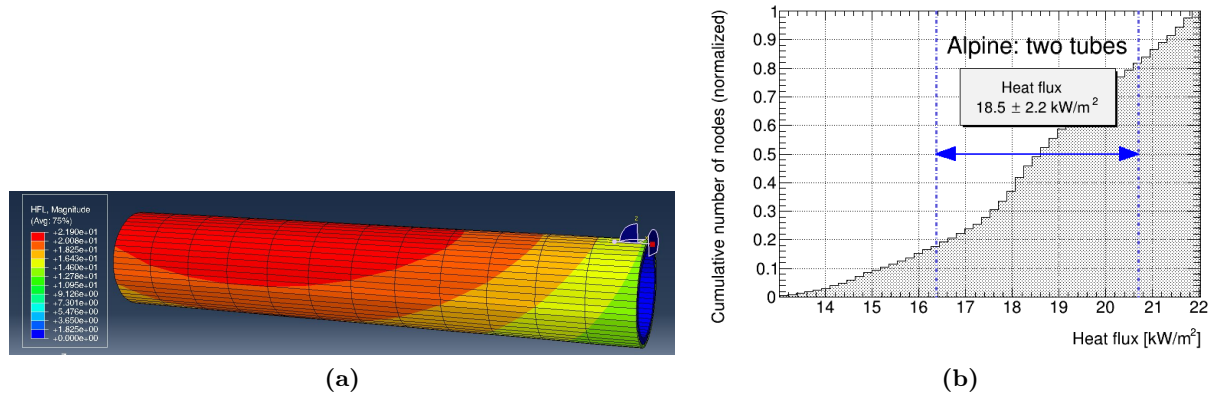


Figure 6.5: (a) Graphical representation of heat flux distribution over the titanium pipe for the Alpine prototype with two tubes and a total heat load of 6W. (b) Cumulative histogram with heat flux measured at each node of the FEA model to calculate the mean heat flux across the pipe.

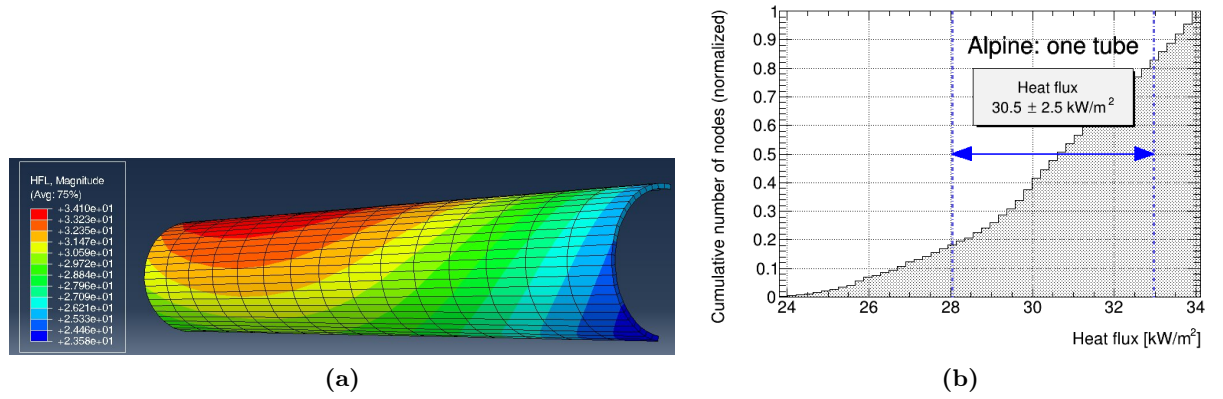


Figure 6.6: (a) Graphical representation of heat flux distribution over the titanium pipe for the Alpine prototype with one tube with a total heat load of 6W. (b) Cumulative histogram with heat flux measured at each node of the FEA model to calculate the mean heatflux across the pipe.

Predicted thermal performance

The predicted thermal performance shown in Figure 6.7 combines the conductive simulation (without any consideration about the heat pick-up from the environment either from natural convection or radiative heat pickup) and the heat transfer coefficient given by the Cheng et al. predictive method with respective error of $\pm 30\%$.

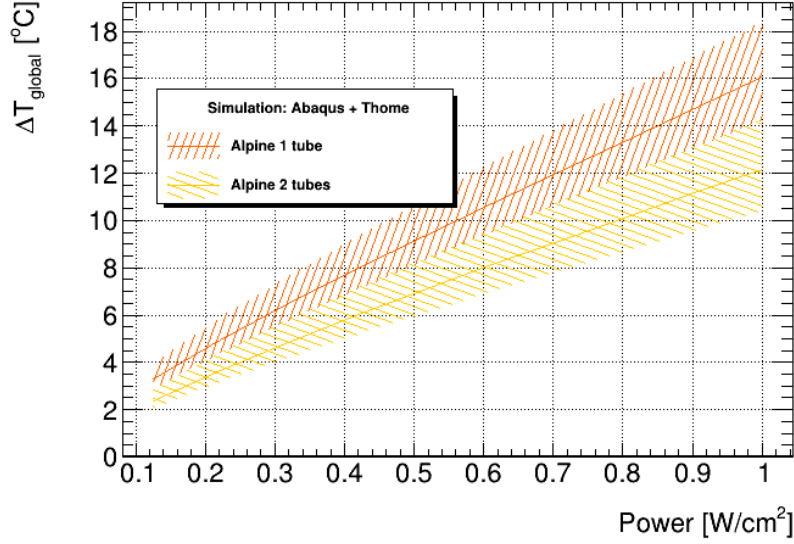


Figure 6.7: *Simulated global thermal performances for Alpine prototypes with one and two tubes.*

6.2 SLIM prototypes

The SLIM local support for a pixel module cell (pixel module + local support) is composed of composite blocks (base block) brazed onto a titanium tube 150 μm thick, with an inner diameter of 2 mm. Each cell has a base block screwed to a second block (cooling block) which hosts the sensor module. The base blocks are the same for all cells while inclined and flat cells have cooling blocks with different geometry (Figure 6.8). For the measurements campaigns were built prototypes with inclined cells as they were the most critical structures given their shape and highest heat fluxes at the pipe (due to lower braze length).

Differently from Alpine, the SLIM design has no foam surrounding the tube. The thermal contact between the tube and base block is through the top half of the tube, brazed along only 8 mm. Hence, the heat flux at the top tube will be not only much higher than for Alpine configuration but the tube wall will also experience a non negligible temperature gradient.

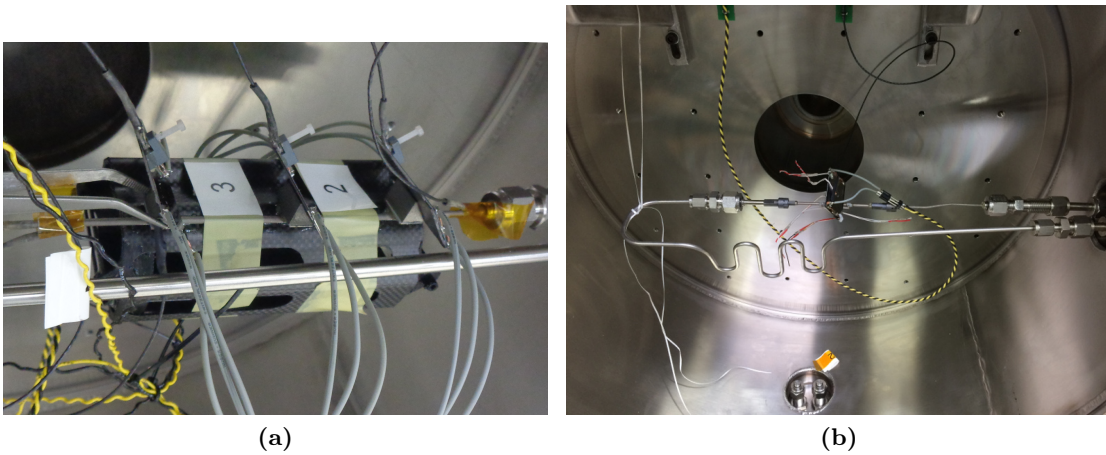


Figure 6.8: (a) Triplet prototype with three SLIM v1 cells (b) and single cell prototype tested at LAPP.

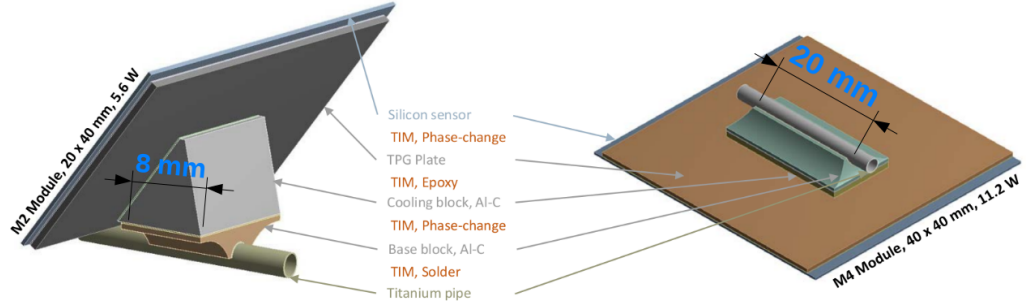


Figure 6.9: Materials and interfaces present in the SLIM inclined (*left*) and flat cells (*right*). Each cell is composed of a common base block, brazed onto the titanium tube.

Materials

The first two SLIM prototypes shared the same design (changed afterwards) and the materials listed in Table 6.2. One prototype was tested from a group of three cells, mounted onto a longeron section (Triplet) and a second prototype was tested from a single cell sample - see Figure 6.8.

The base and cooling blocks are both made of a carbon alloy with good thermal conductivity, interfaced by a layer of MPC315 as temperature interface material. The front-end chip glued to a TPG faceplate with epoxy is glued to the cooling block also with epoxy. The braze of 96.5Sn3.5Ag was chosen because its thermal expansion coefficient (CTE) matched the materials being brazed and presented reasonable thermal conductivity.

FEA uncertainties

In a similar approach to that followed for the Alpine prototypes, the uncertainty on the materials thermal conductivity was also considered to estimate the ΔT_{cond} range. This first version of SLIM presented an expected conductive contribution (ΔT_{cond}) of $1.83 \pm 0.45^\circ\text{C/W}$ ($TFM_K = 14.7 \pm 3.6 \text{ Kcm}^2/\text{W}$), see Figure 6.11. The list of materials and corresponding thermal conductivity are described in Table 6.2.

Material	Thermal conductivity [W/(m K)]	Uncertainties [W/(m K)]	Sensitivity [$^\circ\text{C/W}$]	Uncertainty @6W [$^\circ\text{C}$]
Carbon alloy	(230, 230, 120)	(10,10,10)	[-0.02, +0.03]	[-0.1, +0.03]
MPC 315	4	$\pm 10\%$	[-0.1, +0.3]	[-0.6, +2]
Loaded epoxy	1.5	$\pm 10\%$	[-0.03, +0.05]	[-0.2, +0.3]
Braze (96.5Sn3.5Ag)	55	± 5	[0, +0.01]	[0, +0.06]
TPG	(1500, 1500, 15)	$\pm 10\%$	[-0.02, +0.03]	[-0.1, +0.2]
Silicon	190	± 40	[0, +0.01]	[0, +0.06]
Titanium	23	± 2	[-0.02, +0.03]	[0.1, +2]

Table 6.2: List of materials and corresponding thermal conductivity considered in the FEA model for SLIM first prototypes. For the materials with lack of information on their thermal conductivity, an uncertainty of $\pm 10\%$ is considered.

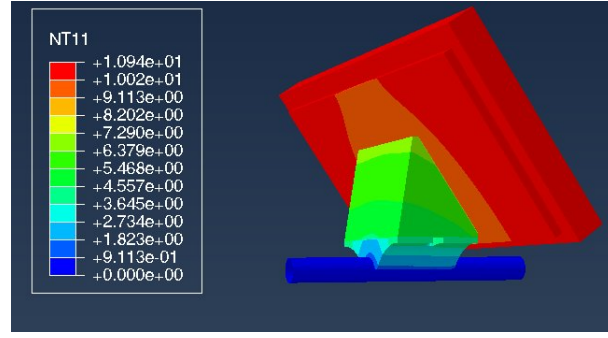


Figure 6.10: ABAQUS simulation results for the SLIM prototype, for an applied heat load of 6W.

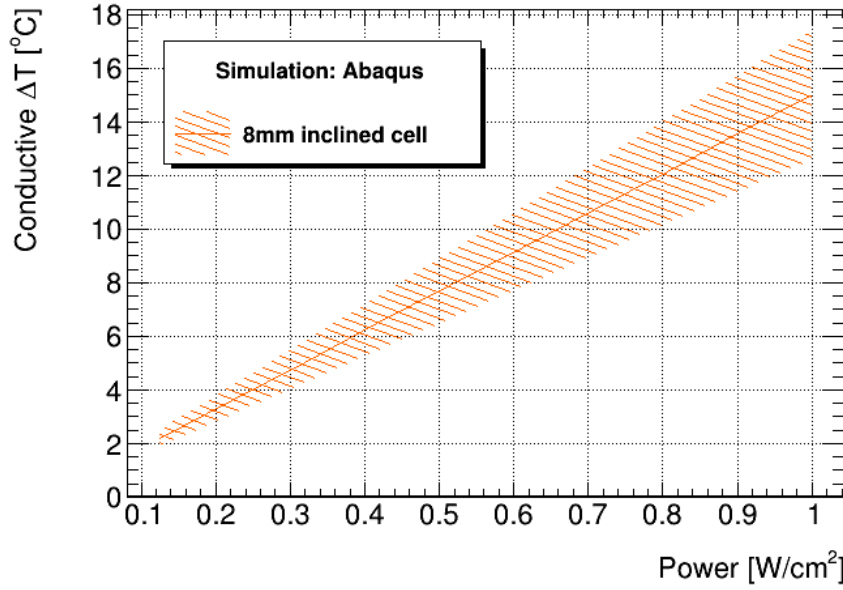


Figure 6.11: Expected thermal performance from heat conduction of SLIM prototype.

Modelling the heat flux on cooling pipe

In a similar approach to that followed for the Alpine prototype, Figure 6.12 shows the expected heat flux profile for the SLIM prototype.

Neglecting the heat drained along the tube walls might lead to a heat flux overestimation at the top part of the wall. This is particularly relevant in the case of SLIM where the heat fluxes are quite high due to the reduced contact area. The heat conduction through the tube walls will depend mainly on the tube thickness and its thermal conductivity. This was investigated in Chapter 8 for an improved version of the local support (ITk OB prototype), with a 100 μm thick titanium tube with 2.5 mm inner diameter. The impact of this effect was investigated by taking different considerations for the HTC at the bottom part of the tube, which will directly affect the mean heat flux at the top half and its homogeneity.

The results are presented in Figure 6.13. It was considered a HTC of 20 kW/(m² K) at the top part of the inner tube wall and a different HTC at the bottom part for each scenario ($HTC_{bottom} = HTC_{top} \times HTC_{cf}$). It is shown that the ΔT_{cond} , considered as the difference between the average temperature of the top half of the tube wall in contact with the braze, is not affected by the heat transfer coefficient i.e the tube wall temperature. Hence, it is indeed valid to treat separately the ΔT_{cond} from the ΔT_{conv} contribution to estimate the ΔT_{global} . It also shows that the mean heat flux at the top part of the tube should be around 80% $\pm$ 5% of

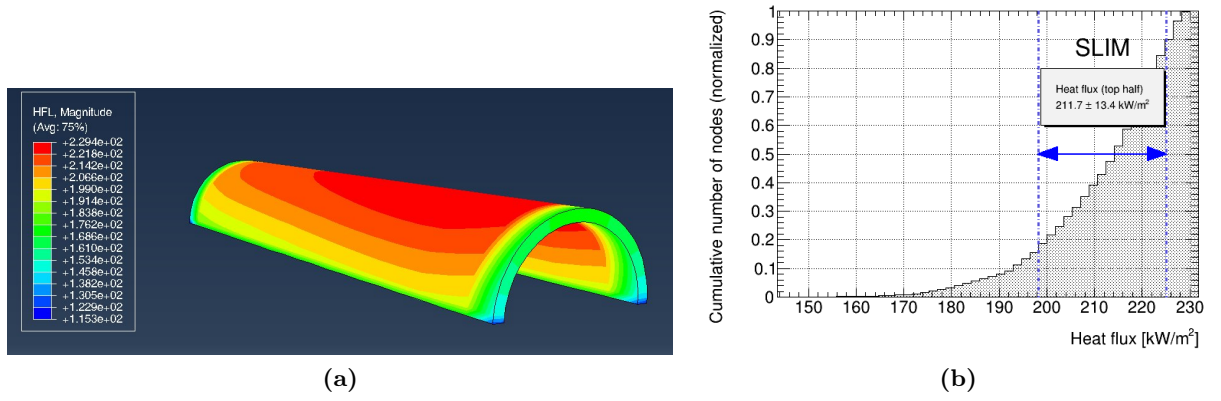


Figure 6.12: (a) Graphical representation of heat flux distribution over the titanium pipe for the SLIM prototype for a 6 W heat load (b) Mean heat flux across calculated from the cumulative histogram with heat flux measured at each surface node.

the maximum heat flux¹. Even though this study was performed for the future design, it was considered here for the SLIM prototypes. The corrected heat flux profile at the top half of the tube, is estimated at $28.2 \pm 2 \text{ kW/m}^2$ per Watt dissipated by the module.

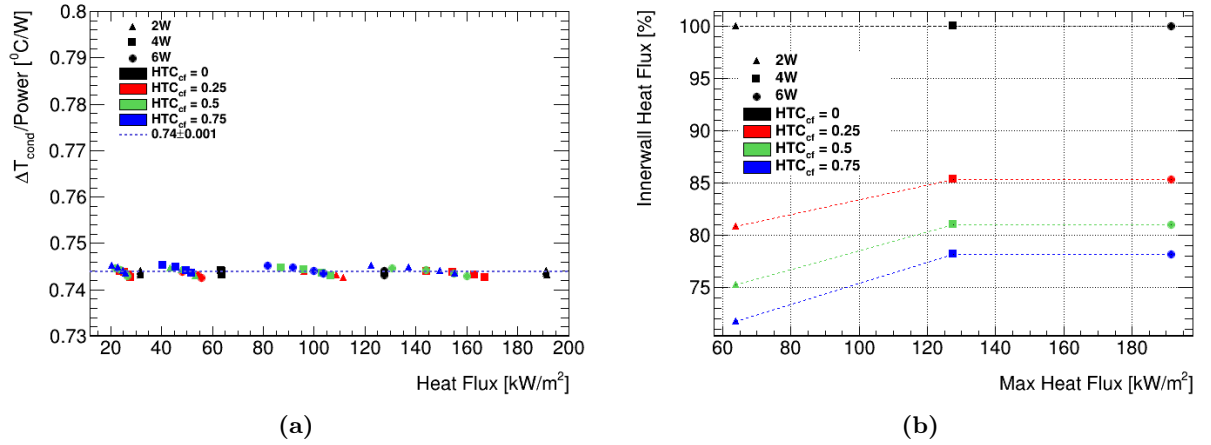


Figure 6.13: (a) Impact of CO_2 heat transfer coefficient on the conductive contribution. At (b) is shown its impact on the top half of the tube flux.

Predicted thermal performance

In a similar approach to that followed for the Alpine prototypes, the thermal performance for the SLIM prototype was simulated. Figure 6.14 shows the result considering the heat flux correction calculated in the previous section.

¹It is assumed that the HTC at the bottom part may be between 25% and 75% of the top inner tube wall's HTC. Parasitic heat pickup ($\approx 0.2 \text{ kW/m}^2$) may lead to a HTC up to $3 \text{ kW/(m}^2 \text{ K)}$ so 80% for the HTC_{cf} was considered as reasonable for the maximum fraction. The maximum heat flux is retrieved when HTC at the bottom part of the tube is neglected.

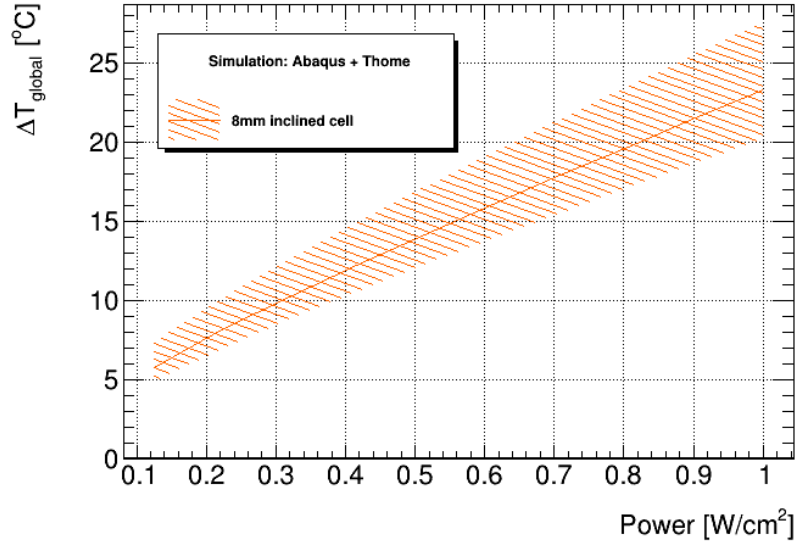


Figure 6.14: *Simulated global thermal performances for SLIM local support.*

6.3 Test setup

The setup presented in Figure 6.15 is composed of hydraulic circuit connecting a bottle with liquefied CO₂ to the blow-off system. The blow-off system is responsible for controlling the CO₂ temperature used to cool the sample which is placed inside the vacuum vessel to minimise air convection. The readout system is based on National Instruments (NI) cDAQ-9179 where up to 14 NI modules (NIM) can be chained on a common serial bus. It is integrated with a LabView human machine interface (HMI) for real time data acquisition, data logging and to interactively control the active elements.

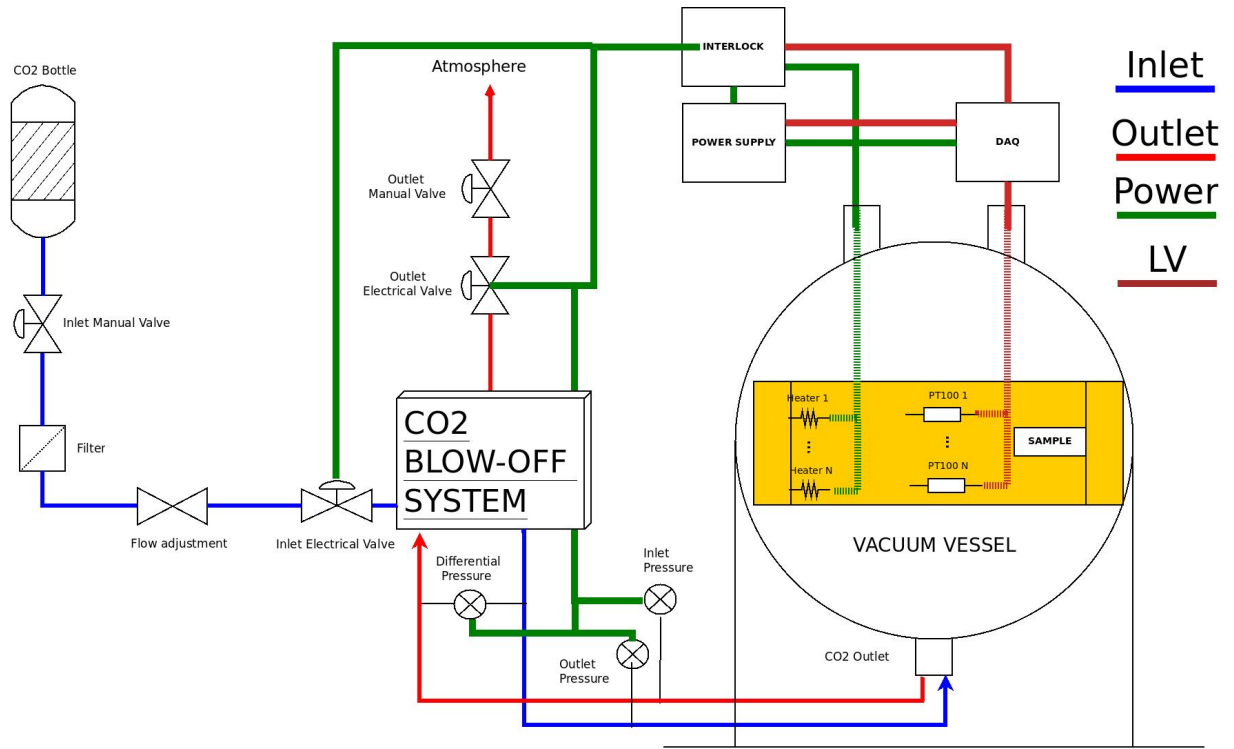


Figure 6.15: Diagram illustrating the interconnection between all the elements composing the experimental setup installed at LAPP used for the thermal management activities.

6.4 CO₂ blow-off system

In 2010, the LAPP team built a CO₂ blow-off system based on the scheme developed at the NIKHEF laboratory [72]. Figure 6.16 displays the CO₂ path. The input CO₂ coming from a bottle is in a saturated state and with a typical pressure of 45-55 bar at room temperature. The pressure drop set at the needle valve is used to place the inlet CO₂ in a two-phase state with temperatures between -40°C to -10°C . The return line coming from the experiment is used to subcool the saturated CO₂ coming from the bottle before being blown into the atmosphere. Thus, it is considered an open loop system. The advantages of this system are the lack of pump, its simplicity to construct and its low cost. On the operation side, the two-phase CO₂ pressure (consequently its temperature) and mass flow can be easily changed manually. Reaching their target values is also fast giving the possibility to run many tests with different conditions in a short period. The main drawbacks are:

- Frequent need to replace the CO₂ bottle;
- The cooling temperature is set manually at the needle valve;
- The system has difficulty to sustain a temperature at the outlet of -40°C with mass flows above 3 g/s. This happens because the pressure drop between point 4 and 5 gets above 5.7 bar, reaching the freezing point at point 5.

The blow-off heat exchanger is dimensioned to feed the evaporator with a vapour quality below 10%. The experimental inlet vapour quality is calculated, considering an adiabatic expansion phase between the heat exchanger and the experiment (2-3) (tube section is rather small and well insulated). A pt100 thermal probe is inserted right after the pre-heat exchanger to compute the enthalpy with Coolprop python package [40] assuming that the liquid is sub-cooled.

To mimic the change of position of the pixel module along a cooling line (avoiding to build a full sized prototype), a cartridge heater is inserted right before the vacuum vessel to change

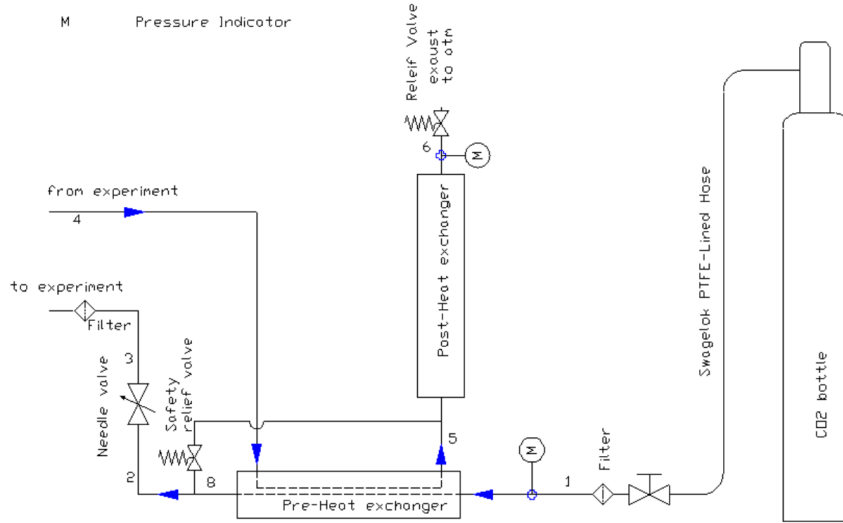


Figure 6.16: Blow-off scheme. *1-2 - Liquid under cooled coming from CO₂ bottle. 2-3 - Expansion phase when theoretically there is an adiabatic process with a pressure fall from the the pressure at the bottle to the pressure set by the relief valve at point 6. 3-4 - Heat exchange phase where energy from the experiment (evaporator) is absorbed by the fluid changing its enthalpy. 4-5 -Pre-cooling of the liquid coming from the bottle using the remaining CO₂ cooling capacity in the experiment return line. 5-6 - The CO₂ is warmed to about 30 °C to avoid freezing. 6 -Venting into the atmosphere.*

the inlet vapour quality. This cartridge can deliver up to 400 W through a JUMO TYA 201 single-phase thyristor power controller.

A Rheonik (RHM 015) mass flow-meter with a precision of 0.001 g/s is placed right before the pre-heat exchanger to compute the necessary heat power to be delivered by the cartridge to reach the target inlet vapour quality. The measurement precision is therefore dominated by the statistical fluctuations of the mass flow measurements around 0.01 g/s.

6.5 Vacuum vessel

Since the measurements are made at low temperatures, the prototypes are inserted into a hermetic vessel which can be emptied of air to minimise the convective heat transfer from the environment. Creation of ice on the prototype is also avoided. A stainless steel vacuum vessel was built with a modular design so its size can be extended to accommodate longer samples. A scroll primary pump is used coupled with a Pfeiffer Vacuum Hipace 300 DN100 turbo pump to achieve a pressure of 10^{-5} mbar. It also includes inlet and outlet feedthroughs to feed the samples with two-phase CO₂ for cooling. It is equipped with a readout system able to read up to sixteen resistive temperature detectors in a four wire configuration, fourteen thermocouples and power eight resistive heaters.

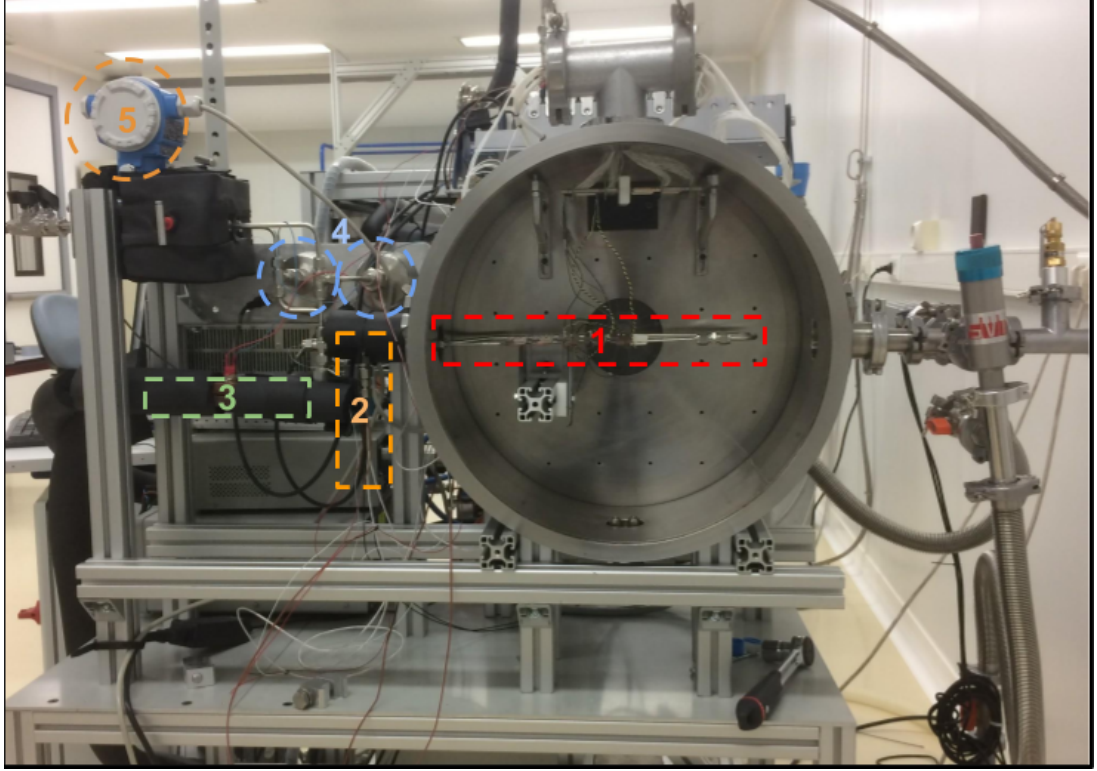


Figure 6.17: Test setup installed at LAPP. (1) - u turn tube connected to CO_2 inlet and outlet integrated with tested sample (2) - pt100 probes inserted at inlet and outlet to measure directly CO_2 's temperature. (3) - cartridge heater inserted before inlet feedthrough to change the CO_2 vapour quality in the experiment (4) - inlet and outlet absolute pressure measurement (5) - differential pressure sensor to measure pressure drop between experiment's inlet and outlet.

6.6 Radiative heat pickup

Although the vacuum vessel greatly reduced the heat pickup from air convection, radiative parasitic heat was identified as an important source of bias at low heat loads, with the temperature sensors measuring a ΔT_{global} of $\approx 2^\circ\text{C}$ with no heat load applied by the heater, on the Alpine prototypes.

The expected sensor net power (q) can be roughly estimated from equation 6.2 where T_e is the warm body (vacuum vessel wall) for an object cooled at T_s . Considering the vacuum vessel's temperature at 20°C , the sensor at -40°C , an emissivity (ϵ) between 0.1 and 0.5 and an exposed area (A_h) overestimated by a factor 1 to 2, the heat pick up estimation is between 0.1 and 0.3 W.

$$q = \epsilon \sigma (T_e^4 - T_s^4) A_h \quad (6.2)$$

Only by considering both conductive and convective contributions, were the measurements successfully explained. With a heat pickup between 0.1 W and 0.3 W, the tube will see locally a heat flux about $2\text{--}6 \text{ kW/m}^2$ (for the prototype Alpine prototype with two tubes). Feeding the CO_2 model with the real cooling conditions (1 g/s and -40°C for the CO_2 temperature) and $2\text{--}6 \text{ kW/m}^2$ for the heat flux, ΔT_{conv} of $1.3 \pm 0.4^\circ\text{C}$ is expected. Summing to the estimated $0.2 \pm 0.1^\circ\text{C}$ for the ΔT_{cond} , the measured 2°C was finally understood. The magnitude of the ΔT_{conv} predicted from the convective model for heat fluxes from 1 kW/m^2 to 5 kW/m^2 is shown in Figure 6.18:

At the beginning, it was not common sense that a non negligible temperature gradient should be expected between the CO_2 and inner tube wall only from the room heat pick up. This result, for a long time not understood, raised the importance to properly factor out the convective from

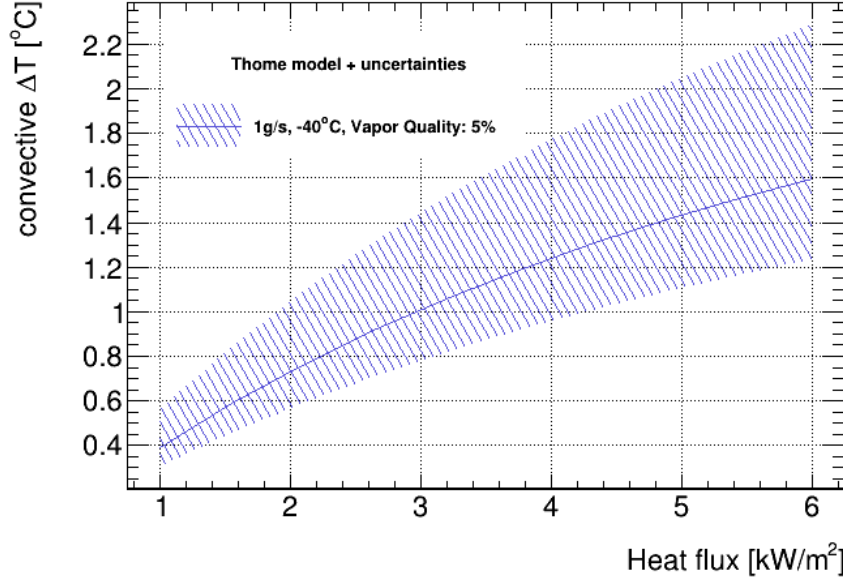


Figure 6.18: Simulation of ΔT_{conv} for low heat fluxes considering Thome model presented in section 2.5 and corresponding uncertainty.

the conductive contribution. Even if both do play a role on the global thermal performance, each one is driven by different parameters.

6.7 RTDs

Platinum temperature sensors (Innovative Sensor Technology, pt 100ohm) with 1/10 DIN accuracy ($<0.05^\circ\text{C}$ at -30°C and 30°C range) are used to measure samples' temperature. A resistive temperature detector (RTD) was preferred over a thermocouple for its better accuracy, long-term stability and repeatability. These pt100 sensing probes are composed of one ceramic plate ($2\text{ mm} \times 2.3\text{ mm}$).

In the early stages of the test setup, thin metal sheathed thermocouples were used to measure the CO_2 temperature. Due to their lower precision when compared to RTDs, they were replaced by 100 mm metal sheathed pt100 probes, placed inside the CO_2 cooling lines to measure the CO_2 sub-cooled temperature in the blow-off and inlet and outlet CO_2 temperatures.

A NI 9213 is plugged to the NI-DAQ to read the pt100s in a 4 wire configuration, a NI 9217 to read thermocouples and NI 9477 which provides a set of relays, to work as an interlock.

6.8 Heaters

For availability reasons, different heaters have been used to mimic the pixel sensors power dissipation in the SLIM and Alpine prototypes. For the Alpine prototypes, silicon heaters were produced with a thickness of $700\text{ }\mu\text{m}$, a size of $20.6\text{ mm} \times 35.6\text{ mm}$ and an electrical resistance of $23\text{ }\Omega$. SLIM prototypes are equipped with $300\text{ }\mu\text{m}$ thick silicon heaters ($19\text{ mm} \times 41\text{ mm}$) with a resistance around $6\text{ }\Omega$.

The heaters are powered in parallel with a PSP-2010 power supply. The selected heaters were powered through a NI 9477 relay module.

To attach these heaters to the support structure, MPC 315 phase change elastomer (TIM) from AmecTermasol, with 0.13 mm thickness and a thermal conductivity of $5\text{ W}/(\text{m K})$ was used. In contrast to paste/grease, its characteristics provide thermal conductivity with low thermal

resistance, control of its thickness and possibility to detach afterwards. Above its transition temperature (around 45 °C) the material softens and fills the surface gaps where it is placed. A warming/cooling cycle is used to both remove and fix the heater.

6.9 Thermal probe attachment

At the very beginning, the temperature probes were bonded with a fast acting cyanoacrylate glue to the silicium heater and epoxy 2011 on top to fix the sensor in place. After conducting thermal tests on the first alpine prototypes, this approach proved afterwards to be the reason why the measurements were not ok.

This bonding method was compared to another technique which used Araldite 2020 instead of cyanoacrylate. This epoxy had lower viscosity and was more suitable for glass and ceramic bonding (gluing pt100's ceramic plate to silicon's surface).

Cyanocrylate vs Araldite 2020

A testing campaign was conducted to quantify the uncertainty brought by the bonding of the thermal probes onto the heaters.

A dedicated experimental setup was built, composed of a water heat pump system capable of driving a water flow circuit at cold and warm temperatures (from $-30\text{ }^{\circ}\text{C}$ to $50\text{ }^{\circ}\text{C}$). This system was coupled with a 3D printed alpine prototype in stainless steel that was conceived to be used as a reference sample for Alpine thermal management studies.

A calibrated thermocouple was placed inside the tube to measure the water temperature and two pt100s were glued to the stainless steel body at the closest location, to measure the difference between the water and stainless steel temperature - see Figure 6.19. This test was performed at warm temperatures close to $50\text{ }^{\circ}\text{C}$ and cold temperatures around $-35\text{ }^{\circ}\text{C}$, inside the vacuum vessel.

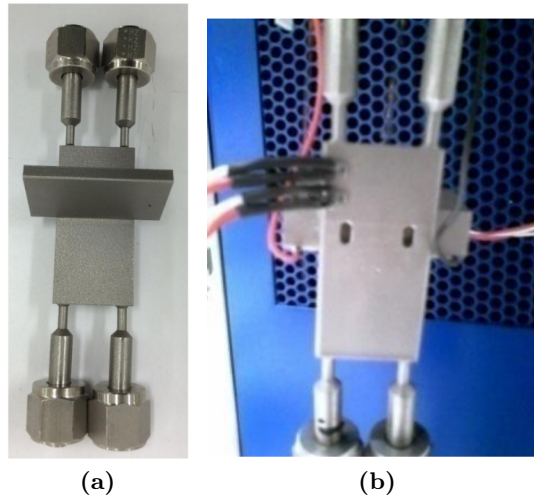


Figure 6.19: Front view of alpine prototype 3D printed in stainless steel and its back view with two pt100s glued next to the tube.

The results showed that the sensors glued with cyanoacrylate measured $3\text{ }^{\circ}\text{C}$ below the warm water temperature and $4\text{ }^{\circ}\text{C}$ above with cold water. When glued with Araldite 2020 the sensors lowered this offset to $0.6\text{ }^{\circ}\text{C}$ at cold temperature. The difference between the temperature measurement of both pt100s was below $0.1\text{ }^{\circ}\text{C}$ which was coherent with their uncertainty of $0.07\text{ }^{\circ}\text{C}$.

The former technique was only used on the Alpine M016 prototype with two tubes discussed in this report while all the others Alpine and SLIM prototypes used Araldite 2020.

Tapped block for bonding temperature probes

One of the disadvantages of gluing the sensor using epoxy was that, once they were glued, it was hard to reuse them without damaging them and the heater. After testing SLIM and Alpine prototypes using the epoxy, as there was still some questioning about how this technique could be biasing the results, a 3D plastic piece with a tapped hole was developed. This block was bonded to the prototype using Araldite 2020 and the pt100 was fixed in place with a screw - see Figure 6.20. Artic Silver thermal interface compound was used for the thermal contact between the sensor and the sample's surface.

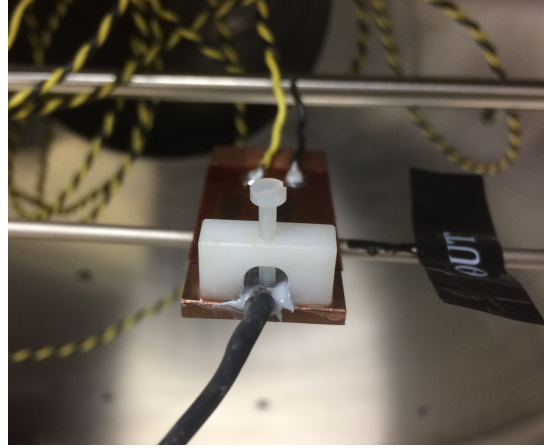


Figure 6.20: Copper block tested sample with bonded pt100 sensor using the screwable system. The 3D printed structure is responsible for housing the pt100 and fixing the sensor in place. Artic silver grease is used for the thermal contact between the pt100 and the sample.

The systematic error of this method was evaluated by repeating six times the process of placing five pt100s and compare their measurements with the same conditions. This was performed using the simplified prototypes of 20 mm and 8 mm wide copper blocks described in section 7.1. They were cooled with -20°C CO_2 at 1.6 g/s and each copper block was heated with 1 W. For a given block, the biggest measured standard deviation for six measurements was 0.2°C . Considering $\sigma_{sys} = \frac{\sigma_{max}}{\sqrt{N}}$, the systematic error due to repeatability was set at 0.1°C .

Systematic errors

The CO_2 temperature cooling the sample is measured by placing pt100 probes at the inlet and outlet vessel fittings (Figure 6.17). As the pressure drop is negligible (small prototypes, with small tube lengths and low power dissipation), the CO_2 temperature is taken from the average temperature between the inlet and outlet.

In Table 6.3 are listed systematic uncertainties considered in the measurements.

Measurement	Uncertainty
T_{CO_2}	$\pm 0.1\text{ }^\circ\text{C}$
$\Delta T_{\text{global}}(\text{Epoxy 2011})$	$\pm 4\text{ }^\circ\text{C}$
$\Delta T_{\text{global}}(\text{Epoxy 2020})$	$\pm 0.6\text{ }^\circ\text{C}$
$\Delta T_{\text{global}}(\text{Tapped block})$	$\pm 0.2\text{ }^\circ\text{C}$
Power	$\pm 0.1\text{ W}$
Mass flow	$\pm 1 \times 10^{-2}\text{ g/s}$

Table 6.3: *List of uncertainties on temperature measurements*

6.10 Thermal performance: Alpine and SLIM

6.10.1 Alpine

The first tested prototypes (Alpine M016 and M017) had two cooling tubes and the sensors of Alpine M016 were glued with cyanoacrylate while the Alpine M017 used Araldite 2020. Every prototype was equipped with two pt100s, one at each of the heater’s top corner.

Heat pick-up from dry air

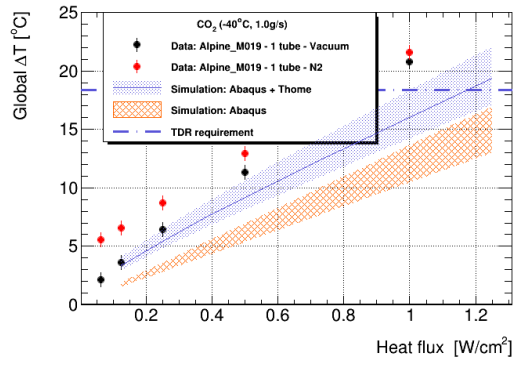
In the future detector, there will be a nitrogen flow at room temperature to keep humidity low, so the prototypes were tested in a nitrogen atmosphere to replicate the same conditions.

Figure 6.21 shows the effect of adding a nitrogen flow for Alpine M017 and Alpine M019 prototypes. The heat pick up observed from nitrogen convection, lead to a raise on the temperature of $[1\text{ }^\circ\text{C}-2\text{ }^\circ\text{C}]$.

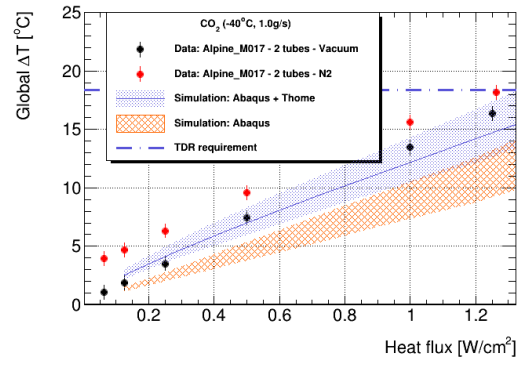
From the results one can compare the performance between both two (Figures 6.21d and 6.21b) and (Figures 6.21c and 6.21a) single pipe prototypes cooled at $-20\text{ }^\circ\text{C}$ and $-40\text{ }^\circ\text{C}$.

As the prototype is colder than the nitrogen flowing around it, there is some heat pickup from natural convection. This becomes more negligible as the temperature difference between sensor and nitrogen’s temperature is less significant, i.e at higher heat loads and higher cooling temperature. The effect in terms of offset seems to be comparable between the two types of prototypes.

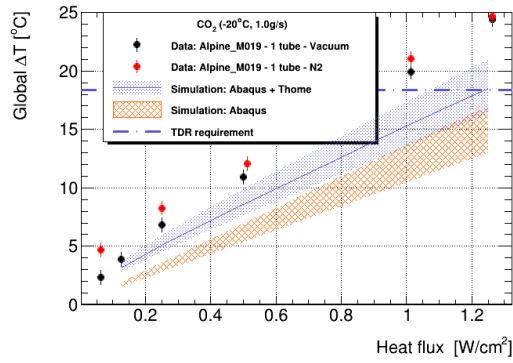
The measurements conducted under vacuum and with nitrogen, fulfilled the TFM design requirement of $32.7\text{ Kcm}^2/\text{W}$ set in Table 5.4 for the outer barrel layers at 0.7 W/cm^2 - see Figure 6.22. However, latest estimates point to a possible dissipation close to 1 W/cm^2 at the end of life of the detector. This would mean that the measured performance on the prototypes with one tube would not respect the maximum ΔT_{global} requirement.



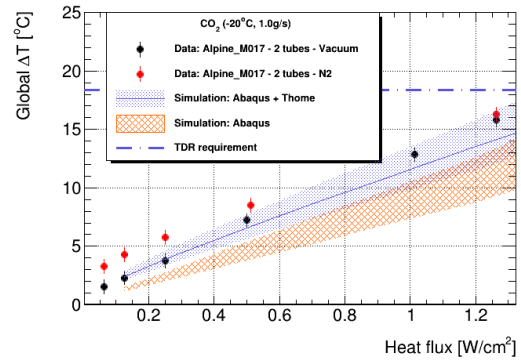
(a)



(b)



(c)



(d)

Figure 6.21: Comparison of thermal performance between Alpine prototypes with one (M019) and two tubes (M017) under vacuum and with a nitrogen flow.

Thermal performance - comparison between prototypes

The magnitude of the TFM hyperbolic shape observed in Figures 6.22c and 6.22d, is explained by CO₂ behaviour due to the radiative heat pickup, that can have a non-negligible effect at low heat loads, as discussed in section 6.6. A parasitic heat pickup of ≈ 0.2 W was considered on the predicted performance.

The Alpine M016 had the worst results due to the use of cyanocrylate to bond the temperature probes. The variability between the prototypes with one tube show the difficulty to control the thermal interfaces in carbon foam prototypes. This was a major drawback for the alpine local support design as the thermal performances margin was considered too tight to afford such variability between similar prototypes.

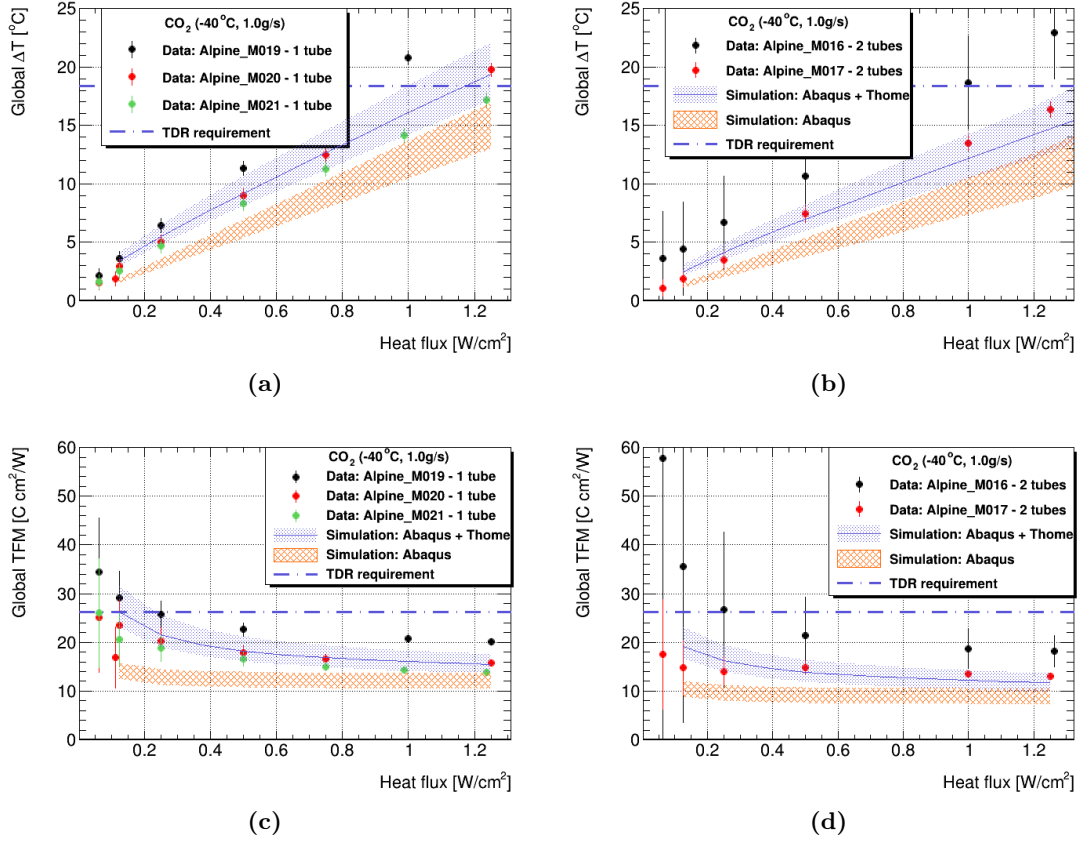


Figure 6.22: Thermal performance of Alpine prototypes with one and two tubes and comparison with the requirement set in the ITK TDR. The measurements were conducted with CO₂ temperature of -40°C and 1 g/s.

6.10.2 SLIM

The angled single cell prototype (see Figure 6.8b at page 97) had its thermal probe glued with cyanoacrylate like the Alpine M016 (thus the big error bars) and tested only with a nitrogen atmosphere. The impact of having nitrogen flow was tested before with the Alpine prototypes. The offset measured between the single cell and the triplet is about 5°C which can not be explained only from convective heat pickup, given that for the alpine prototype it was measured an offset around 1°C to 2°C . The bad performance of the single cell was clearly due the use of cyanoacrylate to bond the pt100.

The second SLIM prototype, composed of three cells (Triplet) was tested with vacuum and cooled at -35°C and 1 g/s CO₂ with pt100s glued with Araldite 2020 epoxy. For technical

reasons only one of the three cells was tested.

The measurements (see Figure 6.23) show that the prototype respected the required ΔT_{global} in the TDR [8] up to 0.75 W/cm^2 . However, one should keep in mind the introduction of nitrogen and a power dissipation of 1 W/cm^2 mean that it would not respect the thermal requirements.

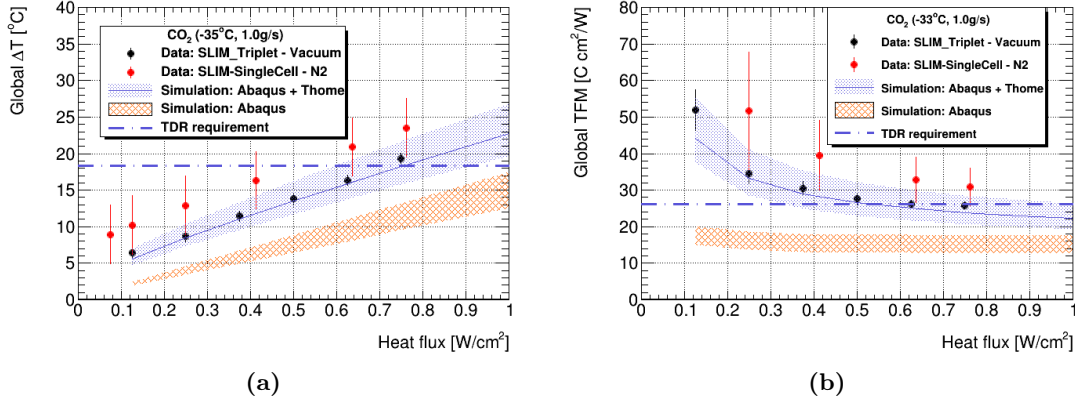


Figure 6.23: Thermal performance of SLIM prototypes measured at LAPP and its comparison with the requirement set in the ITK's TDR.

In the meanwhile, lots of efforts were dedicated to improve the thermal performance of the local support final design with better choice of materials and interfaces.

6.10.3 Discussion on the measurements and next steps

Despite a well controlled test setup, a large variability is observed between the various samples. This shows how important is to control precisely the probe attachment and to investigate the production variability of future prototypes.

The simulations were able to predict the hyperbolic shape of the global TFM, caused by a pure CO₂ convective behaviour due to the radiative heat pickup (assumed to be 0.2 W in the prediction curves).

Globally, the thermal prototypes respected the TDR thermal requirements. However, one can question the safety margin given the large uncertainties we currently have on the estimated performances.

Looking at our measurements, the convective part seems indeed to take a considerable slice on the global thermal performance (close to the 6°C predicted by the CERN cooling group using CoBra). However, as long as the test setup is not validated and the Thome model being used is not proven to work at heat fluxes above $\approx 60 \text{ kW/m}^2$, this result could still be questioned.

Therefore, the decision was to design a prototype to minimise the uncertainty on the estimated ΔT_{cond} and to reproduce the heat fluxes expected for the future local supports of the Pixel Detector. The goal with this new prototype was not anymore to measure the thermal performance of the local support, but to check the validity of the convective model predictions in the pixel outer barrel working region.

NEW PARAMETERIZATION OF THE CO₂ CONVECTIVE MODEL

The local support and any other element in the detector is typically over-designed to respect engineering safety margins. These are based on the operational limits, performance requirements and degree of uncertainty on certain parameters. The pixel sensors will suffer high radiation damage and the minimum temperature needed to operate the sensor will be already a major constraint in the design of the local support (geometry, materials used and manufacturing variability). On top of this, the uncertainty on the thermal performance of a local support is not negligible and the uncertainty on the coolant performance becomes bigger at higher heat loads.

Over-designing the local support in terms of thermal performance, often means the use of denser materials (and good conductors) which will compromise the particle tracking precision. It is therefore of extreme importance to predict well the maximum temperature allowable at the sensor and the thermal performance of the local support so the safety margins do not compromise the tracking performance.

To assess the precision of the modelling methods used to evaluate the thermal performance of the CO₂, simplified local supports were built to minimise the uncertainties due to multiple thermal interfaces and to ease the extraction of the convective and conductive components separately. The main goal of these testing campaigns is to check if the current methods used to calculate the heat transfer coefficient are in agreement with measurements.

Chapter outlook and personal contributions:

This chapter presents the FEA analysis conducted to model the conductive performance of the simplified samples. A novel method based on a nuisance parameters fit is used to disentangle both conductive and convective contributions to the global thermal performance and to verify if the convective model, with a $\pm 30\%$ of uncertainty on the estimated heat transfer coefficient, is valid given the experimental data. The fit results are also put into the context of the Pixel Detector working region and used to evaluate manufacturing variability of the samples.

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7.1 Sample and Test setup

To minimise the impact of the heat conduction on the global thermal performance, simplistic prototypes with few material interfaces were built (end of 2017). They were cooled with two-phase CO₂ with different mass flows, vapour qualities and temperatures. The conductive part of the prototypes is not expected to depend on convective parameters like vapour quality and mass flow so the convective and conductive part are expected to behave differently (uncorrelated). By exploring this, the data gathered from several testing campaigns, plus some a priori knowledge about the modelling uncertainty ranges, the conductive and convective contributions can be disentangled.

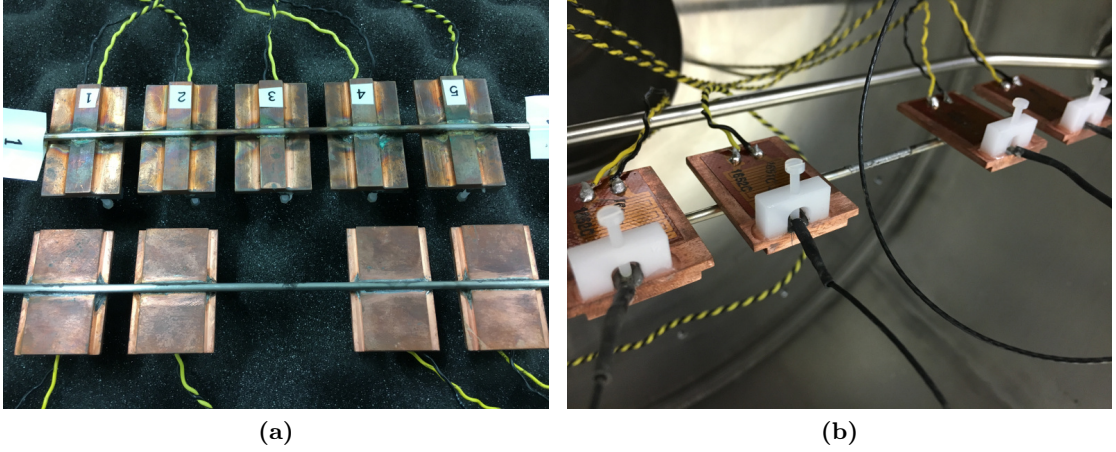


Figure 7.1: (a) Copperblock prototypes with 20 mm and 8 mm of braze length (96.5Sn3.5Ag alloy). (b) Prototype with 20mm copper blocks mounted inside the vacuum vessel.

Two simplified prototypes were built from a 200 μm thick stainless steel tube with 2.35 mm of inner diameter, brazed onto five identical copper blocks. One of the prototypes uses copper blocks with a braze length of 8 mm to mimic the contact area of inclined cells of SLIM local supports presented in section 1.3.3, while the second prototype uses 20 mm wide blocks to mimic the flat cells. The five blocks were tested individually to evaluate the manufacturing reproducibility.

The two samples are placed in a vacuum chamber in order to avoid natural convection (same test setup as described in Chapter 6).

Materials and FEA uncertainties

The ABAQUS model used to predict the heat transfer through conduction shown in Figure 7.2, was subject to a parametric study where the various parameters listed in Table 7.1 were changed to quantify their impact on the measured ΔT_{cond} .

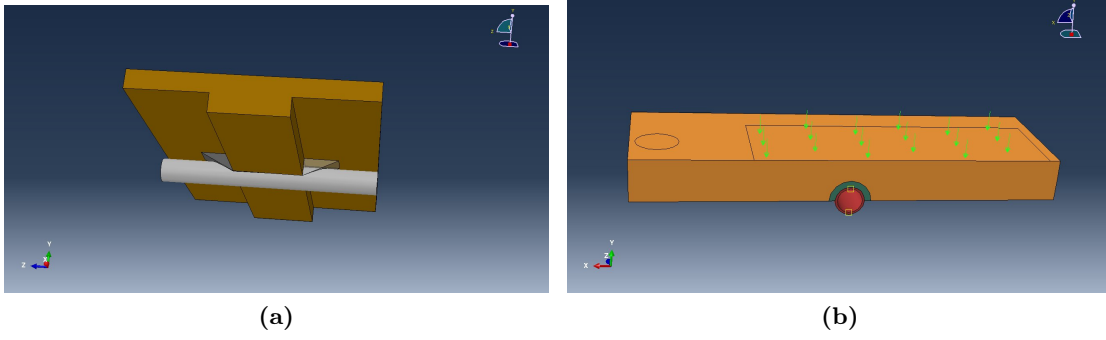


Figure 7.2: FEA model of prototypes with copper blocks brazed onto stainless steel tubes.

Parameter	Central value	Uncertainty	Sensitivity	ΔT @6W
Copper thermal cond.	400 W/(m K)	± 10 W/(m K)	—	—
Stainless steel thermal cond.	15 W/(m K)	± 1 W/(m K)	± 0.02 °C/W	± 0.1 °C
Braze thermal cond.	55 W/(m K)	± 5 W/(m K)	± 0.015 °C/W	± 0.1 °C
Braze thickness	0.5mm	0.1mm	± 0.01 °C/W	± 0.06 °C
Braze extra length	0mm	2mm	± 0.1 °C/W	± 0.6 °C
Heater distance from border	2mm	1mm	± 0.02 °C/W	± 0.1 °C
Stainless steel temp. coefficient	$0.027 K^{-1}$	$0.003 K^{-1}$	—	—
Braze temp. coefficient	$-0.15 K^{-1}$	$0.1 K^{-1}$	—	—

Table 7.1: Input parameters to the ABAQUS simulation. The sensitivities on the expected ΔT_{cond} , were estimated from ABAQUS, by varying the input parameters within their uncertainties per Watt dissipated by the heater.

Special attention was given to the impact of extra braze along the tube as it was not always well controlled due to the brazing process - see Figure 7.3 - and this has a non negligible impact on the thermal performance of the copper blocks, especially the most narrow ones (8 mm).

As the positioning of the tapped blocks lacked some precision, the possible uncertainty due to their true location was also investigated.

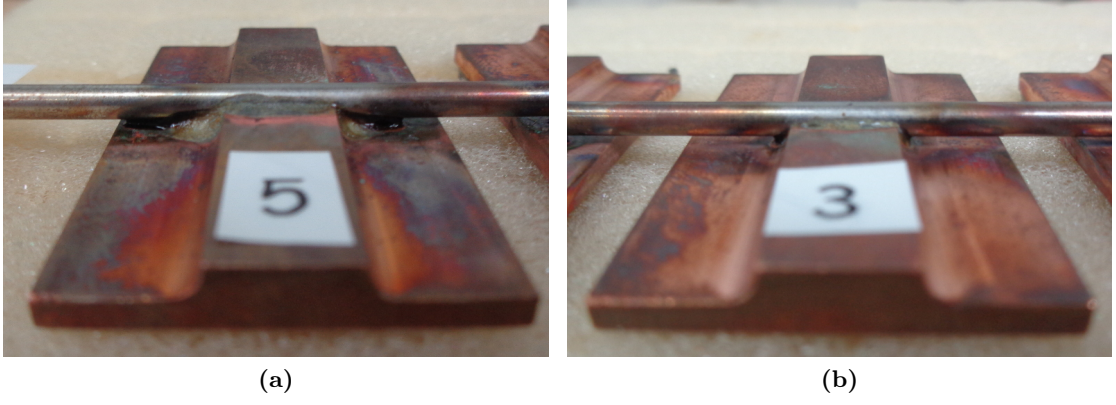


Figure 7.3: Manufacturing variability in the brazing process observed in the blocks 8 mm wide.

7.2 Modelling heat flux on the cooling pipe

In a similar approach to that followed for the SLIM prototypes (discussed in section 6.2), a study was also conducted to estimate the impact on the heat flux at the top half of the tube. In this case, the heat flux was reduced to $75 \pm 5\%$ - see Figure 7.4. The maximum heat flux for the prototype with 8 mm blocks is estimated to 26 kW/m^2 per W applied. For the 20 mm prototypes it is estimated to 10.4 kW/m^2 per W applied.

As shown in Figure 7.4a, the conductive contribution is not significantly affected by the different HTC considerations and is expected at $0.58 \text{ }^\circ\text{C/W}$, for the central values considered in Table 7.1.

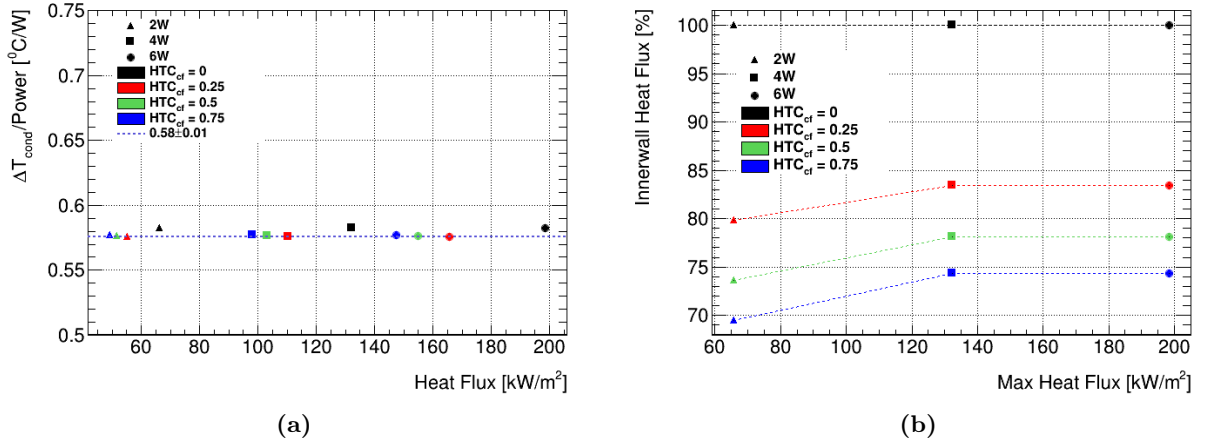


Figure 7.4: (a) Impact of CO_2 heat transfer coefficient on the conductive contribution. (b) shows its impact on the top half of the tube flux, for the prototype with 8 mm copper blocks.

7.3 Datasets and comparisons with baseline simulations

Both prototypes were tested under different CO_2 conditions: mass flows with 2 and 3 g/s, vapour quality from 10 % to 33 % and saturation temperature on the CO_2 blow system between -40°C and -20°C . Each copper block was then heated individually from 1 W, 2 W, 3 W, 4 W, 6 W and 8 W.

20 mm copper blocks

Figure 7.5 shows that the measurements on the 20 mm prototype, closer to the IBL working region, are in agreement with the simulations. However, at higher temperatures and higher heat fluxes, they get closer to the limits of the error.

The measurements at $-20\text{ }^{\circ}\text{C}$ (Figures 7.5a and 7.5b) show better performance than at $-40\text{ }^{\circ}\text{C}$ (Figures 7.5c and 7.5d), which was predicted from the Thome model. Differences between 2 g/s and 3 g/s flows are only observed at $-40\text{ }^{\circ}\text{C}$ and 33% vapour quality.

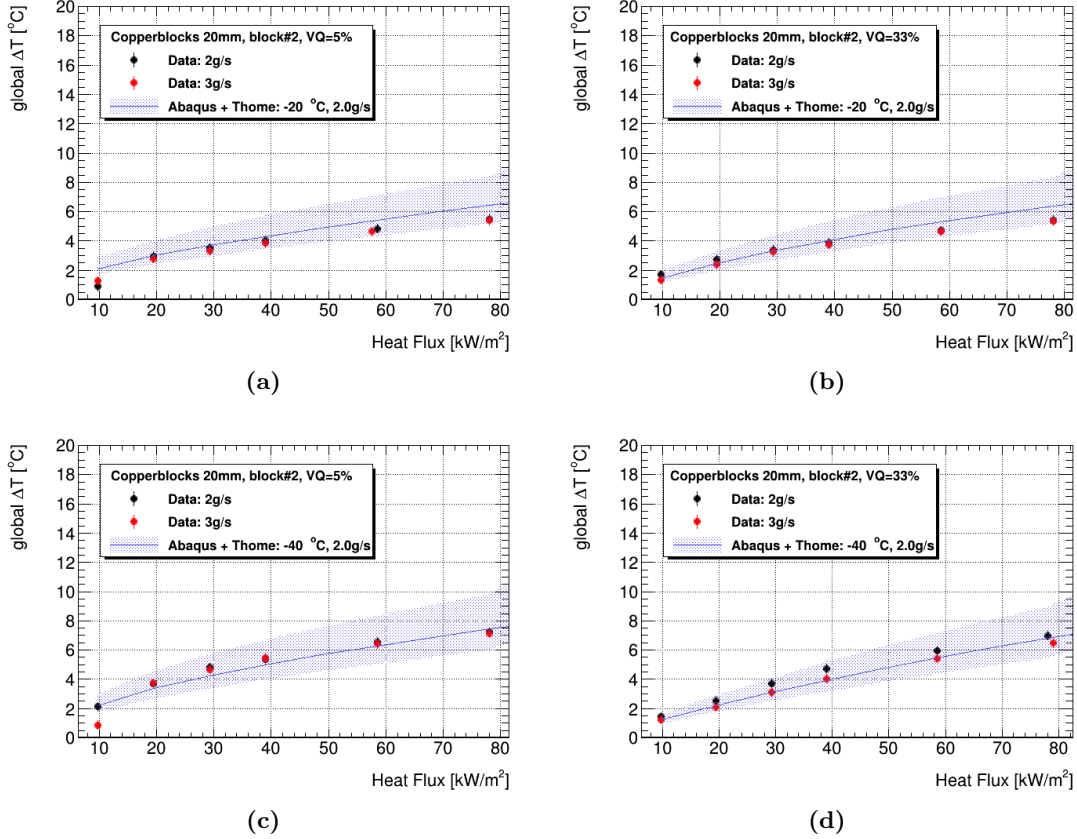
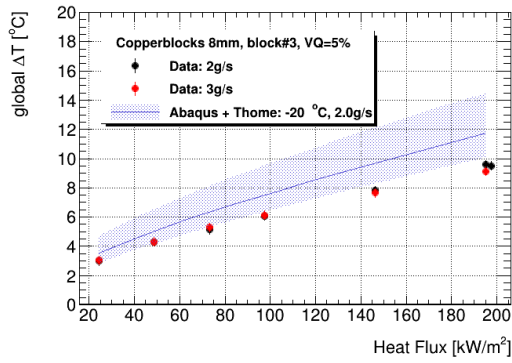


Figure 7.5: Comparison of results for the 20 mm copper blocks sample at 2 g/s and 3 g/s considering different CO_2 temperatures (*top* and *bottom*) and vapour qualities (*left* and *right*).

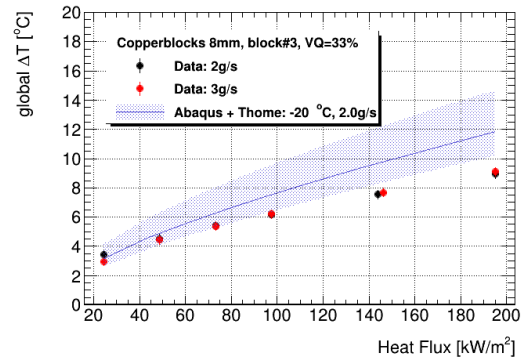
8 mm copper blocks

For the 8 mm prototype, with heat fluxes above $>100\text{ kW/m}^2$, significant discrepancies are observed- see Figure 7.6. The difficulty here is to understand if the discrepancy arises from the local surface description or from the CO_2 convective behaviour.

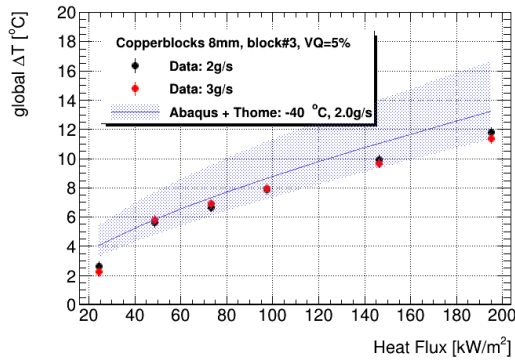
The big uncertainty of the model reaches $\pm 3\text{ }^{\circ}\text{C}$, as the expected ΔT_{conv} goes up to $8\text{ }^{\circ}\text{C}$ at higher heat loads. Still, the measurements do not lay within the confidence region, showing a clear tendency to underestimate the HTC at higher heat loads, especially at higher CO_2 temperatures.



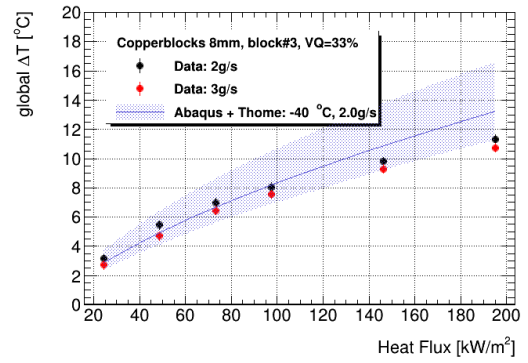
(a)



(b)



(c)



(d)

Figure 7.6: Comparison of results for the 8mm copper blocks sample at 2 g/s and 3 g/s considering different CO_2 temperatures (*top* and *bottom*) and vapour qualities (*left* and *right*).

7.4 Nuisance simulation model

The disagreement between the measurements and the simulation can come either from bad modelling in the FEA or from the convective model with a predicted heat transfer coefficient out of its $\pm 30\%$ confidence region. To properly evaluate where the mismatch is coming from, the conductive and convective components need to be somehow decoupled from the experimental data.

The non-trivial solution is to use the large dataset (108 data points per block) and benefit from the small experimental uncertainties to constrain the modelling parameters. In each dataset, the copper blocks are identical, and only the CO₂ conditions vary. Hence, it is possible to over-constrain the parameters related to the local support to extract precisely their properties (thermal conductivity, temperature dependency, manufacturing variability) and calculate the compatibility with the predicted heat transfer coefficient.

The original approach is based on a nuisance parameter fit, widely used in Particle Physics [73], to decouple the thermal measurements into the conductive and convective components. In the following sections is described how the system is modelled and how the fit is performed.

7.4.1 Model definition

There are Q prototypes with M blocks each. Considering i ($1, \dots, Q$) and j ($1, \dots, M$) as subscripts, it is assumed a priori that each measured temperature of the sensor, y_{ij} is modeled in the following way:

$$y_{ij} = f_i(x_{ij} \mid \theta) \quad (7.1)$$

Where x_{ij} stands for the following experimental measurements:

- Fluid mass flow
- Heater heat load
- Fluid temperature at the inlet and outlet of the experiment
- Fluid temperature before expansion in the blow system

and the parameter θ stands for the model parameters listed in Table 7.2.

Parameter	Description
α_{HTC}	Deviation from predicted HTC
$\epsilon_{radiative}$	Radiative heat pick-up
heater_dist	Probe distance from the block's border [mm]
brazing_t	Brazing thickness
α_{inox_k}	Temperature coefficient stainless steel
$\alpha_{brazing_k}$	Temperature coefficient stainless steel
brazing_l1	Extra length brazing copper block #1
block_l	Copper block width
k_{inox}	Thermal conductivity of stainless steel
$k_{brazing}$	Thermal conductivity of brazing
k_{copper}	Thermal conductivity of copper

Table 7.2: List of model nuisance parameters.

The model $f(x)$ has been defined as the sum of two independent contributions: conductive, f_{cd} and convective, f_{cv} .

$$f(x | \theta) = f_{cv}(x | \theta) + f_{cd}(x | \theta) \quad (7.2)$$

The systematic uncertainty on each experimental measurement, x , and model parameter, θ , is described by a value retrieved from a normal distribution represented by $\nu\delta$. The nuisance parameter ν is retrieved from a normal probability distribution, with $\mu=0$ (mean) and $\sigma^2=1$ (variance) multiplied by the uncertainty δ . Thus, the complete model formulation is defined as:

$$\begin{aligned} y_{ij} &= f(x_{ij} | \theta, \delta, \nu) \\ &= f_{cv}(x_{ij} | \theta, \delta, \nu) + f_{cd}(x_{ij} | \theta, \delta, \nu) \end{aligned} \quad (7.3)$$

7.4.2 Conductive model

The conductive model is retrieved from a numerical heat transfer finite element analysis in ABAQUS. A parametric study considered the following parameters:

- Thermal conductivity for copper, stainless steel and braze alloy
- Contact length between copper block and tube
- Heater distance from the border
- Braze thickness
- Extra length for the braze
- Copper block width

For a given set of measurements, the sensor temperature is expected to vary linearly with the heat flux crossing the contact area. The model considers constant thermal conductivity for the materials along the thermal path. Considering a linear relationship between the temperature of the sensor and the heatflux, Φ_s :

$$f_{cd}(x_{ij} | \theta) = m(x_{ij} | \theta) \times \Phi_s \quad (7.4)$$

The result depends on the model slope m that may vary as a function of the L ($L=10$ + number of blocks) model parameters listed in Table 7.2.

To integrate the nuisance parameters into the model, the slope, $m(x)$, is calculated from a series expansion around the set of central model parameters, θ_0 and nominal experimental conditions, x_0 :

$$m(x_{ij} | \theta) = m(x_0 | \theta_0) \quad (7.5)$$

$$+ \sum_{l=0}^L \frac{\partial m}{\partial \theta} \theta_0 \delta_l \nu_l \quad (7.6)$$

$$(7.7)$$

Thermal conductivity and temperature coefficient

The uncertainty on the thermal conductivity can come from an offset and dependency on temperature. The thermal conductivity temperature coefficient for the copper is neglected ¹ but not for the braze and stainless steel. Both stainless steel and braze are considered to be at the same temperature, calculated from the estimated HTC and local heat flux. The parameter corresponding to the thermal conductivity, k , is therefore corrected to:

$$k + \nu_k \delta_k \Rightarrow k + \nu_k \delta_k + \alpha_k(T - T_0) + \alpha_k \nu_T \delta_T + \nu_\alpha \delta_\alpha(T - T_0) \quad (7.8)$$

where α_k is the thermal conductivity coefficient of the material and $\nu_k \delta_k$ the corresponding nuisance parameter multiplied by its uncertainty. Parameter T is the temperature of the material computed by the model (considers the heat transfer coefficient calculated by the convective model) affected by the uncertainty on the CO2 temperature, ν_T . T_0 is the temperature of reference (typically 25 °C) for the nominal thermal conductivity of the materials.

7.4.3 Convective model

The parameters of interest for the convective model, derived directly from the experimental conditions are the following:

- Inlet vapour quality
- Mass flow
- Fluid temperature
- Tube diameter
- Local heatflux

The fluid temperature and mass flow precision are given by the instruments specifications. The local heatflux depends on the power applied to the heater and contact area between block and tube. The uncertainty must consider possible error on the effective heat load applied and effective contact area that depends on the braze extrusion depth.

The inlet vapour quality is calculated from the temperature of liquid CO2 right before the expansion phase in the blow system. From this point to the inlet of the experiment the pressure drop and heat pick-up are neglected².

The convective component depends on the heat transfer coefficient estimate calculated from the functions presented in section 2.5:

$$f_{cv}(x_{ij} \mid \theta, \nu) = (1 + \nu_{HTC} \delta_{HTC}) \times HTC(x_{ij} \mid \theta, \nu) \quad (7.9)$$

¹Copper presents negligible temperature coefficient for temperatures above 200 K

²The pipe used is 50 cm long with a outer diameter 6 mm. If not insulated, the maximum heat pick-up from the environment (at 20 °C) would be around 5 W (≈ 600 W/m²) for a flow of CO₂ at -40 °C and 1 g/s. In this pessimistic case scenario, one would be committing 1 % of error on the estimated vapour quality.

7.4.4 Fitting parameters

The reliability of the fit for the model given N measurements is tested by minimising the quantity χ^2 :

$$\chi^2 = \sum_{i=0}^Q \sum_{j=0}^M \frac{(y_{ij} - f_i(x_{ij} | \theta))^2}{\sigma_{y_{ij}}^2} + \sum_l \nu_l^2 \quad (7.10)$$

Ideally, the expected result for y_{ij} is within the random fluctuations represented by the uncertainty $\sigma_{y_{ij}}$ so each term on the χ^2 sum will be of order unity. The second term in χ^2 represents the uncertainties on the model parameters. They are assumed a priori, well described by a gaussian distribution with mean value of zero and $\sigma^2 = 1$. Thus, the closer the value of χ^2 is to zero, the better the model describes the data.

Given the fitted set of parameters ν , one can sample over the dataset distribution and estimate the posterior probability function (equation 7.11) marginalizing over the nuisance parameters. The uncertainty on ν is assumed gaussian, and the result from the marginalization will give us an extra information about the uncertainty on the true model parameters.

The sampling over the nuisance parameters distribution, used to estimate the posterior probability function over the dataset is performed with the *emcee* python Monte Carlo (MCMC) Ensemble sampler [74].

$$\mathcal{L}(x_{ij} | \theta, \nu) = \mathcal{L}_{prior} - \frac{\chi^2(x_{ij} | \theta, \nu)}{2} \quad (7.11)$$

$$\mathcal{L}_{prior} = \begin{cases} 0 & \text{if } \nu \text{ realistic e.g. only braze extra lengths } \geq 0 \text{ mm} \\ -\infty & \text{if otherwise} \end{cases} \quad (7.12)$$

7.5 Analytical model vs Abaqus simulation

To introduce the conductive contribution in the fit, the Abaqus simulations were parametrised as explained in section 7.4.2. The most complex feature of the FEA model was the extra braze length affecting the local heat flux. Its impact on the thermal performance was not negligible as shown in Table 7.1. Nonetheless, the effect was successfully included in the model leading to a good agreement between with results from Abaqus as shown in Figure 7.7.

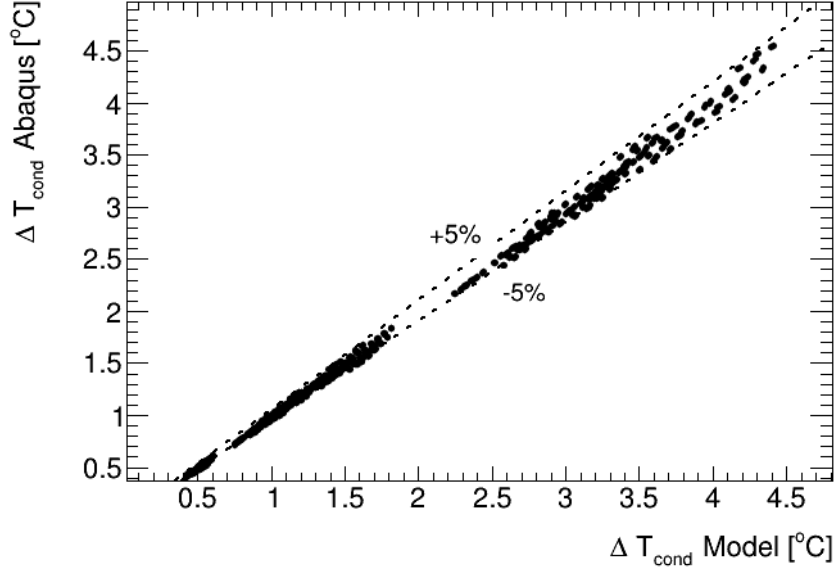


Figure 7.7: Comparison between the results from the FEA simulation and the ones predicted by the proposed conductive model for the prototype with the stainless steel tube.

7.6 Fitting the data: results

The results from fitting the measurements on the stainless steel prototype are illustrated by the pulls on the central values in Figure 7.8, where the x-axis translates the corresponding deviation.

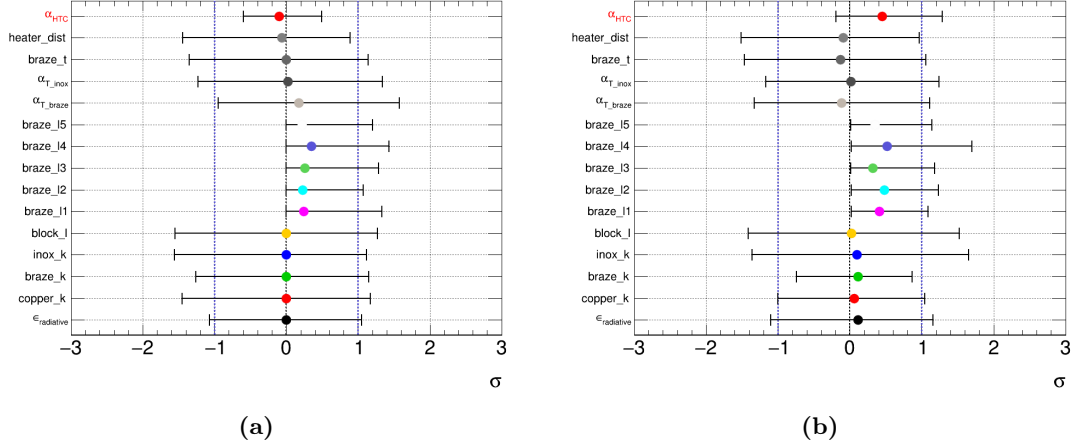


Figure 7.8: Results for the nuisance parameters after fitting the model with the measurements for the 20 mm prototype (a) and 8 mm prototype (b). The χ^2 retrieved were 1.81 and 4.3 respectively.

The results for the prototype with 20 mm block contact length (Figure 7.8a) show no significant pull from the central values i.e. around the central value considered a prior. This is in agreement with the observation as the measurements were shown to be within the confidence region.

However, the same is not true for the 8 mm prototype, where the non agreement between the measurements and the model is translated into the pulls deviation.

The main difference between the two prototypes is the range of heat fluxes considered. With much higher heat fluxes, the central values of the thermal conductivity and correction of the HTC are pulled to compensate the underestimation of the heat transfer coefficient. This happens

because even if the dataset and the prior knowledge about the parameters do constrain the results of the fit, there is some correlation between the parameter considering in the convective model, α_{HTC} , and the thermal conductivity of the materials, as shown in Figure 7.9.

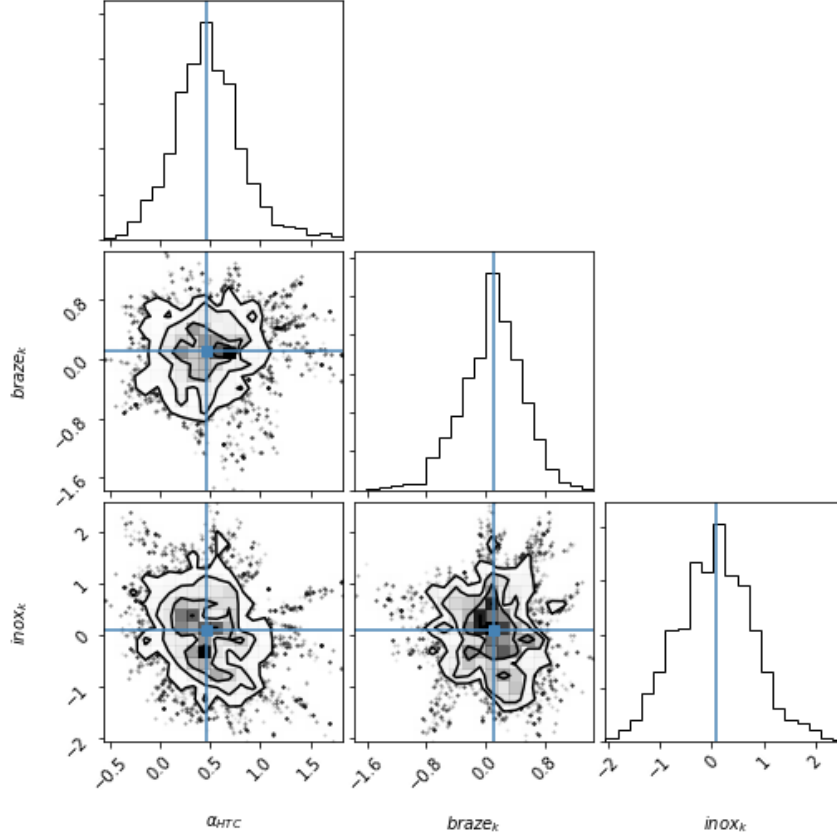


Figure 7.9: Correlation between different model parameters retrieved from the fit.

7.6.1 Deviation from HTC estimates

The fit results for the 8mm prototype shown in Figure 7.8 give a pull on the corrected HTC close to one, meaning a deviation of +30% from the central value.

To improve the model precision, some model floating parameters were defined and fitted to the data (similarly to what was performed by the model authors given their experimental database). As the deviations depended mostly on the CO₂ temperature and heat flux, only the parameters shown in equation 7.13 (calculation of nucleate boiling heat transfer used in the convective model presented in section 2.5) were tuned. To improve the fit constraints, the measurements for each of the blocks with negligible extra braze length (third block for the 8 mm prototype and the fourth of the other) of each prototype were fitted together.

$$h_{nb} = 131 p_r^{N_1} (-\log_{10} p_r)^{N_2} M^{-0.5} q^{N_3} \quad (7.13)$$

The fit results corrected the N_2 from -0.55 to -0.91 and N_3 from 0.58 to 0.59. With the new parameters, the model described well the experimental data with an HTC uncertainty of $\approx \pm 15\%$ for both prototypes - see Figure 7.10.

Better precision is important to reduce the safety margins due to lack of precision in the modelling estimates, especially at higher heat loads where big uncertainties become more problematic.

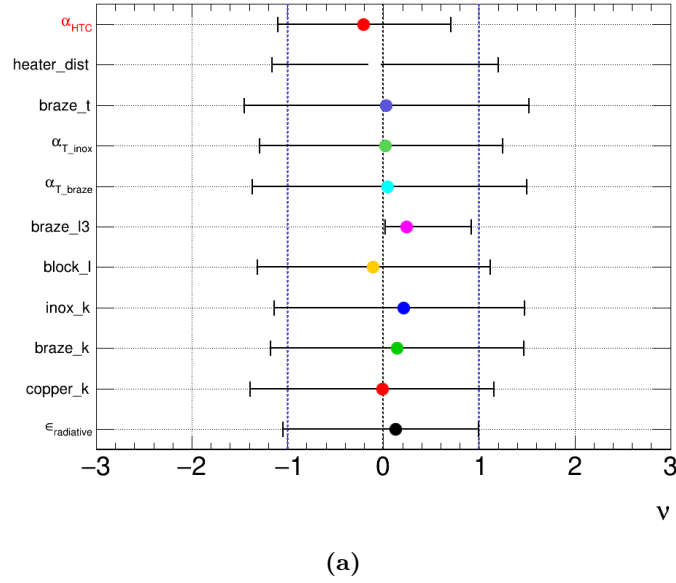


Figure 7.10: Results for the nuisance parameters after fitting the model with corrected parameter N3, considering an uncertainty of $\pm 15\%$ on α_{HTC} ($\chi^2 = 1.6$).

Figure 7.11 shows the deviation of the model after fit from the original Thome model. One can verify the IBL working region ($< 40 \text{ kW/m}^2$) is indeed within the $\pm 30\%$ uncertainty, while the working region of interest for the ATLAS ITk is way out of the confidence region with deviations up to $50 \pm 10\%$.

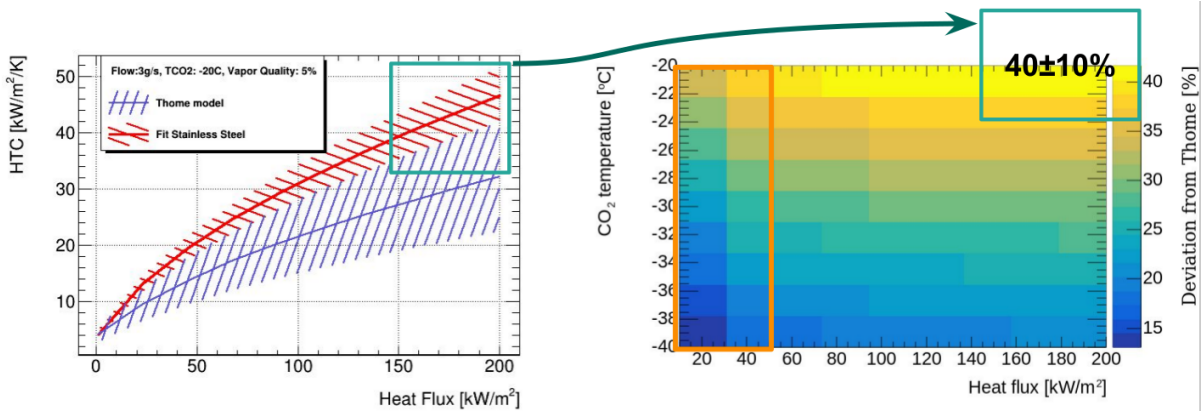


Figure 7.11: Global deviation of the heat transfer coefficient calculated (result from fitting model to data) from the model estimates.

The comparison between the semi-empirical model and the proposed re-parameterization is integrated in the first tab of the web app described in section 4.7.

7.6.2 Radiative heat pick-up

The heat pick up is considered as a measurement and added to the heat load. It is calculated using equation 6.2, with an emissivity, ϵ , of 0.5, the vacuum vessel temperature, T_e , of 20°C and multiplied by a constant. The considered uncertainty of 20% on this parameter is given by its nuisance parameter, ν_ϵ .

As discussed in section 6.6, considering the radiative heat-pick up is particularly important at low heat fluxes where the heat pick-up becomes a non-negligible part of the total heat conducted.

Figure 7.12 shows the radiative heat-pick up fitted values.

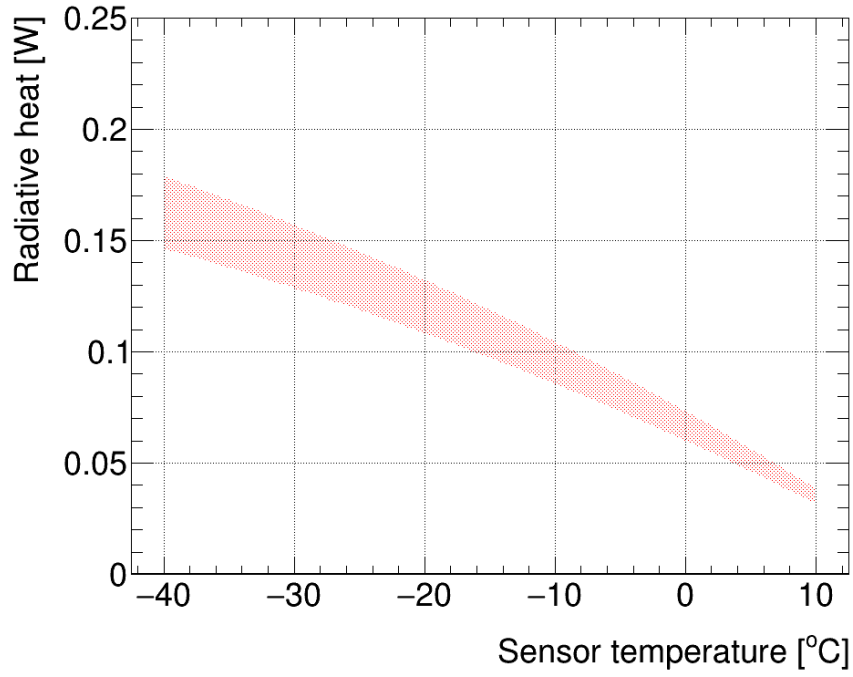


Figure 7.12: Fitted radiative heat pick up as a function of the temperature measured by the thermal probe.

7.6.3 Manufacturing variability

Figure 7.13 shows how the measurements vary between each block³. The variability on the prototype with 20 mm blocks looks within the measurements uncertainty. However, in the case of the 8 mm blocks (Figure 7.13b), the temperature measurements diverge with the increase of the power applied, as a sign of different thermal conductive performance.

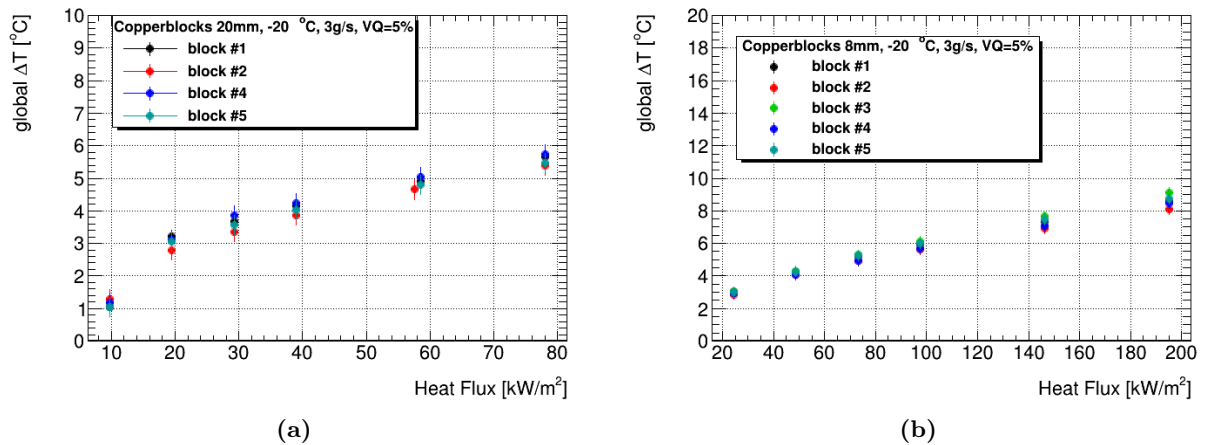


Figure 7.13: Comparison between the temperature measured on every block for each sample under the same conditions (power applied and cooling).

³The third block of the prototype with 20 mm was not well brazed so the copper block was ripped off from the tube during manipulation.

The only parameter in the conductive model particular to each block is the extra braze length. After visual inspection, the 8 mm sample's third copper block and the fourth copper block of the 20 mm sample, were considered to have a negligible extra braze length. This was set as a constraint on the block's braze length nuisance parameter. The fit result for the other brazes will therefore be a relative correction to the one set to be negligible.

The variability between copper blocks will be expressed by their extra braze lengths, assumed a priori to be negligible for every block, as it is the only parameter particular to each copper block in the model.

The results for the prototype of 8 mm blocks, present deviations from the central values for the brazes extra lengths, showing clearly that the variability measured between copper blocks is probably due to the lack of control in the brazing process.

In the case of the 20 mm, the uncertainties assumed a priori are sufficient to describe the data (± 2 mm).

Figure 7.14 shows the consequent prediction for the braze lengths for the copper blocks of prototype 8 mm.

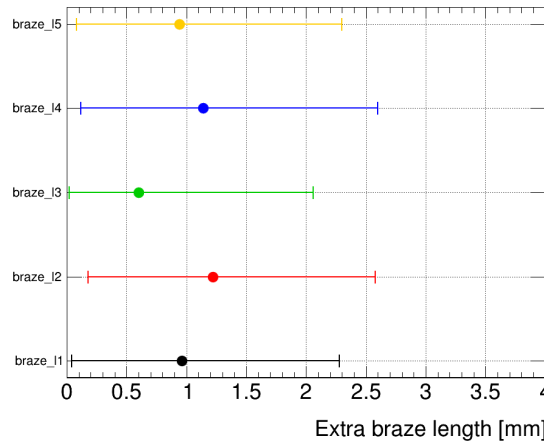


Figure 7.14: Pull results from fit, expressing the variability between extra braze lengths.

7.7 Conclusions: lessons learned and improvement of simulation

In this chapter it is concluded that the model used to calculate the heat transfer coefficient is not valid in some working regions of interest. Uncertainties on the radiative heat-pickup, manufacturing variability of prototypes and materials thermal conductivity were quantified. The right assessment of the modelling uncertainties as crucial to conclude that the model was not in agreement with the measurements at high temperatures (up to -20°C) and high heat fluxes ($>100\text{ kW/m}^2$). This observation is particularly important given that this will correspond to the operating conditions of the Outer Barrel. Also, it is the first time that results from simulations are questioned and evaluated with measurements, conducted with a simplified prototype with special care on the modelling and systematic uncertainties.

Measuring the disagreement between measurements and simulation was determinant to understand whether the deviation is coming from a bad construction of the conductive model or the model used to calculate the heat transfer coefficient.

Inspired by the way physicists analyze big datasets to constrain models, extract modelling parameters and test the compatibility of the measurements with theory, a nuisance parameters fit was implemented. The same principle was used to disentangle the conductive and convective contributions and conclude which of the contributions could explain the data.

Given that the convective model itself is composed by a set of correlations with exponents that come from a fit over a given experimental dataset, the same was performed here to find the exponents that were best suitable to our working region. This re-parameterisation of the model improved the precision on the heat transfer coefficient by a factor of two.

The manufacturing variability was also indirectly evaluated as the extra braze length was also included in as a fit model variable.

The measurements were important to observe how thermal performance of the fluid depends on the cooling conditions. It improves with the vapour quality, gets worse at lower temperatures and does not vary much between 2 g/s and 3 g/s. Even though this was already predicted by the convective model, it was important to raise awareness about how important it is to care about the CO₂ cooling temperature in the thermal qualification campaigns.

The method and the test setup were proven to be robust and the conclusions important to better understand the CO₂ cooling behaviour and limitations of simulations. It was thus decided to repeat the same exercise, on a more precise and realistic prototype with titanium tubes and better controlled manufacturing process. This will be discussed in the next chapter.

CHARACTERISATION OF A REALISTIC LOCAL SUPPORT

A third prototype was also equipped with five copper blocks and tested similarly to the prototypes presented in the previous chapter. It was built (end of 2018) with the elements (base blocks and titanium tube) and methodology (brazing of base block onto titanium tube) intended to be used in the final local support design.

The main goal with this new testing campaign was to perform a thermo-fluidic and manufacturing qualification on a more realistic prototype.

However, odd behaviour of CO₂ had already been observed in titanium pipes, it was also the opportunity to compare with the measurements from the stainless steel prototype and check if the cooling performance was impacted only by changing the tube's material.

The low roughness of titanium pipes was already linked to the wall super heating observed in the IBL [9] responsible for delaying the onset of boiling. As explained in section 2.3.3, the surface roughness does play an important role in the bubble nucleation, improving the heat exchange between the cooling fluid and the heated wall. Better heat transfer coefficient (mainly nucleate boiling) performance in rough surfaces has already been shown to be correlated to the roughness of the surface [75]. A study conducted on titanium surfaces reveals big differences on the measured heat transfer coefficient between untreated titanium surface and titanium oxide thin films ranging from 100 nm to 300 nm [76], so it should be no surprise that if the titanium has a much slicker surface than the stainless steel, one will observe a different cooling behaviour.

Chapter outlook and personal contributions:

The new prototype is described and its predicted thermal performance simulated. The measurements are compared with the stainless steel prototype, the model is tweaked to be in agreement with the measurements and it is discussed how the different cooling behaviour will affect the temperature profile of the longerons in the future Outer Barrel. Simulations predicting the real temperature profile at the beginning and end of operation of the detector are shown with discussion about the future testing qualification testing campaigns.

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8.1 Titanium local support

The prototype is composed of a 150 μm wall titanium tube with 2.5 mm inner diameter, brazed to five composite base blocks (composed of AL-FC fiber and 8 mm wide) - see Figure 8.1. Differently from the previous prototypes, the copper blocks were not brazed directly onto the tube. Instead, they were machined separately and screwed to the base blocks as shown in Figure 8.1b. The thermal interfaces between the base and copper block used Artic Alumina¹ with the same torque (10 mN.m) as planned for the final local support.

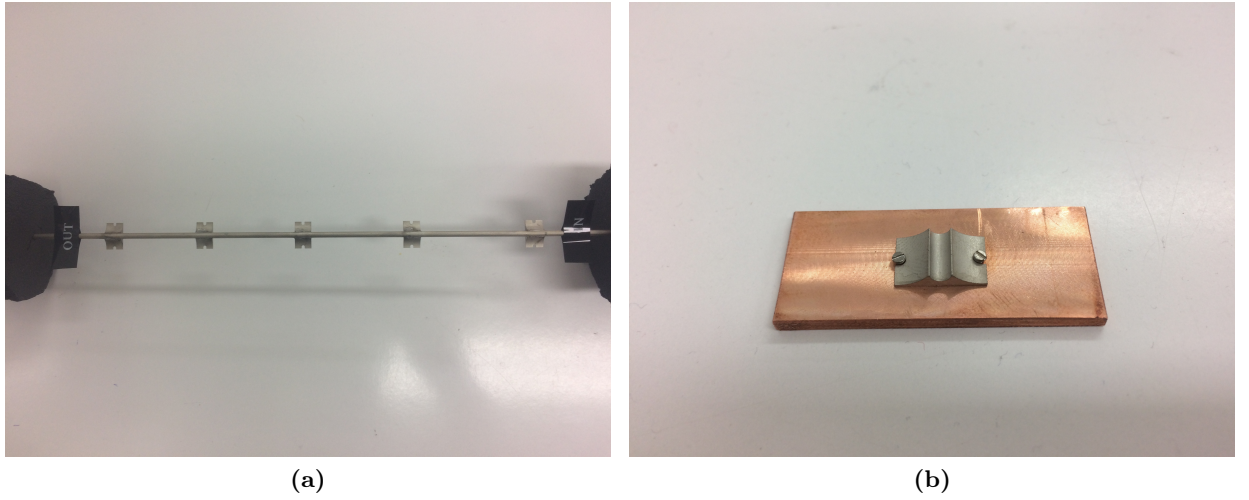


Figure 8.1: At (a) the titanium base blocks brazed onto titanium pipe and at (b) the cooling block brazed onto copper blocks before assembly.

8.2 Heat Transfer FEA and conductive model

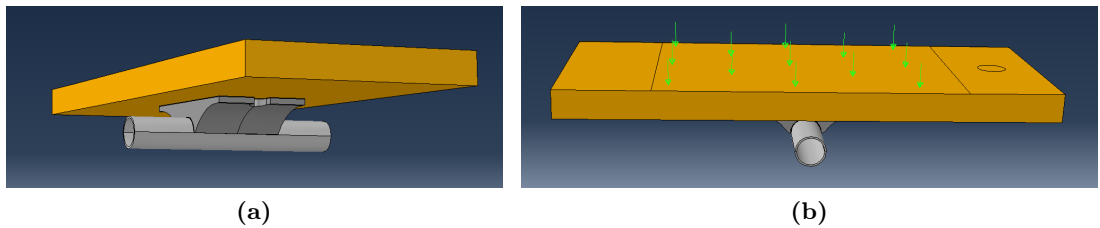


Figure 8.2: FEA model of ITK OB prototype with the copper block mounted onto the titanium tube.

Similarly to what was performed for the previous two prototypes with the stainless steel tube, parametric studies were conducted to infer the effective heat flux at the top part of the tube and to define an analytical model to describe the expected ΔT_{cond} .

¹Ceramic thermal adhesive compound from Artic silver widely used between modern high-power CPUs and high performance heatsinks or water-cooling solutions.

The results on the impact of the heat transfer coefficient on the maximum heat flux on the tube were already discussed on page 100 where it was shown that the mean heat flux at the top part of the tube should be around $80\% \pm 5\%$ of the maximum heat flux, with an expected heat flux of $28.2 \pm 2 \text{ kW/m}^2$ per Watt dissipated in the sensor.

This result is higher than the 75% on the stainless steel prototype because the titanium has better thermal conductivity and also because the stainless steel tube is thicker.

After visual inspection the braze was considered to be well controlled since its extra braze length was under $100 \mu\text{m}$. The analytical model was therefore, defined as a function of the heat load in spite of the heat flux. Figure 8.3 shows that the results from the FEA simulation were well described by equation 8.1 so its uncertainty was neglected.

$$\Delta T_{\text{cond}} = 0.66 \text{ }^\circ\text{C/W} \times \text{Heat load[W]} \quad (8.1)$$

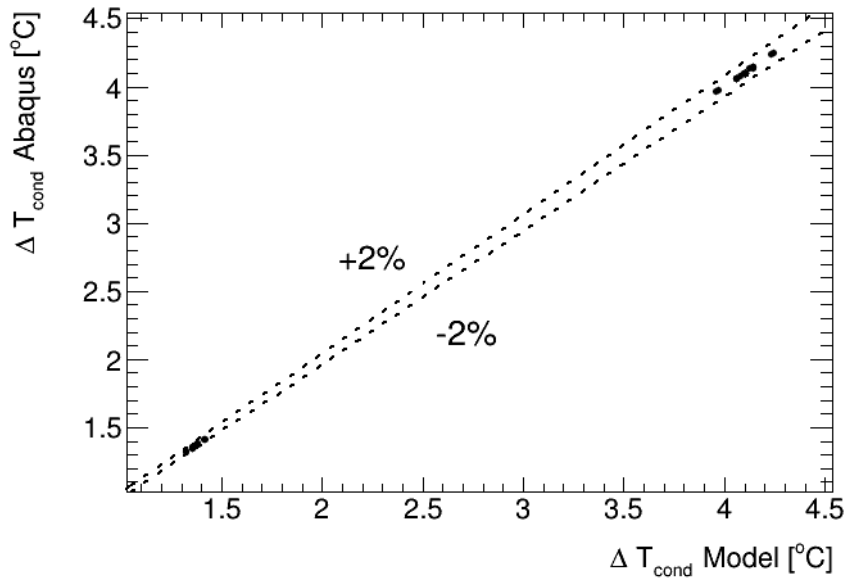


Figure 8.3: Comparison between the results from the FEA simulation and the results predicted by the proposed conductive model for the prototype with the titanium tube.

8.3 FEA uncertainties

The sensitivity to each parameter is retrieved from a parametric study, considering the listed nominal values and uncertainties in Table 8.1.

The most critical parameters for the first prototypes were the thermal conductivity of the stainless steel tube, the thermal conductivity and extra length for the braze. The braze is not anymore a critical parameter as the current one has a much higher thermal conductivity (409 W/(mK)) than the other materials and its extra length is negligible. The titanium has higher thermal conductivity compared to the stainless steel and the tube has half the thickness, so the impact due to its thermal conductivity is also strongly reduced.

Taking into account all the uncertainties on the parameters listed in Table 8.1, the model gives an uncertainty around $\pm 5\%$ on the linear relationship defined in equation 8.1 which will have an impact smaller than the uncertainty of the systematic error of the measurements.

Parameter	Central value	Uncertainty	Sensitivity [°C/W]	@6W [°C]
Copper thermal cond. [W/(m K)]	400	±10	± 0.01	± 0.06
Titanium thermal cond. [W/(m K)]	23	±2	± 0.02	± 0.12
Braze thermal cond. [W/(m K)]	409	±5	—	—
AL-FC thermal cond.[W/(m K)]	(230,230,120)	±(10,10,10)	±0.02	± 0.12
A. Alumina thermal cond. [W/(m K)]	6	±2	—	—
A. Alumina thickness [μm]	5	±3	—	—
Braze thickness [mm]	1.45	0.1	—	—
Heater distance from border [mm]	1	1	± 0.02	± 0.12
Titanium temp. coefficient [K ⁻¹]	0.027	0.003	—	—

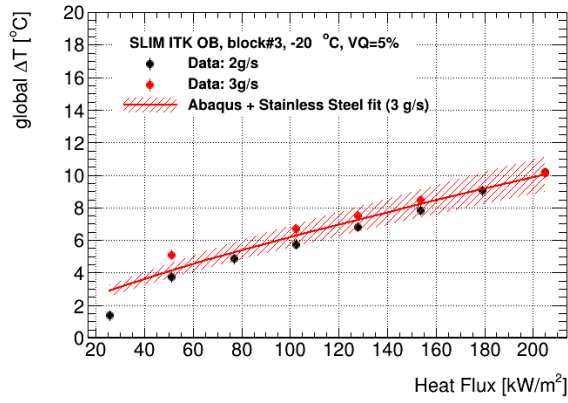
Table 8.1: List of model physical parameters considered to define in the Abaqus model of the realistic local support.

8.4 Results and analysis

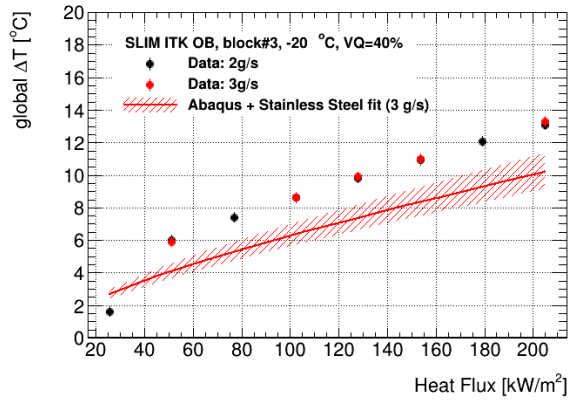
8.4.1 Saturation temperature and vapour quality

Figure 8.4 shows the ΔT_{global} (defined at page 73) as function of the heat flux with a titanium. The measurements at -20°C and -40°C tube demonstrate that the thermal performance is still better at higher saturation temperatures. However, when the measurements are compared with the stainless steel prototype, without applying any heat load on the cartridge heater, a different convective behaviour of the CO_2 is observed. While for the previous prototypes the performance did not vary much at different mass flows (2 g/s and 3 g/s), with a titanium tube the performance is always worse at higher mass flows. This difference is accentuated at lower temperatures.

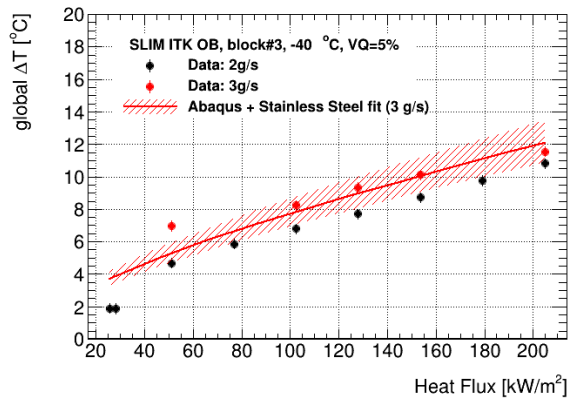
Figure 8.4 shows the thermal performance degradation with the rise of vapour quality. It goes in the opposite direction to what was observed with the stainless steel tube, invalidating what has been long time assumed as a design principle for the Outer Barrel CO_2 cooling system.



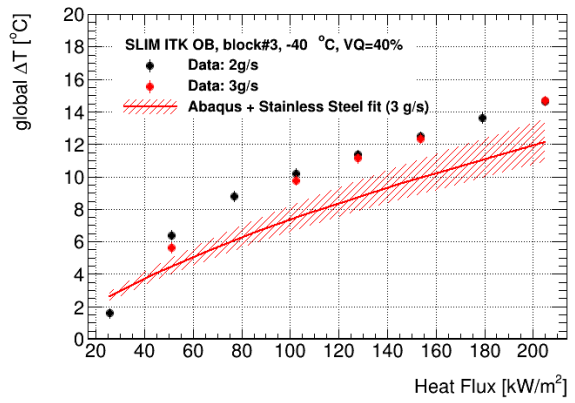
(a)



(b)



(c)



(d)

Figure 8.4: Measurements on the ITK OB prototype at -20°C and -40°C for different vapour qualities.

8.4.2 Manufacturing variability

The variability between measurements with different blocks does not seem to change significantly with the power applied. A constant offset is however observed, that can be explained by, either a systematic offset due to thermal probe attachment or a different contact resistances at the artie alumina layer between the base block and the copper block. The maximum temperature difference between the four² blocks was of $1.5 \pm 0.4^\circ\text{C}$ - see Figure 8.5.

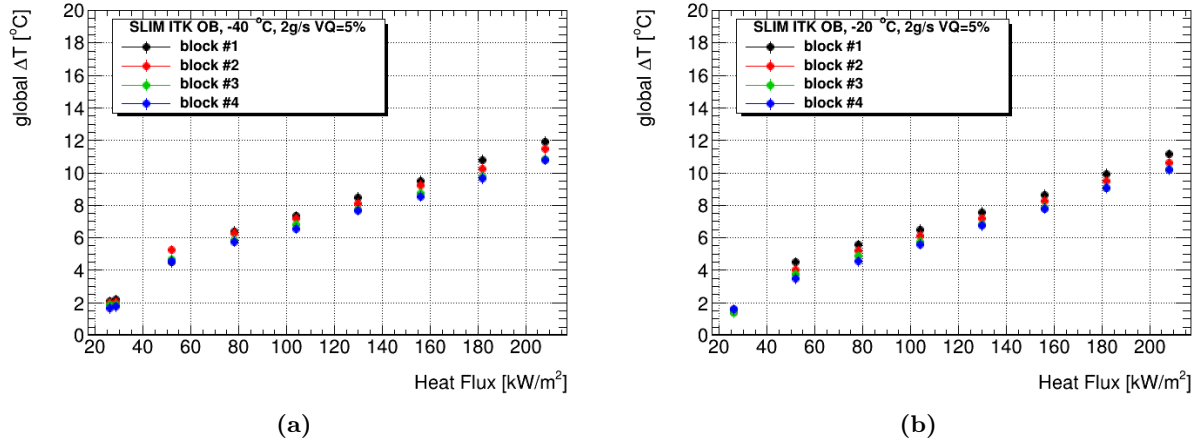


Figure 8.5: Measurements for every block illustrating the low production variability of the local support.

8.4.3 Low mass flows

Including the measurements at 1 g/s in Figure 8.6 demonstrates the different behaviour compared to other higher flux rates. At low vapour qualities, the performance at 1 g/s (Figures 8.6a and 8.6c) varies little with the temperature when compared to higher mass flows. At high vapour quality and high heat fluxes (Figures 8.6b and 8.6d) the performance is more sensitive to the CO_2 temperature, as it was the case for the measurements at 2 g/s and 3 g/s.

²After some temperature cycles, the attachment system placing the sensor on the fifth block detached from the copper block.

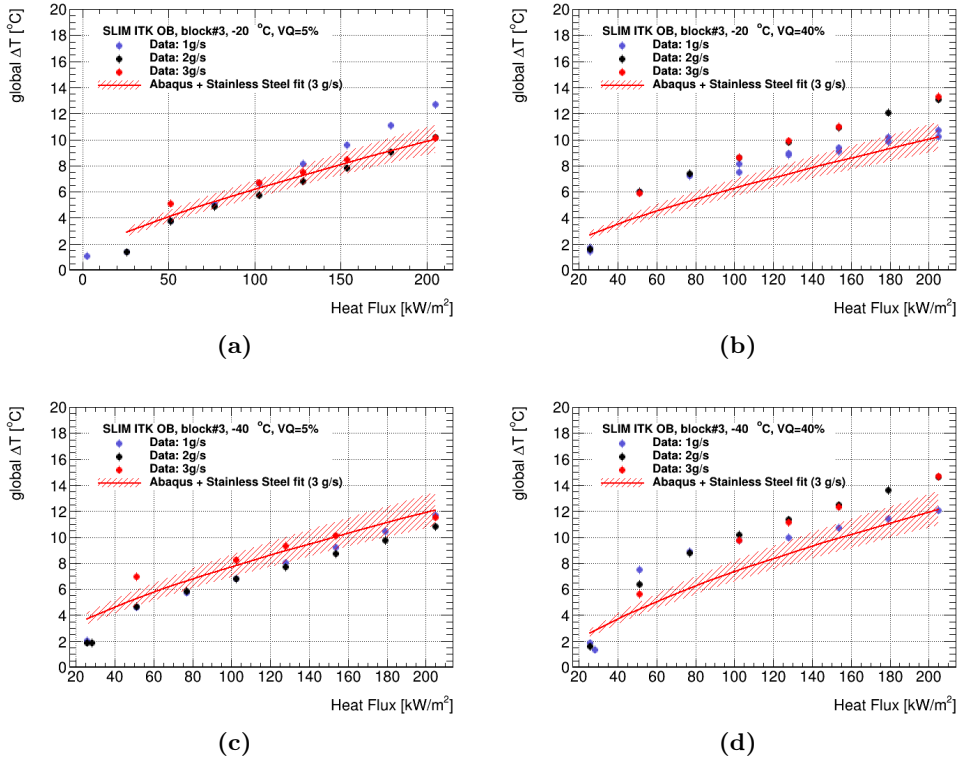


Figure 8.6: Measurements on the ITK OB prototype at -20°C and -40°C , 1 g/s, 2 g/s and 3 g/s at different vapour qualities.

8.5 Discussion: stainless steel vs titanium tubes

Figure 8.4 shows that the fit from the stainless steel prototype (Stainless Steel fit) does not match the titanium measurements. Looking back at the measurements for the stainless steel prototype, one can also observe how the mass flow has little impact on the performance at low vapour qualities, which is not the case for titanium.

The mass flow and the vapour quality are used in the model to estimate the flow pattern and thus, apply the correlation to estimate the heat transfer coefficient accordingly. At 2 g/s, the mass velocity in a tube with 2.5 mm inner diameter is estimated at $407 \text{ kg/m}^2/\text{s}$ which is above G_{wavy}^3 close to $200 \text{ kg/m}^2/\text{s}$. Above this limit, the flow pattern is expected to change from intermittent/bubbly to annular flow around a vapour quality of 10%.

Figure 8.7 shows how the behaviour changes with vapour quality with a clear difference at 3 g/s and 2 g/s between intermittent and annular flow. At 20% and 40% the performance stays the same at 3 g/s while at 2 g/s it only "saturates" at the same level as 3 g/s at 40% of vapour quality. Even if the performance remains better at -20°C than at -40°C , the offset between the Fit Stainless Steel and the measurements at 3 g/s is around 3°C for both cases.

The rise of vapour quality lowers the thickness of the layer of liquid in contact with the tube walls and the vapour and liquid velocities do increase. The high velocity of the liquid layer in a smooth surface seems to be degrading the heat exchange process between the liquid layer and the tube wall. This effect becomes negligible at higher vapour qualities where the main process of heat exchange is from free convection and not the nucleate boiling at the wall surface.

One may conclude that in smooth surfaces like in these titanium pipes, higher mass flows have a negative impact on the nucleate boiling process leading to lower cooling performance at

³Mass velocity defining the boundary between from intermittent/bubbly/annular from the stratified wavy flow - see section 2.5.2.

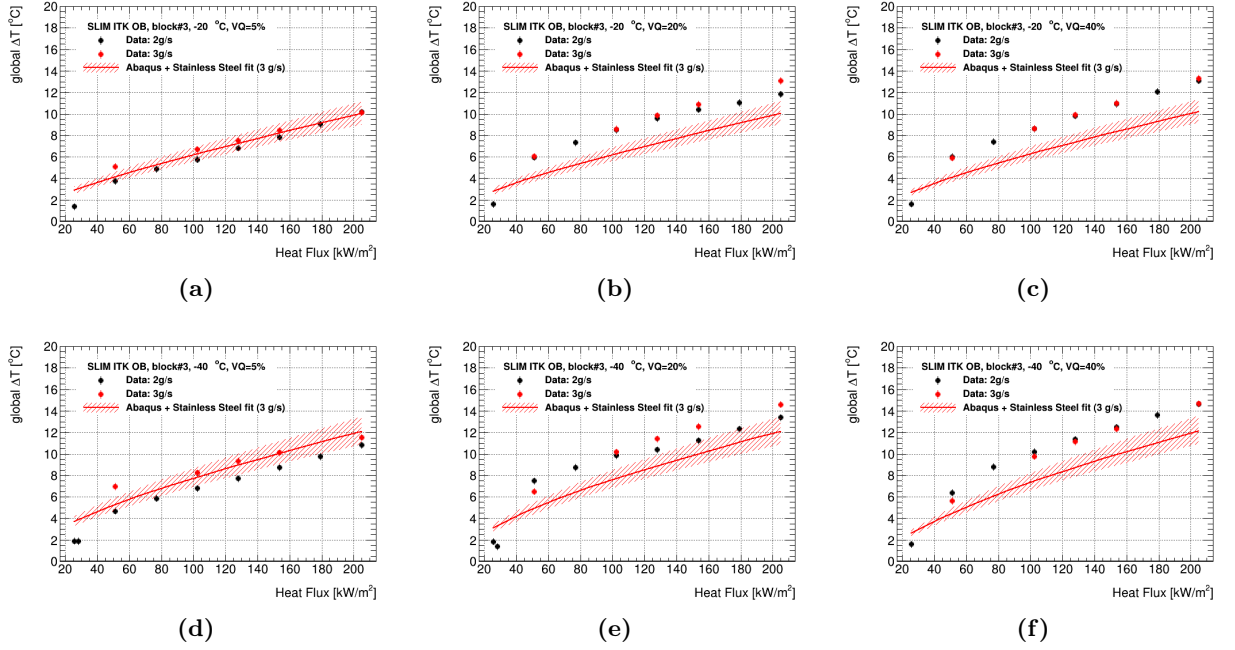


Figure 8.7: Measurements on the ITK OB prototype at $-20\text{ }^{\circ}\text{C}$ and $-40\text{ }^{\circ}\text{C}$, 2 g/s and 3 g/s at different vapour qualities (5%, 20% and 40%).

higher vapour qualities, when compared to rougher surfaces.

This reasoning behind the impact on the nucleate boiling may also explain the difficulties to trigger boiling in IBL. As described in section 2.3.3, to start boiling there is a level of super-heat needed to be surpassed, to trigger the boiling process. It is also shown how the level of super-heat depends on the roughness of the surface.

Figure 8.8 shows the high super-heat was observed in IBL dummy tests. The boiling was only triggered at the lowest mass flows, where no annular flow is expected. This was either because only in that cases the level of super-heat needed was reached or simply because a lower mass flows ease the bubble nucleation.

Another interesting observation is that, in the evaporator (IBL) section, the tube temperature is increasing along the tube instead of decreasing monotonically due to pressure drop (regardless the mass flow).

With 25 W applied over the active length of 66 cm, the vapour quality at the dummy load reached the vapour quality for intermittent-annular transition estimated around 10% before the end of evaporator at all mass flows considered.

In light of our measurements, the rise of temperature can be seen as a degradation of the heat transfer coefficient until the intermittent-annular transition that would afterwards stabilise. Unfortunately, the evaporator section is not long enough to verify the stabilisation of the temperature profile.

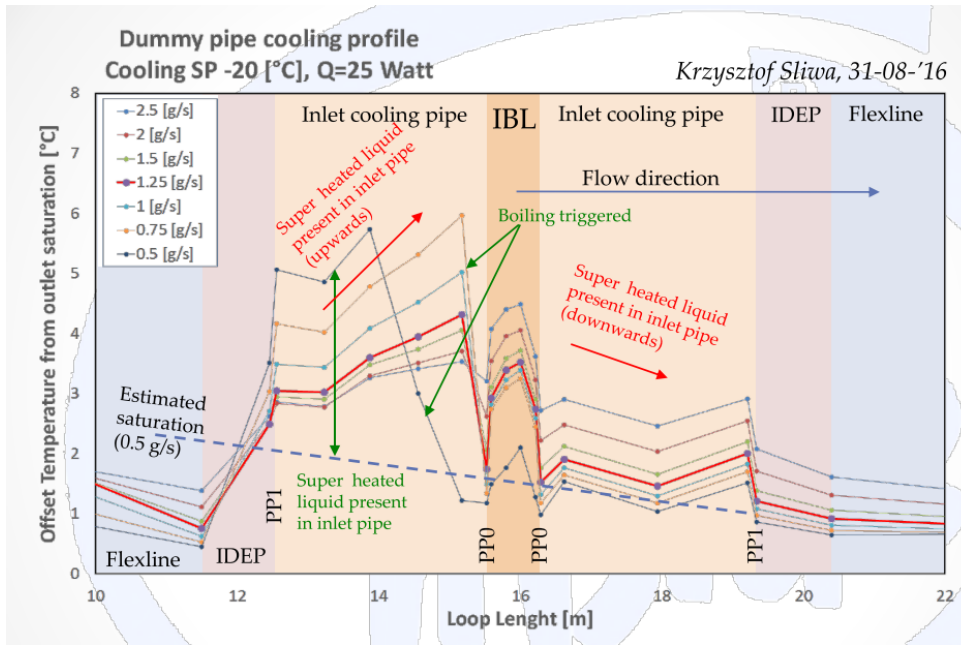


Figure 8.8: Tests performed by the CERN cooling group to investigate the CO_2 super-heating and problems triggering boiling in IBL [77] titanium tubes.

8.6 Consequences for ITk OB design

Based on the previous reasoning, the Stainless Steel Fit was slightly tweaked to estimate well the heat transfer coefficient for flows of 3 g/s or more (assuming above 3 g/s the performance will be similar). For that, it was assumed that the performance predicted in the previous fit will degrade gradually in titanium tubes until reaching the intermittent-annular transition with a thermal performance degradation of 3°C from 50 kW/m^2 to 200 kW/m^2 .

Figure 8.9 compares the heat transfer coefficient calculated with the original model, with the Stainless Steel fit and the proposed Titanium fit. At low vapour qualities the Stainless Steel fit is assumed to describe well the results in titanium, while at higher vapour qualities it is expected a lower heat transfer coefficient in the titanium tubes.

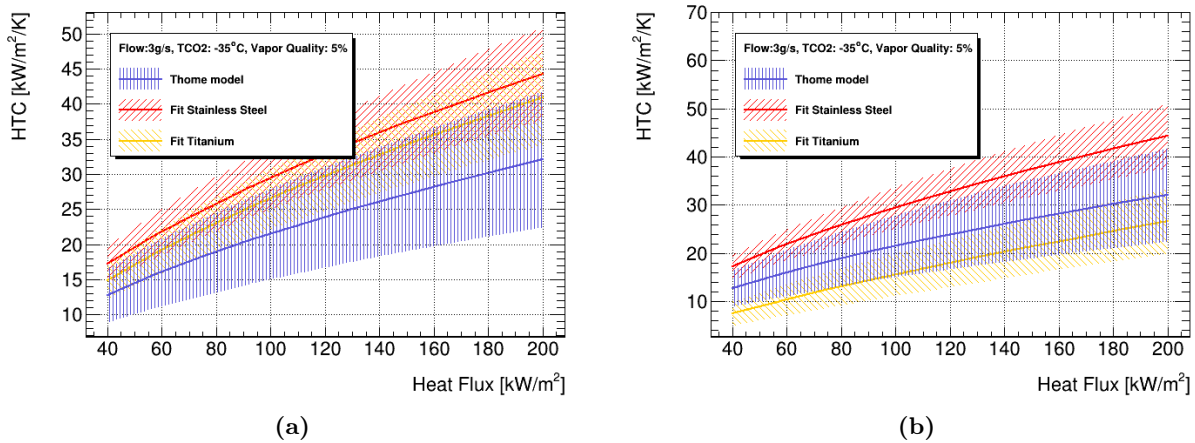


Figure 8.9: Comparison of HTC estimated for a 2.5mm ID titanium tube tube, using the Thome model, the Stainless steel and Titanium fit.

In the future ATLAS ITK Outer Barrel, the local heat fluxes are expected to be between

100 kW/m² and 150 kW/m². Hence, the ΔT_{conv} in the titanium tubes can go up to 9 °C considering our estimates - see Figure 8.10.

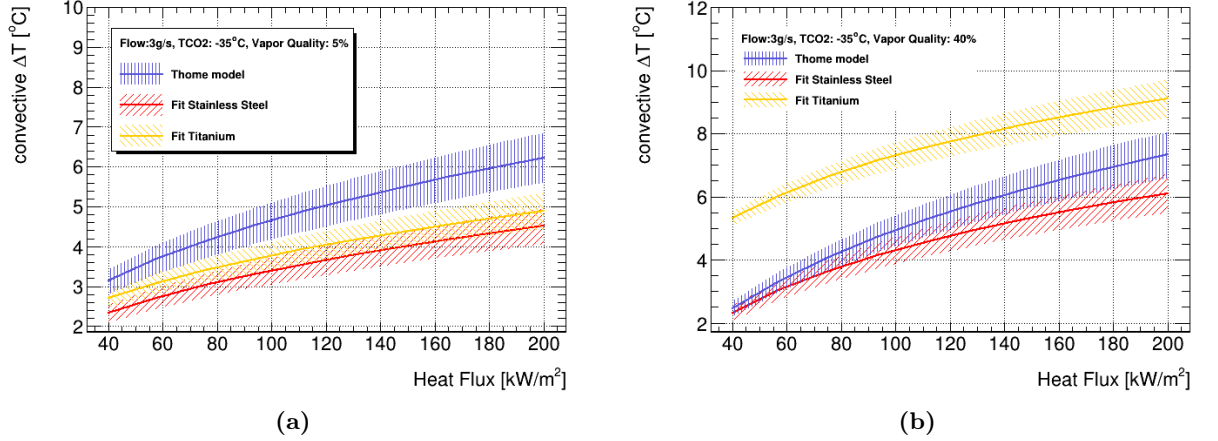


Figure 8.10: Comparison of estimated ΔT_{conv} for a 2.5mm ID titanium tube tube, using the Thome model, Stainless Steel and Titanium fit.

The most challenging thermo-fluidic conditions are found towards the end of life of the detector, when the sensors must be cooled to the lowest temperature possible to achieve. Predicting the cooling performance at the lowest temperature possible is therefore critical to foresee the true performance of the cooling system and accurately estimate the safety margins in advance. Looking at the results for the titanium tubes, the model has, finally, been able to predict reasonably well the cooling performance for the Outer Barrel - see Figure 8.9a. Nonetheless, further steps are needed to improve the predictions given the mismatch between the estimates and measurements on a real prototype.

The different cooling performance between titanium and stainless will consequently affect the temperature profile expected in the longerons of the Outer Barrel as illustrated in Figure 8.11. It replicates the same exercise conducted in section 5.11 to simulate the temperature profile along an Outer Barrel longeron considering the estimated heat transfer coefficient for titanium tubes. In the same section it was also shown how the critical CO₂ temperature will change over time, as the integrated luminosity increases. At the beginning, with a critical CO₂ temperature close to -10 °C, we will assume the CO₂ temperature will be set at -20 °C and -40 °C at the end of life of the detector.

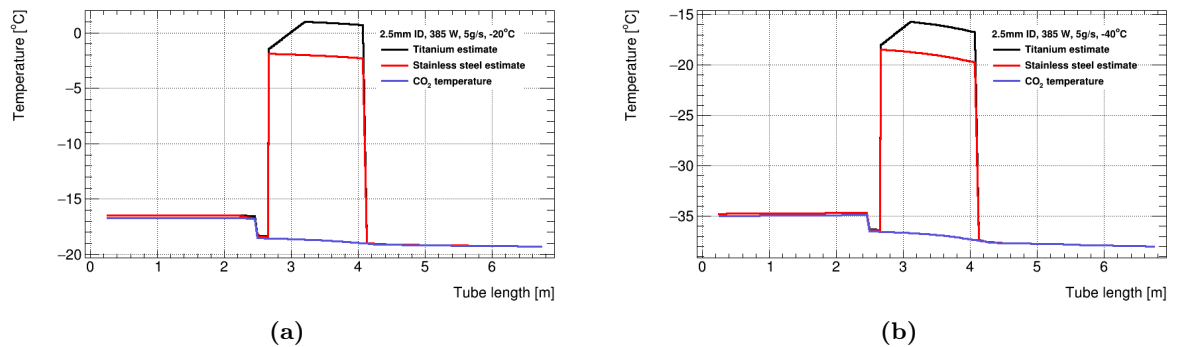


Figure 8.11: (a) Temperature profile expected for a longeron 1.47m long with 36 quad modules, dissipating a total of 385 W at the beginning of operation of the detector (cooled at -20 °C) and (b) at the end of operation. The simulation considers a local support with a TFM of 18 Kcm²/W, 2.45 m long inlet capillary and outlet exhaust with inner diameters 1 mm and 4 mm.

8.7 Final considerations on future qualification campaigns

Most of the early thermal qualification testing campaigns on small local support prototypes for the Outer Barrel considered mass flows of 1 g/s or lower and temperatures of -20°C or even higher. The present results show that testing at such low mass flows and high temperatures is not a good approach to reproduce the effective cooling performance of the future detector with conservative estimates.

The manufacturing variability observed in the local support was a systematic offset independent from the heat applied in the sensor. It was probably a combination of different contact resistances in Artic Alumina paste used as interface between the copper block, base block and the thermal probe attachment. The local support itself is believed to have a negligible impact on the variability in the measurements. In the future design, the use of soft pyrolytic graphite sheet (PGS) is being considered instead of the Artic Alumina paste, which should allow much better control of the thermal layer thickness and thermal variability between cells. To investigate in detail the manufacturing variability of future prototypes, it is important to conduct thermo-fluidic testing campaigns with different heat loads applied at the front-ends (or heaters mimicking their behaviour), to check if the dispersion between all the cells is coming from an offset or different conductive thermal performance.

Nonetheless, the most delicate observation in this chapter is the difference of cooling performance between titanium and stainless steel tubes. With titanium tubes, the performance degrades with increasing mass flow and vapour quality. The convective temperature gradient seems to saturate with the rise of the vapour quality and not depend on the mass flow. Taking data at 5 g/s (roughly the mass flow intended to cool the the future longerons) is vital to confirm or not.

The future Outer Barrel will experience heat fluxes between 100 kW/m^2 and 150 kW/m^2 . Assuming that a similar performance at 5 g/s as for 3 g/s, the ΔT_{conv} is expected to take around up to 9°C of the global temperature gradient (ΔT_{global}). This would take a big slice of the total margin of $\approx 35^{\circ}\text{C}$ allowed between the CO_2 temperature (-35°C at the inlet of evaporator) and the maximum allowable temperature by design of ($\approx 1^{\circ}\text{C}$ in layer 2), listed in Table 5.4.

The convective part of the evaporative system has never been seen as a risky component for the design of the local support. However, due to the particularities of the Outer Barrel, new challenging conditions arised highlighting how important it is to properly predict the convective performance of CO_2 cooling systems in particle detectors.



CONCLUSION

The plan at the beginning of the thesis was to conduct several testing campaigns in small prototypes to improve the simulations and finish by testing a full scale prototype to compare the measurements with the predictions. Unfortunately, the qualification tests for the Pixel Detector were postponed to the end of 2019. Nonetheless, the precision of the simulations has been improved and integrated in the custom simulation tools developed to calculate the thermal requirements of the Pixel Detector. Also, an odd behaviour was observed with titanium tubes and its impact on the thermal profile in a longeron is already anticipated.

Cooling the IBL with CO₂ has been a story of success in ATLAS. The cooling system has experienced very little downtime since its startup and no thermal runaway issues in the pixel sensors were observed. However, the IBL needs to respect a target for operating temperature of -15°C [12]. Interestingly the Pixel Detector, even though it will be under much harsher radiation environment, has target temperatures higher than for the IBL. It happens that this has no relation with the better radiation hardness of the new pixel technology. Not only will the leakage current be a more critical parameter, but also the local support will have many more interfaces: the IBL had a global TFM of $14\text{Kcm}^2/\text{W}$ while the Pixel Outer Barrel will count on a TFM close to $30\text{Kcm}^2/\text{W}$. To cope with the larger expected thermal figure of merit, the safety margins have simply been reduced.

Thanks to the experience acquired during the production of the IBL, having a re-workable local support was of great importance given the large number of pixel modules one might need to re-install in the ITk Pixel Detector. The solution using base and cooling blocks greatly reduced the contact area between the tube and pixel module. The increased heat flux at the tube was assumed to benefit the temperature performance as it was known that the heat transfer coefficient increases with the heat flux. Today, thanks to the work developed during the thesis, the teams responsible for the design of the local support are well aware that the reduction of the contact area has a negative impact on the thermal performance of the CO₂.

Moreover, the set of correlations used to predict the heat transfer coefficient was tested for the first time in the working region of the future ATLAS Pixel Detector. As the models used had never been tested at such high heat fluxes ($>100\text{kW}/\text{m}^2$) there were concerns that the estimates of heat transfer coefficient were too optimistic.

After comparing the model and the experimental data with an innovative approach based on a nuisance parameters fit where the modelling and systematic uncertainties were taken into account, the model was proved to not be valid in the working region of interest and a re-parameterization was proposed. A first set of tests conducted with stainless steel tube was afterwards compared with other measurements in a titanium tube. For the first time, it was observed that the performance in titanium tubes is worse than with stainless steel. The consequences on the temperature profile of the future local support of the Pixel OB are discussed.

The extensive testing campaigns presented anticipate several thermo-fluidic aspects. For example, the prototype used to test the thermal performance of the local support must be built with titanium tubes. Otherwise, the measurements risk to overestimate the real thermal performance and disregard possible super-heating effects. It happens that titanium tubes are a costly component, so prototypes are usually built using stainless steel tubes. This aspect could be easily overlooked if there had been no observation of different behaviour between the two

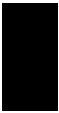
materials.

There are strong signs indicating that the increase of mass flow will worsen the global thermal performance in titanium tubes at low vapour qualities (beginning of the evaporator). Unfortunately, due to the limitations of the CO₂ blow-off system, it was not possible to test the prototypes with a flow above 3 g/s. Forthcoming thermal qualification tests will soon use a realistic CO₂ cooling plant capable of cooling the prototypes with the mass flows needed. This will be the opportunity to check if there is indeed a big change in the thermal performance of the fluid by changing the mass flow. Moreover, the observation of a sloppy temperature profile along the evaporator instead a monotonic decrease of module temperatures (as foreseen in Figure ??) would be inline with the odd behaviour discussed in chapter 8.

Most of the design requirements are driven by the continuous increase of the leakage current of silicon sensors throughout the lifetime of the detector. This will not be possible to replicate in a prototype equipped with real pixel modules unless they are entirely populated with irradiated sensors (rather unlikely). This experimental limitation must be taken into account as a non-controlled parameter in the measurements. The most important measurements from the thermal management standpoint is the absolute temperature difference between the hottest location of the un-radiated modules and the coolant temperature. This measurements will indicate the expected global performance of the local support considering the estimated heat transfer coefficient. In the case of the Outer Barrel, the most important is to check that worst performing modules do respect the production thermal figure of merit for the local support (TFM_K) of $\approx 25 \text{ Kcm}^2/\text{W}$ (see Table 5.4). To foresee how the measured performances will change throughout the life of the detector one can use the tools developed and made available to the collaboration.

Despite several misconceptions in the cooling properties of the CO₂ discussed in this thesis, the expected performances are within the design requirements. Moreover, based in previous experience with the IBL, it is shown in Table 5.4 that 10 TFM units are considered as safety margin to cope for the manufacturing variability. I find this too generous (around 10 °C) given that not only the thermal interfaces will be better controlled than for the IBL (IBL used porous carbon foam glued to other elements with epoxy) but also because badly assembled cells can be easily replaced thanks to the re-workability of the future local support. The first testing campaigns will be a good indicator of how good is the manufacturing variability. If it happens to be lower than what was expected a priori, we will have an extra safety margin to cope with unexpected issues appearing during the staves installation in the final detector.

The thermal testing campaigns will be crucial to evaluate the true thermal performance of the final local support. The worst measured performances can be used afterwards to derive the sensor power dissipation expected throughout the lifetime of the detector and the corresponding critical CO₂ temperature, using the tools here presented.



SUMMARY IN FRENCH

Chapter 1 - Mise à jour d'ATLAS Phase II

Ce chapitre présente le Grand collisionneur de hadrons, le détecteur ATLAS et ses sous-systèmes. Ce travail se concentre sur la description du détecteur de traces actuel (Inner Detector) et sa future mise à jour pour le programme haute luminosité (ITk).

Les arguments en faveur du choix d'une disposition inclinée du détecteur de pixels dans le futur détecteur de traces intérieur sont présentés. Le sujet principal de ce travail est la conception du support local pour le sous-système du tonneau extérieur du détecteur à pixels, chargé de maintenir en place les modules à pixels et d'établir le contact thermique entre le système de refroidissement et les éléments actifs (modules à pixels). Les choix de conception sont présentés et discutés.

Chapter 2 - Débits monophasés et diphasiques dans des minicanaux

Les fluides s'écoulant à l'intérieur des canaux sont largement utilisés comme systèmes de transport de chaleur. Les systèmes de réfrigération utilisent le fluide pour absorber la chaleur dissipée en cours de route et transporter l'énergie accumulée à travers les canaux de distribution, loin des sources de chaleur, pour éviter une élévation localisée de la température.

Dans les installations frigorifiques monophasées, le flux de vapeur/liquide voit sa pression chuter et sa température augmenter en cours de route, de la source de réfrigérant à l'extrémité du dissipateur thermique. D'autre part, les systèmes de réfrigération diphasiques utilisent des flux saturés pour extraire la chaleur de la source de chaleur dans un processus d'ébullition isothermique. Dans les deux cas, les parois du canal sont responsables de l'établissement du contact thermique entre le fluide et la source de chaleur.

L'efficacité des processus d'échange dans les écoulements monophasés entre le fluide et la paroi du canal dépend de la relation entre la surface d'échange thermique ($\propto$ diamètre, D) et le débit massique ($\propto D^2$). Par conséquent, les canaux avec des diamètres plus petits et un rapport surface/volume plus grand ($\propto 1/D$) [14] ont généralement une meilleure performance de refroidissement (cas des capillaires sanguins).

Les détecteurs de particules ont exploré des systèmes de refroidissement utilisant des écoulements diphasiques où les processus d'échange de chaleur sont plus difficiles à prévoir qu'avec des liquides de refroidissement monophasés. Il y a un grand intérêt à minimiser la taille de leurs systèmes de refroidissement, d'où l'importance de bien comprendre le processus d'échange de chaleur qui a lieu dans les flux diphasiques à l'intérieur des petits canaux.

En ce qui concerne le système de classification des canaux proposé par Kandlikar and Grande [15], les détecteurs au silicium ont utilisé des minicanaux pour leurs systèmes de réfrigération avec des diamètres intérieurs entre 0.5 mm. (pour capillaires) et 3 mm (pour évaporateurs).

Ce chapitre sert à donner un aperçu de plusieurs concepts utilisés dans la littérature pour décrire les différents types de débits monophasés et diphasiques. Il présente les propriétés de transfert de chaleur et les prédictions sur les propriétés thermo-physiques de l'écoulement, con-

sidérées comme cruciales pour la conception d'un système de refroidissement. Une section est consacrée à commenter l'origine de la surchauffe, sa relation avec le nucléate et l'ébullition convective et les limites théoriques données par la théorie des fluides de Van der Waals. Même s'il n'est pas crucial pour le sujet de ce travail, il a été un sujet d'actualité dans la communauté du refroidissement des détecteurs de particules. Les méthodes de prévision semi-empiriques pour calculer la perte de charge et le coefficient de transfert thermique utilisés pour concevoir le futur système de refroidissement par évaporation de l'ATLAS ITk sont présentées en détail et leur validité est examinée.

Chapter 3 - Refroidissement au CO₂ pour les détecteurs de traces

Les détecteurs au silicium doivent être refroidis pour maintenir leur température stable, maximiser le rapport signal/bruit et prévenir l'emballement thermique dépendant des dommages causés par le rayonnement. Dans la génération précédente de détecteurs au silicium pour la physique des particules, construits dans les années 1990, ils étaient refroidis à l'eau en mode monophasé dans des systèmes de pression sous-atmosphérique "sans fuite". Le principal inconvénient était le point de congélation élevé de l'eau (0 °C) et son faible rendement, ce qui rend difficile le refroidissement des détecteurs sous le point de congélation de l'eau [36] et la conductivité électrique, même avec de l'antigel. L'un des avantages du refroidissement biphasé est que l'extraction de la chaleur ne modifie pas la température sauf en case de chute des pression de la chute de pression le long de l'évaporateur. Comme le rapport signal/bruit du détecteur dépend de la température du capteur, l'uniformité de la température entre tous les capteurs favorise l'uniformité du comportement du capteur.

Pour refroidir ses détecteurs au silicium autour de -7°C , l'ATLAS ID a d'abord considéré une solution aqueuse avec un dépresseur de point de congélation (méthanol) qui utilisait la chaleur latente de fusion de cristaux de glace microscopiques en suspension pour améliorer la capacité thermique et fonctionnait dans un procédé quasi isotherme [36]. Cette idée a ensuite été abandonnée au profit d'un système de refroidissement par évaporation à base de perfluorocarbones (C₃F₈ ou R218) grâce à leurs propriétés thermo-physiques plus favorables, leur non-toxicité, leur inflammabilité et leur résistance au rayonnement. Cette solution a été implémentée pour les couches SCT et Pixel des détecteurs ATLAS installées dans les années 2000.

Entre-temps, une nouvelle solution de refroidissement par évaporation à base de CO₂ (réfrigérant R744) a gagné en popularité dans la communauté de la physique des hautes énergies. Cette nouvelle technologie a déjà été utilisée pour refroidir une nouvelle couche de Pixel plus interne que les précédents (IBL) d'ATLAS installé en 2014 et a été choisie comme référence pour refroidir l'ensemble du système ATLAS ITk qui sera installé en 2024.

Le présent chapitre décrit en détail les systèmes de refroidissement actuels et futurs utilisés pour refroidir les capteurs en silicium, en mettant en contexte les points d'état du diagramme P-H avec les processus thermodynamiques actuels des systèmes de refroidissement ATLAS (actuels et futurs). Alors que la présentation des arguments justifiant le passage du C₃F₈ installées à un système de refroidissement basé sur CO₂, plusieurs propriétés thermodynamiques sont comparées. Certains aspects des systèmes de distribution de flux, particulièrement importants pour le futur détecteur à pixels ATLAS ITk, sont présentés.

Chapter 4 - Outil de simulation du refroidissement au CO₂

Pour concevoir les systèmes de refroidissement a CO₂, le groupe de refroidissement du CERN a réalisé des simulations prédisant le comportement du liquide de refroidissement avec un outil développé dans MATLAB appelé CoBra (pour CO₂ Branch calculator) [51].

Entre autres applications, cet outil a été déterminant pour l'optimisation des dimensions des

tubes IBL [52] et pour prédire correctement le profil de température de la paroi du tube le long de ses lignes de refroidissement.

Néanmoins, il y avait un besoin récurrent de calculer le coefficient de transfert de chaleur, afin de le comparer avec les données expérimentales extraites de plusieurs campagnes d'essais menées au LAPP. Ainsi, un outil sur mesure, inspiré de CoBra, a été développé en langage de programmation python, mettant en œuvre les méthodes semi-empiriques décrites dans la section 2.5 pour estimer le coefficient de transfert thermique et la perte de charge.

L'implémentation en python était polyvalente et facilement couplée avec les mesures, ce qui était déterminant pour comparer d'énormes ensembles de données avec les estimations et tirer des conclusions sur la capacité d'expliquer les données avec le modèle actuellement utilisé.

Ce chapitre se concentre sur la description de l'outil personnalisé développé, avec quelques exemples de la façon dont il peut être utilisé pour simuler des scénarios réels. L'outil est validé avec des données expérimentales, utilisées pour optimiser la conception du diamètre du tube du futur détecteur à pixels et simuler la répartition du débit dans les collecteurs. L'interface web développée qui permet faire des calculs simples et mise à disposition de la collaboration ATLAS est également présentée.

Le backend de l'application s'est démontré assez polyvalent et on pourrait facilement inclure d'autres implémentations, utiles pour concevoir des systèmes de refroidissement par évaporation.

Comme futures améliorations, on pourrait prévoir l'inclusion d'autres coefficients de transfert de chaleur et d'autres méthodes de prévision des pertes de charge par frottement qui pourraient être mieux adaptées à d'autres liquides de refroidissement. De plus, dans la version actuelle, il n'y a toujours aucune considération sur la chute de pression due à des coudes dans les et aux restrictions d'entrée.

Chapter 5 - Gestion thermo-fluidique du détecteur à pixels d'ATLAS ITk

En raison des conditions extrêmes rayonnement attendues, les capteurs à pixels devront fonctionner à basse température (en dessous de -10°C). La température maximale de fonctionnement dépend principalement des dommages causés par le rayonnement et de la technologie du capteur.

La structure mécanique supportant les capteurs et les puces de lecture doit donc non seulement pouvoir les maintenir en place, mais aussi être couplée à un système de refroidissement, capable d'évacuer toute la chaleur produite, afin que les capteurs pixel soient maintenus en dessous de leur température maximale. L'évaluation correcte de la gestion thermo-fluidique du détecteur Pixel est donc d'une extrême importance, pour guider la conception des structures mécaniques et des services de refroidissement.

Le présent chapitre traite de l'importance de la gestion thermique des capteurs planaires et 3D en silicium, des limites du refroidissement du CO_2 et de l'impact sur la conception du support local des couches extérieures de la partie tonneau du détecteur à pixels.

Afin de comprendre quelle est la puissance totale produite dans le détecteur à pixels à prendre en compte pour concevoir le système de refroidissement, il est expliqué en détail comment estimer la puissance dissipée dans tous les composants d'un module Pixel tout au long de la vie du détecteur. Un outil personnalisé développé pour calculer l'ensemble des exigences de conception thermique et estimer les limites critiques (puissance et température de refroidissement) du système est décrit. Les résultats sont complétés par l'outil décrit dans le chapitre précédent pour une caractérisation thermo-fluidique complète du système de refroidissement CO_2 du tonneau extérieur du futur détecteur à pixels.

Chapter 6 - Qualification thermo-fluidique des premiers supports locaux

Lors de la phase de conception du détecteur ITk Pixel, deux solutions mécaniques ont été envisagées pour accueillir les modules inclinés : les "montagnes alpines" et les blocs de base SLIM. Le premier présentait l'avantage de performances thermiques très élevées, basées sur la même conception de mousse de carbone monobloc que le détecteur IBL, tandis que le second concept, assez innovant, présentait l'avantage significatif d'être " retravaillable ", pour faciliter le remplacement des modules défectueux lors du montage du détecteur.

Pour comparer les simulations avec les premiers essais sur prototypes réels, un banc d'essai expérimental a été construit au LAPP, capable de reproduire les conditions du refroidissement CO_2 de minimiser l'erreur systématique induite par la convection d'air en l'intégrant dans une enceinte à vide.

L'objectif des campagnes de mesure était de vérifier la compatibilité des deux concepts de supports locaux avec les exigences thermiques. C'était aussi l'occasion de vérifier la validité de la simulation thermique utilisée pour optimiser les matériaux thermiques et les interfaces afin d'obtenir les meilleures performances tout en minimisant la quantité de matériaux passifs.

La performance thermique attendue de chaque prototype est estimée à partir d'une analyse par éléments finis réalisée avec ABAQUS (*Dassault Systemes*) [70]. Les simulations sont présentées et la modélisation discutée. Une attention particulière est portée aux incertitudes liées à la simulation des performances thermiques des prototypes. Cette nouvelle approche est à la base des campagnes suivantes d'essais où chaque échantillon a été précédé d'une évaluation approfondie des incertitudes du modèle conducteur et convectif utilisé pour estimer la performance thermique globale. Le dispositif d'essai a été conçu selon un processus itératif tout au long de plusieurs campagnes d'essai menées. L'effort continu d'amélioration de la fiabilité de la mise au point en minimisant les sources d'erreurs expérimentales a été décisif dans le choix des instruments, des capteurs et des méthodes utilisées pour la fixation des sondes thermiques. L'utilisation d'une enceinte sous vide a été déterminante pour comprendre pourquoi le captage de la chaleur radiative est important à prendre en compte lors de la qualification des échantillons refroidis par des systèmes évaporatifs. Ces premiers tests étaient importants pour valider la fiabilité et la robustesse du dispositif de test et des outils utilisés pour traiter les données expérimentales.

La conclusion est, qu'en dépit d'un montage sous contrôle, une grande variabilité est observée entre les différents échantillons. Ce montre bien qu'il est important de contrôler précisément la fixation de la sonde et d'étudier la variabilité de la production des futurs prototypes.

Les simulations ont permis de prédire la forme hyperbolique du TFM global, causée par un comportement purement convectif du CO_2 en raison de l'absorption de chaleur radiative (estimée à 0.2W dans les courbes de prévision).

Globalement, les prototypes thermiques ont respecté les exigences thermiques du TDR. Cependant, on peut s'interroger sur la marge de sécurité étant donné les grandes incertitudes que nous avons actuellement sur les performances estimées.

En regardant nos mesures, la partie convective semble en effet prendre une part considérable sur la performance thermique globale (proche du 6°C prédit par le groupe de refroidissement du CERN utilisant CoBra). Cependant, tant que le montage d'essai n'est pas validé et qu'il n'est pas prouvé que le modèle de Thome utilisé fonctionne avec des flux thermiques supérieurs à $\approx 60\text{ kW/m}^2$, ce résultat peut encore être remis en question.

Il a donc été décidé de concevoir un prototype pour minimiser l'incertitude sur l'estimation ΔT_{cond} et de reproduire les flux de chaleur attendus pour les futurs supports locaux du détecteur à pixels. Le but de ce nouveau prototype n'était plus de mesurer la performance thermique du support local, mais de vérifier la validité des prédictions du modèle convectif dans la zone de travail du détecteur à pixels.

Chapter 7 - Nouveau paramétrage du modèle convectif CO₂

Le support local et tout autre élément du détecteur est généralement surdimensionné pour respecter les marges de sécurité techniques. Celles-ci sont fondées sur les limites opérationnelles, les exigences de rendement et le degré d'incertitude de certains paramètres. Les capteurs à pixels subiront des dommages dus au rayonnement et la température minimale nécessaire au fonctionnement du capteur sera déjà une contrainte majeure dans la conception du support local (géométrie, matériaux utilisés et variabilité de fabrication). De plus, l'incertitude sur la performance thermique d'un support local n'est pas négligeable et l'incertitude sur la performance du liquide de refroidissement augmente avec des charges thermiques plus élevées.

Surdimensionner le support local en termes de performances thermiques signifie souvent l'utilisation de matériaux plus denses (et de bons conducteurs) qui vont compromettre la précision du suivi des particules. Il est donc extrêmement important de bien prédire la température maximale admissible au niveau du capteur et la performance thermique du support local afin que les marges de sécurité ne compromettent pas la performance de suivi.

Pour évaluer la précision des méthodes de modélisation utilisées pour évaluer les performances thermiques du CO₂, des supports locaux simplifiés ont été construits pour minimiser les incertitudes dues aux multiples interfaces thermiques et pour faciliter l'extraction séparée des composants convectifs et conducteurs. L'objectif principal de ces campagnes d'essais est de vérifier si les méthodes actuelles utilisées pour calculer le coefficient de transfert thermique sont conformes aux mesures.

Ce chapitre présente l'analyse par éléments finis effectuée pour modéliser la performance conductrice des échantillons simplifiés. Une nouvelle méthode basée sur un ajustement des paramètres de nuisance est utilisée pour séparer les contributions conductrices et convectives à la performance thermique globale et pour vérifier si le modèle convectif, avec une incertitude de $\pm 30\%$ sur le coefficient de transfert thermique estimé, est validé par les données expérimentales. Les résultats d'ajustement sont également placés dans le contexte de la zone de travail du détecteur à pixels et utilisés pour évaluer la variabilité de fabrication des échantillons.

Dans ce chapitre, nous concluons que le modèle utilisé pour calculer le coefficient de transfert thermique n'est pas valide dans certaines régions de travail d'intérêt. On a quantifié les incertitudes sur le captage de la chaleur radiative, la variabilité de fabrication des prototypes et la conductivité thermique des matériaux. La bonne évaluation des incertitudes de la modélisation est cruciale pour conclure que le modèle n'était pas en accord avec les mesures à des températures élevées (jusqu'à -20°C) et à des flux thermiques élevés ($>100\text{ kW/m}^2$). Cette observation est d'autant plus importante qu'elle correspond aux conditions de fonctionnement du fût extérieur. C'est aussi la première fois que les résultats de simulations sont remis en question et évalués à l'aide de mesures, réalisées avec un prototype simplifié avec un soin particulier pour la modélisation et les incertitudes systématiques.

Mesurer le désaccord entre les mesures et la simulation a été déterminant pour comprendre si l'écart provient d'une mauvaise construction du modèle conducteur ou du modèle utilisé pour calculer le coefficient de transfert thermique.

Inspiré par la façon dont les physiciens analysent les grands ensembles de données pour contraindre les modèles, extraire les paramètres de modélisation et tester la compatibilité des mesures avec la théorie, un ajustement des paramètres de nuisance a été implémenté. Le même principe a été utilisé pour démêler les contributions conductrice et convective et conclure laquelle de ces contributions pourrait expliquer les données.

Étant donné que le modèle convectif lui-même est composé d'un ensemble de fonctions avec des exposants qui proviennent d'un ajustement sur un ensemble de données expérimentales donné, la même chose a été effectuée ici pour trouver les exposants qui conviennent le mieux à notre région de travail. Ce nouveau paramétrage du modèle a permis d'améliorer la précision du coefficient de transfert thermique d'un facteur deux.

La variabilité de fabrication a également été évaluée indirectement, car la longueur de brasage supplémentaire a également été prise en compte comme variable du modèle d'ajustement.

Les mesures étaient importantes pour observer comment la performance thermique du fluide dépend des conditions de refroidissement. Il s'améliore avec la qualité de la vapeur, se détériore à basse température et ne varie pas beaucoup entre 2 g/s et 3 g/s. Même si le modèle convectif l'avait déjà prédit, il était important de faire prendre conscience de l'importance de la prise en compte de la température de refroidissement dans les campagnes de qualification thermique.

La méthode et le dispositif d'essai se sont avérés robustes et les conclusions importantes pour mieux comprendre le refroidissement avec CO₂ et les limites des simulations. Il a donc été décidé de répéter le même exercice, sur un prototype plus précis et réaliste avec des tubes en titane et un procédé de fabrication mieux maîtrisé.

Chapter 8 - Caractérisation d'un support local réaliste

Un troisième prototype a également été équipé de cinq blocs de cuivre et testé de la même manière que les prototypes présentés dans le chapitre précédent. Il a été construit (fin 2018) avec les éléments (blocs de base et tube en titane) et la méthodologie (brasage du bloc de base sur tube en titane) destinés à être utilisés dans la conception finale du support local.

L'objectif principal de cette nouvelle campagne d'essais était de réaliser une qualification thermo-fluidique et de fabrication sur un prototype plus réaliste.

Cependant, un comportement étrange de CO₂ avait déjà été observé dans des tuyaux en titane, c'était aussi l'occasion de comparer avec les mesures du prototype en acier inoxydable et de vérifier si la performance de refroidissement était affectée uniquement par le changement du matériau du tube.

La faible rugosité des tuyaux en titane était déjà liée au retardement de l'ébullition observé dans l'IBL [9]. Comme expliqué dans la section 2.3.3, la rugosité de surface joue un rôle important dans la nucléation de la bulle, améliorant l'échange thermique entre le fluide de refroidissement et la paroi chauffée. Il a déjà été démontré qu'un meilleur coefficient de transfert thermique (principalement l'ébullition nucléée) dans les surfaces rugueuses est corrélé à la rugosité de la surface [75]. Une étude menée sur des surfaces en titane révèle de grandes différences sur le coefficient de transfert de chaleur mesuré entre une surface en titane non traitée et des couches minces d'oxyde de titane allant de 100 nm à 300 nm [76]. Il n'est donc pas étonnant que si le titane a une surface beaucoup plus lisse que l'acier inoxydable, on observe un comportement différent au refroidissement.

Le nouveau prototype est décrit et sa performance thermique prévue est simulée. Les mesures sont comparées avec le prototype en acier inoxydable, le modèle est ajusté pour être en accord avec les mesures et il est discuté comment les différents comportements de refroidissement affecteront le profil de température des longerons dans le futur tonneau externe. Des simulations prédisant le profil de température réelle au début et à la fin du fonctionnement du détecteur sont présentées avec une discussion sur les futures campagnes d'essais de qualification.

La plupart des premières campagnes d'essais de qualification thermique sur de petits prototypes de supports locaux pour le tonneau extérieur du détecteur à pixels ont considéré des débits massiques de 1 g/s ou moins et des températures de -20 °C ou même plus. Les résultats actuels montrent que les essais à des débits massiques aussi faibles et à des températures aussi élevées ne constituent pas une bonne approche pour reproduire la performance de refroidissement efficace du futur détecteur avec des estimations prudentes.

La variabilité de fabrication observée pour le support local était un décalage systématique indépendant de la chaleur appliquée dans le capteur. Il s'agissait probablement d'une combinaison de différentes résistances de contact dans une pâte d'alumine arctique utilisée comme interface entre le bloc de cuivre, le bloc de base et la fixation de la sonde thermique. Cela nous fait penser que le support local lui-même a un impact négligeable sur la variabilité des mesures.

Lors de la prochaine réalisation, l'utilisation de feuilles de graphite pyrolytique tendre (PGS) est envisagée à la place de la pâte thermique, qui devrait permettre un bien meilleur contrôle de l'épaisseur de la couche thermique et de la variabilité thermique entre cellules. Pour étudier en détail la variabilité de fabrication des futurs prototypes, il est important de mener des campagnes d'essais thermofluidiques avec différentes charges thermiques appliquées aux faces avant (ou des réchauffeurs imitant leur comportement), pour vérifier si la dispersion entre toutes les cellules provient d'une performance thermique décalée ou conductrice différente.

Néanmoins, l'observation la plus délicate dans ce chapitre est la différence de performance de refroidissement entre les tubes en titane et en acier inoxydable. Dans le cas des tubes en titane, la performance se dégrade avec l'augmentation du débit massique et de la qualité de vapeur. Le gradient de température convective semble saturer avec l'augmentation de la qualité de la vapeur et ne dépend pas du débit massique. Faire des mesures avec un débit de 5 g/s (à peu près le débit massique destiné à refroidir les futurs longerons) est critique pour confirmer cet effet.

Le futur tonneau extérieur subira des flux de chaleur entre 100 kW/m² et 150 kW/m². en supposant une performance similaire entre 3 et 5 g/s, le ΔT_{conv} devrait prendre environ jusqu'à 9 °C du gradient global de température (ΔT_{global}). Cela prendrait une grosse tranche de la marge totale de ≈ 35 °C permise entre la CO₂ température (−35 °C à l'entrée de l'évaporateur) et la température maximale admissible de (≈ 1 °C en couche 2), indiquée dans Table 5.4.

La partie convective du système évaporatif n'a jamais été considérée comme un élément à risque pour la conception du support local. Cependant, en raison des particularités du tonneau extérieur, de nouvelles conditions difficiles sont apparues, soulignant l'importance de prédire correctement la performance convective des systèmes de refroidissement CO₂ pour les détecteurs de particules.

Conclusions

Au début de la thèse, il était prévu de mener plusieurs campagnes de mesure sur de petits prototypes afin d'améliorer les simulations et de finaliser les mesures avec un prototype pour comparer les mesures avec les prédictions. Malheureusement, les tests de qualification pour le détecteur à pixels ont été reportés à fin 2019. Néanmoins, la précision des simulations a été améliorée et intégrée dans les outils de simulation développés pour calculer les besoins thermiques du détecteur à pixels. De plus, un comportement étrange a été observé avec les tubes en titane et son impact sur le profil thermique dans un longeron est déjà anticipé.

Le refroidissement de l'IBL a été un succès, avec très peu de temps d'arrêt depuis sa mise en service et aucun problème d'emballage thermique dans les capteurs à pixels n'a été observé. Toutefois, l'IBL doit respecter un objectif de température de fonctionnement de −15 °C [12]. Il est intéressant de noter que le détecteur à pixels, même s'il est soumis à un environnement de rayonnement beaucoup plus rude, a des températures cibles plus élevées que celles de l'IBL. Il se trouve que cela n'a aucun rapport avec la meilleure tenue aux radiations de la nouvelle technologie des pixels. Non seulement le courant de fuite sera un paramètre plus critique, mais le support local aura aussi beaucoup plus d'interfaces : l'IBL aura un TFM global de 14 Kcm²/W tandis que le tonneau externe comptera sur un TFM proche de 30 Kcm²/W. Pour faire face à la plus grande valeur thermique attendue, les marges de sécurité ont tout simplement été réduites.

Grâce à l'expérience acquise lors de la production de l'IBL, avoir un support local retravaillable était d'une grande importance étant donné le grand nombre de modules à pixels qu'il a fallu réinstaller dans le détecteur à pixels ITk. La solution utilisant des blocs de base et de refroidissement a considérablement réduit la surface de contact entre le tube et le module à pixels. On a supposé que l'augmentation du flux de chaleur au niveau du tube favorisait la performance de la température, car on savait que le coefficient de transfert de chaleur augmente avec le flux de chaleur. Aujourd'hui, grâce aux travaux développés au cours de la thèse, les équipes en charge de la conception du support local sont bien conscientes que la réduction de la surface de contact

a un impact négatif sur la performance thermique du CO₂.

De plus, l'ensemble des corrélations utilisées pour prédire le coefficient de transfert thermique a été testé pour la première fois dans la zone de travail du futur détecteur à pixels ATLAS. Comme les modèles utilisés n'avaient jamais été testés avec des flux thermiques aussi élevés ($>100 \text{ kW/m}^2$), on craignait que les estimations du coefficient de transfert thermique soient trop optimistes.

Après avoir comparé le modèle et les données expérimentales avec une approche innovante basée sur une adaptation des paramètres de nuisance où la modélisation et les incertitudes systématiques ont été prises en compte, le modèle s'est avéré non valide dans la région de travail d'intérêt et un nouveau paramétrage a été proposé. Une première série d'essais réalisés avec un tube en acier inoxydable a ensuite été comparée à d'autres mesures dans un tube en titane. Pour la première fois, il a été observé que les performances des tubes en titane sont pires qu'avec l'acier inoxydable. Les conséquences sur le profil de température du futur support local sont discutées.

Les vastes campagnes d'essais présentées anticipent plusieurs aspects thermofluidiques. Par exemple, le prototype utilisé pour tester la performance thermique du support local doit être construit avec des tubes en titane. Sinon, les mesures risquent de surestimer les performances thermiques réelles et d'ignorer d'éventuels effets de surchauffe. Il arrive que les tubes en titane soient un composant coûteux, c'est pourquoi les prototypes sont généralement construits avec des tubes en acier inoxydable. Cet aspect pourrait facilement être négligé s'il n'y avait pas eu d'observation de comportement différent entre les deux matériaux.

Il y a de forts signes indiquant que l'augmentation du débit massique va dégrader les performances thermiques globales des tubes en titane à basse qualité de vapeur (début de l'évaporateur). Malheureusement, en raison des limites du système de production de CO₂ froid, il n'a pas été possible de tester les prototypes avec un débit supérieur à 3 g/s. Pour les essais de qualification thermique à venir, on utilisera bientôt une installation de refroidissement réaliste CO₂ capable de refroidir les prototypes avec les débits massiques nécessaires. Ce sera l'occasion de vérifier s'il y a effectivement un changement important dans la performance thermique du fluide en modifiant le débit massique. De plus, l'observation d'un profil de température négligé le long de l'évaporateur au lieu d'une diminution monotone des températures des modules serait conforme au comportement étrange discuté au Chapitre 8.

Pour bien comparer les mesures avec les simulations, il est important de modéliser le changement sur les propriétés thermo-fluidiques le long de la longueur du tube. Ceci est possible à l'aide des outils de simulation proposés ici.

BIBLIOGRAPHY

- [1] ATLAS Collaboration. *ATLAS : letter of intent for a general-purpose pp experiment at the large hadron collider at CERN, CERN-LHCC-92-004*. Tech. rep. Geneva: CERN. Geneva. LHC Experiments Committee, 1992, p. 108. URL: <http://cds.cern.ch/record/291061>.
- [2] G Aad et al. “Observation of a new particle in the search for the Standard Model Higgs boson with the ATLAS detector at the LHC”. In: *Physics Letters B* 716.1 (Sept. 2012), pp. 1–29. ISSN: 0370-2693. DOI: [10.1016/J.PHYSLETB.2012.08.020](https://doi.org/10.1016/J.PHYSLETB.2012.08.020). URL: <https://www.sciencedirect.com/science/article/pii/S037026931200857X#!>.
- [3] B. Abbott et al. “Production and integration of the ATLAS Insertable B-Layer”. In: *Journal of Instrumentation* 13.5 (2018). ISSN: 17480221. DOI: [10.1088/1748-0221/13/05/T05008](https://doi.org/10.1088/1748-0221/13/05/T05008). URL: <https://arxiv.org/abs/1803.00844>.
- [4] Peter Vankov. “ATLAS Upgrade for the HL-LHC: meeting the challenges of a five-fold increase in collision rate”. In: *EPJ Web of Conferences* 28 (Jan. 2012), p. 12069. ISSN: 2100-014X. DOI: [10.1051/epjconf/20122812069](https://doi.org/10.1051/epjconf/20122812069). URL: <http://www.epj-conferences.org/10.1051/epjconf/20122812069%20http://arxiv.org/abs/1201.5469%20http://dx.doi.org/10.1051/epjconf/20122812069>.
- [5] *ATLAS Phase-II Upgrade Scoping Document*. Sept. 2015. URL: <https://cds.cern.ch/record/2055248?ln=fr>.
- [6] A Ferrari et al. “FLUKA : A multi-particle transport code (program version 2005)”. In: (2005). DOI: [10.5170/cern-2005-010](https://doi.org/10.5170/cern-2005-010). URL: <https://cds.cern.ch/record/898301>.
- [7] Torbjörn Sjöstrand, Stephen Mrenna, and Peter Skands. “A Brief Introduction to PYTHIA 8.1”. In: (Oct. 2007). DOI: [10.1016/j.cpc.2008.01.036](https://doi.org/10.1016/j.cpc.2008.01.036). URL: <http://arxiv.org/abs/0710.3820%20http://dx.doi.org/10.1016/j.cpc.2008.01.036>.
- [8] Collaboration ATLAS. “ATLAS Inner Tracker Pixel Detector - Technical Design Report”. In: (2017). URL: <https://cds.cern.ch/record/2285585/files/ATLAS-TDR-030.pdf>.
- [9] Collaboration ATLAS. *ATLAS Inner Tracker Strip Detector - Technical Design Report*. Tech. rep. 2017. URL: <https://cds.cern.ch/record/2257755/>.
- [10] Collaboration ATLAS. *Letter of Intent for the Phase-II Upgrade of the ATLAS Experiment*. Dec. 2012. URL: <https://cds.cern.ch/record/1502664?ln=fr>.
- [11] Andreas Salzburger et al. *Final Report of the ITk Layout Task Force*. Dec. 2016. URL: <https://cds.cern.ch/record/2239573?>.
- [12] M Capeans et al. *ATLAS Insertable B-Layer Technical Design Report*. Sept. 2010. URL: <https://cds.cern.ch/record/1291633?ln=fr>.
- [13] Diego Feito. *Private communication*. Geneva.
- [14] Satish G. Kandlikar et al. *Heat Transfer and Fluid Flow in Minichannels and Microchannels*. Elsevier, 2014. ISBN: 9780080983462. DOI: [10.1016/C2011-0-07521-X](https://doi.org/10.1016/C2011-0-07521-X). URL: <https://linkinghub.elsevier.com/retrieve/pii/C2011007521X>.

- [15] Satish G. Kandlikar and William J. Grande. “Evolution of Microchannel Flow Passages—Thermohydraulic Performance and Fabrication Technology”. In: *Heat Transfer Engineering* 24.1 (Jan. 2003), pp. 3–17. ISSN: 0145-7632. DOI: [10.1080/01457630304040](https://doi.org/10.1080/01457630304040). URL: <http://www.tandfonline.com/doi/abs/10.1080/01457630304040>.
- [16] Lixin Cheng et al. “New prediction methods for CO₂ evaporation inside tubes: Part I – A two-phase flow pattern map and a flow pattern based phenomenological model for two-phase flow frictional pressure drops”. In: *International Journal of Heat and Mass Transfer* 51.1-2 (Jan. 2008), pp. 111–124. ISSN: 00179310. DOI: [10.1016/j.ijheatmasstransfer.2007.04.002](https://doi.org/10.1016/j.ijheatmasstransfer.2007.04.002). URL: <https://linkinghub.elsevier.com/retrieve/pii/S0017931007002839>.
- [17] James B. Taylor, Andres L. Carrano, and Satish G. Kandlikar. “Characterization of the effect of surface roughness and texture on fluid flow—past, present, and future”. In: *International Journal of Thermal Sciences* 45.10 (Oct. 2006), pp. 962–968. ISSN: 12900729. DOI: [10.1016/j.ijthermalsci.2006.01.004](https://doi.org/10.1016/j.ijthermalsci.2006.01.004). URL: <https://linkinghub.elsevier.com/retrieve/pii/S1290072906000056>.
- [18] F.W. Dittus and L.M.K. Boelter. “Heat transfer in automobile radiators of the tubular type”. In: *International Communications in Heat and Mass Transfer* 12.1 (Jan. 1985), pp. 3–22. ISSN: 07351933. DOI: [10.1016/0735-1933\(85\)90003-X](https://doi.org/10.1016/0735-1933(85)90003-X). URL: <https://linkinghub.elsevier.com/retrieve/pii/073519338590003X>.
- [19] Satish G. Kandlikar. “Nucleation characteristics and stability considerations during flow boiling in microchannels”. In: *Experimental Thermal and Fluid Science* 30.5 (May 2006), pp. 441–447. ISSN: 08941777. DOI: [10.1016/j.expthermflusci.2005.10.001](https://doi.org/10.1016/j.expthermflusci.2005.10.001). URL: <https://linkinghub.elsevier.com/retrieve/pii/S0894177705001202>.
- [20] Lixin Cheng, Gherhardt Ribatski, and John R. Thome. *Two-Phase Flow Patterns and Flow-Pattern Maps: Fundamentals and Applications*. Vol. 61. 5. Applied mechanics reviews, 2008, p. 050802. DOI: [10.1115/1.2955990](https://doi.org/10.1115/1.2955990). URL: <http://appliedmechanicsreviews.asmedigitalcollection.asme.org/article.aspx?articleid=1399061>.
- [21] Lixin Cheng et al. “New flow boiling heat transfer model and flow pattern map for carbon dioxide evaporating inside horizontal tubes”. In: *International Journal of Heat and Mass Transfer* 49.21-22 (Oct. 2006), pp. 4082–4094. ISSN: 00179310. DOI: [10.1016/j.ijheatmasstransfer.2006.04.003](https://doi.org/10.1016/j.ijheatmasstransfer.2006.04.003). URL: <https://linkinghub.elsevier.com/retrieve/pii/S0017931006002444>.
- [22] R W Lockhart and R C Martinelli. “Proposed correlation of data for isothermal two-phase, two-component flow in pipes”. In: *Chemical Engineering Progress* 45.1 (1949), pp. 39–48.
- [23] John C Chen. “Correlation for Boiling Heat Transfer to Saturated Fluids in Convective Flow”. In: *Industrial & Engineering Chemistry Process Design and Development* 5.3 (July 1966), pp. 322–329. ISSN: 0196-4305. DOI: [10.1021/i260019a023](https://doi.org/10.1021/i260019a023). URL: <http://pubs.acs.org/doi/abs/10.1021/i260019a023>.
- [24] K.E. Gungor and R.H.S. Winterton. “A general correlation for flow boiling in tubes and annuli”. In: *International Journal of Heat and Mass Transfer* 29.3 (Mar. 1986), pp. 351–358. ISSN: 00179310. DOI: [10.1016/0017-9310\(86\)90205-X](https://doi.org/10.1016/0017-9310(86)90205-X). URL: <https://linkinghub.elsevier.com/retrieve/pii/001793108690205X>.
- [25] Sung Min Kim and Issam Mudawar. “Universal approach to predicting saturated flow boiling heat transfer in mini/micro-channels – Part II. Two-phase heat transfer coefficient”. In: *International Journal of Heat and Mass Transfer* 64 (2013), pp. 1239–1256. ISSN: 00179310. DOI: [10.1016/j.ijheatmasstransfer.2013.04.014](https://doi.org/10.1016/j.ijheatmasstransfer.2013.04.014). URL: <http://dx.doi.org/10.1016/j.ijheatmasstransfer.2013.04.014>.

- [26] David C. Johnston. “Thermodynamic Properties of the van der Waals Fluid”. In: (2014). ISSN: 1943-2909. DOI: [10.1088/978-1-627-05532-1](https://doi.org/10.1088/978-1-627-05532-1). URL: <http://arxiv.org/abs/1402.1205v0><http://dx.doi.org/10.1088/978-1-627-05532-1>.
- [27] Lixin Cheng, Gherhardt Ribatski, and John R. Thome. “New prediction methods for CO₂ evaporation inside tubes: Part II—An updated general flow boiling heat transfer model based on flow patterns”. In: *International Journal of Heat and Mass Transfer* 51.1-2 (Jan. 2008), pp. 125–135. ISSN: 00179310. DOI: [10.1016/j.ijheatmasstransfer.2007.04.001](https://doi.org/10.1016/j.ijheatmasstransfer.2007.04.001). URL: <https://linkinghub.elsevier.com/retrieve/pii/S0017931007002840>.
- [28] John R Thome and Jean El Hajal. “Two-Phase Flow Pattern Map for Evaporation in Horizontal Tubes: Latest Version”. In: *Heat Transfer Engineering* 24.6 (Nov. 2003), pp. 3–10. ISSN: 0145-7632. DOI: [10.1080/714044410](https://doi.org/10.1080/714044410). URL: <http://www.tandfonline.com/doi/abs/10.1080/714044410>.
- [29] S Z Rouhani and Axelsson E. “Calculation of void volume fraction in the subcooled and quality boiling regions”. In: 13 (1969), pp. 383–393. DOI: [10.1016/0017-9310\(70\)90114-6](https://doi.org/10.1016/0017-9310(70)90114-6).
- [30] Dag Biberg. “An explicit approximation for the wetted angle in two-Phase stratified pipe flow”. In: *The Canadian Journal of Chemical Engineering* 77.6 (Dec. 1999), pp. 1221–1224. ISSN: 00084034. DOI: [10.1002/cjce.5450770619](https://doi.org/10.1002/cjce.5450770619). URL: <http://doi.wiley.com/10.1002/cjce.5450770619>.
- [31] Jesús Moreno Quibén and John Richard Thome. “Experimental and analytical study of two-phase pressure drops during evaporation in horizontal tubes.” In: *Faculté des sciences et techniques de l’ingénieur STI* Ph.D. (2005), p. 159. DOI: [10.5075/epfl-thesis-3337](https://doi.org/10.5075/epfl-thesis-3337). URL: <http://library.epfl.ch/theses/?nr=3337>.
- [32] Jesús Moreno Quibén and John R Thome. “Flow pattern based two-phase frictional pressure drop model for horizontal tubes. Part I: Diabatic and adiabatic experimental study”. In: *International Journal of Heat and Fluid Flow* 28.5 (Oct. 2007), pp. 1049–1059. ISSN: 0142727X. DOI: [10.1016/j.ijheatfluidflow.2007.01.003](https://doi.org/10.1016/j.ijheatfluidflow.2007.01.003). URL: <https://linkinghub.elsevier.com/retrieve/pii/S0142727X07000057>.
- [33] Jesús Moreno Quibén and John R Thome. “Flow pattern based two-phase frictional pressure drop model for horizontal tubes, Part II: New phenomenological model”. In: *International Journal of Heat and Fluid Flow* 28.5 (Oct. 2007), pp. 1060–1072. ISSN: 0142727X. DOI: [10.1016/j.ijheatfluidflow.2007.01.004](https://doi.org/10.1016/j.ijheatfluidflow.2007.01.004). URL: <https://linkinghub.elsevier.com/retrieve/pii/S0142727X07000069>.
- [34] S. Kutateladze. “On the transition to film boiling under natural convection”. In: (1948), pp. 10–12.
- [35] N. Kattan, J. R. Thome, and D. Favrat. “Flow Boiling in Horizontal Tubes: Part 3—Development of a New Heat Transfer Model Based on Flow Pattern”. In: *Journal of Heat Transfer* 120.1 (Feb. 1998), p. 156. ISSN: 00221481. DOI: [10.1115/1.2830039](https://doi.org/10.1115/1.2830039). URL: <http://heattransfer.asmedigitalcollection.asme.org/article.aspx?articleid=1443089>.
- [36] ATLAS Collaboration. “ATLAS Inner Detector: Technical Design Report - Vol I”. In: *Design I*. April (1997), pp. 27–30. URL: <http://cdsweb.cern.ch/record/331063/files/cer-0254563.pdf>.
- [37] D Attree et al. “The evaporative cooling system for the ATLAS inner detector”. In: (2008). URL: <https://iopscience.iop.org/article/10.1088/1748-0221/3/07/P07003/meta>.
- [38] M. Battistin et al. “The Thermosiphon Cooling System of the ATLAS Experiment at the CERN Large Hadron Collider”. In: *International Journal of Chemical Reactor Engineering* 13.4 (2015), pp. 511–521. ISSN: 15426580. DOI: [10.1515/ijcre-2015-0022](https://doi.org/10.1515/ijcre-2015-0022).

- [39] V. Vacek et al. “Perfluorocarbons and their use in cooling systems for semiconductor particle detectors”. In: *Fluid Phase Equilibria* 174.1-2 (Sept. 2000), pp. 191–201. ISSN: 03783812. DOI: [10.1016/S0378-3812\(00\)00426-X](https://doi.org/10.1016/S0378-3812(00)00426-X). URL: <https://linkinghub.elsevier.com/retrieve/pii/S037838120000426X>.
- [40] Ian H. Bell et al. “Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp”. In: *Industrial Engineering Chemistry Research* 53 (2014), pp. 2498–2508. DOI: [10.1021/ie4033999](https://doi.org/10.1021/ie4033999). URL: <http://pubs.acs.org/doi/abs/10.1021/ie4033999>.
- [41] Bart Verlaet. “Dimensioning of CO₂ cooling pipes in detector structures”. In: *Forum on Tracking Detector Mechanics*. 2013. URL: <https://indico.cern.ch/event/233332/%20contributions/1546088/>.
- [42] Paolo Petagna, Bart Verlaet, and Andrea Francescon. “Two-Phase Thermal Management of Silicon Detectors for High Energy Physics”. In: May 2018, pp. 335–412. DOI: [10.1142/9789813229471{_}0005](https://doi.org/10.1142/9789813229471{_}0005). URL: https://www.worldscientific.com/doi/abs/10.1142/9789813229471_0005.
- [43] L. Friedel. “Improved friction pressure drop correlations for horizontal and vertical two-phase pipe flow”. In: *Meeting, European Two-Phase Flow Group, Ispra, Italy* 18.7 (1979), pp. 485–491. URL: https://jglobal.jst.go.jp/en/detail?JGLOBAL_ID=201002057597311499&rel=0.
- [44] Satish G. Kandlikar. “A General Correlation for Saturated Two-Phase Flow Boiling Heat Transfer Inside Horizontal and Vertical Tubes”. In: *Journal of Heat Transfer* 112.1 (1990), p. 219. ISSN: 00221481. DOI: [10.1115/1.2910348](https://doi.org/10.1115/1.2910348). URL: <http://heattransfer.asmedigitalcollection.asme.org/article.aspx?articleid=1440331>.
- [45] Yung-Su Tsai. “Pair production and bremsstrahlung of charged leptons”. In: *Reviews of Modern Physics* 46.4 (Oct. 1974), pp. 815–851. ISSN: 0034-6861. DOI: [10.1103/RevModPhys.46.815](https://doi.org/10.1103/RevModPhys.46.815). URL: <https://link.aps.org/doi/10.1103/RevModPhys.46.815>.
- [46] Mukund Gupta. *Calculation of radiation length in materials*. Tech. rep. Geneva: CERN, 2010. URL: <https://cds.cern.ch/record/1279627>.
- [47] Bart Verlaet, A Van Lysebetten, and M Van Beuzekom. “CO₂ cooling for HEP experiments”. In: *Topical Workshop on Electronics for ...* (2008), pp. 328–336. DOI: [10.5170/CERN-2008-008.328](https://doi.org/10.5170/CERN-2008-008.328). URL: <https://cds.cern.ch/record/1158652>.
- [48] Johannes van Es et al. “AMS02 Tracker Thermal Control Cooling System commissioning and operational results”. In: *43rd International Conference on Environmental Systems*. Reston, Virginia: American Institute of Aeronautics and Astronautics, July 2013. ISBN: 978-1-62410-215-8. DOI: [10.2514/6.2013-3389](https://doi.org/10.2514/6.2013-3389). URL: <http://arc.aiaa.org/doi/10.2514/6.2013-3389>.
- [49] B. Verlaet et al. “The ATLAS IBL CO₂ cooling system”. In: *Journal of Instrumentation* 12.02 (Feb. 2017), pp. C02064–C02064. ISSN: 1748-0221. DOI: [10.1088/1748-0221/12/02/C02064](https://doi.org/10.1088/1748-0221/12/02/C02064). URL: <http://stacks.iop.org/1748-0221/12/i=02/a=C02064?key=crossref.aaeb765387f104cfd3a49e61714c55c>.
- [50] J. Daguin et al. “Evaporative CO₂ cooling system for the upgrade of the CMS pixel detector at CERN”. In: *13th InterSociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems*. 188. IEEE, May 2012, pp. 723–731. ISBN: 978-1-4244-9532-0. DOI: [10.1109/ITHERM.2012.6231499](https://doi.org/10.1109/ITHERM.2012.6231499). URL: <http://ieeexplore.ieee.org/document/6231499/>.
- [51] Bart Verlaet and Joao Noite. *Design considerations of long length evaporative CO₂ cooling lines*. 2012, pp. 934–940. ISBN: 9781622762736. URL: <http://www.proceedings.com/15739.html>.

- [52] Z Zhang. “Optimisation de la Performance Thermique du Detecteur Pixel Alpine”. PhD thesis. 2015. URL: <https://cds.cern.ch/record/2142809/files/CERN-THESIS-2015-320.pdf>.
- [53] O. Kunz et al. “The GERG-2004 Wide-Range Equation of State for Natural Gases and Other Mixtures”. In: *VDU Verlag GmbH* (2007).
- [54] O. Kunz and W. Wagner. “The GERG-2008 Wide-Range Equation of State for Natural Gases and Other Mixtures: An Expansion of GERG-2004”. In: *Journal of Chemical & Engineering Data* 57.11 (Nov. 2012), pp. 3032–3091. ISSN: 0021-9568. DOI: [10.1021/jc300655b](https://doi.org/10.1021/jc300655b). URL: <http://pubs.acs.org/doi/10.1021/jc300655b>.
- [55] E. W. Lemmon and R. T Jacobsen. “A Generalized Model for the Thermodynamic Properties of Mixtures”. In: *International Journal of Thermophysics* 20.3 (1999), pp. 825–835. ISSN: 0195928X. DOI: [10.1023/A:1022627001338](https://doi.org/10.1023/A:1022627001338). URL: <http://link.springer.com/10.1023/A:1022627001338>.
- [56] Eric W. Lemmon et al. “Thermodynamic Properties of Air and Mixtures of Nitrogen, Argon, and Oxygen From 60 to 2000 K at Pressures to 2000 MPa”. In: *Journal of Physical and Chemical Reference Data* 29.3 (May 2000), pp. 331–385. ISSN: 0047-2689. DOI: [10.1063/1.1285884](https://doi.org/10.1063/1.1285884). URL: <http://aip.scitation.org/doi/10.1063/1.1285884>.
- [57] Eric W. Lemmon and Richard T Jacobsen. “Equations of State for Mixtures of R-32, R-125, R-134a, R-143a, and R-152a”. In: *Journal of Physical and Chemical Reference Data* 33.2 (June 2004), pp. 593–620. ISSN: 0047-2689. DOI: [10.1063/1.1649997](https://doi.org/10.1063/1.1649997). URL: <http://aip.scitation.org/doi/10.1063/1.1649997>.
- [58] A.P. Colijn and Bart Verlaat. “Evaporative CO2 Heat Transfer measurements for cooling systems of particle physics detectors”. In: *7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, Antalya, Turkey*. 2010. URL: <https://hdl.handle.net/11245/1.333237>.
- [59] Plotly Technologies Inc. *Collaborative data science*. Montreal, QC, 2015. URL: <https://plot.ly>.
- [60] Pierre Barroca. *ATLAS ITk CO2 cooling*. URL: <https://atlas-sfb.cern.ch/>.
- [61] Michael Moll. “Radiation Damage in Silicon Particle Detectors”. PhD thesis. University Hamburg, 1999. URL: <http://inspirehep.net/record/513308>.
- [62] Stephen L Shapiro et al. “Silicon pin diode array hybrids for charged particle detection”. In: *Nuclear Instruments and Methods in Physics Research Section A: Accelerators, Spectrometers, Detectors and Associated Equipment* 275.3 (Mar. 1989), pp. 580–586. ISSN: 01689002. DOI: [10.1016/0168-9002\(89\)90747-X](https://doi.org/10.1016/0168-9002(89)90747-X). URL: <https://linkinghub.elsevier.com/retrieve/pii/016890028990747X>.
- [63] Leonardo Rossi. “Pixel detectors hybridisation”. In: *Nuclear Instruments and Methods in Physics Research Section A: Accelerators, Spectrometers, Detectors and Associated Equipment* 501.1 (Mar. 2003), pp. 239–244. ISSN: 01689002. DOI: [10.1016/S0168-9002\(02\)02041-7](https://doi.org/10.1016/S0168-9002(02)02041-7). URL: <https://linkinghub.elsevier.com/retrieve/pii/S0168900202020417>.
- [64] RD53 Collaboration. *The RD53A Integrated Circuit*. Tech. rep. 2018. URL: https://cds.cern.ch/record/2287593/files/%20RD53A_Manual_V3-42.pdf.
- [65] Mattias Hammer. *Private communication*. Geneva.
- [66] Mattias Hammer. *Private communication*. Geneva. URL: https://indico.cern.ch/event/774777/attachments/1754396/2843832/FE_power_distributions.pdf.
- [67] Paul S Miyagawa and Ian Dawson. *Radiation background studies for the Phase II inner tracker upgrade*. Tech. rep. 2014. URL: <https://cds.cern.ch/record/1753332/files/ATL-UPGRADE-PUB-2014-003.pdf>.

- [68] Danilo Giugni. *Private communication*.
- [69] Graham Beck and Georg Viehhauser. “Analytic model of thermal runaway in silicon detectors”. In: *Nuclear Inst. and Methods in Physics Research, A* 618 (2010), pp. 131–138. DOI: [10.1016/j.nima.2010.02.264](https://doi.org/10.1016/j.nima.2010.02.264). URL: www.elsevier.com/locate/nima.
- [70] Michael (Simulia) Smith. *ABAQUS/Standard User’s Manual, Version 6.14*. 2009. URL: <https://www.sharcnet.ca/Software/Abaqus/6.14.2/v6.14/books/usi/default.htm>.
- [71] Henkel Technologies. *LOCTITE STYCAST 2850FT, Technical Datasheet*. 2014. URL: <http://tds.loctite.com/tds5/Studio/ShowPDF/STYCAST%202850FT-EN?pid=STYCAST%202850FT&format=MTR&subformat=HYS&language=EN&plant=WERCS>.
- [72] Berend Munneke. *Nikhef CO2 cooling blow system - ATU-SYS-ER-0001*. 2008. URL: https://espace.cern.ch/CO2/Public%20Documents/Presentations%20and%20Publications/NIKHEF/080806_CO2coolingBlowsystem.pdf.
- [73] Glen Cowan et al. “Asymptotic formulae for likelihood-based tests of new physics”. In: *The European Physical Journal C* 71.2 (July 2010), pp. 1–19. ISSN: 1434-6044. DOI: [10.1140/epjc/s10052-011-1554-0](https://doi.org/10.1140/epjc/s10052-011-1554-0). URL: <http://arxiv.org/abs/1007.1727>.
- [74] Daniel Foreman-Mackey et al. “emcee : The MCMC Hammer”. In: *Publications of the Astronomical Society of the Pacific* 125.925 (Mar. 2013), pp. 306–312. ISSN: 00046280. DOI: [10.1086/670067](https://doi.org/10.1086/670067). URL: <http://iopscience.iop.org/article/10.1086/670067>.
- [75] Benjamin J. Jones, John P. McHale, and Suresh V. Garimella. “The Influence of Surface Roughness on Nucleate Pool Boiling Heat Transfer”. In: *Journal of Heat Transfer* 131.12 (2009), p. 121009. ISSN: 00221481. DOI: [10.1115/1.3220144](https://doi.org/10.1115/1.3220144). URL: <http://heattransfer.asmedigitalcollection.asme.org/article.aspx?articleid=1449769>.
- [76] Sudev Das and Swapan Bhaumik. “Enhancement of Nucleate Pool Boiling Heat Transfer on Titanium Oxide Thin Film Surface”. In: *Arabian Journal for Science and Engineering* 39.10 (Oct. 2014), pp. 7385–7395. ISSN: 1319-8025. DOI: [10.1007/s13369-014-1340-z](https://doi.org/10.1007/s13369-014-1340-z). URL: <http://link.springer.com/10.1007/s13369-014-1340-z>.
- [77] Krzysztof Sliwa. *Private communication*.