CO₂ evaporative cooling for the ATLAS detector

— Afroditi Koutoulaki —

CO₂ EVAPORATIVE COOLING FOR THE ATLAS DETECTOR

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Contents

Nomenclature	1
Acronyms	9
Introduction	11
1 The Large Hadron Collider	15
1.1 CERN and Large Hadron Collider	15
1.2 ATLAS experiment	15
1.3 Inner Detector	15
1.4 Large Hadron Collider Upgrade	16
1.5 ATLAS Inner Detector upgrade	17
1.5.1 Pixel detector	17
1.5.2 Strip detector	17
1.5.2.1 Barrel and Stave	18
1.5.2.2 Endcap and Petal	18
1.6 Cooling	18
1.6.1 CO ₂ evaporative cooling	19
1.6.2 ATLAS Inner Tracker	19
1.6.3 Future Pixel Trackers	20
2 Cooling theory	21
2.1 Heat Transfer	21
2.1.1 Conductive heat transfer	21
2.1.2 Convective heat transfer	22
2.1.3 Radiation heat transfer	23
2.2 Single-phase flow	24
2.2.1 Heat exchange	24
2.2.2 Pressure drop	24
2.2.3 Pressure losses through enlargements, contractions and bends	25
2.2.4 Pressure losses across pillar structures	26
2.2.5 Heat Transfer Coefficient	26
2.3 Two-phase flow	27
2.3.1 Flow regimes in mini-channels	28
2.3.2 Heat transfer mechanisms	29

2.4	Evaporation models in mini-channels	30
2.4.1	Pressure losses through enlargements, contractions and bends	30
2.5	Evaporation models in Micro-channels	31
2.5.1	Dimensionless numbers in micro-channels models	31
2.5.2	Flow regimes in micro-channels	32
2.5.3	Pressure drop	33
2.5.4	Heat transfer coefficient	36
2.6	Two-phase cooling methodology in HEP	37
3	Petal Description	39
3.1	Sensor	39
3.2	Front-end electronics	40
3.3	Powering scheme	40
3.4	High voltage supply	41
3.5	Mechanical structure	41
3.6	Radiation damage and heating sources	42
3.6.1	Silicon sensor	42
3.6.2	ASICs	44
3.7	UTOPIA layout design	44
3.7.1	Geometry	45
3.7.2	Silicon leakage power	45
3.7.3	ASICs power	46
3.8	Conclusion	47
4	Petal Prototype Measurements	49
4.1	Experimental Apparatus	49
4.1.1	Mechanical structure	49
4.1.2	Silicon wafers	51
4.1.3	Heaters	52
4.1.4	Power supply for heaters	53
4.1.5	Temperature sensors	53
4.1.6	CO ₂ plant	54
4.1.7	Insulation box	55
4.1.8	Cabling	56
4.1.9	Data acquisition and Control system	56
4.2	Measurements	57
4.2.1	Study cases	58
4.2.2	Cooling system parameters	58
4.2.3	Example of measuring heat exchange with surroundings	59
4.2.4	Example of a typical measurement: nominal power (case a)	60
4.3	Estimating heat leak from measurements	63
4.4	Discussion of results for the study cases	64

5	Thermal model of the prototype petal	67
5.1	CoBra CO ₂ calculations	67
5.1.1	Nominal power (case <i>a</i>)	68
5.1.2	Nominal power during commissioning (case <i>d</i>)	70
5.2	FEA Thermal model	72
5.2.1	Geometry	72
5.2.2	Material properties	75
5.2.3	Mesh	76
5.2.4	Boundary conditions	76
5.2.5	Heat transfer coefficient between components	77
5.2.6	Step module	78
5.2.7	Output	78
5.3	Comparison FEA prediction with measurements	79
5.3.1	Heat exchange with the surroundings	79
5.3.2	Nominal power (case <i>a</i>)	81
5.3.3	Above nominal power (case <i>b</i>)	84
5.3.4	Nominal power without EoP (case <i>c</i>)	85
5.3.5	Nominal power during commissioning (case <i>d</i>)	87
5.4	Discussion	89
6	Prediction model for latest design	91
6.1	The ITk layout design	91
6.1.1	Geometry	92
6.1.2	Sensors	93
6.1.3	ASICs	94
6.2	Thermal modeling	95
6.2.1	Geometry	97
6.2.2	ASICs Power	98
6.2.3	Sensor power	99
6.2.4	DC/DC Converter Power	100
6.2.5	EoP board	101
6.2.6	Materials properties	101
6.3	CoBra CO ₂ calculations	104
6.3.1	TID case	104
6.3.2	End-of-life case	106
6.4	Running the FEA	108
6.4.1	Modeling of the sensors	108
6.4.2	Convective and radiative heat transfer	109
6.4.3	Boundary conditions	109
6.4.4	Thermal runaway methodology	109
6.5	FEA results	110
6.5.1	Temperature predictions	110
6.5.2	Thermal runaway	112
6.6	Optimization studies	113
6.6.1	DC/DC converters issue	113

6.6.2	Cooling tube diameter	115
6.6.3	Glue	118
6.6.4	Facesheet material	120
6.6.5	Addition of carbon foam “ribs”	120
6.6.6	Change of tube configuration	122
6.6.7	Modeling CoBra with non-uniform heat flux	123
6.7	Conclusion	124
7	Single micro-channel studies	127
7.1	Blow system	127
7.2	Modification of blow system	128
7.2.1	By-pass and experiment	129
7.2.2	Mass flow	129
7.2.3	Pressure	130
7.2.4	Vacuum insulation	131
7.3	Test section	131
7.3.1	Heating	132
7.3.2	Temperature sensors	133
7.3.3	Flow visualization	135
7.4	Data acquisition and Control system	135
7.5	Accuracy	136
7.6	Check of the temperature sensors	137
7.7	Analytical estimation of heat leak	138
7.8	Measurements	139
7.9	Discussion	143
8	Single micro-channel results	145
8.1	Temperature calibration	145
8.2	Measurement of heat leak	146
8.3	Single-phase	147
8.3.1	Pressure drop	147
8.3.2	Heat transfer coefficient	149
8.4	Two-phase	151
8.4.1	Flow Regimes	152
8.4.2	Pressure drop	157
8.4.3	Heat transfer coefficient	160
8.5	Empirical Parameters	167
8.6	Conclusions	169
9	Studies on silicon-glass multi-channels	171
9.1	Silicon-glass multi-channel device	171
9.2	Set-up	174
9.3	Test section	175
9.4	Experimental results	177
9.4.1	Measurements	177

9.4.2	Mass flow fluctuations	181
9.4.3	Flow distribution	182
9.4.4	Flow Regime	185
9.4.5	Pressure drop	190
9.4.6	Heat transfer coefficient	193
9.5	Conclusions	198
Outlook		201
A Heat Analysis I		203
A.0.1	Non-uniform heating	203
A.0.1.1	Solutions for the ODE	206
A.0.1.2	Application of boundary conditions	207
B Heat Analysis II		211
B.0.1	Uniform heating	211
C Error analysis		217
C.1	Error analysis	217
D Petal Information		219
D.1	Heater Information	219
D.2	Petal model	220
E Radiation damage at TID bump		223
F Silicon electrical conductivity		225
Bibliography		227
Summary		233
Samenvatting		237
ΠΕΡΙΛΗΨΗ		241
Acknowledgements		245

Nomenclature

Greek letters

α	damage coefficient	
ΔT	Temperature difference	$[K]$
Δh_{vap}	Latent heat of evaporation	$[J/kg]$
δ_{min}	Minimum liquid film thickness	$[\mu m]$
ϵ	absolute roughness	$[\mu m]$
ϵ_{emit}	Emissivity coefficient	
ϵ_{void}	Void fraction	
λ	Thermal conductivity of the fluid	$[W/mK]$
μ	Dynamic viscosity	$[Pa]$
Φ_{eq}	Particle fluence	$[MeV\ neq]$
ρ	Density	$[kg/m^3]$
σ	Surface tension	$[N/m]$
σ_e	electrical conductivity	$[Ohm^{-1}m^{-1}]$
$\sigma_{St-Boltz}$	Stefan-Boltzmann constant	$[W/m^2K^4]$
τ	Pair period	$[s]$
θ	Angle	$[^\circ]$

Latin letters

A	Area	$[m^2]$
a_q	Empirical parameter in EB regime	

Nomenclature

An	annular regime	
Aw	Empirical parameter in Alashkar equation	
BO	Boiling number	
$Bond$	Bond number	
C	Empirical parameter in air convection	
C_2	Empirical parameter in two phase pressure drop	
c_p	Specific heat of the fluid	[J/kg K]
C_{cooper}	Empirical parameter in isolated bubbly heat transfer coefficient	
C_{LM}	Empirical parameter in Lockhart-Martinelli equations	
cdo	Empirical parameter in EB regime	
Co	Confinement number	
D	Inner diameter	[m]
D_h	Hydraulic diameter	[m]
D_{th}	Diameter used for the critical heat flux	[m]
dp	Pressure drop	[bar]
e	electron energy	[eV]
E_g	Effective band-gap energy	[eV]
E_p	Relative roughness	
ELF	entrained liquid fraction	
f_d	Darcy–Weisbach friction factor	
f_L	Liquid Fanning friction factor	
f_V	Vapor Fanning friction factor	
f_{tp}	Two phase Fanning friction factor	
Fr	Froude number	
G	Mass flux	[kg/m ² s]

g	Gravitational constant	$\left[\frac{m^3}{kg s^2}\right]$
Gr	Grashof number	
h	Heat transfer coefficient	$[W/m^2 K]$
h_a	Empirical parameter in Annular regime	
I	Leakage current density	$[A/cm^3]$
I_o	Leakage current density at zero fluence	$[A/cm^3]$
J	Superficial velocity	$[m/s]$
j	areal current density	$[A/cm^2]$
k	Thermal conductivity	$[W/mK]$
k_{bend}	Bend coefficient	
k_{con}	Contraction coefficient	
k_{exp}	Expansion coefficient	
L	Length	$[m]$
L_{cap}	Capillary length	$[m]$
L_{ch}	Characteristic length	$[m]$
L_{ev}	Evaporator length	$[m]$
M	Molar mass	$[kg/mol]$
m	Mass	$[kg]$
M_{par}	Lockhart-Martinelli parameter	
M_{ult}	Liquid multiplier	
n	Empirical parameter in air convection	
n_a	Empirical parameter in Annular regime	
Nu	Nusselt number	
P	Perimeter	$[m]$
Pr	Prandtl number	

Nomenclature

Pre	Reduced pressure	
Q	Heat transfer	$[J]$
Q_c	heating areal power density	$[W/m^2]$
R	Resistance	$[K/W]$
R_b	Radius bend	$[^\circ]$
R_T	Correction factor for leakage current	
Re	Reynolds number	
T	Temperature	$[K]$
t	average liquid film thickness	$[m]$
t^+	dimensionless liquid film thickness	
U	Flow velocity	$[m/s]$
V	Voltage	$[V]$
W	Wavy regime	
w	Width	$[m]$
w_w	Channel wall thickness	$[m]$
We	Weber number	
x	Vapor quality	

Special letters

$\%XO$	radiation length	$[g/cm^2]$
$\dot{m}$	Mass flow	$[kg/s]$
$\dot{H}$	Power	$[J/s]$
$\dot{h}$	Enthalpy	$[J/kg]$
$\dot{Q}$	Rate of heat transfer	$[W]$
h	Heat transfer coefficient	$[W/m^2K]$
$\dot{q}$	Rate of heat transfer per unit area	$[W/m^2]$

subscripts

$2ph$ two-phase

ah after heater

An annular regime

ave average

$B - An$ bubbly-Annular regime

$bend$ bend

c core

ch characteristic

$CO2_{in}$ Fluid temperature measured at the inlet

$CO2_{out}$ Fluid temperature measured at the exit

con contraction

$cond$ conductive

$conv$ convective

$D - M$ dryout-Mist transition

dry dryout regime

EB Elongated bubbles regime

eff effective value

exp expansion

$film$ film

$frict - inlet$ frictional contribution at the inlet section

$frict - outlet$ frictional contribution at the exit section

H homogeneous

$I - An$ intermediate-Annular regime

IB Isolated bubbles regime

Nomenclature

<i>in</i>	at inlet
<i>in</i>	at the inlet
<i>inevap</i>	inlet of evaporator
<i>L</i>	liquid
<i>l</i>	laminar regime
<i>LF</i>	liquid film
<i>M</i>	mist regime
<i>meas</i>	measured value
<i>out</i>	at the outlet
<i>outevap</i>	exit of evaporator
<i>outmicro</i>	exit of micro-channel
<i>p</i>	pair of liquid slug and vapor
<i>pre</i>	preheater
<i>pred</i>	predicted value
<i>rad</i>	radiative
<i>ref</i>	reference
<i>satexitp</i>	saturated value derived from the exit pressure
<i>satp</i>	saturated value derived from the inlet pressure
<i>si – m</i>	derived from measurement value at the silicon device
<i>si – p</i>	predicted value at the silicon device
<i>t</i>	transitional/turbulent regime
<i>tc</i>	thermocouple
<i>tot</i>	total value
<i>V</i>	vapor
<i>Void</i>	void fraction

wall – inner at the inner wall of the tube

wall – out at the outer wall of the tube

wet   wetted

Acronyms

2-PACL	2-phase accumulator controlled loop
ALICE	A large ion collider experiment
AMS	Alpha magnetic spectrometer
ATLAS	A toroidal LHC apparatus
ASIC	Application-specific integrated circuit
BLS	Best fit straight line
CAD	Computer aided design
CERN	Conseil Européen pour la Recherche Nucléaire
CFHX	Counter flow heat exchanger
CFD	Computational fluid dynamics
CHF	Critical heat flux
CMOS	Complementary metal-oxide-semiconductor
CMS	Compact muon solenoid
CO ₂	Carbon dioxide
CoBra	CO ₂ branch calculator
DAQ	Data acquisition
DC	Direct current
DC3D4	4-node linear heat transfer tetrahedron
DC3D8	8-node linear heat transfer brick
EB	Elongated bubbles
EC	Endcap
EoP	End of petal
EPFL	École polytechnique fédérale de Lausanne
FEA	Finite element analysis
FLUKA	FLUktuierende Kaskade
FS	Full scale
HCC	Hybrid controller circuit
HEP	High energy physics
HL-LHC	High luminosity LHC
HTC	heat transfer coefficient

HV	High voltage
HX	Heat exchanger
IB	Isolated bubbles
IBL	Insertable b-layer
IBM	International business machines
ID	Inner detector
ISS	International space station
ITk	Inner Tracker
LHC	Large hadron collider
LHC-b	LHC-beauty
LOI	Letter of intent
LMpar	Lockhart-Martinelli parameter
LV	Low voltage
MS	Medium strip
NIKHEF	Nationaal Instituut voor Subatomaire Fysica
NIST	National institute of standards and technology
OD	Outer diameter
PCB	Printed circuit board
PID	Proportional integrated derivative
PXD	Pixel Vertex detector
PT	Platinum (resistance thermometer)
REFPROP	Reference fluid properties
RSS	Root of sum of squares
R&D	Research and development
SCADA	Supervisory control and data acquisition
SCT	Semiconductor tracker
SS	Short strip
SS (material)	stainless steel
SVD	Singular Value Decomposition
TC	thermocouple
TID	Total ionizing dose
TRT	Transition radiation tracker
TTCS	Tracker thermal control system
USB	Universal serial bus
UV	Ultra violet
VELO	Vertex locator
WinCC-OA	SIMATIC WinCC Open architecture

Introduction

Mankind has been questioning the nature of matter since the ancient times. Up until now a few of the fundamental questions have been answered but there are many more on the list. Technological advances have helped scientists to build experimental setups, throughout the years, in order to prove and validate their theories concerning the nature of matter.

CERN and the Large Hadron Collider (LHC) is the highest energy particle collider built to date to study particle physics. The ATLAS detector is one of the two general purpose experiments around the LHC. The ATLAS experiment focuses on studies to understand the fundamental constituents of matter [1]. The ATLAS detector requires technological advances in engineering in order to meet its goal. Cooling of the detector is one of the main engineering challenges. This thesis is concerned regarding evaporative CO₂ cooling, as an advanced cooling method to enable ATLAS detector to meets its purpose.

At the outset of this thesis work, the decision was made to adopt evaporative CO₂ cooling for the petal support module of the ATLAS detector. Remaining tasks were to optimize the cooling design and prove that the design cools well. Nikhef had available a prototype petal and a CO₂ cooler for this investigation. The petal had to be equipped with heaters and dummy modules. A thermal finite element model of the petal had to be built and verified against measurements of the prototype. The aim was to use this model to optimize the petal design, and to reliably demonstrate that the design would perform safely at the high luminosity LHC upgrade. The optimized parameters are the fractional radiation length of the petal, minimizing hot spots such as electrical chips, and maximizing the thermal safety margins, such as the distance from the thermal runaway.

To go beyond the current standard practice in thermal engineering, a second goal was to evaluate the performance of CO₂ cooling in micro-channels. The models for the mini-channel (diameter > 1 mm) cooling required adaptation for this goal. An extensive list of empirical models exist for the mini-channel (diameter > 1 mm) CO₂ cooling [1], [2]. Keniar et al in ref. [3] studied experimentally CO₂ flow boiling in a single channel of 1.55 mm hydraulic diameter (considered as mini-channel than micro-channel) and Mastrullo et al in ref. [4] investigated the influencing factors of collected data of flow boiling of CO₂ in smooth and enhanced tubes in a range 0.8 mm up to 8.92 mm. The smallest

channel (0.5 mm, which could be closer to micro-channel size) investigated with CO₂ flow boiling was presented by Ducoulombier et al in refs.[5], [6]. Ducoulombier et al studied the heat transfer of CO₂ boiling flow at four saturation temperatures ranging from -10°C up to 0°C and mass fluxes ranging 200 to 1200 kg/m²s. Hellenschmidt et al studied the effects of saturation temperature on the boiling properties of CO₂ in small diameter pipes of 2.15 mm, 1 mm and 0.5 mm in ref. [7]. Prediction models exist for micro-channels, as described by Dupont et al in refs. [8] and [9], and by Cioncolini et al in ref. [10] but the validated database include only studies on temperatures > -10°C and evaporation in channels greater than 0.5 mm diameter. Significant effort has taken place over the last years to fill in the gap in this field of research (see AIDA-2020 project [11]). The goal of the research presented here is to extend the range of application of micro-channel evaporation modeling to below 0.30 mm diameter and to temperatures down to -20°C, which is a typical diameter for cooling channels and temperature of interest for the particle physics community. Experimental validation of existing models is a major goal of this research.

The thesis focuses on the evaporative CO₂ cooling in the ATLAS detector. In chapter 1 the ATLAS experiment is described. The main subsystems of the ATLAS detector and the motivation for the upgrade are presented. Chapter 2 presents the theoretical-empirical models for mini-channels and micro-channels that are used in this thesis. Chapter 3 describes the petal detector. Real scale prototypes are useful during the *R&D* phase. The detectors in the strip endcap tracker have never been tested with evaporative CO₂. A prototype was built and its cooling performance with CO₂ tested. The tests were performed under various conditions. Chapter 4 covers the measurement of the cooling performance of the prototype petal with CO₂ evaporation. The prototype is designed based on the UTOPIA layout, which is the original layout design. Chapter 5 describes the Finite Element Analysis (FEA) thermal model of the prototype based on the UTOPIA layout. FEA in combination with experimental results can be accurate enough to predict the mechanical and thermal behavior of the structures. A FEA model is favorable for the advance of complicated structure designs mainly because it saves a lot of effort, time and money compared to prototyping each design phase. Further studies and optimization plans are described in chapter 6 based on the FEA model. The thermal model is adapted based on the most recent layout design (which is different compared to the UTOPIA layout).

Evaporative CO₂ cooling in micro-channels is foreseen for high power density detectors. For this reason, experiments on a single straight micro-channel are conducted in order to validate the existing empirical models. Chapter 7 presents the experimental facilities and the design parameters. The results are discussed in chapter 8. Finally, a silicon-glass multi-micro-channel heat exchanger is chosen to be tested as a more realistic example of a detector cool-

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ing structure. Chapter 9 provides the experimental results of the silicon-glass multi-micro-channel device.

Chapter 1

The Large Hadron Collider

1.1 CERN and Large Hadron Collider

The Large Hadron Collider (LHC) is the highest energy particle collider built to date. It is a two ring superconducting circular collider. It accelerates protons at its 27 km circumference and provides proton-proton collisions at four main points. The LHC hosts four major experiments around those collision points — two general purposes experiments, ATLAS and CMS and two specialized experiments, ALICE and LHC-b.

1.2 ATLAS experiment

The ATLAS detector is one of the two general purpose experiments around LHC. The ATLAS experiment focuses on studies to understand the fundamental constituents of matter [12]. The ATLAS detector has a forward-backward symmetry covering almost a 4π angle around the interaction point. It consists of several sub-detectors installed in layers around the interaction point. Starting from the interaction point and moving towards the outer components, ATLAS consists of the inner detector, calorimeters and muon detector.

1.3 Inner Detector

The inner detector (ID) shown in figure (1.1) is the sub-detector closest to the collision point. It consists of three independent systems, the pixel tracker, the semiconductor tracker (SCT) and the transition radiation tracker (TRT). The purpose of the ID is to provide tracking information about charged particles. The pixel tracker is arranged in a central barrel region of 3 layers and 2 endcap regions with 3 disks each. The semiconductor tracker (SCT) is a strip detector

arranged in 4 barrels and 9 disks. The transition radiation tracker (TRT) consists of straw detectors interspersed with material that gives off transition radiation when traveled by electrons. It uses xenon gas to detect the transition radiation photons.

Figure 1.1: Inner detector of ATLAS experiment.

1.4 Large Hadron Collider Upgrade

The LHC has worked successfully since 2008, delivering a luminosity of 150 fb^{-1} for the ATLAS experiment and leading to outstanding discoveries such as the Higgs boson.

To go beyond these results, the LHC will be upgraded in phases over the coming years to deliver an even higher luminosity. During the phase-II High Luminosity LHC (HL-LHC) upgrade scheduled for around 2025, the LHC will deliver an instantaneous luminosity that is seven times higher compared to the current one ($7 \times 10^{34} \text{ cm}^{-2} \text{ s}^{-1}$ compared to $1 \times 10^{34} \text{ cm}^{-2} \text{ s}^{-1}$).

This translates to an integrated luminosity of 3000 fb^{-1} over 10 years of operation for the ATLAS experiment. Radiation damage will be 10 times larger than in the current runs. Therefore, along with the LHC, the sub-detectors of the ATLAS experiment need to be upgraded.

1.5 ATLAS Inner Detector upgrade

The ID will be replaced with a new tracker called inner tracker, ITk. The ITk will be an all silicon based detector. The current ID will be completely removed. The design for the new ITk aims for a life expectancy of 15 years. It includes a new sensor layout; all silicon based new electronics components to cope with the high hardware trigger rate of up to 500 kHz at L0; and new cooling and support structures able to withstand an integrated luminosity of 3000 fb^{-1} [13].

The first upgrade layout design is known as the UTOPIA layout [14]. Over several years, changes on the design, mainly on the number of barrel layers and endcap disks led to the LOI layout design [13]. The latest to date upgrade layout is known as the ITk layout design [15]. Based on the ITk layout design, the ITk consists of silicon pixel detectors close to the collision point surrounded by silicon strip detectors. Figure (1.2) shows the inner tracker. The pixel and strip detector are indicated.

Figure 1.2: Inner tracker of ATLAS experiment after HL-LHC upgrade phase II.

1.5.1 Pixel detector

The pixel detector is arranged in 5 barrel layers with endcap ring structures. In the intermediate region, modules are tilted to face the interaction point, reducing the number of modules needed to cover all tracks.

1.5.2 Strip detector

The strip detector is also arranged in barrels and endcaps. The basic detector unit is a module. The module consists of the sensor, the ASICs, and their

power boards. Multiple modules are placed on a local support.

1.5.2.1 Barrel and Stave

The Layout has a central barrel with 4 layers of detectors. The innermost layers have short- strips (24.1 mm), whereas the outermost ones have long strips (48.2 mm). The stave is the modular component of the barrel section.

1.5.2.2 Endcap and Petal

The ITk layout strip layout has two identical endcaps (EC) rotated 180 degrees with respect to each other. Each EC consists of 6 disks, and each disk has 32 identical petals.

The petal is the modular component of the EC. Modules are mounted on both sides of a wedge-shaped mechanical support structure, which reaches from inner to outer radius of the endcap. The layout document [15] provides a description of the sensors and the arrangement on each disk of the EC. The arrangement refers to the position of the sensors with respect to the beam line. Furthermore, it gives the dimensions and the number of short and long strips for each sensor. In total, there are nine sensors on each side of the support, located in six different rings. The three outermost rings have two modules each, side by side. The sensors have reverse-biased diodes and produce a signal when a charged particle passes through. Read-out electronics collect this signal.

The readout electronics are mounted on a multi-layer flexible printed circuit board (PCB) known as a hybrid. The hybrid is mounted directly on the silicon sensor. The mechanical support provides the necessary stiffness to keep the sensors reliably stable during the runs. The support hosts all the electronics services necessary to run and control the sensors.

1.6 Cooling

Cooling of the detectors is vital for particle physics experiments. Detectors receive significant radiation. Radiation damages the sensors, inducing high leakage currents. Heating and electronics noise due to radiation-induced leakage-current are minimized by running the sensors cold (about -20°C). The front-end electronics dissipate a considerable amount of heat, and this has to be efficiently removed by the cooling system, both to protect the front-end electronics and to avoid heating the sensors.

An important requirement for the cooling is to minimize the material in the detector. The services on and around the detectors contribute to multiple scattering, electron-positron pair production from photons, bremsstrahlung of

electrons, and hadronic interactions. All these decrease the tracking performance. The radiation length is the mean length to reduce the energy of an electron by $1/e$. It is crucial to minimize the fraction of a radiation length (usually expressed as %XO) presented by the material in the ITk. This is achieved by using materials with long radiation length (in g/cm²), low density, and minimizing how much of material is present. Stable temperatures contribute to positional stability.

1.6.1 CO₂ evaporative cooling

The current ID is cooled by evaporation of octafluoropropane (C_3F_8) [16]. Fluorocarbons are non-flammable, non-conductive and radiation-resistant, making them favorable for HEP experiments. However, the use of C_3F_8 in the ITk would require large diameter tubes to remove the expected heating power, and would not reach low enough temperatures, especially for the pixel detectors. Instead, the ITk will have CO₂ evaporative cooling to reach the required temperatures. CO₂ evaporative cooling has been used in LHC-b for its Vertex Locator (VELO) detector, and it has been successfully operated since 2010 [12]. More information on the CO₂ cooling system for VELO can be found in ref. [17]. Another experiment that uses CO₂ for cooling is the Tracker Thermal Control System (TTCS) in the Alpha Magnetic Spectrometer (AMS) of the International Space Station (ISS) which is as well running successfully. Moreover, CO₂ with 2-PACL [17] technique has been adopted by several other experiments, among them the CMS pixel, the Belle2 PXD and SVD and the Insertable B Layer (IBL).

CO₂ is chemically stable, non-flammable, radiation hard. CO₂ has a high latent heat of evaporation (313 J/kg at -30 °C) so requires less mass flow to remove a given heat power. The triple point of CO₂ is at -55 °C. Evaporation above -40 °C (safely away from triple point) takes place at relatively high pressure (10 bar). The benefits of employing CO₂ are the usage of much smaller diameter tubes, the reduction of the fractional radiation length, and the general facilitation of the assembly. Furthermore, evaporation at high pressures gives small ratio of the pressure drop along the line to the pressure of the CO₂, and therefore has almost isothermal behavior. Finally, the evaporation of high pressure CO₂ has the advantage of higher heat transfer coefficient, because high-density gas bubbles carry more heat.

1.6.2 ATLAS Inner Tracker

Both the pixels and strips of the ITk will assemble many detector modules onto mechanical supports with embedded cooling. The cooling tubes have an

inner diameter of around 2 mm, and are categorized as mini-channels (tubes with inner diameter in the range of 1-10 mm are considered mini-channels). The heat transfer coefficient between the CO₂ and the inner wall can be up to 20.000 W/m²K as shown in ref. [18].

1.6.3 Future Pixel Trackers

HEP experiments of future colliders and possibly the inner layers of the ATLAS pixel tracker will require replacement and even better cooling. These layers are located close to the collision point and suffer very high radiation damage. Reduced mass at these areas is even more important for good vertex tracking performances. Channels with an inner diameter of less than 1 mm down to 0.1 mm, known as micro-channels, are favorable. Micro-channels are etched directly in the silicon substrate of the front-end readout chip, contributing very little to radiation length compared to conventional cooling tubes. They cool the readout chips directly, yielding very low temperature differences between the heating source and the coolant.

Chapter 2

Cooling theory

This chapter presents the formulas and models used during this study. The basic formulas of heat transfer are summarized at the beginning. Then, the theory for single-phase flow is presented, that was used for the validation of the experimental set-up with single-phase data as described later in chapter 7. The empirical models for two-phase flow are introduced including a description of the different flow patterns that have been observed in the past. A reference to the evaporation models for the mini-channels is given. The CoBra model, that is used to calculate the petal cooling parameters, uses these models. And finally, a detailed description is given of the equations for the heat transfer and flow characteristics for the different flow patterns observed in micro-channels. These models are used in chapters 8 and 9 to estimate the flow properties in the single micro-channel and silicon-glass multi-channels studies.

2.1 Heat Transfer

2.1.1 Conductive heat transfer

Conduction is the mode of heat transfer in which the energy exchange happens from a high temperature to a low temperature region due to the molecular interactions and due to the drift of electrons. The rate of heat conducted ($\dot{Q}_{cond}$ in W) within a body is estimated from the Fourier law of heat conduction:

$$\dot{Q}_{cond} = -\frac{k}{L}A\Delta T \quad (2.1)$$

where A is the area in m² that is constant over length L in m, ΔT the temperature difference in K between the two regions, k is the thermal conductivity of the material in W/mK. The conductive thermal resistance is the ratio of the temperature difference over the heat transferred. The conductive thermal

resistance (R_{cond}) is then:

$$R_{cond} = \frac{L}{kA} \quad (2.2)$$

For multi-layer structures, the effective thermal conductivity k_{eff} is estimated based on the total thermal resistance.

$$k_{eff} = \frac{L_{tot}}{R_{tot}A_{tot}} \quad (2.3)$$

where A_{tot} is the area of the structure, R_{tot} is the total thermal resistance, and L_{tot} is the total thickness of the structure. The above equation is used to find the components (k_x, k_y, k_z) for the multi-layer structures described in section 6.2.6 in chapter 6. The components (k_x, k_y, k_z) are derived from the conductivities k_i of the layer for a constant area as:

$$k_x = k_y = k_z = \frac{\sum t_i}{\sum \frac{t_i}{k_i}} \quad (2.4)$$

where the conductivity k_x is in the through thickness direction, k_y is at the in-plane direction, k_z is at the transverse direction, and t_i is the thickness of the i^{th} layer.

2.1.2 Convective heat transfer

Convection is the mode of heat transfer that takes place due to diffusion (random molecular motion) and advection (macroscopic motion of the fluid). The rate of heat convected ($\dot{Q}_{conv}$ in W) between a fluid and a solid surface is estimated from Newton's cooling law.

$$\dot{Q}_{conv} = hA\Delta T \quad (2.5)$$

where A is the area in m^2 , ΔT the temperature difference in K between the coolant and the surface, and h the heat transfer coefficient in W/m^2K .

The resulting convective thermal resistance (R_{conv}) is then:

$$R_{conv} = \frac{1}{hA} \quad (2.6)$$

Empirical relations for free air convection can be found for different geometries in ref. [19]. In chapter 5, the heat transfer that takes place around the petal during the measurements is estimated using these empirical equations. The estimation of the average heat transfer coefficient is calculated as:

$$h = C \left(\frac{\Delta T}{L_{ch}} \right)^n \quad (2.7)$$

where C and n are empirical values, ΔT is the temperature difference between T_{wall} and T_{fluid} in K, and L_{ch} is the characteristic dimension in m usually expressed as:

$$L_{ch} = \frac{A}{P} \quad (2.8)$$

where A is the heat exchange contact area (in this case the outer surface areas of the petal) in m^2 and P is the perimeter in m. As an illustration, table (2.1) provides the empirical values for the C and n in eq. (2.7).

Table 2.1: Empirical values, according to Holman [19], of C and n for the calculation of the heat transfer coefficient during free convection from a surface towards air at atmospheric pressure for $10^4 < Gr \ Pr < 10^9$.

Surface	C	n
Vertical plate or cylinder	1.42	1/4
Horizontal plate or cylinder	1.32	1/4
Horizontal plate isothermal heated from below	1.32	1/4
Horizontal plate isothermal heated from above	0.59	1/4

2.1.3 Radiation heat transfer

Thermal radiation refers to the heat energy emitted by a body due to its temperature. All bodies at a temperature above absolute temperature zero emit energy by the process of electromagnetic radiation. The rate of heat radiated ($\dot{Q}_{rad}$ in W) from a body is estimated from Stefan-Boltzmann Law as:

$$\dot{Q}_{rad} = \epsilon_{emit} \sigma_{St-Boltz} A T^4 \quad (2.9)$$

where ϵ_{emit} is the emissivity coefficient of the object, A is area of the emitted body in m^2 , T is the temperature of the body in K, and $\sigma_{St-Boltz}$ is the Stefan-Boltzmann constant in $W/m^2 K^4$.

2.2 Single-phase flow

The equations described in this section were used to estimate heat and flow parameters for single-phase flows in the petal cooling described in chapter 5, and to validate the experimental set-up described in chapter 8.

2.2.1 Heat exchange

In single-phase flow, the power that is needed to increase the temperature of the fluid by ΔT is found with eq. (2.10).

$$\dot{Q} = c_p \dot{m} \Delta T \quad (2.10)$$

where $\dot{m}$ is the mass flow in kg/s, ΔT is the temperature difference of the fluid between the inlet and the exit, and c_p is the specific heat of the fluid in J/kg K.

2.2.2 Pressure drop

The total pressure drop experienced in a single-phase fluid when flowing through a tube is the sum of the static, momentum, and frictional pressure drop. The frictional term of the pressure drop per unit length is estimated using the equation Darcy–Weisbach :

$$\frac{dp}{dz} = f_d \frac{G^2}{2D_h \rho} \quad (2.11)$$

where f_d is the Darcy–Weisbach friction factor, G is the mass flux ($G=\rho U$) in kg/m²s, D_h is the internal hydraulic diameter in m, and ρ is the density of the fluid in kg/m³. The f_d Darcy–Weisbach friction factor for laminar flow ($Re < 2300$) in circular tubes is determined from Poiseuille’s law in eq. (2.12), whereas the Blasius correlation in eq. (2.13) is used for turbulent flows ($Re > 2300$) in smooth tubes.

$$f_d = \left\{ \frac{64}{Re}, \quad \text{for } Re \leq 2300 \right\} \quad (2.12)$$

$$f_d = \left\{ \frac{0.3164}{Re^{0.25}}, \quad \text{for } 2300 < Re \leq 10000 \right\} \quad (2.13)$$

where Re is the Reynolds number and is calculated as:

$$Re = \frac{\rho U D_h}{\mu} \quad (2.14)$$

and ρ is the density, U is the velocity of the fluid, D_h is the hydraulic diameter, and μ the dynamic viscosity. The hydraulic diameter is calculated as:

$$D_h = \frac{4A}{P_{wet}} \quad (2.15)$$

where A is the area in m^2 , and P_{wet} the wetted perimeter in m. For rough tubes and turbulent flow, the Colebrook correlation [20] is used, where it is considered a function of the relative roughness ($E_p = \frac{\epsilon}{D_h}$). When solved iteratively, the Colebrook equation gives the friction factor for higher Reynolds numbers.

2.2.3 Pressure losses through enlargements, contractions and bends

The pressure losses through enlargements, contractions, and bends can be found in ref. [21]. The pressure drop due to the contraction from diameter D_1 to D_2 is estimated as:

$$\Delta p_{con} = k_{con} \frac{G^2}{2\rho} \quad (2.16)$$

with k_{con} is the contraction coefficient as in eq. (2.17), and G is the mass flux in kg/m^2s .

$$k_{con} = \begin{cases} 1.20 + \frac{38}{Re}, & \text{for } Re \leq 4000 \\ \frac{1}{2} \left(1 - \left(\frac{D_2}{D_1} \right)^2 \right), & \text{for } Re > 4000 \end{cases} \quad (2.17)$$

The pressure drop due to expansion from D_1 to D_2 is given by eq. (2.18):

$$\Delta p_{exp} = k_{exp} \frac{G^2}{2\rho} \quad (2.18)$$

where k_{exp} the expansion coefficient given by (2.19).

$$k_{exp} = \begin{cases} 1.20 + \frac{38}{Re}, & \text{for } Re \leq 4000 \\ 1 - \left(\frac{D_1}{D_2} \right)^4, & \text{for } Re > 4000 \end{cases} \quad (2.19)$$

The pressure drop per unit length due to a bend in single-phase flow is estimated with eq. (2.20):

$$\frac{dp}{dz} = 0.5f_d \frac{G^2}{\rho} \pi \frac{R_b}{D_h} \left(\frac{\theta}{180} \right) + 0.5k_{bend} \frac{G^2}{\rho} \quad (2.20)$$

where R_b is the bend radius, θ is the bend angle in $^\circ$, k_{bend} is the bend coefficient found in ref. [21], and f_d is found from the Colebrook or Blasius equations depending on the Reynolds number.

2.2.4 Pressure losses across pillar structures

In many applications, pillar structures embedded in a channel are used as heat exchangers or as regenerators. The pillars lie perpendicular to the direction of the flow and can be of various shapes. The silicon-glass micro-channel heat exchanger in chapter 9 consists of a pillar structure of circular pillars in a staggered pattern. The pressure drop of a laminar gas flow through a circular pillar structure in a staggered pattern is estimated using eq. (2.21), [22]:

$$\Delta p_{pill} = fN \frac{G^2}{2\rho} \quad (2.21)$$

where f the friction factor estimated using eq. (2.22), and N the number of tube rows.

$$f = 70.20Re^{-0.44} \quad (2.22)$$

2.2.5 Heat Transfer Coefficient

The heat transfer coefficient for single-phase flow in a circular pipe is found using eq. (2.23), where λ is the thermal conductivity of the fluid, D_h the hydraulic inner diameter, and Nu is the Nusselt number [19] :

$$h = \frac{Nu\lambda}{D_h} \quad (2.23)$$

The Nu for fully developed laminar single-phase flow in a circular tube with uniform heat flux is a constant parameter (2.24).

$$Nu = 4.36 \quad (2.24)$$

The Nu for fully developed laminar single-phase flow in a circular tube with uniform surface temperature is also a constant parameter (2.25).

$$Nu = 3.66 \quad (2.25)$$

For fully developed turbulent flow ($Re > 10^4$), the Nu in a smooth circular pipe is found using Dittus-Boelter [23] correlation with $z=0.4$ for heating and $z=0.3$ for cooling. The Dittus-Boelter correlation is:

$$Nu = 0.023 Re^{0.8} P_r^z \quad (2.26)$$

where P_r is the dimensionless Prandtl number, which is the ratio of the momentum diffusivity to the thermal diffusivity:

$$P_r = \frac{c_p \mu}{k} \quad (2.27)$$

where c_p is the specific heat in J/kgK, μ is the dynamic viscosity in Pa s, and k is the thermal conductivity in W/mK.

For a transient and fully developed turbulent flow in a circular pipe with a rough surface, the Nu is found using the Gnielinski correlation [24].

$$Nu = \frac{(f_d/8)(Re - 1000)P_r}{1 + 12.7(f_d/8)^{1/2}(P_r^{2/3} - 1)} \quad (2.28)$$

2.3 Two-phase flow

The equations described in this section were used to estimate heat and flow parameters for the two-phase flow cooling in the petal described in chapter 5 and 6. In two-phase flow, the power ($\dot{H}$) needed to change the state of a fluid is given:

$$\dot{H} = \Delta h_{vap} \dot{m} \quad (2.29)$$

where $\dot{m}$ is the mass flow in kg/s, and Δh_{vap} the latent heat of evaporation, which is the enthalpy difference of the fluid between the start and the end of the process expressed in J/kg.

2.3.1 Flow regimes in mini-channels

In two-phase flow, the characteristic distribution of the liquid-vapor interface differentiates the flow regimes and the different flow patterns. There can be bubbles suspended in a liquid stream, liquid droplets in a vapor stream, or both phases existing intermittently. A full hydrodynamical description of the flow pattern requires the knowledge of the void distribution, velocity distribution, and shear distribution of the flow field [25]. It is clearly impossible to specify in detail the distribution in time and space. However, a few widely accepted general flow regimes can be predicted by independent variables, such as flow rate or physical properties of the flow [26]. This macroscopic approach is based on visualization studies.

The flow regimes that have been observed in horizontal mini-channels are summarized below (see also figure (2.1)). For a vertical flow in mini-channels, differences have been observed due to gravity effects.

1. Bubbly flow. The gas phase is distributed in discrete bubbles within a liquid continuum. Due to gravity the bubbles have the tendency to move to the upper part of the channel.
2. Plug flow. The bubbles have the same cross section as the channel and move along in a bullet shape. The layer on the bottom is thicker than on the top and the nose of the bubble is asymmetric.
3. Stratified flow. There is clear separation of the vapor and liquid phase, vapor on the top and liquid on the bottom.
4. Wavy flow. As the gas velocity increases, large wavy surface is built up on the interface.
5. Slug flow. For even further increase in the gas velocity the waves are big enough to reach the top of the channel.
6. Annular flow. A liquid layer or film on the channel walls presents a continuous structure interface with a gas stream in the center of the channel. Thicker layer of liquid at the bottom than at the top.

The transition from one regime to the other has been presented in refs. [1] and [2]. These transitions will be used in chapters 5 and 6 to determine the flow characteristics.

Figure 2.1: Flow regimes observed in horizontal tubes [27]

2.3.2 Heat transfer mechanisms

The different flow patterns determine different modes of heat transfer. The heat transfer mechanics are summarized starting from the subcooled liquid up to the saturated vapor area. To begin with, in the sub-cooled liquid pure convective heat transfer takes place, whereas the temperature of the liquid at the wall is below that for bubble nucleation. As the heat flux increases, bubbles are formed adjacent to the wall in the subcooled liquid. This is known as subcooled boiling, because the temperature of the bulk fluid is increasing but still is below the saturation temperature (ΔT_{sub}), and only the wall temperature stays constant above the saturation temperature (ΔT_{sat}). The next transition takes place when the saturation temperature is reached for the bulk liquid. This is known as saturated nucleate boiling, and is related to the thermodynamic quality. The quality from this on defines the heat transfer mechanisms. The vapor quality is defined as the fraction of the fluid that is vapor.

$$x = \frac{\dot{h} - \dot{h}_L}{\Delta h_{vap}} \quad (2.30)$$

where $\dot{h}$ is the enthalpy at the point of interest, $\dot{h}_L$ is the saturated liquid enthalpy and Δh_{vap} the enthalpy of evaporation. As the quality increases,

there is a point where the heat transfer mechanism changes to evaporation. This is accompanied by a change in the flow pattern from bubbly/slug to annular flow [27]. The thickness of the liquid film on the wall is such that does not allow the liquid in contact with the wall to reach the temperature level for bubble nucleation. Forced convection is the heat transfer mechanism, and the region beyond this point is known as two-phase forced convective region. As the heat flux and quality increase further, there is a point at which full evaporation of the liquid film takes place and the temperature of the wall starts rising. This is the transition known as dryout in the liquid deficient region, which is followed by the dry saturated vapor region. No superheating conditions are taken into account in the above description.

2.4 Evaporation models in mini-channels

The CoBra model is used for the prediction of the flow regime, the estimation of the pressure drop, and the calculation of the heat transfer coefficient in the petal cooling described in chapters 5 and 6. CoBra is a Matlab based program that solves the one dimensional flow problem and uses the empirical models for the prediction of CO₂ flow and heat characteristics, as presented in refs. [1] and [2]. The detailed formulas are not repeated here. A detailed description of CoBra can be found in ref. [28].

2.4.1 Pressure losses through enlargements, contractions and bends

The pressure losses through enlargements, contractions and bends during two-phase flow are found in refs. [27] and [29]. The two-phase pressure drop due to contraction from diameter D_1 to D_2 is estimated as

$$dp = \frac{G_2^2}{2\rho_L} \left(1.45 - \left(\frac{D_2}{D_1} \right)^4 \right) \left(1 + \frac{\rho_L}{\rho_V} x \right) \quad (2.31)$$

where G_2 is the mass flux through the D_2 , ρ_L and ρ_V the liquid and vapor density, respectively, and x the vapor quality. The value of 1.45 in eq. (2.31) is a factor that depends on the specific geometry (ratio of the two diameters), used in chapter 8. The analytical expression is found in ref. [27].

The two-phase pressure drop due to expansion from D_1 to D_2 is found in [27].

$$dp = \frac{G_1^2}{2\rho_L} \left(1 - \frac{D_1}{D_2} \right)^4 \left(1 + \frac{\rho_L}{\rho_V} x \right) \quad (2.32)$$

where G_1 is the mass flux through the D_1 .

The two-phase pressure drop due to bend is found in ref. [29].

$$\frac{dp}{dz} = 1 + \left(\frac{\rho_L}{\rho_V} - 1 \right) \left[\left(1 + C_2 \right) x(1 - x) + x^2 \right] dP_L \quad (2.33)$$

where dP_L the pressure drop in liquid phase, x the vapor quality, ρ_L and ρ_V the liquid and vapor density. The coefficient C_2 is calculated by Chisholm [29].

$$C_2 = 1 + \left(\frac{2.2}{k_{bend}(2 + R_b/D_{bend})} \right) \quad (2.34)$$

where k_{bend} is the bend coefficient in single-phase flows, R_b is the radius of the bend, and D_{bend} is the diameter of the bend.

2.5 Evaporation models in Micro-channels

The equations described in this section are used to estimate heat and flow parameters for the single micro-channel cooling described in chapter 8. At the micro-meter level, some macro-scale phenomena may be suppressed, whereas other fluid and thermal phenomena may be dominant due to the decreasing channel size. During two-phase flows in micro-channels, the flow is mainly laminar, and capillary forces become more dominant, whereas the stratification of the liquid is suppressed. Therefore, at the micro-channel level, different equations and new dimensionless numbers are used. The dimensionless numbers can be found in ref. [27]. These dimensionless numbers for the micro-channels are summarized next.

2.5.1 Dimensionless numbers in micro-channels models

The confinement number (Co) provides the transition from the macro-scale to micro-scale:

$$Co = \frac{L_{cap}}{D_{th}} \quad (2.35)$$

where D_{th} is the hydraulic diameter at which the transition from macro-scale to micro-scale occurs, and L_{cap} is the capillary length defined as

$$L_{cap} = \sqrt{\frac{\sigma}{g(\rho_L - \rho_V)}} \quad (2.36)$$

where σ is the surface tension, g is the gravity constant, and ρ_L and ρ_V is the liquid and vapor density, respectively. Ong describes in ref. [30] that the

confinement number above which the micro-scale zone exists is $Co > 0.5$. The boiling number (BO) describes the effect of heat flux on the boiling and is calculated as:

$$BO = \frac{\dot{q}}{G\Delta h_{vap}} \quad (2.37)$$

where $\dot{q}$ is the heat flux, G the mass flux, and Δh_{vap} the latent heat of evaporation.

The Bond number (Bond) gives the importance of the gravitational forces with respect to the surface tension forces. It is calculated as:

$$Bond = \frac{g(\rho_L - \rho_v)D_h^2}{\sigma} \quad (2.38)$$

where D_h is the hydraulic diameter, σ is the surface tension, g is the gravity constant, and ρ_L and ρ_v are the liquid and vapor densities.

Ong and Thome [30] presented a flow-pattern map that is discussed in section 2.5.2. They use the Weber liquid number We_L and the Froude liquid number Fr_L . The We_L gives the effect of the inertial forces with respect to the surface tension forces and it is calculated as:

$$We_L = \frac{G^2 L_{ev}}{\sigma \rho_L} \quad (2.39)$$

where L_{ev} is the evaporator length, where the heat exchange takes place during evaporation. The Froude liquid number gives the effect of the inertia to gravitational forces and is calculated as:

$$Fr_L = \frac{G^2}{\rho_L^2 g D_h} \quad (2.40)$$

2.5.2 Flow regimes in micro-channels

Revellin and Thome [31] categorize the flow regimes in micro-channels into three patterns; isolated bubbles, elongated bubbles, annular flow. They also mention the dryout regime. The isolated bubble regime refers to the mini-channels regimes of bubbly flow (defined as having bubbles shorter in length than the channel diameter), slug flow (defined as bubbles longer than the channel diameter), and mixed bubble flows displaying both long and short bubbles. In this regime, the bubble frequency increases with heat flux and, thus, vapor quality at a fixed mass velocity increases. The elongated bubble regime refers to the mini-channel regimes of slug flows and churn flows (the latter are long bubbles followed by liquid plugs) but where some short bubbles may still exist and where the frequency of the bubbles decreases with increasing heat flux and thus vapor quality at a fixed mass velocity. The annular flow

regime refers to the mini-channel regimes of both smooth annular flows with nearly no interfacial waves and wavy annular flow where interfacial waves are highly evident. The dryout regime refers to the dryout condition encountered after passing through the critical heat flux (CHF), see eq. (2.43). The critical heat flux is the value of heat flux, at which the system shows a sharp reduction in the heat transfer, due to the replacement of liquid by vapor adjacent to the heat transfer surface. As mentioned above, Ong and Thome proposed a flow-pattern map [30], [32] that is summarized next. The equation for the vapor quality at the transition of two regimes is given as a function of various dimensionless numbers. The vapor quality that defines the transition between the isolated bubble to the elongated bubble (IB/EB) regimes is calculated as:

$$X_{IB/EB} = 0.36Co^{0.20} \left(\frac{\mu_V}{\mu_L} \right)^{0.65} \left(\frac{\rho_V}{\rho_L} \right)^{0.9} Re_V^{0.75} BO^{0.25} We_L^{-0.91} \quad (2.41)$$

where μ_L and μ_V are the liquid and vapor dynamic viscosities, Re_V is the Reynolds number at vapor phase, and ρ_L and ρ_V are the liquid and vapor densities. The vapor quality that defines the transition between the elongated bubble to annular (EB/A) regime is calculated as:

$$X_{EB/A} = 0.047Co^{0.05} \left(\frac{\mu_V}{\mu_L} \right)^{0.7} \left(\frac{\rho_V}{\rho_L} \right)^{0.6} Re_V^{0.8} We_L^{-0.91} \quad (2.42)$$

The dryout occurs when the heat flux is larger than the critical heat flux. The critical heat flux (CHF) $\dot{q}_{CHF}$ is given by Ong [30], [32] as:

$$\frac{\dot{q}_{CHF}}{G\Delta h_{vap}} = 0.12 \left(\frac{\mu_V}{\mu_L} \right)^{0.183} \left(\frac{\rho_V}{\rho_L} \right)^{0.062} We_L^{-0.141} \left(\frac{L_{ev}}{D_h} \right)^{-0.7} \left(\frac{D_h}{D_{th}} \right)^{0.11} \quad (2.43)$$

where L_{ev} is the evaporator length, and D_{th} is calculated using eq. (2.35) for $Co=0.5$ at which the bubble growth is confined by the channel and the bubbles grow in axial [30],[32].

2.5.3 Pressure drop

The total pressure drop is the sum of the static (gravity), momentum (acceleration), and frictional pressure drops. For horizontal channels, the static pressure drop is zero. The frictional pressure drop for the isolated bubble and the elongated bubble flow is found by Lockhart and Martinelli [33]. The frictional pressure drop gradient for two-phase flow is calculated based on the

liquid pressure gradient as:

$$\left(\frac{dp}{dz}\right)_f = \left(\frac{dp}{dz}\right)_L Mult \quad (2.44)$$

where the liquid only pressure gradient is:

$$\left(\frac{dp}{dz}\right)_L = 2f_L \frac{((1-x)G)^2}{(\rho_L D_h)} \quad (2.45)$$

In the above equation ((2.45)), the f_L is the Fanning friction factor for the liquid, which is one fourth of the Darcy friction factor. The Mult is the liquid only multiplier is as:

$$Mult = 1 + \frac{C_{LM}}{LMpar} + LMpar^{-2} \quad (2.46)$$

The LMpar is the Lockhart-Martinelli parameter and is calculated based also on the vapor only pressure gradient as:

$$LMpar = \sqrt{\frac{\left(\frac{dp}{dz}\right)_L}{\left(\frac{dp}{dz}\right)_V}} \quad (2.47)$$

The vapor only pressure gradient is calculated as:

$$\left(\frac{dp}{dz}\right)_V = 2f_V \frac{(xG)^2}{(\rho_L D_h)} \quad (2.48)$$

where the f_V is the Fanning friction factor for the vapor. The C_{LM} parameter and the f_L, f_V friction factors for liquid and vapor depend on the Reynolds number. The f_L and f_V are the liquid and vapor friction factors, which are calculated below with the liquid Re_L and vapor Re_V Reynolds number.

$$f_L = \left\{ \begin{array}{ll} \frac{16}{Re}, & \text{for } Re_L \leq 2000 \\ 0.0791 Re_L^{-0.25}, & \text{for } 2000 \leq Re_L \end{array} \right\} \quad (2.49)$$

$$f_V = \left\{ \begin{array}{ll} \frac{16}{Re}, & \text{for } Re_V \leq 2000 \\ 0.0791 Re_V^{-0.25}, & \text{for } 2000 \leq Re_V \end{array} \right\} \quad (2.50)$$

The C_{LM} parameter is calculated as :

$$C_{LM} = \left\{ \begin{array}{ll} 2.16(Re_L^{0.047} We_L^{0.60}), & \text{for } Re_L < 2000 \text{ \& } Re_V < 2000 \\ 10, & \text{for } Re_L > 2000 \text{ \& } Re_V < 2000 \\ 1.45(Re_L^{0.25} We_L^{0.23}), & \text{for } Re_L < 2000 \text{ \& } Re_V > 2000 \\ 20, & \text{for } Re_L > 2000 \text{ \& } Re_V > 2000 \end{array} \right\} \quad (2.51)$$

The frictional pressure drop in the annular regime is calculated in ref. [10]. In the next section, core is defined as the vapor flow that occupies most of the cross section in the pipe. The frictional pressure drop in the annular regime is summarized:

$$\frac{dp_f}{dz} = \frac{2f_{tp}}{D_h} \rho_c U_c^2 \quad (2.52)$$

where U_c is the core flow velocity which is the flow velocity of the vapor given in eq. (2.56), and f_{tp} is the Fanning frictional factor. The Fanning frictional factor f_{tp} is calculated as:

$$f_{tp} = \left\{ \begin{array}{ll} 0.172 We_c^{-0.372}, & \text{for } Bond > 4 \\ 0.0196 We_c^{-0.372} Re_{LF}^{0.318}, & \text{for } Bond < 4 \end{array} \right\} \quad (2.53)$$

The Weber core We_c number is calculated as:

$$We_c = \frac{\rho_c J_V^2 D_h}{\sigma} \quad (2.54)$$

where J_V is the vapor superficial velocity calculated as

$$J_V = \frac{xG}{\rho_V} \quad (2.55)$$

The flow velocity of the vapor core is calculated as

$$U_c = \frac{J_V}{\epsilon_{void}} \quad (2.56)$$

with ϵ_{void} the void fraction:

$$\epsilon_{void} = \frac{h_a x^{n_a}}{1 + (h_a - 1)x^{n_a}} \quad (2.57)$$

The void fraction is valid for

$$0 < x < 1$$

$$0.7 < \epsilon_{void} < 1$$

$$10^{-3} < \frac{\rho_v}{\rho_l} < 1 .$$

In eq. (2.57) the h_a and n_a are empirical parameters that are calculated as:

$$h_a = -2.129 + 3.129 \left(\frac{\rho_v}{\rho_l} \right)^{-0.2186} \quad (2.58)$$

$$n_a = 0.3487 + 0.6513 \left(\frac{\rho_v}{\rho_l} \right)^{0.5150} \quad (2.59)$$

The following dimensionless and geometrical numbers are used in the above equations: The ρ_c core density

$$\rho_c = \frac{(x + ELF(1 - x))}{\frac{x}{\rho_v} + \frac{ELF(1-x)}{\rho_L}} \quad (2.60)$$

where ELF the entrained liquid fraction which is calculated as

$$ELF = \left\{ (1 + 279.6 W e_c^{-0.8395})^{-2.209}, \quad \text{for } 10^1 < W e_c < 10^5 \right\} \quad (2.61)$$

The liquid film Reynolds number is estimated then with the ELF:

$$Re_{LF} = \frac{(1 - ELF)(1 - x)GD_h}{\mu_L} \quad (2.62)$$

where μ_L is the dynamic viscosity in liquid, and G the mass flux.

2.5.4 Heat transfer coefficient

In chapter 8, the heat transfer coefficient is estimated based on the flow regime, the applied heat flux, and the pressure, which are given as input. The heat transfer coefficient for the different regimes is summarized next. The heat transfer coefficient for the isolated bubble regime is calculated from Cooper [34] as:

$$h_{IB} = C_{cooper} P_{re}^{0.12-0.2 \log_{10}(\epsilon)} (-\log_{10}(P_{re}))^{-0.55} M^{-0.5} \dot{q}^{0.67} \quad (2.63)$$

where C_{cooper} is a constant ($C=55$), ϵ is the absolute roughness in μm , M is the molecular mass in kg/kmol , P_{re} is the reduced pressure (ratio of saturation pressure and critical pressure), and $\dot{q}$ is the heat flux in W/m^2 .

The heat transfer coefficient for the elongated bubble regime is described in John's three zone model [8], [9]. The time-averaged local heat transfer coefficient is estimated as:

$$h_{EB}(z) = h_L(z) \frac{t_L}{\tau} + h_{film}(z) \frac{t_{film}}{\tau} + h_{dry}(z) \frac{t_{dry}}{\tau} \quad (2.64)$$

where τ is the pair period (the period when a pair of liquid slug and vapor passes) in s, t_L is the time for the liquid slug to pass a fixed location, t_{dry} is the maximum time when the film exists till dryout occurs, and t_{film} is the time that the film is present.

The heat transfer coefficient for the annular regime is calculated by the Cioncolini and Thome model that is described in ref. [10]. The heat transfer equation for convective evaporation in annular flow is summarized.

$$h_{An} = \frac{k_L Nu}{t} \quad (2.65)$$

where k_L is the liquid thermal conductivity of the fluid, and t the average liquid film thickness in [m]. The Nusselt number (Nu) is calculated as :

$$Nu = 0.0776t^{+0.9} Pr_L^{0.52} \quad (2.66)$$

where Pr_L is the liquid Prandtl number, and t^+ is the dimensionless liquid film thickness that is calculated as:

$$t^+ = \max\left(\sqrt{\frac{Re_{LF}}{2}}, 0.0165 Re_{LF}\right) \quad (2.67)$$

The Re_{LF} is the liquid film Reynolds number which is calculated in eq. (2.62). Further details can be found in ref. [10].

2.6 Two-phase cooling methodology in HEP

In this section, the methodology followed for the petal design, described in chapter 6, is explained. In evaporative cooling systems, the design parameters are the mass-flow rate ($\dot{m}$) and the latent heat of evaporation (see eq. (2.29)). In the petal design, the known parameters are the evaporation temperature (T_{evap}) or pressure (P_{evap}) and the expected dissipated power in the sensor and ASICs. Based on the above, the vapor quality is selected as follows. CoBra, a Matlab based program that solves the one dimensional flow problem, is used in order to select the vapor quality at the exit of the tube. Giving as input to CoBra the saturated temperature and expected power, CoBra calculates the flow regime and heat transfer coefficient along the tube for varying mass flow rates. The vapor quality is chosen to be safely away from dryout, and usually where the heat transfer coefficient is maximum. Finally, the mass flow is calculated to achieve the desired vapor quality as follows.

The pressure-enthalpy diagram in figure (2.2) shows the evaporation process. Looking at the pressure-enthalpy diagram, the experiment starts at the saturated liquid line at the inlet tube.

Figure 2.2: Pressure-enthalpy diagram for CO₂ [35]

This requires that the inlet conditions are known and controlled. On the saturated liquid line the vapor quality is zero with enthalpy $\dot{h}_{sat}$. Having a desired vapor quality target (B), for example 0.6, the enthalpy $\dot{h}_{out}$ is found in a pressure-enthalpy diagram following an isothermal line. The latent heat of evaporation $\Delta\dot{h}_{vap}$ [kJ/kg], which is the difference between the enthalpy at the saturated liquid line and the enthalpy of interest, is the energy needed in order to evaporate the liquid and give vapor quality 0.6. The nominal power added from the heaters is known and along with the latent heat of evaporation are used in eq. (2.29) to find the mass flow.

Chapter 3

Petal Description

As discussed in chapter 1, the endcap of the ATLAS detector consists of 7 disks, each of which has 32 identical petals. The petal is a double-sided structure that hosts the sensor modules and the electrical services. It provides mechanical support and cools the modules. This chapter describes the petal and its components: the modules, the mechanical structure, the heating sources, and the cooling. The information is used in chapters 4 and 5 to build and evaluate the thermal model of the petal.

3.1 Sensor

The sensor is $320\text{ }\mu\text{m}$ thick AC coupled p-type silicon with n+ strips built from 6 inch wafers. The sensors are wedge shaped; inner and outer edges are arcs of circles. Figure (3.1) shows a petal from the UTOPIA layout (originally designed in 2010) with the wedge-shaped sensors at each ring, where the outer three rings have two of the sensors side by side. There are six sensor shapes in total.

Figure 3.1: CAD drawing of petal detector support according to the UTOPIA layout. The cooling tube passes from the two ‘ears’ of the petal.

3.2 Front-end electronics

The front-end readout electronics will be 130 nm technology ASICs known as ABC130. The ABC130 ASIC serves 256 strips, which are connected to the ASICs through bond wires. Data is sent from ABC130 ASIC to the Hybrid Controller Chip (HCC). The ABC130 and HCC ASICs are mounted on low mass, polyimide based multi-layer flexible printed circuit boards (PCBs) called hybrids. Each hybrid carries a number of ABC130 ASICs and HCC ASICs depending on the number of strips on each sensor. The newest iteration of ASICs for the ITk are the ABC130*. Table (3.1) gives the dimensions of each ASIC used later in the thermal model of the petal to estimate the power density of the ASICs.

Table 3.1: Dimensional information of the ASICs.

ASIC	Dimensions
	mm x mm
ABC130	7.9 x 6.8
ABC130*	7.9 x 6.8
HCC	2.86 x 4.7

The HCC sends data to the End of Petal (EoP) device at up to 640 Mbps. The EoP is the interface between the on-detector and off-detector electronics. Figure (3.2) shows a closer picture of an endcap hybrid.

Figure 3.2: Example of an endcap hybrid with ABC130 ASICs and HCC ASIC.

3.3 Powering scheme

The Low Voltage (LV) powering scheme has one power supply for all modules on one side of the petal. This scheme considerably reduces material in the

services compared to the current ID which has one power supply per module. The baseline powering scheme is a Direct Current to Direct Current (DC/DC) conversion scheme. It includes one power conversion step from the 11 V supply line to 1.5 V at each module. Most of the modules have one power board while some have two. The power board has a DC/DC radiation-hard converter with a custom toroidal air-core inductor and a buck converter ASIC, the FEAST2. The inefficiency in the DC/DC converter, when reaching the current limit, gives a high power density in the FEAST2 ASIC.

Table 3.2: Dimensional information of the FEAST2 ASIC.

FEAST2 size	package size	max current
mm ²	mm ²	A
3.8 x 2.8	5 x 5	4

3.4 High voltage supply

All modules on one side of the petal are supplied with one High Voltage (HV) supply bias line considerably reducing the material used. The bias voltage for each line is 500 V or even higher.

3.5 Mechanical structure

The mechanical structure provides the necessary stiffness in order to keep the sensors stable during operation. It consists of carbon fiber facesheets separated by honeycomb core. The carbon fiber facesheets offer high thermal conductivity in plane for good heat spreading and high stiffness for mechanical stability. The core provides high shear modulus. Finally, carbon fiber closeouts secure the outer parts of the core. A titanium cooling tube is embedded in the mechanical structure. The tube is placed in a V-shaped line in the mid-plane of the petal with the entrance and exit of the line at the broader end of the petal. Around the tube there is carbon foam. The carbon foam around the cooling tube provides a low thermal impedance between the coolant and the heat source, keeping the temperature drop between them low. Figure (3.3) gives the cross section of a petal. In chapters 4 and 5 the bus-tape is not taken into account, whereas in chapter 6 the bus-tape is included in the thermal model.

Figure 3.3: Cross section of a petal with embedded cooling [36]. The ABC ASICs are on both sides of the structure where the outermost ASIC is the HCC.

3.6 Radiation damage and heating sources

Particles transversing the modules cause radiation damage. Charged particles cause ionizing radiation damage especially in insulators in electronic chips and sensors. Hadrons cause non-ionizing damage in the bulk material [37].

3.6.1 Silicon sensor

Hadronic interactions with nuclei in the sensor crystal cause lattice dislocations. This can change the effective doping, reduce charge collection efficiency and increase the leakage current [37]. Irradiated silicon has the ability to anneal; to self-heal. The annealing process is temperature dependent. It anneals faster at higher temperature. On a longer time scale though, it has been seen that the crystal gets worse; it reaches a reverse annealing stage. Reverse annealing can be avoided by keeping the sensors cold. In the next chapters, the reverse annealing is predicted with FEA for the petal and is referred as thermal runaway.

The leakage current density (A/cm^3) depends on the fluence and is strongly dependent on temperature. The increase in reverse bias current I due to bulk damage has been investigated by Moll (equation 5.1 in [38]), and follows

$$I_{(T_{ref})} = I_o + \alpha_{(T_{ref})} \Phi_{eq} V \quad (3.1)$$

where I_o is the leakage current density at zero fluence which is negligible compared to the final leakage current, Φ_{eq} is the particle fluence (usually given in terms of 1 MeV neq dose) and α the damage coefficient. In practice, the damage coefficient is measured at some reference temperature T_{ref} , usually at 293 K. The leakage current depends on the absolute temperature T and the effective band-gap energy E_g via the following formula where k is the Boltzmann

constant [38].

$$I(T) \propto T^2 e^{\left(\frac{-E_g}{2kT}\right)} \quad (3.2)$$

The leakage current at another absolute temperature T is found from (3.2).

$$I(T) = R(T)I(T_{ref}) \quad (3.3)$$

where the dimensionless correction factor $R(T)$ is

$$R(T) = \frac{T^2}{T_{ref}^2} \exp\left(\frac{E_g}{2k} \left(\frac{1}{T_{ref}} - \frac{1}{T}\right)\right) \quad (3.4)$$

The correction factor $R(T)$ is used later to estimate the power produced in the sensors. The effective band-gap energy E_g is also temperature dependent. We use the description of Varshini [39]. The volume current density I can be converted to an areal current density j (A/cm^2) scaling by the sensor thickness (t). This leakage current produces a heating areal power density $\dot{Q}_c$ (W/m^2)

$$\dot{Q}_c = jV \quad (3.5)$$

where V is the sensor's bias voltage. The areal current density can be used to define a conductivity σ_e via Ohm's law. Assuming current only flows in the thickness direction, Ohm's law can be written as :

$$j = \sigma_e E = \sigma_e \frac{V}{t} \quad (3.6)$$

where E is the electric field, and t is the thickness of the sensor. Combining eqs. (3.5) and (3.6) gives the heating power $\dot{Q}_c$

$$\dot{Q}_c = \frac{\sigma_e V^2}{t} \quad (3.7)$$

which is used in chapter 6 to model the temperature dependence of the sensor power. From eq. (3.6) follows that $\sigma_e(T)$ scales with the same factor $R(T)$ as the leakage current I .

Sensors must operate at temperatures well below zero degrees Celsius. The leakage current at these temperatures is reduced and the effect of reverse annealing is avoided for longer time. Even without thermal runaway, a high leakage current increases shot noise in the readout ASICs. Eq. (3.2) explains also the runaway situation described earlier. The leakage current increases with higher temperature, doubling roughly for every 7 K. The increase in current creates further heat and can cause “thermal runaway” if not controlled.

3.6.2 ASICs

Considerable heat is dissipated in the ASICs (ABC130 and HCC) due to the high data rates. The power board for the DC/DC powering scheme that includes DC/DC converter dissipate also heat with the main contributor the buck converter ASIC; FEAST2. The DC/DC converter has a maximum efficiency around 75-80% which varies with the current. The EoP consists of several chips and contributes as well in heat production.

The heat produced in the ASICs tends to increase with radiation damage due to extra leakage current in transistors through the damage of the silicon dioxide in the insulators. In the thesis, the small effect of the radiation damage on the ASICs is not taken into account and it is assumed that the heat produced in the ASICs stays constant over time.

Regarding the IBM 130 nm CMOS technology ASICs, an increase has been seen in the digital current and not in the analog current after receiving a certain Total Ionization Dose (TID), known as TID bump. The TID bump depends on the dose rate history and the temperature of the ASICs while its shape is a function of the integrated dose at a given time. In chapters 4 and 5 the TID bump effect is not taken into account but in chapter 6 it is included in the calculations.

3.7 UTOPIA layout design

The layout design gives the number and position of strip and pixel detectors in the ITk. Information on the UTOPIA layout can be found in ref. [14]. The layout continues to evolve along with the design of the ITk. The prototype presented in chapter 4 is from the UTOPIA layout designed in 2010. This layout has a barrel part with three cylinders of short-strip staves followed by two layers of long-strip staves, and two endcap parts each consisting of five discs of endcap modules. The next sections provide information on the geometry and heating sources for the prototype petal described in chapter 4.

3.7.1 Geometry

The sensors used in the prototype petal follow the geometry described here. Table (3.3) summarizes the positions of the sensors for the UTOPIA layout in respect to the beam line and collision point, and the area of each sensor from the innermost to the outermost ring.

Table 3.3: Sensor area and position for the UTOPIA layout; R_i is the radii of the inner ring and R is the translation along the sensor strip direction from the collision point.

Ring	R_i mm	R mm	Area mm ²
0	336.344	392.244	8182
1	443.778	494.470	9327
2	540.720	581.131	8600
3	616.790	677.650	7925
4	737.290	794.714	7925
5	850.650	902.068	8838

3.7.2 Silicon leakage power

The silicon leakage power is estimated based on the radiation dose expected at the different rings. Figure (3.4) shows the radiation dose (or fluence) in neutron equivalent n/cm² over the different locations in the endcap disks (vertical lines) and the barrel (horizontal lines) for the UTOPIA layout. The radiation dose multiplied with the radiation damage coefficient gives the current density. The current density multiplied with the area and the thickness of the sensor gives the sensor current. The assumed bias voltage to calculate the power is 500 V. Table (3.4) summarizes the radiation dose received in the sensors and the power generated at 293 K. Disc 4 in UTOPIA layout (the last one) receives the most radiation because it is located closest to the calorimeter where most neutrons are produced. In the thesis, the fluence prediction for this disc is used as the worst case. In addition an extra ring 0 is included which was not in the UTOPIA layout.

Figure 3.4: Estimated radiation dose in neutron equivalent n/cm^2 for the innermost segment of each strip disk ring [40].

Table 3.4: Estimated leakage power dissipated at 293 K in the sensors based on the fluence for disc 4 and a radiation damage factor $3.99 \cdot 10^{-17} \text{ A/cm}^3/(n_{eq}/\text{cm}^2)$ [37]. The assumed bias voltage is 500 V.

Ring	Dose neq/cm^2	Current density A/cm^3	Current A	Power W
0	$7 \cdot 10^{14}$	0.0279	0.073	36.55
1	$5.5 \cdot 10^{14}$	0.219	0.065	32.75
2	$4.5 \cdot 10^{14}$	0.179	0.049	24.70
3	$4 \cdot 10^{14}$	0.195	0.040	20.23
4	$3.5 \cdot 10^{14}$	0.139	0.039	19.55
5	$3.1 \cdot 10^{14}$	0.123	0.035	17.49

3.7.3 ASICs power

Table (3.5) gives the maximum expected current for the ABC130 for Short Strips (SS) and Medium Strips (MS) and HCC ASICs. Table (3.6) gives a summary of the number of chips for each sensor and the estimated power dissipated from the hybrid. No TID bump effect was included for the UTOPIA studies.

The hybrid power refers to the power in the HCC, the ABC130 ASICs and the DC/DC converter. Here, the DC/DC converter is assumed to have the

80% maximum efficiency. When modeling a more realistic efficiency is used. The power in the EoP board is estimated to be around 2-3 W per side. The efficiency in the DC/DC converter is defined as the following ratio

$$\epsilon = \frac{H_{hybrid}}{H_{DC/DC} + H_{hybrid}} \quad (3.8)$$

where the H_{hybrid} and $H_{DC/DC}$ are the hybrid and DC/DC power board power. The power in the DC/DC board can be calculated from this.

Table 3.5: Maximum current expected for each type of ASIC for 1.3 V [41]

ASIC	Type	Current A/ASIC
ABC130	Analogue SS	0.036
ABC130	Analogue MS	0.064
ABC130	Digital	0.136
HCC	—	0.2

Table 3.6: Number of chips for each hybrid and strip type and estimated power dissipated in hybrid and the DC/DC power converter for each ring for 1.3 V assumed 80% DC/DC efficiency.

Wafer	Hybrid	No. ASIC	Strip type	Hybrid Power W	Power (DC/DC) W	Total power W
1	1	7	SS	1.80	0.45	2.26
1	2	8	SS	2.03	0.50	2.53
2	1	9	SS	2.25	0.56	2.81
2	2	10	SS	2.47	0.61	3.09
3	1	11	MS	3.12	0.78	3.90
4	1	6	MS	1.82	0.45	2.27
5	1	7	MS	2.08	0.52	2.60
6	1	9	MS	2.60	0.65	3.25

3.8 Conclusion

The petal is a mechanical structure with embedded cooling that supports and cools the modules. The main heat contributor is the ASICs. The silicon sensor can be sensitive to high temperatures especially in heavily radiated environment. The petal is cooled in order to remove the heat from the ASICs, avoid “thermal runaway” of the silicon sensors, keep the signal-to-noise ratio above 10 and prolong the lifetime of the detector. In the next chapter, the geometry and the power estimations for the UTOPIA geometry will be used to

study the thermal performance of the petal and to determine the power that has to be taken by the cooling system.

Chapter 4

Petal Prototype Measurements

A petal prototype has been built to study its thermal performance. The prototype is a mock-up of the petal with heaters to simulate the various heating sources. It includes (dummy) silicon wafers, electrical heaters, and CO₂ cooling. The prototype is cooled down to the evaporation temperature, and then the heaters are turned on. The temperature of many points on the silicon and petal is measured. This chapter describes the realization of the petal in a prototype and presents its thermal performance. The prototype is based on an earlier design, known as UTOPIA. Since then, the design continued to evolve to the current ITk layout design. Nevertheless, reliable conclusions can be drawn since all the designs consider CO₂ evaporation as the cooling method and use mostly the same materials.

4.1 Experimental Apparatus

4.1.1 Mechanical structure

The petal prototype is a sandwich structure with embedded cooling tube. As shown in figure (4.1), the prototype consists of carbon fiber pre-impregnated (prepreg) facings on a honeycomb core. There is carbon foam around the titanium tube for cooling the core. Carbon fiber closeouts prevent the facesheets from peeling off. The silicon sensors are glued on the facesheets. Figure (4.2) shows the top view of the petal prototype where the heaters cover most of the area. It indicates the total width and length of the prototype as well as the distance between the inlet and outlet section of the tube. The bottom figure shows the U-shape of the cooling tube which does not pass at the ‘ears’ of the petal as it was shown in figure (3.1). The shape is chosen such that it passes from all the silicon sensors.

Figure 4.1: Drawing of cross section of the petal prototype. The circular cooling tube is surrounded with carbon foam and the silicon wafers are on the top of carbon fiber facings.

Figure 4.2: Top view of the FEA geometry of the prototype petal with the heaters placed above the silicon wafers. Two EoP heaters are placed on one of the ‘ears’ of the petal prototype. The back side is identical but with the EoP heaters placed on the opposite ‘ear’. The cooling tube follows a U-shape and is surrounded with carbon foam.

A full description of the manufacturing process of the prototype can be found in ref. [42]. Table (4.1) gives the thickness dimensions, while table (4.2) presents the cooling tube dimensions. Table (4.3) gives the material information for each component.

Table 4.1: Thickness of the main components of the petal prototype as it is built.

Part	Thickness [mm]
Facesheet	0.20
Carbon foam	5.00
Honeycomb	5.00
Silicon wafer	0.42
Adhesive	0.07
Silicon-facesheet	
Adhesive	0.05
Facesheet-honeycomb	

Table 4.2: Dimensions of the U-shape cooling tube in the petal prototype according to the UTOPIA layout.

Inner Diameter [mm]	2
Outer Diameter [mm]	2.30
Total length [mm]	1340

Table 4.3: Component and material specifications of the petal prototype.

Part	Material	Type
Facesheets	Carbon Fiber prepreg 0/90/0	K13D2U, RS-3 resin [43]
Cooling pipe	Titanium alloy	type 17
Carbon Foam	Carbon Foam 0.55 g/cm ³	POCOFOAM HTC
Closeouts	Carbon Fiber prepreg 0/90	M55j
Core	Polyaramid Honeycomb	AR312715
Adhesive		
Honeycomb-Facesheet	Epoxy	Araldite AY 103-1/ HY 991
Adhesive		
Silicon- Facesheet	Thermal Gel	Dow Corning SE4445 CV

4.1.2 Silicon wafers

Plain silicon wafers are used in place of the silicon sensors in the prototype. There are six different shapes of silicon wafers with dimensions according to the UTOPIA layout design [14]. The wafers are not real sensors and therefore they do not produce any heat themselves. Their main purpose is to account for the heat transfer in the silicon and to provide a mechanical structure on which the temperature can be measured.

The silicon wafers are glued on the top of the facesheets without any cable-bus in between. The glue is a two-component thermal conductive silicone gel. Care was taken to avoid air bubbles that could build up pressure, and prevent

the control of the final thickness of the glue. For this reason, a glue pattern was designed for each of the silicon wafers to give 80% coverage, and avoid the creation of air bubbles. The final thickness of the glue was controlled by adding thin wires (70 μm diameter) between the facings and the silicon wafers during the gluing process.

4.1.3 Heaters

Custom designed heaters are glued on the top of the silicon wafers in the prototype [44]. They simulate the heat produced in the front-end electronics and the silicon sensors. The heaters were designed by the author and were manufactured by Heatron. They consist of narrow meandering copper traces on polyimide with pressure sensitive glue on the rear and a cover layer of polyimide with openings for soldering leads.

In total, 9 heater circuits were designed to simulate the heat produced on a petal. Each heater has two or three different heater circuits. There is a heater circuit that simulates the front-end electronics for each hybrid, which will be referred to as hybrid-heater, whereas the other heater circuit simulates the silicon sensors, which will be referred to as sensor-heater. The shape of each

Figure 4.3: The different shapes of the polyimide flexible heaters. Each heater has with different heater circuits for the sensor and electronics (hybrid, power converter).

heater circuit is defined by the shape of the heat source it simulates. The sensor heater additionally features a tab to be glued on the facesheets. This tab represents the DC/DC converter. Each heater circuit is designed with a different resistance so that when all heaters are connected in series, their power

distribution gives the expected heat load. Table (D.1) in Appendix D presents the resistance specifications for each heater circuit. Figure (4.3) shows 6 of the different heaters. A separate heater set takes the place of the End-of-Petal (EoP) circuit board. It is a 10 ohm resistor in an aluminum heat sink. A heater set is placed on each of the ‘ears’ of the prototype petal. Table (4.4) gives the information provided by Heatron Inc.

Table 4.4: Etched foil polyimide heater information (Heatron Inc.)

Copper layer	FR 9110R
Cover layer	FR 0110
Glue	3M 966 PSA
Max Current [A]	2
Temperature range [$^{\circ}C$]	-30 to +30

4.1.4 Power supply for heaters

In order to minimize the number of power supplies, the heater circuits are grouped and connected in series. Voltage is supplied by two three-channel power supplies, one for each side of the petal. On each side of the petal, all the sensor-heaters are connected in series to one channel sharing a common current, whereas all the hybrid-heaters are connected in series to another channel. Therefore, the heat-load for the silicon and hybrids can be controlled independently for each side. For each side then the EoP heater is connected to the last channel of the power supply. The measurement of the voltage drop along each heater circuit is done with a Keithley multimeter via a multiplexer.

4.1.5 Temperature sensors

Temperature sensors read the temperature on the silicon and the facesheets at various locations to investigate the thermal performance of the detector. Platinum Resistance Thermometers (PT100) were selected for their high accuracy and connected with a 4-wire configuration. Figure (4.4) shows the positions of the sensors. Most of the silicon wafers have two PT100 sensors, whereas the innermost two have three. The sensors are placed next to each DC/DC converter tab and on one of the corners of the wafer. Additional PT100 sensors are placed on the surface of the heaters while PT100 are placed also on the facesheets (yellow squares). The PT100 sensors are attached with a thermally conductive glue. The wires of the sensors are glued to the petal to prevent heat conduction and minimize undesired strains.

Thermocouples type K measure the temperature of the CO_2 at the inlet and

outlet of the petal both at the wall and in direct contact with the CO₂. Another thermocouple was placed on a heater and connected to a safety device that switches off the power supply when the temperature exceeds a certain limit.

Figure 4.4: Numbering and positioning of the sensors on the petal top side (side A). Bottom side (side B) is identical except for the inlet and outlet that are reversed. Each wafer has a temperature sensor at the corner that has a long path to the cooling line. Also, each wafer has a temperature sensor near each hybrid-heater.

4.1.6 CO₂ plant

A 1.5 kW CO₂ cooling system called ‘Green Cooler’ exists at NIKHEF and uses a 2-Phase Accumulator Controlled Loop (2PACL) method. The 2PACL and the Green Cooler are described in ref. [17]. The primary cooling unit condenses and cools down the CO₂ of the secondary cooling unit and the accumulator. Two pumps in series circulate the sub-cooled CO₂. A Coriolis flow meter measures the mass flow rate. A heater increases the temperature of the subcooled CO₂ to reach the saturated liquid state. The secondary cooling unit also has an accumulator to control the pressure and therefore the temperature of the evaporative CO₂ at the outlet of the experiment. A PID controller adjusts the accumulator pressure, mass flow and enthalpy heater power. The pressure is measured before the inlet and after the outlet of the petal with two pressure transmitters. Moreover a Digital Transmitter Indicator measures directly the pressure drop. Figure (4.5) presents the experimental set-up with the cooling plant and the primary cooling.

Figure 4.5: Schematic drawing of the cooling set-up. The pressure at the outlet is controlled with the accumulator. Pressure and temperature measurements give the conditions at the inlet and outlet of the petal. More details in ref. [17]

4.1.7 Insulation box

An insulated box was built at NIKHEF to minimize environmental leaks during cold tests. The insulation box consists of layers of wood and insulating foam. Figure (4.6) shows the insulation box in the lab. An opening at one side of the top part of the box gives access to a needle valve for adjusting the mass flow during measurements while connectors are placed next to the box for easy access to the thermocouples and pressure sensor connections.

Figure 4.6: The insulation box in which the prototype is placed during measurements to minimize heat leaks to/from the ambient environment. There is access to the mass flow setting via an opening on the top surface of the box while built-in connections on the side of the box offer easy access to the temperature and pressure sensors.

4.1.8 Cabling

A number of cables connect the petal with the measuring devices and power supplies. Figure (4.7) shows the petal in the insulation box connected with all the cabling. In order to have a modular prototype, easy to connect and disconnect, metal frames with connectors were added at the bottom and top edge of the petal. Heavy duty cables were soldered between the in-series heaters group, minimizing heat production in the cables. The end of these in-series groups are connected to circular connectors at the inner (narrow) end of the petal. Cables run from these circular connectors to the power supplies. In addition to the current carrying cables, each heater pad has a voltage sensing wire soldered on to it. These are collected in a DB50 connector at the outer end of the petal. A ribbon cable connects to the Keithley multiplexer, enabling the voltage drop across each heater to be measured. Each PT100 lead is soldered to a twisted pair cable. These are collected in a DB37 connector at the outer end of the petal. From there, cables lead to the HAPTAS (see next section) PT100 temperature readout system. A document describing all the cabling connections to and from the petal can be found in ref. [45].

Figure 4.7: Prototype petal placed in the insulation box features polyimide heaters, temperature sensors and power wires. Metal frames with connectors at the edges of the petal offer modularity and ease of access to the wires.

4.1.9 Data acquisition and Control system

The read-out system for the temperature sensors, called HAPTAS was produced at Nikhef. This reads the resistance of each PT100 sensor and converts

it to temperature. A DT800 data logger reads-out the thermocouple sensors. A Keithley multimeter with a multiplexer reads-out the voltage across each heater circuit. It has a RS-232 interface connection to the computer for remote programming. The pressure sensors are read-out with a WAGO I/O system and then connected to the computer via Ethernet. Table (4.5) summarizes the instruments connected to the control system that is built based on the WinCC-OA SCADA tool. The control system is in control of the Green cooler, the power supplies, flow meter, and multimeter. The control system records pressure, temperature, mass flow, voltage, and current. Table (4.6) gives the accuracy of the readings.

Table 4.5: Communication protocol information for the read-out devices used in the experimental set-up of the prototype petal.

Unit	Type	Communication protocol
Power supplies	TTi PL330TP	RS-232
PT100 readout	HAPTAS	RS-232
Thermocouple	DT800 data	MODBUS over TCP/IP
Voltage drop	Keithley 7702 multiplexer	RS-232
Pressure	WAGO I/O system	CANbus
Mass flow	RHEONIK Coriolis	Analog signal to PLC (0...10V)

Table 4.6: Accuracy of the sensors, meters and power supply during the measurements

Measurement	Type	Uncertainty
Temperature	PT100 class A	± 0.3 °C
Temperature	PT100 class B	± 0.6 °C
Temperature	Thermocouple type K	± 1.5 °C
Absolute Pressure	PTX7511 GE Druck	± 0.07 bar
Differential Pressure	CD23 Transmitter indicator	± 0.07 bar
Mass flow	RHEONIK Coriolis	15% of reading
Power	TTi PL330TP	$0.1\% \pm 10$ mV

4.2 Measurements

In the next section, a description of the measurements of the petal is summarized. Examples of the single-phase and two-phase measurements are provided at the end.

The real detector will operate below 0°C during data-taking, detector tests and the standby mode. On the other hand, during commissioning and silicon annealing mode the detector will operate at around room temperature. In

order to examine the performance of the petal during these phases the prototype is measured under various evaporation temperatures. The cold operation refers to a CO₂ evaporation temperature between -35°C and -30°C, whereas the warm operation refers to a CO₂ evaporation temperature at +20°C.

The experiment starts by switching on the cooling. The petal is cooled down and the power of the heaters is turned on when stable temperatures are reached. The mass flow is adjusted to reach the desired outlet vapor quality. A measurement takes place when steady state conditions are reached. The condition for steady state is achieved when the temperatures are stable to 0.1 °C. Typical time to reach equilibrium is approximately 3 hours.

4.2.1 Study cases

The cases selected to study include four different power levels; the nominal power expected in UTOPIA (case *a*); 1.5 times the nominal power as a safety margin (case *b*); the nominal power without the EoP to see its effect on the thermal performance (case *c*) and warm running for assembly tests and commissioning (case *d*). The indicated power in case *a* is the total power expected in the petal due to ASICs and silicon at the end of the experiment operation at an integrated luminosity of 3000 fb⁻¹.

Table 4.7: Study cases consisting of different evaporation temperatures and power levels.

Case	T	Power
	[°C]	[W]
<i>a</i>	-30	76
<i>b</i>	-30	114
<i>c</i>	-35	70.6
<i>d</i>	20	76

4.2.2 Cooling system parameters

The design parameters of each of the study cases in table (4.7) are defined. These include the mass flow (m), evaporation temperature (T_{evap}), nominal power (P) and vapor quality (x).

The vapor quality at the exit of the petal is chosen to be around 0.4-0.6, safely away from dryout. Table (4.7) gives the coolant temperature and power. The mass flow is calculated using eq.(2.29) in order to achieve the desired vapor quality at a specific power. Table (4.8) summarizes the mass flow and the difference in the latent heat of evaporation between inlet and outlet.

Table 4.8: Mass flow and latent heat of evaporation for the different case studies for a vapor quality x between 0.4-0.6 at the exit of the petal.

Case	Latent heat of evaporation	Mass flow
	[kJ/kg]	[g/s]
<i>a</i>	182.09	0.45-0.5
<i>b</i>	182.09	0.60-0.75
<i>c</i>	187.91	0.35-0.45
<i>d</i>	91.2	1.30

4.2.3 Example of measuring heat exchange with surroundings

The heat exchange with the surroundings is examined prior to the measurements of the study cases presented in table (4.7). Heat flows into the petal from the surroundings and via the petal into the coolant. When there is a temperature difference between the petal and the surroundings during mono-phase cooling, the coolant temperature rises with enthalpy so that from mass flow, heat capacity and temperature rise, the heat leak from the surroundings can be estimated. It is assumed that there is no heat leak when the petal is at room temperature.

Tests are conducted in order to estimate the environmental heat leak. The petal is placed inside the insulation box and tested. In addition to the sensors placed on the petal shown in figure (4.4), some extra sensors were added inside the insulation box near the petal. Figure (4.8) shows the position of the extra sensors.

Figure 4.8: Position of the additional sensors used during the mono-phase cooling test to determine the parasitic heat load.

Figure (4.9) shows the temperature measurements during the heat leak test at a mass flow of 0.5 g/s. The TT367 is the environmental temperature and the readings labeled as CO₂ in SAT and CO₂ out SAT are the saturated temperatures calculated from the measured pressures. The accumulator was

set at 20 °C. The heater was used this time as a cooler and set to a lower temperature to provide sub-cooled liquid CO₂ in the petal. Initially, the petal was at room temperature. As soon as the cooling was switched on, the petal started cooling down. The CO₂ at the inlet was at -20 °C and the parasitic heat increased the temperature of the CO₂ along the length of the cooling tube. The temperature at the outlet is around -7 °C and still in the liquid phase. In section (4.3) the heat leak during this measurements is estimated.

Figure 4.9: Temperature evolution during the single-phase cooling test to determine the heat leak to the petal at a mass flow of 0.5 g/s and coolant temperature -20°C. There is 6 to 7°C increase in the CO₂ temperature from inlet to outlet.

4.2.4 Example of a typical measurement: nominal power (case *a*)

Figure (4.10) shows the evolution of measurements (mass flow, heater power, coolant temperatures, pressure drop along the test section and silicon temperatures) during the day for case *a*. Other cases are similar. The accumulator was tuned to give -30 °C. Cooling was turned on at 9:00 am. At 9:30 am the mass flow was set to 2.5 g/s in order to achieve fast cooling of the petal. Power was turned on at 11:00 am, tripped at 11:15 due to a short and was switched on at 11:25. At 11:50 the mass flow was set to 0.5 g/s in order to achieve a vapor quality x in the range of 0.5-0.6. Steady state conditions in the silicon temperatures were reached after approximately 3 hours. The measurements are taken at around 13:00.

The temperature of the CO₂ at inlet and outlet follows the fluctuations of the pressure due to the two-phase region. The CO₂ temperature at the outlet is lower than at the inlet, despite the heat flux applied. In the two-phase flow

the heat is used to change the phase of the medium and not to increase the temperature as it happens in mono-phase flow. Any temperature drops are related solely to the pressure drop.

Switching on the power at 11:00, shows an immediate reaction of the temperature sensors. The pressure drop increases due to two-phase flow in the petal and possibly due to a change of the flow pattern regime. Since the pressure at the outlet is fixed by the accumulator, any pressure drop is seen as a rise in pressure at the inlet, in turn causing the CO₂ temperature at the inlet to increase. Generally, the power fluctuations affect rapidly the CO₂ temperatures. For example, during the short time of the power disruption the CO₂ temperature show some peaks of a few degrees. When the power is set to nominal and the mass flow is reduced at 11:25, the CO₂ inlet reaches a lower value due to a lower pressure drop. Later, the steady state measurements will be used to validate the prediction in chapter 5.

Figure 4.10: Mass flow, power, pressure drop, CO₂ temperatures at inlet and outlet and silicon temperatures for case *a* at evaporation temperature -30 °C. Steady state conditions are reached after 3 hours from the last adjustment of the parameters. At the steady state conditions most of the temperatures are close to each other except 91 and 92 that are closer to the EoP heater.

4.3 Estimating heat leak from measurements

Heat leak towards the petal takes place when the petal is cooler than the surroundings. The heat leak is examined using liquid CO₂ at various inlet temperatures and mass flows. An example of those measurements was presented in section (4.2.3). In the following section, the heat leak is estimated based on the measurements. Eq.(2.10) is used to estimate the heat leak. The coolant heat capacity c_p is calculated for the different inlet and outlet temperatures of the CO₂ using software REFPROP from NIST and the mean heat capacity c_p is used in Eq.(2.10). Table (4.9) summarizes the measured parameters utilized to calculate the heat leak. Case 1 of table (4.9) is the one discussed in section (4.2.3). Table (4.10) shows the heat leak calculated from measurements for the different cases (1-4).

Table 4.9: Parameters used for the calculation of the heat leak. The mass flows and the coolant temperatures at inlet and outlet are measured values. The specific heat (c_p) is estimated based on the coolant temperatures and the pressure.

case	mass flow [g/s]	CO ₂ inlet [°C]	CO ₂ outlet [°C]	c_p inlet [J/kg K]	c_p outlet [J/kg K]
1	0.5	-19.4	-7.3	2077	2214
2	1.0	-16.0	-9.2	2108	2187
3	1.0	-25.8	-16.7	2029	2101
4	1.0	-21.3	-13.5	2064	2134

Table 4.10: Heat leak from the ambient environment to the CO₂ estimated from the single-phase cooling measurements.

case	measured heat load [W]
1	14
2	15
3	19
4	16

It is assumed that a mean silicon temperature from the measured ones is representative of the overall petal temperature. Therefore, the mean silicon temperature from measurements is plotted against the estimated heat leak in table (4.10). An extra point is added for the case where the mean silicon temperature equals the room temperature and then the heat leak should be zero. A straight line fit is used to obtain the heat leak expected at different temperatures.

Figure 4.11: Heat leak estimation for lower coolant temperatures based on a linear extrapolation of the measured heat leaks when the heaters are off.

From the fit, it can be concluded that the expected heat leak for a petal with a mean silicon temperature of $-20\text{ }^{\circ}\text{C}$ is 20 W. This heat leak is much smaller than the heater power but still significant.

4.4 Discussion of results for the study cases

The thermal performance of the prototype petal was measured at various conditions. The measurements for the different study cases in table (4.7) are presented in table (4.11) for side A of the petal. Side B gives similar results and therefore isn't presented here separately. The labeling in table (4.11) is according to figure (4.4).

Generally, the results show that the areas near the petal "ears" have a long heat path to the coolant giving high measured silicon temperatures at the locations 91A and 92A of figure (4.4). On the other hand, the silicon temperatures are uniform within a maximum range of $4\text{ }^{\circ}\text{C}$ when the heaters in the ears are switched off. Looking at the results, the following can be summarized:

The areas close to the EoP are not cooled adequately in this design. With the EoP heaters on at nominal, the maximum temperature difference (ΔT_{max}) between the CO_2 and silicon is up to $17\text{ }^{\circ}\text{C}$, whereas the mean temperature difference (ΔT_{mean}) between the CO_2 and silicon is around $9\text{ }^{\circ}\text{C}$. Switching off of the EoP brings the ΔT_{mean} down to $7.5\text{ }^{\circ}\text{C}$ and the ΔT_{max} down to $11\text{ }^{\circ}\text{C}$. Despite the $5\text{ }^{\circ}\text{C}$ difference in the CO_2 between nominal (case a) and nominal

without EoP (case *c*), switching off the EoP shrinks the temperature range spreading on the silicon and minimizes the hot spots. The difference in power between these two cases gives 2-3 °C lower values on the mean temperatures. With the EoP heaters on at 50% above the nominal, the ΔT_{mean} rises to 11°C. Finally, at the warm running, some temperatures reach 27°C. The glue below the ASICs in the real detector will give much higher temperatures and therefore care should be taken during warm running to avoid overheating of the chips.

Table 4.11: Measured temperatures for top side (side A) of the petal for the four study cases. The ΔT_{mean} gives an indication of the general performance while the ΔT_{max} raises concerns on the possible issues with the ASICs.

Power case:	a) Nominal	b) Nominal + 50%	c) Nominal no EoP	d) Warm
CO ₂ inlet	-30.0	-29.9	-34.9	19.8
CO ₂ outlet	-30.2	-30.3	-35.3	19.8
11A	-22.6	-20.3	-27.8	24.8
12A	-24.0	-22.0	-29.2	24.6
13A	-23.7	-21.9	-29.0	24.8
21A	-25.0	-23.4	-30.2	23.9
22A	-23.5	-21.5	-28.7	25.5
23A	-20.9	-18.6	-26.6	26.1
31A	-23.2	-20.9	-28.9	24.4
32A	-23.8	-22.0	-28.2	25.8
41A	-23.3	-21.3	-29.8	23.7
42A	-24.5	-23.1	-28.6	25.5
51A	-24.7	-22.6	-29.9	24.8
52A	-22.6	-19.9	-27.7	27.0
61A	-21.7	-19.3	-28.6	24.9
62A	-22.7	-20.7	-27.4	26.5
71A	-19.6	-16.1	-25.8	28.6
72A	-19.8	-16.6	-27.5	26.6
81A	-17.1	-13.6	-25.6	26.3
82A	-18.5	-15.7	-23.9	28.5
91A	-14.6	-10.5	-25.6	27.0
92A	-12.7	-7.8	-23.8	29.3
f1A	-19.0	-15.1	-27.8	20.4
f2A	1.3	7.9	-27.8	21.6
f3A	-9.2	-4.4	-27.8	21.3
f4A	-24.0	-22.1	-27.8	20.5
f5A	-15.5	-12.6	-27.8	21.2
f6A	-22.3	-19.9	-27.8	20.2
f7A	-21.3	-18.5	-27.8	20.1
f8A	-21.5	-19.2	-27.8	20.8
T_{mean}	-21.4	-18.9	-27.6	25.9
T_{max}	-12.7	-7.8	-23.8	29.3
$\Delta T (T_{mean} - CO_2)$	8.8	11.4	7.6	6.1
$\Delta T (T_{max} - CO_2)$	17.5	22.4	11.5	8.9

The nominal cases *a* and *d* of table (4.7) examine the same power but at

different temperatures. It is expected to have the same ΔT_{mean} effect with respect to the coolant. However, no such effect was seen between the warm and cold test. This can be explained by the fact that the flow regime and therefore the heat transfer coefficient differ on the two cases according to the CoBra predictions in section 5.1. A smaller diameter cooling pipe could bring the flow regime into annular and give better HTC during normal operation. Also a smaller effect on the higher ΔT_{mean} values is expected on the cold test due to the extra heat load from the environment compare to the negligible heat load in the warm test. For the same reason the temperatures measured on the facesheets for the warm cases differ from the cold cases due to the different direction in the heat transfer between ambient environment and the petal. The measured temperatures will be compared later with FEA results. The results allow good verification of a FEA model, which can then be used to improve reliably the petal design without the need of building new prototypes for every change in the design.

Chapter 5

Thermal model of the prototype petal

This chapter presents the Finite Element Analysis (FEA) thermal model of the petal. The FEA results predict the temperature distribution over the sensors which is a result of the heat input and the total thermal resistance between the coolant and the heating sources. The thermal resistance is the sum of the convective term for CO₂ and the conductive term for the mechanical structure. The chapter starts with the ‘CoBra’ results. Then it describes the FEA procedure and presents the input parameters. It then compares the FEA results with the measurements presented in chapter 4, to validate the FEA model.

5.1 CoBra CO₂ calculations

The CO₂ parameters that are used as input in the FEA are estimated using CoBra [28]. CoBra is a MATLAB [46] based program that solves the one dimensional flow problem using the relevant empirical models and predicts the behavior of CO₂ in mini-channels described in chapter 2. The CoBra procedure is described in ref. [28].

CoBra gets as input the tube (diameter and length) and the operation parameters (heat load, mass flow, pressure and coolant temperature at entrance). It divides the tube in small sections and estimates for each section the thermodynamic conditions (pressure, enthalpy and temperature). The mass flow is constant while the heat load is spread uniformly along the tube. The program predicts the flow regime and calculates the pressure drop and heat transfer coefficient along the tube depending on the flow regime. The CO₂ thermodynamical properties (enthalpy, viscosity, specific heat, surface tension etc.) are calculated using REFPROP [47] from NIST. Table (5.1) summarizes the input parameter used in CoBra. The pressure is derived from the saturated temperature (T_{sat}) given in table (5.1). The inner diameter and the tube length used

in CoBra as input are found in table (4.2).

Table 5.1: Summary of the input parameters for CoBra.

Case	T_{sat} [°C]	Mass flow kg/s	Heat load [W]
<i>a</i>	-30	0.45	76
<i>b</i>	-30	0.70	114
<i>c</i>	-35	0.40	70.6
<i>d</i>	19.80	1.30	76

5.1.1 Nominal power (case *a*)

The CoBra results for case *a* of table (4.7) at evaporation temperature -30°C are summarized below. Figure (5.1) shows the coolant temperature and pressure along the tube length. The coolant temperature decreases by 0.3°C due to the pressure drop along the evaporator.

Figure 5.1: CoBra results for case *a* of table (4.7). The temperature of the fluid along the tube and the pressure along the evaporator. There is a small pressure drop along the evaporator and therefore a small temperature decrease in the coolant from inlet to outlet.

Figure (5.2) shows the heat transfer coefficient along the tube length and the vapor quality. The sharp changes in the heat transfer coefficient correspond

Figure 5.2: CoBra results for case *a* of table (4.7). The heat transfer coefficient (HTC) and vapor quality along the evaporator. There is an initial reduction in the heat transfer coefficient in the first section of the evaporator and then an increase up to 6500 W/m²K. The vapor quality reaches 0.5 at the exit off the evaporator due to the heating.

to the transition from the intermittent to the stratified-wavy regime. Within the stratified-wavy regime there is another smaller change in the heat transfer coefficient behavior due to the two terms h_v and h_{wet} in the calculation of the heat transfer coefficient in ref. [2]. Figure (5.3) presents the flow pattern map. The flow regime map indicates the transition lines from one regime to the other depending on the flow properties. The flow properties are expressed in vapor quality and mass flux. The red lines indicate the transition to the dry-out or mist flow regime which is considered undesired. The green lines shows the results for case *a*. The flow regime switches from intermittent to stratified-wavy. The heat transfer coefficient change seen in figure (5.2) is due to the switching from one flow regime to the other. Cases *b* and *c* of table (4.7) give similar results.

Figure 5.3: The flow pattern map with varying vapor quality as it is predicted by CoBra for case *a*. The blue lines indicate different regimes, whereas the red lines indicate the dry-out regime as predicted by the theory [1], [2]. The green line indicates the measurements. Particularly, it shows that during this evaporation process the flow regime changes from intermittent to stratified-wavy.

5.1.2 Nominal power during commissioning (case *d*)

During commissioning the detector will operate at a relatively higher coolant temperature of approximately 20°C. There is a slightly increasing temperature at the entrance because the first few millimeters in the evaporator are in single-phase flow. As soon as the two-phase flow takes place the temperature decreases due to the pressure drop. For the conditions given in case *d* of table (4.7), the CoBra results are summarized below. Figure (5.4) shows the coolant temperature and pressure along the tube length. The coolant temperature decreases by 0.1 °C due to the pressure drop along the evaporator. Figure (5.5) shows the vapor quality increasing from 0 to 0.4 due to the heating. The heat transfer coefficient stays quite constant at around 13,000 W/m²K.

Figure (5.6) presents the flow pattern map. During the warm test the regime moves from intermittent to annular. The heat transfer coefficient is quite high around 13,000 W/m² due to the intermittent and annular flow.

Figure 5.4: CoBra results for case *d* of table (4.7). There is a small pressure drop along the evaporator therefore a small temperature decrease in the coolant from inlet to outlet.

Figure 5.5: CoBra results for case *d* of table (4.7). The heat transfer coefficient is around 13,000 W/m²k. The vapor quality reaches around 0.4 at the exit of the evaporator due to the heating. The initial sharp rise in heat transfer coefficient is due to single-phase cooling at the start.

Figure 5.6: The flow pattern map with varying vapor quality as it is predicted by CoBra for case *d*. The blue lines indicate the various regimes, whereas the red lines indicate the dry-out /mist regimes as predicted by the theory [1], [2]. The green line indicates the measurements.

5.2 FEA Thermal model

An FEA thermal model is built using Abaqus software [48] to simulate the thermal behavior of the prototype petal. The FEA procedure is described in this section. The FEA includes a pre-processing and a post-processing part. The pre-processing part includes the design of the geometry, the material definition, the meshing, the contact definitions and boundary conditions. An uncoupled thermal analysis is selected that focuses on mainly solid heat conduction with temperature dependent conductivity and convection. The post-processing part includes mainly the visualization of the results.

5.2.1 Geometry

The first step in the pre-processing analysis is the modeling of the geometry. The FEA model includes all the parts that have been reported in the previous sections. The mechanical structure is modeled with solid parts of which the properties can be isotropic or orthotropic. The honeycomb is modeled as isotropic assuming the main heat transfer contribution is via air. Each layer of glue was modeled separately as a solid part. The heater components like copper, polyimide layers and glue are modeled as well. The CO_2 is defined in

Abaqus through a film condition interaction [49]. Table (5.2) gives a summary of the material characteristics. Figure (5.7) gives the petal geometry built in Abaqus based on the prototype.

Table 5.2: Geometrical properties for the components of the petal thermal model.

Component	Part	Properties
Heater	Solid	Isotropic
Glue heater	Solid	Isotropic
Sensor	Solid	Isotropic
Glue Sensor-FC	Solid	Isotropic
Facesheet	Solid	Orthotropic
Core	Solid	Isotropic
Carbon foam	Solid	Orthotropic
Closeouts	Solid	Isotropic
Tube	Solid	Isotropic

Figure 5.7: Geometry built in Abaqus for the prototype petal. Flex heaters are colored red and the heaters for the EoP are colored blue.

Figure 5.8: View of honeycomb core material (green) and carbon-foam (khaki).

Figure (5.8) shows the modeling of the honeycomb core material with green color and the carbon foam with khaki color. There is an empty space at the end of the petal where there is no material present. Figure (5.9) shows the facesheets placed on top of the honeycomb and carbon foam, whereas figure (5.10) shows the heaters and the closeouts.

Figure 5.9: FEA geometry of the petal with the facesheets (beige).

Figure 5.10: FEA geometry of the petal with the heaters (red), silicon wafers (grey) and closeouts. (blue)

5.2.2 Material properties

The next step in an FEA is the material assignment for each part. An uncoupled thermal analysis requires the definition of material properties such as thermal conductivity. Most of the properties are based on the manufacturer's data-sheet. There are some properties based on the measurements from previous studies for the Stave, which is the equivalent detector in the Barrel of the Silicon Strip Tracker [50]. Carbon foam and carbon fiber are defined as orthotropic, whereas honeycomb is defined as isotropic, assuming it consists mainly of air. The geometrical parameters defined for the thermal model are summarized in table (5.3).

Table 5.3: Thermal conductivity (at 20°C) of the materials used and the thickness of each component for the petal thermal model. The heater parts are modeled separately. The x coordinate is in the thickness direction, y coordinate is in the longitudinal direction and z coordinate is the vertical direction.

Part	Material	Thermal conductivity	Thickness
		X/Y/Z W/mK	mm
Heater	Polyimide-Copper-Polyimide	0.46-385-0.46	0.135 *total
Glue heater	Acrylic PSA	0.14	0.05
Sensor	Silicon	148	0.32
Glue Si-Fc	Dow Corning 340	1.26	0.06
Facesheet (Fc)	prepreg RS3 resin and carbon fiber K13D2U	1.3/294/148	0.20
Core	Honeycomb-air	0.25	5.00
Carbon foam	Pocofoam HTC	235/70/70	5.00
Closeouts	M55j fiber	156	0.25
Tube	Titanium T40	16.4	0.15
Glue	Aradilte AW 103-1/HY 991	0.22	varying

5.2.3 Mesh

The next step is the discretization of the geometry into finite elements, known as ‘mesh elements’. The geometry is meshed with diffusive heat transfer elements with the temperature the only degree of freedom. The diffusive heat transfer elements in Abaqus/Standard can use either first order (linear) or second order (quadratic) interpolation. It has only three dimensional diffusive heat transfer elements. Each part has its own mesh size and type and depends on the geometry itself.

The naming of the element type in Abaqus/Standard gives a description of the family, degrees of freedom, number of nodes, formulation and integration of the elements. A typical diffusive heat transfer element is a DC3D8. The letter D tells that it is a diffusive element, the letter C gives the family type, which in this case is a solid element and the number 3 tells it is 3-dimensional element. The last part D8 gives information about the integration and the nodes. In this case, the element has eight nodes at its corners and uses linear integration. The mesh properties for each part are tabulated in table (5.4). The mesh size is proportional to the component size and complexity. Critical parts are modeled with finer mesh. For example, the tube is modeled with finer mesh to give more accurate results.

Table 5.4: Mesh properties for petal thermal model.

Part	Element type	Element shape	Element max size mm
Heater-Glue	Heat Transfer DC3D4	Linear tetrahedron	2.5
Sensor (Si)	Heat Transfer DC3D4	Linear tetrahedron	2.5
Glue Si-Fc	Heat Transfer DC3D8	Linear hexahedron	4
Facesheet (Fc)	Heat Transfer DC3D8	Linear hexahedron	4
Core	Heat Transfer DC3D8	Linear hexahedron	1.5
Carbon foam	Heat Transfer DC3D8	Linear hexahedron	1.5
Closeouts	Heat Transfer DC3D8	Linear hexahedron	1
Tube	Heat Transfer DC3D8	Linear hexahedron	0.03

5.2.4 Boundary conditions

The boundary conditions are as follows:

Prescribed surface heat flux

The production of heat from the heaters is modeled by defining a surface heat flux. Table (5.5) shows the heat input for the different sets of hybrids and sensors.

Prescribed temperature

The temperature of the CO₂ is fixed over time via the film condition thermal interaction module. The temperature is varying though along the tube. The assigned values are based on the measured values at the inlet and outlet and on the simulated CoBra results that give the evolution along the length for each case. It is assumed that the heat load is uniformly distributed for now. In chapter 6, a non-uniform heat load is given as input to CoBra which gives 1-2 °C discrepancy with the uniform case. The ambient temperature is also defined via this interaction. The ambient temperature is set to the value obtained from the measurements.

Table 5.5: Heat input for thermal FEA model for UTOPIA prototype for the different power cases.

Heater type	Case a	Case b	Case c	Case d
	nominal W/m ²	above nominal W/m ²	no EoP W/m ²	no EoP W/m ²
Hybrid 1A	3538	4121	3445	3457
Hybrid 1B	4515	4615	3859	4430
Sensor 1	86	134	76	97
Hybrid 2A	3887	4548	3808	3812
Hybrid 2B	3863	4537	3800	3790
Sensor 2	67	105	59	76
Hybrid 3	4820	5660	4730	4750
Sensor 3	54	86	47	61
Hybrid 4	4240	4800	4070	3965
Hybrid 5	4240	5100	4260	4020
Sensor 4	49	76	42	54
Sensor 5	49	78	43	55
Hybrid 6	4350	5075	4240	4250
Hybrid 7	4350	5210	4340	4365
Sensor 6	43	68	37	48
Sensor 7	43	69	38	49
Hybrid 8	4680	5490	4560	4440
Hybrid 9	4680	5620	4600	4560
Sensor 8	38	59	33	42
Sensor 9	38	59	33	42

5.2.5 Heat transfer coefficient between components

Convection

Convection is defined via the film condition thermal interaction module. The film condition interaction assigns on a surface a film coefficient (heat transfer coefficient).

To simulate the convection from the inner tube wall to the coolant, a heat

transfer coefficient is defined. The heat transfer coefficient for the CO₂ is calculated from CoBra as presented in section 5.1. In a similar way, to simulate the convection from the ambient environment to the petal in the box, the heat transfer is modeled with a film condition interaction. Based on the measured heat leak, the heat transfer coefficient is set to 2-3 W/m²K based on figure (4.11).

Gap conductance

The conductivity heat transfer between two contact surfaces is defined by the gap conductance assuming perfect contact between the surfaces (i.e. there is no temperature gradient between the contact nodes from one surface to the other).

Radiative heat exchange with the environment is not considered in this analysis.

5.2.6 Step module

A steady-state analysis is selected. Automatic incrementation is selected. Information on the incrementation is shown in table (5.6). The equation solver is set to direct. The solution technique uses the full Newton method. Fluxes and boundary conditions ramp linearly with time during a step, reaching the total magnitude at the end of the time interval. The starting condition at each increment is derived by extrapolating the previous stage.

Table 5.6: Incrementation parameters in heat transfer step steady state analysis

Type	Max No increments	Minimum Increment size
Automatic	100	1e-5

5.2.7 Output

The requested outputs are temperature (TEMP) at integration points, nodal temperatures (NT11) and components of the heat flux vector (HFL).

5.3 Comparison FEA prediction with measurements

The comparison of the FEA results with experimental measurements is presented. Firstly, the comparison during the single-phase cooling is summarized and followed by the comparison for the two-phase cooling. All cases of table (4.7) are considered for validation of the FEA model. Results are presented for side A only for the two-phase cases. Side B is similar to side A (see figure (4.4)).

5.3.1 Heat exchange with the surroundings

As a first step, an FEA model was built to verify the modeling of the resistance between the ambient environment and the petal using a heat transfer coefficient h_{air} . The value of h_{air} is tuned to give the measured heat leak based on figure (4.11). The temperature distribution has been measured on the carbon fiber facesheets and on the silicon wafers. The position of the measured points is shown in figure (4.4). Figures (5.11) and (5.12) show the comparison for the third case in table (4.9) when liquid CO_2 enters the petal at -25.8°C , for sides A and B. The limits in FEA are determined by a $1\text{-}2^\circ\text{C}$ spread due to the uncertainty of the heat convection and possible uncertainties on the measured locations.

Figure (5.11) shows that the FEA and measured values in Side A of the petal give a uniform temperature distribution within a spread range of $1\text{-}2^\circ\text{C}$. There are a few measured points higher than the rest (1.3 and 3.1). The reason can be that there is not very good thermal contact of the sensors with the silicon. Similarly, in figure (5.12) the overall temperature distributions measured and predicted for side B are within a spread range of $1\text{-}2^\circ\text{C}$. There are a few measured points higher than the rest (9.1, 9.2, 1.1 and 1.2) due to the poor thermal contact or because they are in contact with the box.

In general, the FEA results show the same trend for both sides; the temperature from the inlet towards the outlet drops in the sections right after and before the inlet and outlet and on the bend of the tube (5, 7 1, 4 and 6). These points are closer to the tube therefore are cooled better.

Despite the temperature difference between the inlet and outlet coolant temperature due to the heat leaks, the conductive heat transfer and the silicon establish a fairly uniform temperature distribution. In general, the comparison shows that the air convection is modeled within an acceptable range. Moreover, the heat flux that enters the tube derived from the FEA results is in close agreement with the measured heat leak values. Table (5.7) shows the results of the tuning of the h_{air} as described in section 5.2.5.

Figure 5.11: Comparison of the FEA results and measurements at side A with CO_2 inlet temperature -25.8°C without any power added from the heaters (case 3 in table (4.9)).

Figure 5.12: Comparison of the FEA results and measurements at side B with CO_2 inlet temperature -25.8°C without any power added from the heaters (case 3 in table (4.9)).

Table 5.7: Comparison of the FEA predicted and measured total heat flux on the tube due to the ambient heat leaks during single-phase cooling for the cases presented in table (4.9).

Case	Measured heat load	Predicted heat load
	W/mm ²	W/mm ²
2	0.0018	0.0019
3	0.0022	0.0028
4	0.0019	0.0023

5.3.2 Nominal power (case *a*)

An FEA thermal model of the petal during cold operation at nominal power is simulated according to the parameters described above. The heat transfer coefficient of the CO₂ to the tube is determined using CoBra model and is presented in figure (5.2).

Figure (5.13) presents the temperature of the silicon at the integration points (TEMP). In comparison with the case (5.3.1) we now have two-phase cooling and the temperature is established by the evaporation of the liquid phase. As a result the petal has a quite uniform temperature with a large portion at around -25 °C. In order to compare the FEA results with the measurements,

Figure 5.13: FEA silicon temperature results of side A for case *a*. The distribution is rather uniform apart from those close to the DC/DC converters heaters at the facesheets. Also at the outer ring the silicon sensor is warmer at the sides close to the EoP. A smaller effect is seen on the opposite side of the EoP where the temperatures are affected by the EoP on the other side.

again the same points were considered as shown in figure (4.4). Figure (5.14) shows the comparison of the FEA results with the measurements at side A for

the case where the coolant temperature at the inlet is $-30\text{ }^{\circ}\text{C}$ and the power is set at 76 W . Due to the very small pressure drop along the tube as seen in figure (5.1), the temperature of the coolant can be assumed fairly constant along the tube. The measured and predicted values show the same trend. The temperature close to the inlet and outlet are higher than the rest temperatures on the petal. The reason is that these points are further away from the cooling tube. The temperatures measured close to the inlet (9.1, 9.2) are even higher due to the presence of the EoP heater. In general, there is good agreement between the measurements and the simulation. A discrepancy at the points close to the inlet (9.1, 9.2) are due to the not accurate modeling of the EoP heater.

Figure (5.15) shows the ratio of difference between simulated silicon temperatures and coolant temperature compared to difference between measured silicon temperature and coolant temperature. This is shown for both sides A and B. Ratios close to 1 indicate good agreement within $\pm 10\%$. The EoP region is modeled less well and is not included in figures (5.13) - (5.15).

Figure 5.14: Comparison of the FEA results and measurements for case *a*. Majority of the points fall within the error range while differences exist near the edges.

Figure 5.15: Temperature difference ratio of FEA and measured values for evaporation temperature of $-30\text{ }^{\circ}\text{C}$ at nominal power. The mean ratio is around 1.05 for both side A and B.

5.3.3 Above nominal power (case *b*)

The same FEA model is used to evaluate case *b* where the power is adjusted to 50% above nominal. This case gives an estimation on whether the petal can be still adequately cooled and operate if an increased power above the nominal occurs. Figure (5.16) shows that the silicon temperatures. The same trend is seen as in case *a*, with increased temperatures at the sides of the outermost sensors at Ring 5 close to the EoP.

Figure 5.16: FEA silicon temperature distribution for case *b* with coolant inlet temperature -30°C at 50 % above nominal power (side A).

Figure 5.17: Comparison of the FEA results and measurements for coolant inlet temperature -30°C at 50 % above nominal power.

Figure (5.17) presents the comparison of the FEA results with the measurements for the case where the coolant temperature is -30°C and the power is

Figure 5.18: Temperature difference ratio between the FEA and the measured results for evaporation temperature -30°C at 50 % above nominal power. The mean ratio is on average 0.90 for both sides.

set at 114 W. The temperature close to the inlet and outlet are higher than the rest temperatures on the petal as seen in section 5.3.3. The measured and predicted values show the same profile. Apart from the values close to the EoP (9.1, 9.2) there is good agreement between the measurements and the simulation. Figure (5.18) shows the ratio of difference between simulated silicon temperatures and coolant temperature compared to difference between measured silicon temperature and coolant temperature; ratios close to 1 indicate good agreement. The same trend is seen as in figure (5.15).

5.3.4 Nominal power without EoP (case c)

This time the EoP heaters are off, the coolant temperature is -35°C and the power is set to 70.6 W. Figure (5.19) shows the temperature distribution at the silicon. Note that the temperature scale is far more detailed than in figures (5.13) and (5.16). With the EoP heaters off, the hot spots are seen now on both sides on the points that are further from the cooling tube.

Figure (5.20) presents the comparison of the FEA and the measurements. There is better temperature uniformity compared to the figures (5.15) and (5.17) due to the fact that the EoP heaters are off. The points closer to the EoP heater are only $+5^{\circ}\text{C}$ hotter than rest, whereas in figures (5.15) and (5.17) there was a $+15^{\circ}\text{C}$ difference between these points and the rest. In general, measured and predicted temperatures follow the same profile and agree within the error range. Figure (5.21) shows the ratio of difference between simulated silicon temperatures and coolant temperature compared to difference between

measured silicon temperature and coolant temperature.

Figure 5.19: FEA silicon temperature results for coolant temperature -35°C at nominal power without EoP heaters (side A).

Figure 5.20: Comparison between FEA predictions and measurements for evaporation temperature -35°C at nominal power without EoP heaters.

Figure 5.21: Temperature difference ratio between the FEA and the measured results for an evaporation temperature of -35°C at nominal power without EoP heaters. The mean ratio is 1.07 for both side A and B.

5.3.5 Nominal power during commissioning (case *d*)

Case *d* of table (4.7) is studied at an evaporation temperature of 20°C and at nominal power. Figure (5.22) shows the temperature distribution at the silicon. The hot spots are seen on the points further away from the cooling tube and not close to the EoP as seen in figure (5.13). The relatively high temperatures in the coolant and on these points cover the effect of the EoP heater.

Figure (5.23) shows the comparison between FEA prediction and measure-

Figure 5.22: Prediction of the silicon temperature results case *d* with evaporation temperature of 20°C at nominal power (side A).

ments. There is better temperature uniformity compared to the figures (5.15) and (5.17) mainly because of the high heat transfer coefficient compared to

the other cases. In general, measured and predicted temperatures follow the same profile and agree within the error range. Only the points 4.1 and 6.2 that are on the same side of the petal are outside the range. These points may not be adequately cooled by air convection. Also, due to the relatively high heat transfer coefficient in the commission case, the error range in the FEA is narrower.

Figure 5.23: Comparison between FEA prediction and measurements for a coolant temperature of 20 °C at nominal power without EoP heaters.

Figure 5.24: Difference ratio between FEA and measured temperatures for an evaporation temperature of 20 °C at nominal power without EoP heaters. The mean ratio is 1.05 and 1.15 for side A and B respectively.

Figure (5.24) shows the ratio of difference between simulated silicon temper-

atures and coolant temperature compared to difference between measured silicon temperature and coolant temperature; ratios close to 1 indicate good agreement. It can be seen that the two sides have different results. This can be due to the different heat transfer coefficients for the air convection inside the box for side A and side B. Side A agrees closer with the measurement compared to side B.

5.4 Discussion

Different cases were examined to validate the FEA thermal model. The comparison between measurements and FEA predictions shows good agreement except for the edges where the EoP is located and especially when the EoP heaters are switched on. In some cases the FEA model expects the petal to be warmer than the measured temperature. In reality there is considerable heat transfer from the EoP to the air inside the box. The examined design of the EoP with a simple resistor is not realistic giving poor prediction of the EoP. However, away from the EoP or without the EoP, the model predicts temperatures very well. The differences are within the error range, and error for the ratio of the measured to predicted temperature difference above the coolant temperature is about 10%, ie the FEA correctly predicts ΔT at the 10% level.

The validated model can therefore be applied to study the behavior of the detector in new designs.

Chapter 6

Prediction model for latest design

Chapters (4) and (5) discussed the cooling performance and the thermal modeling of a petal prototype which was based on an early petal design. The outcome was a verified thermal model by comparison with measurements. In this chapter, the model is adapted to the latest layout design, called ITk layout. The petal is modeled now including ASICs and hybrids, whereas the silicon sensors are modeled with a temperature dependent self-heating. The tube characteristics are similar to that of the prototype petal discussed in chapter 4, therefore, CoBra is used confidently here to characterize the flow characteristics. The aim of this chapter is to examine whether in the ITk layout design the petal will be adequately cooled during two important time periods of its lifetime; at the TID ‘bump’ and at the end-of-life. It was also investigated whether the petal at the end-of-life will be safe from thermal runaway. Recommendations for improvement are discussed at the end of the chapter.

6.1 The ITk layout design

At the time of this work, the ITk layout was evolving with a few different pixel layouts, while the strip layout was fixed and known as ITk layout. Figure (6.1) shows the strip layout along with one of the under consideration pixel layouts. The Pixel Tracker is in blue and the Strip tracker is in red. The Strip Tracker has four barrel layers and six disks in each End-cap. The Barrel for both Pixel and Strip Tracker is the horizontally positioned lines and the Endcaps the vertically positioned lines. In the thesis the focus is mainly on the Endcap where the petal is located. In chapter 1 the geometrical definitions of the detector were summarized.

Figure 6.1: Quarter segment of the Inner Tracker layout [51] consisting of the Pixel and Strip Trackers.

Figure 6.2: Petal CAD drawing from DESY showing the length and width of the petal. The silicon is shown in blue that covers most of the petal area while the ASICs are shown as white.

6.1.1 Geometry

As far as the petal is concerned, the main changes from the LoI/UTOPIA design are that the petal becomes longer, has only one ‘ear’ to hold the EoP board, and the DC-DC conversion components have moved from the facesheets to the silicon sensors. Dimensions and numbers of the readout ASICs (ABCs and HCCs) have also changed. The dimensions of the ABCstar and HCCstar ASICs discussed in chapter 3 are used in this chapter.

The local support includes wider core structure and facesheets in order to fit the larger sensors compared to the first design. The cooling tube is arranged in a V-shape and passes below the central middle region of sensors while entering and exiting in the EoP area. The geometry and dimensions of each component in the local support can be found in [52]. Figure (6.2) shows a CAD drawing of the petal including the ASICs, hybrids and DC/DC converters. The readout ASICs (white) are sitting on the hybrids (green) while the DC/DC converters (orange) are placed also on the silicon sensors (blue).

6.1.2 Sensors

The geometrical information and the predictions of the radiation damage of the sensors are given. Table (6.1) summarizes the sensors geometry in the latest petal design. According to figure (6.2) and table (6.1) the inner two rings and fourth ring (R0, R1 and R3) have two modules. Compared to the previous layout (table (3.3)) the sensors in ring 3 to 5 in the new layout are rotated by a few rads. More information on the layout can be found in [15].

Table 6.1: Sensor area , position and rotation for latest design; R_i and R_o are the radii of the inner side and outer side from the beam line, R is the translation along the sensor strip direction from the beam line according to the latest layout and θ the rotation of the sensor in plane. The area is used later for the calculation of the heat flux per sensor.

Ring	R_i mm	R mm	R_o mm	Area mm ²	Rotation rad
0	384.500	438.610	488.420	9179	0
1	489.820	534.640	574.190	9093	0
2	575.590	609.410	637.200	7595	0
3	616.790	697.900	755.500	8202	0.098
4	756.900	812.470	866.060	8933	0.098
5	867.460	918.750	967.780	9302	0.099

The leakage power produced in the sensor depends on the radiation dose/fluence. The radiation dose/fluence has been studied at different positions in the ATLAS detector. Figure (6.3) shows the predictions of the 1 MeV n_{eq} fluences normalized to 3000 fb⁻¹ of HL-LHC running for the different geometric coordinates of the detector. These predictions were made using the FLUKA package [53]. In figure (6.3) the axes r and z are the radial and longitudinal distance in respect to the center of the detector and correspond to that of figure (6.1). It is shown that the innermost part of the last disk reaches the highest fluence in the Endcap. The radiation dose prediction will be used later to estimate

the radiation induced leakage power in the sensors which is needed as input for the thermal model.

Figure 6.3: The FLUKA predictions of the 1 MeV n_{eq} fluences normalized to 3000 fb⁻¹ of HL-LHC running ([54],[55]), which reach 10¹⁵cm⁻² at the innermost part of the last disk. The last disk is taken into account as a worst case scenario in the thesis.

6.1.3 ASICs

The geometrical information and the predictions of the radiation damage of the ASICs are given. Table (6.2) gives the number of hybrids on each ring and the ASICs on each hybrid. Sensors that are side by side on the same ring share the same hybrids. The DC/DC converter includes the FEAST2 ASIC. Rings 3 to 5 have two modules (right and left) that share the DC-DC converter located on the left module.

The Total Ionizing Dose (TID) is an important quantity for understanding damage in electronics and to predict the power consumption as discussed in section 3.6.2. Figure (6.4) gives the FLUKA predictions of the TID normalized to 3000 fb⁻¹ for the extended layout and shows the position dependency of the TID. It can be seen that the modules will have more damage on the innermost part with the last disk being the worst of all the disks. The positional dependency of the TID per disk is neglected in the thesis and instead a peak dose is taken into account for each disk.

Table 6.2: Number of ASICs for the hybrids on each ring according to the latest design [15]. The three outermost rings have two modules each and are referred as right and left module. The DC-DC converter which is common for those modules is located on the left module.

Ring	Hybrid	No. ABCs	No. HCCs	No. FEAST2
0	1	8	1	1
	2	9	1	
1	1	10	1	1
	2	11	1	
2	1	12	2	1
	2	14	2	1
3 (Right+Left)	1	14	2	1
	2	14	2	
4 (Right+Left)	1	16	2	1
	2	18	2	1

Figure 6.4: The FLUKA predictions of the total ionizing dose (TID) normalized to 3000 fb^{-1} for the extended layout ([54],[55]), reaching $4 \times 10^5 \text{ Gy}$ at the innermost part of the last disk. There is a positional dependency of the TID per disk but it is neglected in the thesis.

6.2 Thermal modeling

The modeling of the petal aims to investigate the thermal performance of the structure during its lifetime and consider the possible impacts on its perfor-

mance resulting from the increase in the digital current of the ASICs. Two particular situations are examined during the ten years of operation of the detector under specific assumptions. The first case, which will be referred in the thesis as TID case, represents a situation early in the operation when there is an increase in the digital current in the ASICs (ABCs, HCC, FEAST2) as discussed in section 3.6.2. The TID bump was first seen in the IBL detector [56]; the ITk design shares the same ASIC technology, and so it should be prepared to deal with this effect. At the TID case the sensor leakage power is negligible mainly because it occurs at the beginning of the operation with minimum radiation accumulation. Figure (6.4) showed the position dependency of the TID, but for simplicity it is assumed that there is no position dependency of the TID for the ASICs early in the operation. As a result, in this thesis, all ASICs are modeled at their peak dose simultaneously as a worst case scenario. The second case, which will be referred in the thesis as end-of-life case, represents the situation at the end of the ten years of operation, with sensors having received around 3000 fb^{-1} integrated luminosity. For the end-of-life case, the positional dependency of the fluence is taken into account for the sensors (seen in figure (6.3)). It is interesting to understand which sensor is critical or not, therefore, the dependency is important to be included. As discussed in section 3.6.2, it is assumed the TID bump has disappeared for the ASICs, leaving a small lasting increase in digital and analogue power. For this scenario, the last disk is selected as the worst case scenario with the highest fluence and TID effect, as it was seen in figure (6.4).

6.2.1 Geometry

The modeled geometry includes all the components of the real detector in order to simulate its thermal performance. Particularly, the bus tape (shown in figure (3.3)), the ASICs and the glue layers are modeled. Figures (6.5) and (6.6) show the details of the model as implemented in Abaqus based on the DESY drawing [52].

Figure 6.5: Petal dimensions and drawing in Abaqus. The petal is flipped on the opposite side compared to figure (6.2). The ASICs are shown in yellow, the hybrid in green, the DC/DC converter in pink and the sensors in blue.

Figure 6.6: Transparent view of the petal and the tube passing mid-plane of the sensors, and entering and exiting at the EoP. The tube passes over the readout ASICs in R5 as it is expected to receive more radiation than the rest since they are located on the last disk.

More pictures can be found in Appendix (D). In the latest design the ‘ear’ is on one side only, to minimize the material used. The cooling tube therefore enters and leaves the structure through the ‘ear’ where the EoP board is. The facesheets are initially K13D2U 80 gsm (carbon fiber/resin). In the optimization, K13C2U 45 gsm (carbon fiber/resin) material is explored instead.

6.2.2 ASICs Power

The expected ASICs power is estimated for the different power cases (TID and end-of-life) and will be used later as input in the thermal model.

TID case

All the readout ASICs are modeled at TID peak simultaneously as a worst case scenario. The increase in the digital current compared to non-irradiated ASICs is estimated with a 2.5 scale factor according to the latest studies [15]. Based on the above, table (6.3) gives the maximum expected ASIC current expected from the FEAST2 ASIC. The details for the FEAST2 are not included here since the only available information for the FEAST2 ASIC is the dependency of the efficiency with the total power.

Table 6.3: Maximum current and power per ASIC expected at the TID peak [57]. The total power per ASIC type is the sum of the analogue and digital power.

Chip	Power type	Current A	Power W/chip
ABC130*	Analogue	0.055	0.0825
	Digital	0.0875	0.2137
HCC*	Analogue	0.075	0.11
	Digital	0.3125	0.468

Table (6.4) gives the total module power at the TID case. The power is estimated using tables (6.2), (6.3) and a voltage of 1.5 V. The module power includes the ABC (digital and analog power) and HCC ASICs, whereas the total module power includes the module power and the DC/DC converter power. There is one DC/DC converter per ring and it is considered to have one hybrid for two sensors side by side on the same ring. During the TID case a degradation in power conversion efficiency in the DC/DC converter by 8% is expected [58]. The total module power per petal face is around 63 W excluding the power in EoP. It is clear that ring 3 uses too much power for a single DC/DC converter, leading to a very low efficiency and increased power consumption. One of the early recommendations is to go to two DC/DC converters in ring 3.

Table 6.4: Estimated total module power (including ABC, HCC and DC/DC converter) for each hybrid in each ring for 1.5 V at TID case. The total module power per petal is around 126 W at TID case excluding the EoP power.

Ring	Hybrid	Module	Efficiency	DC/DC	Total module
		power	DC/DC conversion	power	power
		W	%	W	W
0	1	2.30	-	-	-
	2	2.50	0.59	3.30	8.07
1	1	2.72	-	-	-
	2	2.94	0.54	4.70	10.40
2	1	3.73	0.64	2.10	5.78
3	1	4.15	-	-	-
	2	4.15	0.36	14.40	22.70
4	1	4.60	0.60	3.00	7.57
5	1	5.00	0.58	3.60	8.60

End-of-life

Table (6.5) presents the estimated maximum current expected for the ABC130* and HCC* for the end-of-life case.

Table 6.5: Maximum current and power at the end of the 10 years of operation [57] at end-of-life case. The total power per ASIC type is the sum of the analogue and digital power.

ASIC	Power type	Current A	Power W/ASIC
ABC130*	Analogue	0.055	0.0825
	Digital	0.035	0.0525
HCC*	Analogue	0.075	0.11
	Digital	0.125	0.187

Table (6.6) gives the estimated total module power for the end-of-life case at maximum dose and at inner radius of disk 5. The power is estimated using tables (6.2), (6.5) and a voltage of 1.5 V. The total module power per petal face is around 27 W excluding the power in EoP. Here, the power conversion efficiency is higher than in the TID case due to the lower current.

6.2.3 Sensor power

The expected sensor power is estimated for the two power cases (TID and end-of-life) and is used later as input for the thermal model.

Table 6.6: Estimated total module power (including ABC, HCC and DC/DC converter) for each hybrid in each ring for 1.5 V at end-of-life case. The total module power per petal is around 54 W at end-of-life case excluding the power produced in the EoP board.

Ring	Hybrid	Module power W	Efficiency DC/DC conversion %	DC/DC power W	Total module W
0	1	1.38	-	-	-
	2	1.51	0.75	0.95	3.85
1	1	1.65	-	-	-
	2	1.78	0.74	1.24	4.67
2	1	2.22	0.76	1.50	2.92
3	1	2.50	-	-	-
	2	2.50	0.66	2.50	7.50
4	1	2.76	0.75	0.90	3.65
5	1	3.03	0.75	1.00	4.05

TID case

The integrated luminosity received by the sensors during the TID case is quite low and around $10^{14} \text{ n}_{eq}\text{cm}^{-2}$ as it is described in Appendix (E). The current density is much lower than that of the end-of-life case in table (6.7). In this thesis, the sensor power is ignored at the TID peak.

End-of-life

The expected sensor power for the end-of-life case is estimated similarly as in chapter 3. Table (6.7) summarizes the radiation dose expected in the sensors and the power generated at 293 K. The bias voltage applied in the sensors is 500 V and the thickness of the sensor is 320 μm . The non-ionizing dose is taken from figures (6.3) and (6.4).

6.2.4 DC/DC Converter Power

The DC/DC converter with the FEAST2 ASIC convert the high voltage applied to the total petal to a lower voltage for each sensor. The DC/DC converter exhibits an efficiency that varies with temperature while at increased loads (>4 Amps) the efficiency drops due to increased conduction losses. The efficiency of the DC/DC converter is calculated based on the total current that the converter handles each time. Assuming an average temperature for the DC/DC converters the relationship is given in [58]. During the TID case a degradation in power conversion efficiency in the DC/DC converter by 8% is taken into account.

Table 6.7: Estimated power dissipated in the sensors for the latest design based on the fluence for disc 5 and a radiation damage factor 3.99^{-17} A/cm³/(neq/cm²) [37]. The calculation of the energy band gap E_g is based on the Varshni temperature dependent formula [39] and the assumed bias voltage is 500 V.

Ring	Non-ionizing Dose	TID	Current density	Current (293 K)	Q_{ref} (293 K)
	neq/cm ²	MGy	A/cm ³	A	W/mm ²
0	$8.17 \cdot 10^{14}$	0.366	0.0316	0.092	0.00522
1	$6.74 \cdot 10^{14}$	0.217	0.0265	0.077	0.00430
2	$5.81 \cdot 10^{14}$	0.155	0.0235	0.057	0.00371
3	$5.34 \cdot 10^{14}$	0.129	0.0217	0.057	0.00341
4	$4.53 \cdot 10^{14}$	0.094	0.0191	0.054	0.00289
5	$4.00 \cdot 10^{14}$	0.079	0.0173	0.051	0.00255

6.2.5 EoP board

The EoP board is located at the ‘ear’ of the petal and receives the signal from all the sensors and transmits it to outside the petal. With no data on the EoP board, we apply a fixed power (around 2.5 W per chip) independent of dose, which is a typical value for the ASICs of the existing EoP boards.

6.2.6 Materials properties

The FEA thermal model for the latest petal design is presented. The methodology is the same as described in chapter 3. Only deltas are mentioned here. Most of the materials for the latest design model are the same as the model for the prototype presented in chapter 3. The high thermal conductivity Pocofoam is replaced with Allcomp since Pocofoam is not produced anymore. Allcomp is also a high thermal conductivity carbon foam material. In addition, the ASICs and hybrids are modeled. The material for the ASICs is silicon. The bustape, the readout hybrid (for the ABCs and HCCs) and power hybrid (for the DC/DC converter) are printed circuit boards with multi-layers of copper and polyimide. The materials used for each component are given in table (6.10). The FEA geometric parameters for the thermal petal model are summarized in table (6.8). To determine accurately the cooling of the ASICs the thermal conductivity for each circuit board is calculated based on its composition.

Bus tape composition

The bus tape consists of 2 layers of copper (50% coverage) 18 μm thick, and 3 layers of polyimide, 25 μm thick. Also, glue layers exist between the different material layers, 25 μm thick. Figure (6.7) shows a simplified version of the cross section of the bus tape.

Table 6.8: Modeling of the different components in the petal thermal model

Component	Part	Properties
ASICs	Solid	Isotropic
Hybrid	Solid	Isotropic
Sensor	Solid	Isotropic
Glue Sensor-FC	Solid	Isotropic
Facesheet	Solid	Orthotropic
Core	Solid	Isotropic
Carbon foam	Solid	Orthotropic
Closeouts	Solid	Isotropic
Tube	Solid	Isotropic

Figure 6.7: View of a simplified version of the cross section of a bus tape for the petal (not in scale). Green is copper and it is shown as a continuous layer while in reality it has only a 50% coverage.*Power hybrid composition*

The power hybrid consists of 4 layers of copper 35 μm thick and 3 layers of polyimide 50 μm thick. Under the FEAST2 ASIC [59] there are blind thermal vias that end before the last copper layer. There are in total 25 thermal vias per total FEAST2 ASIC area with a pitch around 0.55 mm. Glue layers exist between the different material layers, 50 μm thick.

Readout hybrid composition

The readout hybrid consists of 4 copper layers of 18 μm and 3 polyimide layers of 50 μm . Under the ABC ASICs there are blind thermal vias till the second copper layer. There are 20 thermal vias under the ABC ASICs with pitch around 1 mm. Glue layers exist between the different material layers 50 μm thick.

In the model, each circuit board is replaced by a homogeneous material with isotropic thermal conductivity. Each layer is treated as an isotropic material, with properties given in table (6.9). The through-conductivity is modeled based on the copper layers (50% coverage) and using a thermal conductivity of half of the actual copper value. This is a good estimation in the thickness direction but it is far over-estimating the heat-spreading effect. Therefore, for the in-plane conductivity the copper is not taken into account and only the glue and polyimide values are used. The components (k_x, k_y, k_z) of the

homogeneous material are derived from the conductivities k_i of the layer based on eq. (2.4). For the more complicated geometry of the vias, a simple cuboid FEA model of the via region of the circuit board was created. Heat flux was applied across opposite faces of the cuboid to determine the k_x and k_z via the resulted temperature difference. Table (6.9) gives the thermal conductivity of the single materials used for the calculation of the total effective thermal conductivity in the different directions.

Table 6.9: Thermal conductivity of the single materials within the hybrids and bus tape. These values are used to estimate the effective conductivity.

Material	Thermal conductivity
	W/mK
polyimide	0.17
copper	400
glue	0.17

Finally, table (6.10) summarizes the thermal properties of each material as it used in the FEA. The intermediate conductivity of the in between values of the silicon are linearly interpolated.

Table 6.10: Material information for the latest ITk petal thermal model. The x refers to the length direction, the y refers to the through thickness direction of the petal and the z refers to the cross direction of the petal.

Part	Material	Thermal conductivity x/y/z W/mK	Thickness mm
ABC ASIC	Silicon	191(250K)..148(300K)	0.3
ABC ASICs to Hybrid	tra-duct-2902	2.99 (50% coverage)	0.08
Readout Hybrid	Cu/polyimide	72/0.23/72	0.2
		72/0.54/72 (all vias)	
Hybrid to sensor	FH5313 Epolite	0.23	0.12
FEAST2 ASIC	silicon	191(250K)..148(300K)	0.3
Power Hybrid	Cu/polyimide	120/0.34/120 no vias	0.3
Sensor	silicon	191(250K)..148(300K)	0.2
Sensor to bus	DC SE4445	2	0.2
EoP	silicon	91(250K)..148(300K)	0.3
EoP substrate	Cu/polyimide	72/0.23/72	1.6
Bus tape	PolyI/Cu/Al	0.17/0.24/0.17	0.2
Facesheet	CFRP K13C2U 45gsm	180/1/90	0.15
Facesheet	carbon fiber K13D2U	294/1.3/148	0.15
	prepreg RS3 resin,80 gsm		
Honeycomb	air	-	-
Facing to foam	Hysol+BN	1.33	0.1
Carbon foam	Allcomp	30	-
Foam to pipe	Hysol+BN	1.33	0.1 -
Cooling Tube	Titanium Grade 2	16.4	0.15

6.3 CoBra CO₂ calculations

The CO₂ heat transfer coefficient and the CO₂ temperature need to be included as well in the thermal model. The CO₂ properties for the two power cases are estimated using CoBra. The results are given in sections (6.3.1) and (6.3.2). The strategy is described next. The CO₂ at the entrance of the tube is assumed to be saturated liquid or two-phase at very low vapor quality ($x < 0.01$). The mass flow rate is chosen to give a vapour quality of 0.35 at the exit of the tube for the highest power case (TID case). For an evaporation temperature of -30 °C and with 131 W power for the TID case, the mass flow is 1.25 g/s in order to reach 0.35 vapor quality at the exit. For the same mass flow rate at 59 W power (the end-of-life case) the vapor quality is 0.16 at the exit of the tube. Table (6.11) gives a summary of the input parameters for the CoBra calculations for the two power cases.

Table 6.11: Input parameters for CoBra to calculate the CO₂ properties used later in the thermal modeling.

Parameter	Power case TID	Power case end-of-life
Inner diameter	2 mm	2 mm
Length	1.4 m	1.4 m
Total power	131 W	59 W
Mass flow	1.25 g/s	1.25 g/s
Evaporation Temperature	-30 °C	-30 °C

6.3.1 TID case

Figure (6.8) gives the CO₂ temperature and pressure along the tube for the TID case. There is a small pressure drop of 0.1 bar and therefore there is a very small drop in the coolant temperature from inlet to outlet. For simplicity, this temperature drop is neglected in the thermal modeling. Figure (6.9) shows the flow regime map. The flow regime map indicates the transition lines from one regime to the other depending on the flow properties. The flow properties are expressed in vapor quality and mass flux. The red lines indicate the transition to the Dry-out or Mist flow regime which is considered undesired. For the TID case (indicated with the green line) the flow regime changes from intermediate to annular pattern at around 0.1 vapor quality. The vapor quality is 0.3 at the outlet of the tube due to the increasing heat flux seen by the coolant from inlet to outlet.

Figure (6.10) shows the development of the heat transfer coefficient and vapor quality along the tube for the TID case. Assuming a uniform heat load, the

Figure 6.8: CoBra results for the TID case, temperature and pressure of the fluid along the tube. There is a small pressure drop along the evaporator therefore a small temperature decrease in the coolant from inlet to outlet.

Figure 6.9: The flow pattern map with varying vapor quality as it is predicted by CoBra for the TID case. Depending on the diameter and the mass flow, different regimes can take place. The green line indicates that during this specific evaporation process it changes from intermittent to annular regime.

heat transfer coefficient increases and with a fixed coolant temperature the surface temperature will get closer to that of the CO₂. At 0.1 vapor quality

where the annular regime starts the heat transfer coefficient becomes steeper.

Figure 6.10: CoBra results for the TID case, heat transfer coefficient and vapor quality along the evaporator. The heat transfer coefficient increases up to 10000 W/m²K. The vapor quality reaches 0.3 at the exit of the evaporator due to the heating.

6.3.2 End-of-life case

Figure (6.11) gives the CO₂ temperature and pressure along the tube for the end-of-life case. The temperature and pressure profile follows the boiling line and corresponds well with figure (6.8). The small temperature drop is neglected in the thermal modeling. Figure (6.12) shows the flow regime map for the end of life case. For the end-of-life case (indicated with the green line) the flow changes from intermediate to annular regime around 0.1 vapor quality. Due to the lower heat load the vapor quality is <0.2 at the exit of the tube. Figure (6.13) gives the development of the heat transfer coefficient and vapor quality along the tube for the end-of-life case. At 0.1 vapor quality where the annular regime starts the heat transfer coefficient becomes slightly steeper as in (6.10). Due to the low heat load it reaches only up to 6500 W/m²K.

Figure 6.11: CoBra results for the end-of-life case, temperature of the fluid along the tube and the pressure along the evaporator. There is a small pressure drop along the evaporator therefore a small temperature decrease in the coolant from inlet to outlet.

Figure 6.12: The flow regime map with varying vapor quality as it is predicted by CoBra for the end-of-life case. Depending on the diameter and the mass flow, different regimes can take place. The green line indicates that during this specific evaporation process it changes from intermittent to annular regime.

Figure 6.13: CoBra results for the end-of-life case, heat transfer coefficient and vapor quality along the evaporator. The heat transfer coefficient increases up to $6500 \text{ W/m}^2\text{K}$. The vapor quality reaches 0.15 at the exit of the evaporator due to the heating.

6.4 Running the FEA

6.4.1 Modeling of the sensors

The temperature dependent heat production in the sensors is included in the thermal model. The heat produced by the sensors is due to the production of leakage current as described in section 3.6.1. The equation describing the mechanism behind the production of leakage current could be fed directly into Abaqus using subroutines. But this needs access to specific compilers, which were not available. Another way to simulate the heat generated in the sensor due to the leakage current, is by defining the sensor with thermal-electric mesh elements. The process is summarized. The thermal-electric mesh elements require the definition of electrical conductivity σ_e (in $\text{Ohm}^{-1}\text{m}^{-1}$) and an electrical potential ϕ (in V) across the element. Abaqus calculates the current density j (A/m^2) and power density Q_c (W/m^2) using Ohm's law. For this reason, the sensors are modeled with a temperature-dependent electrical conductivity to produce the desired power density allowing for the fluence and bias voltage. In Appendix (F) the calculation of the electrical conductivity for the silicon sensor is described. The bias voltage is always taken as 500 V.

6.4.2 Convective and radiative heat transfer

Modeling of the heat transfer by convection to and from the ambient environment and by radiation would require the modeling of a large EC part. The environment around the detector is occupied with nitrogen and there is natural air convection due to the various heat sources. Limited information is available on the ambient temperature around the detector during operation, while the presence of EC disks makes any analytical estimation inaccurate. The simplified estimation of the heat leaks described in chapter 5 is not representative of the EC environment. Study of the ITk system as a whole is needed based on computational fluid dynamics (CFD). A CFD study can be found in ref. [60] and initial results predict only small effects of the heat leaking to and from the environment. For this reason, heat loads leaking in from the outside and from the Barrel are ignored. Also, air convection that would make the top part of the EC warmer than the bottom is ignored. In the model, all heat produced from the ASICs and the sensors goes to the coolant.

6.4.3 Boundary conditions

The boundary conditions defined in the thermal model include prescribed heat flux for the ASICs, convective heat transfer between the coolant and the tube inner surface, constant coolant temperature and the electrical potential over the sensors required to predict the temperature dependent leakage current. Particularly, the electrical potential is defined by keeping one surface of the sensor grounded (zero voltage) while the other side is subject to the bias voltage. The bias voltage is set to 500 V. Finally, the heat transfer coefficient between the coolant and the inner surface of the tube varies along the tube length and is set according to the CoBra results described in section 6.3.

6.4.4 Thermal runaway methodology

The ‘thermal runaway’ is undesired in HEP and is signified by the uncontrolled increase in the leakage current as described in section 3.6.1. The thermal runaway can be predicted using FEA. FEA can give an estimation of the margin of operation in terms of radiation induced power and coolant temperature, known as power headroom and coolant temperature headroom respectively. In the thesis only the coolant temperature headroom is discussed.

The “coolant temperature headroom” is estimated by setting the sensor power q per unit sensor area at the value expected at the end-of-life and by running several FEA at increasing coolant temperature. The “coolant temperature headroom” is the thermal runaway coolant temperature minus the design coolant temperature and indicates the coolant temperature safety margin at

the end-of-life. Above thermal runaway the FEA fails to converge.

6.5 FEA results

The FEA results are presented in figures (6.14) and (6.15) for the TID and end-of-life power cases respectively. Note that the figures have different temperatures scales from each other.

6.5.1 Temperature predictions

Regarding the TID case, the hottest spot is predicted at the FEAST2 of the DC/DC converter at R3 at about $+55^{\circ}\text{C}$. As a result the worst silicon sensor in terms of temperature is at ring 3 (R3) which has a hot spot temperature of -7.5°C . On average the silicon temperature is -22°C with the lowest value at the sensor in R3 which is at -11°C . This is expected due to the reduced efficiency in R3 (seen in table (6.6)). The ABC ASIC temperature range from -25 to -12°C . The HCC ASIC temperature range is -20 to -12°C , whereas the other FEAST2 ASIC temperature range from 4 to 41°C . The outermost sensors are warmer at the EoP side compared to the one further away from the EoP. This is expected due to the increased heat load for the EoPs located both on the one ‘ear’.

Figure 6.14: Surface temperatures at TID. The FEAST2 ASICs are the hottest areas as it is expected due to the high heat load and low efficiency. The EoP ASICs are also warmer than the readout ASICs due to the increase heat load concentrated on that spot.

Regarding the end-of-life, the hottest spot again is predicted at the FEAST2 of the DC/DC converter at R3, here at about $+21^{\circ}\text{C}$. As a result the worst silicon sensor in terms of temperature is at R3 which has a hot spot

temperature of -19°C . On average the silicon temperature is -25°C with the lowest value at the sensor in R3 which is at -23°C . The ASIC temperatures range from -26 to -20°C . The HCC temperature range is -24 to -20°C , whereas the rest of the FEAST2 ASICs are -7 to -14°C . The outermost sensors are warmer at the EoP side compared to the one further away from the EoP. This is again expected due to the increased heat load for the EoPs located both on the one ‘ear’.

Figure 6.15: Surface temperatures at end-of-life. The FEAST2 ASICs are the hottest areas as it is expected due to the high heat load and low efficiency. The EoP ASICs are also warmer than the readout ASICs due to the increase heat load concentrated on that spot.

Table (6.12) summarizes the maximum temperature difference within a ring for both scenarios. It can be seen that there is up to 20°C temperature difference across the sensors in R3 which is considerable hot spot that could lead to thermal runaway. Table (6.12) also gives the mean silicon temperatures for both scenarios. It can be seen that the sensors in R3 are considerably warmer than the rest, which is also not desired due to thermal runaway.

Table 6.12: Results of maximum silicon temperature difference across a sensor and results of mean silicon temperature for each sensor for the TID and end-of-life.

Ring	$\Delta T_{max}^{\circ}\text{C}$	$\Delta T_{max}^{\circ}\text{C}$	$T_{mean}^{\circ}\text{C}$	$T_{mean}^{\circ}\text{C}$
	TID	End-of-life	TID	End-of-life
0	12.30	4.40	-23.50	-26.50
1	10.60	5.40	-20.60	-25.30
2	9.80	4.20	-23.00	-25.50
3	19.60	8.90	-11.00	-23.00
4	17.20	6.70	-21.40	-25.60
5	18.30	5.70	-22.50	-26.30

The summary table (6.12) showed that there is considerable temperature difference of the sensor in R3 compared the rest of the rings which could lead to

thermal runaway. The reduced efficiency in the DC-DC converters is a major issue that needs attention.

6.5.2 Thermal runaway

Running the FEA with several coolant temperatures predicts the thermal runaway. The heat transfer coefficient was kept constant (negligible impact). Figure (6.16) shows the FEA prediction of the temperature difference between the coolant and the sensor for various coolant temperatures and when the thermal runaway is expected to start for the 2 mm inner diameter tube. The thermal runaway is at around -17°C coolant temperature. This means that the coolant temperature can be increased up to 13°C before the silicon starts heated uncontrollably. Generally, it can be seen that the thermal resistance due to the silicon is increased at elevated coolant temperatures up to the thermal runaway temperature. This is very much expected according to section 3.6.1 where the sensor leakage current is temperature dependent.

Figure 6.16: Thermal runaway FEA prediction for the 2 mm inner diameter tube. The coolant temperature headroom is 13°C . At -17°C coolant temperature the silicon temperature starts increasing sharply, declaring the thermal runaway.

This initial design, even without optimization, has sufficient safety margin before thermal runaway. This is due to the low temperature of the CO_2 coolant and the low thermal resistance from the sensor to coolant. An even larger coolant headroom would allow to run at higher coolant temperature. That would give more space for optimization like use thinner face-sheets which could worsen the safety margin.

6.6 Optimization studies

The results so far have shown that there is some safety margin before thermal runaway, therefore, there is margin for mechanical and electrical improvements in the petal design. A series of optimization steps is presented below in an attempt to improve the design.

6.6.1 DC/DC converters issue

The DC/DC converters in R1 and R3 showed a considerable loss in efficiency (see tables (6.4) and (6.6)) resulting in increased waste heat. The reason for this is that in R1 and R3 the FEAST2 ASIC runs with up to 6 Amps, while, it has been designed to run with a maximum current of 4 Amps. We need to use two FEAST2 ASICs on R1 and R3. In this way the total current supplied by each FEAST2 ASIC is reduced, increasing the efficiency.

The power expected per ring with two FEAST2 ASICs in R1 and R3 is pre-

Table 6.13: Estimated total module power (including ABC, HCC and DC/DC converter) for each hybrid in each ring for 1.5 V at TID case with 2 FEAST2 for R1 and R3. The total module power per petal is around 103 W at TID case excluding the EoP power, significantly lower than the original design (126 W).

Ring	Hybrid	Module power W	Efficiency %	DC/DC power W	Total module power W
0	1	2.30	-	-	-
	2	2.50	0.59	3.30	8.07
1	1	2.72	0.68	1.31	-
	2	2.94	0.67	1.40	8.40
2	1	3.73	0.64	2.10	5.78
3	1	4.15	0.62	2.50	-
	2	4.15	0.62	2.50	13.30
4	1	4.60	0.60	3.00	7.58
5	1	5.00	0.58	3.60	8.60

sented in tables (6.13) and (6.14) for the TID and end-of-life cases respectively. Comparing tables (6.13) and (6.14) with tables (6.4) and (6.6) (only the values in R1 and R3 are different, shown in bold letters in the table) it can be seen that the total power module has been significantly decreased for the TID case. This is due to the improved efficiency in R1 and R3 for the TID case. There is no significant improvement for the end-of-life case mainly because the power usage was already within the design limitations of the FEAST2 ASICs (<4 Amps).

Table 6.14: Estimated total module power (including ABC, HCC and DC/DC converter) for each hybrid in each ring for 1.5 V at 3000 fb⁻¹ case with 2 FEAST2 for R1 and R3. The total module power per petal is around 51 W at end-of-life case excluding the EoP power, slightly lower than the original design (54 W).

Ring	Hybrid	Module power W	Efficiency %	DC/DC power W	Total module W
0	1	1.38	-	-	-
	2	1.51	0.75	0.95	3.85
1	1	1.65	0.74	1.10	-
	2	1.78	0.75	1.20	4.60
2	1	2.22	0.76	0.75	2.92
3	1	2.50	0.75	0.80	-
	2	2.50	0.75	0.80	6.60
4	1	2.76	0.75	0.90	3.65
5	1	3.03	0.75	1.00	4.05

Table (6.15) gives the mean silicon sensor temperatures at each ring when 2 FEAST2 are used in R1 and R3. The use of 2 FEAST2 in R3 lowers the mean silicon temperature by roughly 10 °C and by roughly 3 °C in R1, compared with table (6.12).

Table 6.15: FEA prediction of mean silicon temperature for each sensor at the end-of-life and TID case when 2 FEAST2 are used in R1 and R3.

Ring	T _{mean} ° C TID	T _{mean} ° C End-of-life
0	-24.50	-26.50
1	-23.50	-25.50
2	-23.90	-25.80
3	-21.30	-24.50
4	-23.00	-25.80
5	-22.80	-26.40

Thermal Runaway

Figure (6.17) shows the FEA prediction of the temperature difference between the coolant and the sensor for various coolant temperatures and when the thermal runaway is expected to start when 2 FEAST2s are used in R1 and R3. The inner diameter of the tube is 2 mm. The coolant temperature head-room has increased by 5°C compared with figure (6.16). This means that the coolant temperature can be increased even more, up to 18 °C, before the silicon starts heating uncontrollably. Again here, it can be seen from the trend

that the thermal resistance due to the silicon is increased at elevated coolant temperatures up to the thermal runaway temperature.

Figure 6.17: Thermal runaway prediction for coolant headroom for FEA with 2 FEAST2s on R1 and R3 for the 2 mm inner diameter tube. The coolant temperature headroom is 18 °C. At -12 °C coolant temperature the silicon temperature starts increasing sharply, declaring the thermal runaway.

6.6.2 Cooling tube diameter

The studies of the prototype petal presented in chapter 4, showed that the diameter of 2 mm was not the optimum in terms of the flow regime for a specific set of power conditions. Therefore, studies on the choice of the optimum inner diameter for HL-LHC upgrade were conducted. The focus of these studies is mainly on the optimum flow regime for the coolant and an improved thermal performance of the petal. Various diameters between 2 to 1 mm are examined. The option to keep the current inner diameter of 2 mm but add two petals in series is also studied.

In summary the following criteria for the selection of diameter are considered. The optimum pressure drop shall be less than 1 bar in order to keep the uniformity on the coolant temperature along the tube. More specifically, the optimum temperature difference from inlet to outlet shall be less than 2°C. And finally, the optimum regimes are considered the bubbly, bubbly to annular and annular regime due to the high heat transfer coefficient.

Firstly, the different cases are examined based on the CO₂ properties. Table (6.16) gives a summary of the results from Cobra for the different input parameters (power, mass flow and diameter). The results from CoBra include the pressure drop, temperature drop from inlet to outlet due to the pressure drop, the vapor quality at the exit of the petal, and the predicted heat transfer coefficient and the predicted flow regime as the change along the length of the channel. In table (6.16) the cases that are considered optimum are marked with bold letters. A case is considered optimum when it satisfies all the criteria for both power cases. Rejected cases based on the current assumptions and conditions described above are also shown. The reason of rejection is underlined. Please note that in table (6.16) that the pressure drop increases as the diameter increases. Consequently the vapor quality is affected due to the different pressure level. To keep the vapor quality at 0.3 at the exit of the channel at the TID case at decreasing diameter the mass flow is increased. Some of the options are examined further with FEA while the thickness of the core material as well as the wall thickness of the tube were kept constant. Table (6.17) summarizes the FEA results. It presents the maximum temperature of the petal, the maximum temperature among the sensors and the mean temperature of the worst temperature sensor. The location of each is mentioned. It can be concluded that similar thermal performance is expected for all the diameters. For all cases the highest temperature is seen in the FEAST2 ASICs in R5 for the TID cases while for the end-of-life case the highest temperature is in the EoP ASIC. At the end-of-life the efficiency of the FEAST2 in R5 is better therefore less heat is produced making the EoP the hottest spot. For all power cases and diameters the sensor in R3 is the worst. The increased heat transfer coefficient at smaller diameters improved slightly the temperature of the silicon which can be considered almost negligible.

Table 6.16: Comparison of CO₂ flow properties for various diameters evaluated by CoBra. The I stands for the intermediate regime and A for the annular regime and B for the bubbly regime. Cases with bold letters are considered the optimum, whereas for the rest cases the underlined parameter indicates the reason of rejection.

Petal in series	Power W	inner diameter mm	MF g/s	Petal	ΔP bar	ΔT ° C	HTC W/m ² K	Quality at exit	regime
1	108	2	1.25		0.10	0.20	5700-9500	0.30	<u>I-An</u>
		1.9	1.25		0.10	0.20	5900-10300	0.30	<u>I-An</u>
		1.8	1.25		0.15	0.30	6100-11300	0.30	<u>I-An</u>
		1.7	1.25		0.25	0.40	6400-12500	0.30	<u>I-An</u>
		1.6	1.25		0.25	0.50	6600-13800	0.30	<u>I-An</u>
		1.5	1.25		0.30	0.70	6900-15500	0.30	<u>I-An</u>
		1.4	1.25		0.40	0.90	7300-17500	0.30	<u>I-An</u>
		1.3	1.55		0.80	1.70	8800-23400	0.30	B-An
		1.2	1.55		1.00	<u>2.00</u>	8500-23400	0.30	B-An
		1.0	1.65		<u>2.40</u>	<u>4.60</u>	10400-24300	0.30	B-An
	56	2	1.25		0.05	0.10	4000-6600	0.16	<u>I-An</u>
		1.5	1.5		0.40	0.90	5000-10900	0.16	<u>I-An</u>
		1.4	1.5		0.40	0.90	5400-12400	0.16	<u>I-An</u>
		1.3	1.55		0.40	0.80	6700-15500	0.16	B-An
		1.0	1.65		1.30	2.70	8700-25000	0.16	B-An
2	108	2	2.35	1st	0.10	0.20	6000-11000	0.16	<u>I-An</u>
			2.35	2nd	0.40	0.80	11200-15400	0.30	An
	56	2	2.35	1st	0.07	0.15	5000-8300	0.08	<u>I</u>
			2.35	2nd	0.20	0.50	8500-12000	0.16	<u>I-An</u>

Table 6.17: Comparison of diameters based on the FEA predictions. The second pair of columns refers to the location and the value of the highest temperature on the petal while the third pair of columns refers to the location and the value of the highest temperature among the sensors. The last column summarizes the mean temperature of the worst sensor which is in R3.

inner diameter	Power	T max	Location	T max	Ring	T mean
mm	W	° C	on Petal	silicon ° C		silicon ° C
2	108	42.00	FEAST2 R5	-7.50	R3	-21.30
	56	11.00	EoP	-18.50	R3	-24.50
1.5	108	41.50	FEAST2 R5	-7.60	R3	-21.30
	56	11.00	EoP	-19.30	R3	-25.00
1.3	108	43.00	FEAST2 R5	-7.20	R3	-21.40
	56	11.20	EoP	-19.20	R3	-24.90
1	108	41.40	FEAST2 R5	-7.30	R3	-21.20
	56	11.05	EoP	-19.20	R3	-24.90

Table (6.18) presents the thermal runaway results for the different diameters. The thermal runaway at 1.5 mm ID is already better than that of the 2 mm inner diameter by 2 °C. Actually there seems to be a sharp decline for diam-

eters higher than 1.5 mm. Between 1 and 1.5 mm the thermal runaway can be within the computational error range. With two petals in series, the first petal suffers thermal runaway before the second therefore two petals in series are not preferred.

The selection of 1.3 mm inner diameter for the petal compared to the other

Table 6.18: Comparison of coolant temperature headroom for the different diameters.

inner diameter mm	Petal in series	T_{CO_2} runaway
2		-14.00
2	1 st	-14.70
2	2 nd	-11.50
1.5		-12.50
1.3		-13.00
1		-13.00

diameters examined seems favorable in terms of flow optimum regime (bubbly and annular) and of thermal runaway studies as seen in tables (6.17) and (6.18). Since no deterioration in the thermal performance is expected when the inner diameter decreases, the 1.3 mm inner diameter is suggested as it favors in terms of the material budget (compared to the 1.5 mm inner diameter). Further studies and prototypes on 1.3 mm inner diameter are recommended to examine the mechanical performance of the petal, which is expected not to be worst than that of the 2 mm inner diameter.

6.6.3 Glue

Studies were conducted within the ITk community in order to improve the assembly process during the prototyping of the modules. Optimization studies were focused on alternatives for the silver epoxy glue that is used between ASICs and hybrid. The silver epoxy glue needs 6 to 12 hours curing time, slowing down the production.

Various UV cured glues were tested and are expected to have no detrimental performance impact during construction and operation of the modules such as ease of handling, flexibility and rigidity. Table (6.19) gives the material information for the new UV glue. The properties of the silver epoxy glue are also given for reference. Both of the glues show the same thermal performance based on the FEAST2 ASIC temperature. Figure (6.18) shows the FEA results looking at a cross section from the core material towards the FEAST2 ASIC at R5.

Table 6.19: Material information for latest ITk petal thermal model

Part	Material	Thermal conductivity W/mK	Thickness mm
ASIC to Hybrid	epoxy glue	2.99	0.08
	tra-duct-2902	(50% coverage)	
ASIC to Hybrid	UV glue	0.5	0.08
	Loctite 3525	(50% coverage)	

Figure 6.18: FEA mean element temperature results for the silver epoxy and UV cure glues.

The points refer to the same elements in the geometry for both. Similar thermal performance is predicted for both types of glue. Similar behavior is seen also for a cross section between the core material and the readout ASIC but only the cross section results of the FEAST2 are shown here. No error due to the computational modeling is included here since it is difficult to estimate that. The resistance in the glue layer is higher than the rest of the resistances. Someone can argue that due to the geometrical complexity of the petal such a representation of the results is not very accurate. However, since the FEAST2 ASIC occupies a small area and the temperature is uniformly distributed on the FEAST2 ASIC, figure (6.18) is used to give the comparison between the two types of glue.

6.6.4 Facesheet material

An alternative for the K13D2U material is examined to reduce the material mass budget (%X0). The new material is a K13C2U 45 gsm. It is lighter than the K13D2U 80 gsm giving lower %X0. The mechanical performance of the petal is similar for both materials as shown in ref. [61]. The K13C2U 45 gsm material gives a slightly higher mean silicon temperature by 0.7 °C. Despite the slightly warmer silicon the K13C2U 45 gsm material is preferred since it contributes less to the material mass budget.

6.6.5 Addition of carbon foam “ribs”

In an attempt to cool the FEAST2, extra carbon foam is added in place of the core material. Goal is to conduct the heat from the FEAST2 ASICs to the coolant via a shorter thermal path and reduce the hot spots. Figure (6.19) shows the silicon temperatures without (top) and with (bottom) extra carbon foam “ribs”. The addition of a carbon-foam section in place of the honeycomb right below the DC/DC converters decreases the temperatures of the silicon and particularly the hot spot in R3 by roughly 2°C. Figure (6.20) compares the mean temperatures of the FEAST2 ASIC at different rings with and without extra carbon foam ribs. Due to the temperature uniformity on the ASIC the mean temperature of each FEAST2 ASIC is plotted. The FEAST2 ASIC temperatures are slightly improved, but in itself not sufficiently to justify the extra complication in petal manufacture. Therefore, the addition of carbon foam below the DC/DC converters is not proposed.

Figure 6.19: FEA silicon surface temperatures without (top) and with extra carbon foam (bottom) added below the DC/DC converters.

Figure 6.20: FEA FEAST2 ASIC temperatures without and with extra foam “ribs” sections below the DC/DC converters, with two FEAST2 ASICs in R1 and R3.

6.6.6 Change of tube configuration

At the TID case, the predicted temperatures give the hottest spot at R5 under the FEAST2 ASIC in the DC/DC converter as seen in table (6.17). The current tube configuration doesn't cool the FEAST2 ASIC adequately and the addition of extra carbon foam gave insufficient improvement. For this reason, a new tube configuration is examined, with the tube passing right below the DC/DC converters especially for the outermost 2 sensors. Figure (6.21) shows the geometry of the new tube. The main difference with figure (6.6) is that in the new configuration the tube passes right below the FEAST2 ASICs in R5 instead of the ABC ASICs. The heat transfer coefficient is re-calculated for the 1.3 mm inner diameter for the two power cases using the new tube length (1.5 m instead of the old 1.4 m).

Figure 6.21: View of petal with modified tube passing under the DC/DC converter in R5, whereas figure (6.6) had the tube passing right below the readout ASICs. Both sides are visible in this figure.

Figure (6.22) presents the FEAST2 ASIC temperatures at the different rings for the old configuration and the new one where the tube passes below the DC/DC converter in R5. The FEAST2 temperature in R5 drops significantly (about 11 °C). Careful positioning of the tube is important. However, given that the final DC-DC converter designs are not available yet, this is a difficult improvement to guarantee. Furthermore, the tube path cannot be optimized for all FEAST2 chips; R0 remains hot (30 °C) and R3 and R4 get hotter with this change. We recommend continuous improvement of the petal thermal performance as the detailed design evolves.

Figure 6.22: FEA FEAST2 ASIC temperatures at different rings for the old and new tube configuration. The re-routed tube gives lower temperature of about 11 °C at R3 while it gives increased temperatures up to 5 °C for R3 and R4. Continuous optimization of the tube configuration is recommended as the design evolves.

6.6.7 Modeling CoBra with non-uniform heat flux

The hot spots along the detector due to the ASICs, seen in figure (6.19), result in a non-uniform heat flux distribution along the tube. To accurately predict the heat transfer coefficient along the tube based on the non-uniform heat flux, iteration between Cobra and Abaqus is needed.

Abaqus provides to Cobra the geometry of the tube, split in smaller sections, and the wall heat flux for each section. Cobra calculates back, for each section, the heat transfer coefficient and the coolant temperature. The heat transfer coefficient and the coolant temperature are fed back to Abaqus in order to calculate the new values of heat flux along each section. The initial values for Abaqus are based on uniform heat transfer coefficient and coolant temperature. The iteration process stops when the results are stable.

Figure (6.23) shows the FEA temperature results for the sensors at uniform heating and non-uniform heating. The temperature results for the first three iterations of non-uniform heating are presented. It can be seen that the converge is reached already with the first iteration between Abaqus and Cobra. With non-uniform heating, the tube gets locally warmer (hot spots) close to the readout ASICs and FEAST2 ASIC which makes the non-uniform heating more realistic.

The heat transfer coefficient, calculated with CoBra based on the non-uniform heat flux, fluctuates over a short length of tube but increases overall along the length of the tube. A locally lower heat transfer coefficient in combination with a peak in heat flux, will cause locally the temperatures to increase. The non-uniform heating gives higher mean sensor temperature 0.5-1 °C in the end of life case and up to 2 °C in the TID case. Sensor in R3 has the largest temperature increase due to the higher power load from FEAST2 ASIC compared to the other rings.

Figure 6.23: Comparison of the FEA results for uniform and non-uniform heating and for the first 3 Abaqus-CoBra iterations (non-uniform heating 1 to 3). Convergence is reached at the first iteration. Silicon temperatures increase by 1 to 2 °C, i.e. uniform heating is optimistic by this amount.

6.7 Conclusion

In this chapter, a petal is modeled that includes ASICs and hybrids, and the silicon sensors are properly modeled with a temperature dependent self-heating. The CoBra is used confidently to calculate the flow and thermal properties of the CO₂ since the tube characteristics are similar to the prototype presented in chapter 5. An FEA thermal model was built and the current design was examined under two cases during the operation; the TID case and the end-of-life. The results revealed some important issues in the current design that need to be tackled in order to run safely during the 10 years of operation. Firstly, the DC/DC converter ASIC receives more current than it can handle and therefore a considerable drop in the efficiency takes place leading to excess heat at the ASIC. Therefore, a design with 2 DC/DC converters is proposed here instead

of 1 for R1 and R3. The FEA results showed that the presence of 2 DC/DC converters in R1 and R3 significantly reduces the hot spots on the silicon by 10 °C.

Secondly, the study of the CO₂ in terms of flow and heat transfer revealed that in the current design the flow regime expected is not optimum. Optimum flow regime is considered one that has high heat transfer between the coolant and the wall. Particularly, the study of different inner diameters in terms of heat transfer and thermal runaway showed that a decrease in the inner diameter would improve or give at least similar results as a larger one. A smaller diameter is preferred even when the thermal performance is similar because of the better radiation length. Moreover, according to the studies in [61] a smaller diameter will not deteriorate the mechanical performance of the petal, is easier to build and creates less stress during cool down. For these reasons, an inner diameter down to 1.3 mm is recommended.

Furthermore, the facesheet material K13C2U 45 gsm is preferred than the K13D2U 80 gsm. Despite the slightly warmer silicon the K13C2U 45 gsm material is preferred since it contributes less to the material budget, and the mechanical performance of the petal remains adequate [61].

Other optimization steps such as change of the tube configuration, change of the ASIC-hybrid glue and adding carbon foam in places can improve the thermal performance but depend on the positioning of all the power dissipating components which is still unclear. Finally, it is suggested to use CoBra with non-uniform heat flux input as it can improve the precision. It was shown that the uniform assumption is optimistic by 1 to 2 °C.

Chapter 7

Single micro-channel studies

The previous chapters presented studies on the Petal detector that included embedded mini-channel cooling with evaporative CO₂. The empirical models for CO₂ evaporation flow in mini-channels have been studied thoroughly by Cheng et al [1], [2]. However, high power density detectors in future upgrades that are located at critical areas will require the usage of micro-channels in order to save material and offer sufficient thermal performance. Empirical models for micro-channels exist, as described by Dupont et al in refs. [8] and [9], and by Cioncolini et al in ref. [10]. Unfortunately, those have not been verified for CO₂ evaporation in micro-channels (inner diameter <0.5 mm). This chapter describes the experimental set-up including the cooling plant and the test section built for studying flow characteristics and heat transfer during evaporation of CO₂ in a single micro-channel of 0.25 mm inner diameter. Results are presented and a discussion follows on the physical phenomena seen during measurements.

7.1 Blow system

An open loop cooling system was developed at Nikhef and is aimed for CO₂ cooling studies of detector units. It is known as blow system because the coolant is blown-off to the environment at the end. The blow system is an easy to build and use system with relative short start time (20-30 minutes depending on the initial settings of pressure and mass flow) and it does not include any moving parts. A description of the blow system, that is schematically depicted in figure (7.1), can be found in ref. [62] and in ref. [63].

The blow system works with high pressure (>50 bar) liquid CO₂ at room temperature taken from a 50 liter bottle equipped with a syphon type cylinder. The warm CO₂ is pre-cooled in a counter-flow heat exchanger (CFHX) and then expanded to the desired pressure before entering the test section. The coolant is at the saturated liquid line and when it passes the heated test section it evaporates. The 2-phase low pressure (15-20 bar) CO₂ is used afterwards to pre-cool the incoming warm liquid CO₂ in the counter-flow heat exchanger.

Before expanding to atmospheric pressure, the CO_2 is heated externally with circulating warm water (50°C) in a Julabo thermal bath. The operating mass flow range is 0.1 to 3 g/s which can cool the experiment down to -35°C with a cooling capacity up to 600 W. A Rheonik Coriolis flow meter measures the total mass flow and a Swagelok metering needle valve is used to adjust the total flow. The pressure is set using a high pressure proportional relief valve after the thermal bath. A pressure gauge indicates the pressure.

Figure 7.1: Schematic drawing of the blow system. A bottle supplies liquid CO_2 at room temperature to a counterflow heat exchanger where the temperature is decreased. A pressure regulator downstream decreases the pressure. The CO_2 entering the heated section of the experiment evaporates partially.

7.2 Modification of blow system

The blow system has been modified to study the evaporation flow in a single micro-channel. The mass flow in a single micro-channel is an order of magnitude lower than the lowest flow that the blow system can generate in a controllable way. Moreover, increased environmental heat leaks are expected when running at small flows and below 0°C . The control of the inlet conditions are not guaranteed with the existing cooling plant. Therefore, the experimental set-up for testing a single micro-channel includes the blow system with some additions/modifications. Figure (7.2) shows the schematic drawing of the adapted blow system to study the flow in a single micro-channel. The modifications are discussed in the next sections.

Figure 7.2: Schematic drawing of adapted blow system. The inlet CO_2 line is split into 2 lines where most of it is used to cool the incoming CO_2 in the counterflow heat exchanger, whereas the rest is directed to the test section where it evaporates partially due to the heating. The by-pass line (green) pre-cools the inlet of the vacuum section via a heat exchanger (HX) before passing the CFHX.

7.2.1 By-pass and experiment

The single micro-channel will be tested at small flows ($<0.1 \text{ g/s}$). In order to succeed in this, the main flow is split into two lines, the experiment line, and the by-pass line. The experiment line includes the test section/evaporator. In the experiment line the desired mass flow is set, and the remaining flow is directed to the by-pass line. Increased heat leak is expected due to the low flow in the experiment. For this reason, the two lines are set at different pressures using the pressure regulators at the end of each line. The by-pass line is designed at the lowest pressure, and consequently, at lower temperature, and it is used to sub-cool the incoming warm liquid CO_2 . The lowest pressure that can be achieved in the by-pass line is 10 bar to avoid solidification. The experiment line is set at the pressure of interest.

7.2.2 Mass flow

The total flow to the system is measured, as shown in figure (7.1). For the two-phase studies the flow in the micro-channel itself needs to be known. For this reason, a Bronkhorst mini-Coriolis flow meter is added that measures the

mass-flow rate for the experiment line. The flow device is located right after the thermal bath in figure (7.2). The mass flow device has a temperature limit >0 °C for the measuring gas and since the 2-phase CO₂ returning from the experiment is at -30 °C, the returning line must be heated further via the thermal bath before the mass flow measurement. Only the mass flow rate to the experiment is measured directly.

7.2.3 Pressure

The pressure in the experiment line is set close to the outlet of the blow system with a Bronkhorst electronic backward pressure control valve and measured with a Bronkhorst pressure meter. A differential pressure transducer is also used to measure the pressure drop over the test section and the return line. Table (7.1) provides information on the pressure sensors and mass flow meters for the set-up.

Table 7.1: Range specifications of the sensors and devices in the set-up.

Sensor/Device	Type	Range
Pressure absolute meter	GE UNIK 5000	0.07 to 700 bar
Pressure differential meter	Druck LTD PDCR 130/35 WL	0 to 70 bar
Pressure relief valve	Swagelok SS-6R3A-MM	3.4 to 24.1 bar
Pressure regulator	Bronkhorst P-512C-M10A-AAD-19-E	2 to 100 bar
Control valve	Bronkhorst F-033C-LIU-11-E	12 to 40 bar
Mass flow meter	Bronkhorst mini-Coriolis M13	0.01 to 0.5 g/s
Mass flow meter	Rheonik Coriolis RHM015	0.07-10 g/s
Mass flow transmitter	Rheonik Coriolis RHEM07	0 - 5 V

7.2.4 Vacuum insulation

The test section is placed in a vacuum chamber to prevent conductive and convective heat leak and condensation of water vapor or ice on the tube. An estimation of the amount of heat leak to the test section is described later. Early tests showed increased heat leak in the section right after the splitting of the main line and before the test section, making it impossible to control the inlet conditions; to stay at the saturated liquid line or sub-cooled liquid area. For this reason, a custom-designed vacuum feed-through was built to guarantee sub-cooled CO_2 liquid at the inlet of the test-section. The vacuum feed-through is a tube-in-tube heat exchanger and is labeled as HX in figure (7.2). It uses the low pressure CO_2 to sub-cool the experiment line. Figure (7.3) shows the vacuum feed-through CAD drawing and the final product as manufactured at the workshop of the University of Twente.

Figure 7.3: Vacuum flange as it is designed and built to keep the CO_2 at the inlet of the experiment at sub-cooled conditions; left: rendered 3D drawing; right: as built.

7.3 Test section

The test section in the vacuum space is shown in figure (7.4). The experiment includes a preheated section, the evaporator and a glass section for visualization. All the sections are circular tube channels. The preheated section determines the inlet conditions. The diameter of the preheated section is chosen to be in the range of a mini-channel in order to accurately estimate, using known models, the pressure drop over its length. The evaporator is the main section of interest for the study of the flow and heat transfer characteristics. The inner diameter is 0.25 mm, which is in the typical range for micro-channel CERN applications. The length (200 mm) is chosen in order to catch the transitions from one regime to the other and examine high heat loads. The glass section aimed for visualization of the flow patterns is of the same inner

diameter as the evaporator. It is located after the evaporator at a distance of 20 mm. The main assumption is that the heat leak in the glass section is negligible and the flow pattern seen in the glass section is the same as the flow regime at the exit of the evaporator. Table (7.2) presents the material and geometrical information of each section. PEEK connectors (UPCHF-151) were used between the pre-heater and evaporator and between the evaporator and the glass tube. The return line immediately after the glass section has the same diameter as the preheater section.

Figure 7.4: Test section in vacuum where red lines indicate the heated sections.

Table 7.2: The material and geometrical information of the different sections in the test section built to study flow in a single micro-channel.

Section	Material	OD	ID	Length
	mm	mm	mm	mm
Preheater	SS	1.6	0.88	100
Evaporator	SS	1.6	0.25	200
Visualization	Fused Silica	0.35	0.25	50
Exit/return line	SS	1.60	0.88	350

7.3.1 Heating

The heating of the preheated and evaporator sections is realized with manganine heating wire of 0.1 mm OD. The heating wire characteristics (length and resistance) are selected based on the maximum heat load required for the tests.

Table (7.3) summarizes the heating wire information for the two sections and the total heated length. The heater is then extended with manganine wires of 0.2 mm OD and further down with copper wires to minimize heat leaks. The copper wires are soldered to the connector in the vacuum chamber and from there the wires are connected to a Delta SM7020-D power supply. Two power supplies provide the voltage to the heaters. In order to estimate the heat transfer coefficient, temperature readings are needed along the outer wall of the section. Therefore, the heater is wrapped in turns leaving unheated sections of 0.5 mm in between, large enough to place temperature sensors while at the same time keeping a uniform temperature at the inner wall of the channel. The uniform temperature is an important assumption in order to reliably compare the experimental results with the prediction models. In Appendix A it is shown that it is fair to assume uniform temperature. Figure (7.5) shows the heater pattern.

Table 7.3: Wire information for the heater and the pre-heater used in the test section. The heater wire for the preheated section is wrapped around the tube over a total distance 65 mm, whereas the total length for the evaporator heater is 100 mm including the gaps.

Heated section	Heater wire	OD mm	Resistance Ohm	Heated length mm
Preheater	manganine	0.1	148	65
Evaporator	manganine	0.1	118	100

Figure 7.5: Heating manganine wire wrapped around the micro-channel leaving 0,5 mm gaps in between for the temperature measurements.

7.3.2 Temperature sensors

Temperature sensors are placed along the test section to measure the wall temperature at various locations. Figure (7.6) shows the positions of the temperature sensors. A PT1000 class Y resistance temperature detector (RTD) with 4-wire configuration is placed before the evaporator. The PT1000 is used to measure a reference absolute temperature (T_{ref}) since it has better accuracy than the thermocouples. The PT1000 is placed at 5 mm distance from the

heater location to avoid any effect due to backwards heat conduction from the heated section (see figure (A.3)). Thermocouples type E (T1-T6) are placed on

Figure 7.6: Test section including the preheater, evaporator and the glass section.

the outer wall of the evaporator on various positions over its length, in between the heater sections. The thermocouples are custom made type E with chromel and constantan wires connected in such a way that we could measure delta-T directly, as described in ref. [64]. The cold junction reference for the thermocouples is located before the evaporator, where the PT1000 is placed. This is achieved by having one of the junctions of the thermocouple glued on top of the PT1000. In this way, a direct measurement of the ΔT between the tube wall location and the PT1000 is given. The other junction of the thermocouple is placed on the available unheated spaces on the evaporator. The available unheated space on the evaporator is limited by the requirement of uniform heating as discussed earlier. Therefore, thermocouple wires of 0.05 mm OD are selected in order to fit in the available space. The small contact point (junction) of the thermocouples will give fast response making the determination of rapid changes in the flow easier to detect, and due to their low mass it will not affect the sensed temperature by thermal shunting. And finally, two thermocouples are placed also before the pre-heated section and after the evaporator to provide reference temperatures (T_{pre} and T_{ah}). All temperature sensors are wrapped by one turn on the tube to have more accurate reading and minimize heat leaks. The PT1000 sensor is extended with manganine wires that have low conductivity and then with copper wires that are soldered directly to the connector. The thermocouples type E, that consist of chromel and constantan wires, are extended with copper wires that are soldered to the connector. Figure (7.7) shows the connection wires for the different sensors.

Figure 7.7: Illustrative drawing of the test section with the heater and temperature sensors. Manganine and copper wires are used to extend up to the connector to minimize heat leaks towards the test section.

7.3.3 Flow visualization

The observation of the flow pattern is done in the glass section right after the evaporator. A FASTCAM SA-X2 Photron high speed camera is used in combination with front illumination using a fiber optics bundle Olympo ILP1 at maximum illumination. A white paper is placed between the glass and the light source to act as diffuser. The lenses include a Navitar 12x telecentric zoom system.

7.4 Data acquisition and Control system

A Labview program was built to monitor and control the experiment. A PICO USB TC-08 thermocouple data logger is used for reading the thermocouple voltages. A NI DAQ card measures the signals from the pressure sensors, mass flow meters, and PT1000. The absolute pressure sensor signal is scaled to 0-5 V and converted to bar units with a linear relationship. The differential pressure sensor output is a current and, therefore, a resistance is added to give a voltage in the range 0-10 V and then converted to bar units with a linear relationship. The PT1000 is connected to the NI DAQ card and a capacitor to minimize fluctuations in the reading. The power supplies are connected through amplifiers to the NI DAQ card. The voltage is controlled via a Labview interface, whereas the current is set manually and limited to a

max value to avoid overheating the wires. Two temperature sensors are placed on the junction of the heater wires to monitor the temperature increase of the wires during heating. A control loop sets the voltage to zero when the temperature on the heaters exceeds a safety value of +100°C. Table (7.4) summarizes the type of sensors used and their accuracies.

Table 7.4: Accuracy specifications of the sensors and the devices used for the measurements and their accuracies according to the manufacturers/suppliers. FS stands for full scale and BLS the best fit straight line.

Sensor/Device	Type	Accuracy
Pressure absolute	GE UNIK 5000	0.04% FS BSL
Pressure differential	Druck LTD PDCR 130/35 WL	0.1% or 0.05% BSL
Pressure controller	P-512C-M10A-AAD-19-E	0.5% FS
Temperature	TC type E	1.7 °C or 0.5% of reading
Temperature	PT1000 class y	0.10 + 0.0017 x abs(t)
Mass flow	Bronkhorst mini-Coriolis M-13	0.50% of reading for gas
Mass flow	Pheonik - 015	0.20% of reading (0.1*Qmax)
Power	Delta power supply	0.2% + 0 mV to +8 mV (5V)at CV
Current source	Lakeshore model121	0.1%
NI DAQ card	model 6259	0.1%
TC reader	PICO data logger USB T08	Sum of 0.2% of reading and 0.5 °C

7.5 Accuracy

The absolute accuracy of each measurement depends on the sensor and on the measuring device of the signal. The Root of Sum of Squares (RSS) method is used to estimate the total error due to the different contributions since the errors in the sensor and the reading device are uncorrelated. Eq. (7.1) gives the RSS method where m is the number of variables contributing in the total error and x the error of each variable.

$$RSS = \sqrt{\sum_{n=1}^m x_n^2} \quad (7.1)$$

Table (7.5) summarizes the calculated errors based on eq. (7.1). The analysis was performed assuming that error sources such as broken TCs, TC shunting, and other “major” issues have been eliminated. Systematic errors caused by TC junction type and mounting scheme are usually negligible for normal environments.

Table 7.5: Estimation of errors during measurements based on RSS method. The thermocouple error is estimated with the cold junction measured with the PT1000.

Sensor/Reading	Error
Thermocouple Type E	1.70 °C
PT1000	0.14 °C
Pressure drop transducer	7 mbar
Pressure transducer	- mbar
Coriolis sensor	-
Power supply- Voltage	14 mV
Power supply-Current	4 mA
Pressure controller	5.6 mbar

7.6 Check of the temperature sensors

The temperature measurements are examined prior to any analysis to determine their reliability. Data were collected with the micro-channel placed in the vacuum environment, without any flow in the micro-channel and without heating. When the heater is off, all the measurements on the evaporator give similar temperatures within a range of 0.1 °C. The T_{ref} is slightly higher by +0.4 °C but still within the error range as shown in table (7.5). Any difference can be attributed to self-heating. The T_{pre} is slightly higher than the T_{ah} by 1 °C. This temperature can be expected since the T_{pre} measurement is closer to the wall of the vacuum section and is warmed up easier. Next, heating of the channel showed a uniform temperature increase of all the sensors (T_1 to T_6). No change in T_{pre} and T_{ref} and T_{ah} was seen.

Data were collected with flow in the micro-channel. When the heater is off, the temperature of the T_{ref} is higher by +2 °C than the temperatures on the heater section (T_1 to T_6). This is highly suspicious since T_{ref} should measure the temperature at the entrance of the heated section, which always should be lower than the temperature at the heater. A further inspection of the T_{ref} sensor showed a poor contact interface with the tube, resulting in a higher temperature. The T_{pre} is always lower than the T_{ah} by up to 5 °C (depending on mass-flow rate of the sub-cooled CO_2) while the T_1 to T_6 measurements were all close to each other within their error ranges. The T_{ah} was always higher than the T_1 to T_6 , but still no conclusion could be drawn since all were within their error ranges.

When very small heat is applied via the heater, the T_{ref} is higher than the rest, whereas the T_1 to T_6 and T_{ah} follow a clear ascending order from inlet to outlet. All the temperature measurements are logical and qualitatively as expected apart from the T_{ref} as discussed above.

7.7 Analytical estimation of heat leak

The heat leak in the test section is a combination of different parameters resulting in a complicated thermal circuit, which is difficult to solve analytically. Even in a vacuum insulated environment, there is always heat leak from the surroundings due to radiation and conduction through the measuring wires and due to bulky components like connectors that are not cooled adequately. In this section the heat leak due to heat conduction and radiation is examined. All the wires of the sensors and the heaters, used in the test section, are soldered to a metallic connector in the vacuum chamber (see figure (7.7)). The other side of the metallic connector which is outside the vacuum chamber has all the wires going to the data-acquisition devices. Therefore, it is assumed that the wires at one end of the connector are at room temperature while the wires at the other end of the connector are at a lower temperature due to the cold test section. As a worst case the wires in the vacuum chamber are assumed to be at -20°C giving a total temperature difference of 40°C . Table (7.6) gives the length, diameter and thermal conductivity of each wire. The length of the wire that is wrapped around the tube, and therefore is cooled, is not included in table (7.6).

Table 7.6: Material information for the sensors, heaters and the extended wires used for the temperature sensors and the heaters.

Material	Diameter	Length	Thermal conductivity
	mm	mm	W/mK
Chromel	0.12	0.1	19
Costantan	0.12	0.1	19.5
Copper	1.2	0.3	401
Manganine	0.2	0.005	22
Manganine	0.1	0.03	22

Table (7.7) presents the estimated heat leak conducted due to the wires. The estimated worst case heat leak is small compared to the latent heat of evaporation, which is of the order of 3 W. Taking into account a more realistic temperature difference between the one end of the wire and the other end, the heat leak estimated is even smaller. Regarding heat radiation, since the test section as depicted in figure (7.6), is much smaller than the outside wall of the vacuum chamber, the radiative heat load on the test section can be evaluated using eq. (2.9), with A the outer area of the tubes, that face the vacuum chamber inner walls. The heat leak due to radiation is estimated to be less than 0.1 W, assuming an emissivity coefficient of 0.1 for the stainless steel section and 0.85 for the glass section, and an outer wall temperature of -20°C (as it is seen in figure (7.9)). The heat leak due to radiation is smaller

than the heat leak due to conduction and, therefore, it is neglected here. Later in chapter 8 the estimated heat leak is compared with the measured heat leak.

Table 7.7: Estimated parasitic heat due to the wires of the temperature sensors and the heaters.

Sensor	Number of sensors /wires	Parasitic leak W
Thermocouple type E	7/14	0.42
PT1000	1/4	0.26
Heater/Preheater	2/1	0.13
Total		0.80

7.8 Measurements

The measurements of the single straight micro-channel form a two-dimensional matrix in which mass flow and power are varied at a constant saturation pressure. The test section is sub-cooled and then heated with the heater and/or the preheater. Boiling starts, measurements are taken and the flow pattern is monitored in the glass section. This section presents a set of illustrative measurements performed during the tests. Figure (7.8) shows the temperatures as a function of time following changes in the heater power. Figure (7.9) shows the corresponding mass flow and pressure drop. The horizontal axis is the time in seconds and is set to 0 at the start of the recording. Measurements were taken every 2 seconds. The pressure at the by-pass line shown in figure (7.2) was set at 20 bar by adjusting the pressure regulator and the mass flow at 0.045 g/s. The temperatures were at equilibrium after approximately 5 seconds after the experimental valve was opened. The temperatures at the start of the graph show the equilibrium temperatures at $-24\text{ }^{\circ}\text{C}$. The preheater was switched on after 180 s at 0.8 W. Then the heater was switched on in steps, up to 5 W between 250 s and 1190 s keeping the preheater constant. The heater was switched off at 1190 s. Then, at 1690 s only the preheater was increased in steps, reaching 1.25 W and subsequently it was decreased to zero, in steps. At 2300 s, the preheater was switched on at 1.25 W and the heater was increased stepwise reaching 5 W. At 3560 s, the heater was turned off whereas till 3870 s only the preheater was on. The preheater was increased then to 4.8 W while the heater was increased in steps, reaching 5 W. At about 4150 s the preheater was off, and at about 4380 s the heater was switched off. In figure (7.8), the T_{satp} and $T_{satexitp}$ temperatures are the saturation temperatures calculated from the pressures at the inlet and outlet, respectively, using REFPROP. These temperatures give an indication of whether liquid or two-phase cooling

takes place in the channel. The switching on of the preheater at 180 s, gave a sudden reaction on the temperature readings and mass flow. The flow now is no longer sub-cooled but at saturated condition. After an initial sharp increase most of the temperatures, dropped soon after, whereas T_1 and T_2 did not. At 400 s, with the heater on and increased in steps, the temperatures of T_1 and T_2 kept increasing indicating boiling delay. The other temperatures increased but the slope was much smaller compared to that of T_1 and T_2 , indicating better heat transfer than at T_1 and T_2 . At 750 s, with the increase of the heater power to 1.8 W, all temperatures at the evaporator dropped including T_1 and T_2 , indicating a change in the regime and/or switching from single phase to 2-phase cooling.

Figure 7.8: Recording of the wall temperatures and the heater powers during two-phase flow. The saturation temperatures (T_{satp} , $T_{satexitp}$) in the top graph indicate the CO₂ temperature at the inlet and at the outlet of the micro-channel, respectively, based on the pressures at those positions. The sharp jumps in the wall temperatures indicate the transition from single-phase to two-phase flow.

Figure 7.9: Recording of the mass flow settings and the pressure drop evolution during the two-phase flow experiments presented in figure (7.8). During a transition from single-phase to two-phase flow an instant peak reaction in the mass flow is seen as well as in the pressure drop. The pressure drop changes with the two-phase flow and the heating power.

At about 1200 s, after switching off the heaters, the temperatures dropped back to -24°C . Between 1690 s and 1800 s, while the preheater power was increased in steps up to 1.5 W, the temperatures increased reaching -5°C . Then, as the preheater power was decreased in steps, the temperatures also decrease, reaching the -24°C soon after the preheater was set off. No hysteresis was seen. At 2300 s, the preheater was switched on at 1.25 W and the temperatures reacted quickly, reaching the same values as in the previous step. This time the power was on for a longer time, and the switching from single to 2-phase cooling was seen with the temperatures dropping to -18°C . The T_{ah} dropped to -20°C and followed the saturation temperature (T_{satp}). At 2500 s, the heater power was increased in steps and the temperatures at the tube wall increased while the T_{ah} was fairly constant. At 2870 s, with constant power, a drop in the temperature readings was seen. The temperatures and the mass flow started fluctuating indicating the transition to a different regime. As the power of the heater was increased further, the fluctuations decreased. The temperatures were slightly increased, compared to the increased power, indicating a regime with increased heat transfer. At 3690 s, the temperatures increased to a range between -17 to -16°C with the increase of the preheater power.

7.9 Discussion

The measurements show that the temperature recordings clearly indicate the transition from single-phase to two-phase flow. Starting from sub-cooled conditions, heating results in warming up of the liquid flow and boiling delay does occur, resulting in temperatures above saturation value. As soon as boiling starts, the temperature drops to the saturation point and stays constant at further power increase. The temperature readings on sections before and after the heaters (T_{ah}) give also a good indication of the fluid temperature. Particularly, T_{ah} seems to be close to the fluid temperature and it changes along with the pressure. Superheating is quite often seen in the measurements, reaching in many case a ΔT of $+15^{\circ}\text{C}$ which is repeatable. In real-life applications, superheating is an undesired condition and should be prevented. Finally, the pressure drop follows the power input which is also expected because as the power increases, more liquid evaporates and the vapor quality increases leading to higher pressure drops.

Chapter 8

Single micro-channel results

Having described the experimental set-up and presented some preliminary measurements, we proceed with the data derivation and interpretation. In this chapter, the experimental data from the two-phase cooling tests on a single straight micro-channel are processed. The two-phase data are compared with the theory of two-phase flow to check whether the already existing models predict well the behavior of CO_2 evaporative cooling in micro-channels. A discussion on environmental heat leaks and temperature corrections is also presented, and the set up is validated with single-phase data before proceeding with the derivation of the two-phase results.

8.1 Temperature calibration

The temperature sensors, as discussed in section 7.3.2 and depicted in figure (7.6), are calibrated before using them to estimate the thermal behavior of the micro-channel. As discussed in section 7.6, initially the aim was to measure temperature differences directly from the T_{ref} to each thermocouple, to more accurately measure temperature changes. However, since the T_{ref} was not thermally coupled well enough to the fluid, because of a poor thermal contact of the sensor mounting block to the tube, it was reverted to measuring the temperature itself at each point, and deducing the temperature changes. Since all the thermocouple measurements are taken with the T_{ref} as the cold junction reference, they give an accurate measurement of the temperature difference (ΔT_{tc}) between the T_{ref} and the various measurement points (T_1 to T_6) on the micro-channel. The absolute temperatures at the various measurement points on the micro-channel are found by adding the T_{ref} to ΔT_{tc} measurement of the thermocouple in T_1 to T_6 . The section in the evaporator showed negligible heat leak when looking at the temperatures T_1 to T_{ah} as will be discussed in section 8.2. Therefore, the mean temperature of T_1 to T_{ah} without heating applied can be used as the temperature of the fluid prior to the heating, which

is defined as T_{start} . The temperature measured at the outside of the tube at the different positions ($T_1 - T_6$) is defined as:

$$T_{wallout} = \Delta T_{tc} + T_{ref} \quad (8.1)$$

where ΔT_{tc} is the thermocouple reading at the points T1 to T6 of figure (7.6), and T_{ref} is the PT1000 reading. In all cases, when absolute temperatures are used they are corrected for the offset of T_1 to T_{ah} without heat being added via the heaters.

8.2 Measurement of heat leak

Environmental heat leaks to the test section were examined prior to the measurements, where the test section was cooled down to -20 °C. The procedure was as follows. The test section was cooled with CO₂ and no extra heat load was added through the heaters. The CO₂ was subcooled by around 5 °C from the saturated temperature. The difference in the CO₂ temperature between the inlet and outlet thus indicated the amount of heat leak to the test section. A ΔT_{tot} increase between the preheater (T_{pre} measurement) and the exit of the evaporator (T_{ah} measurement) is seen. The temperature difference between T_1 and T_{ah} is only a small fraction of that ΔT_{tot} and it is within the uncertainty error (error due to averaging). Furthermore, the distance between T_1 and T_{ah} is smaller than the total distance between T_{pre} and T_{ah} , and therefore we can assume that all heat is added in the first section (i.e. the section between inlet of preheater and inlet of heater on the evaporator).

The heat leak between the preheater and the end of the heater is derived from eq. (2.10) and is shown in figure (8.1) for different mass flow cases. The temperature difference used in eq. (2.10) is the difference between the temperature measured in the preheater T_{pre} and the average temperature of T_1 and T_{ah} measured on the heater. Figure (8.1) shows that the average measured heat leak is $0.4 \text{ W} \pm 0.2 \text{ W}$. The measured heat leak is in fair agreement with the analytically estimated heat leak presented in table (7.7). The uncertainty in the measured heat leak data points is high (>80%) due to the very low temperature differences measured, which are comparable to the temperature sensor error.

Figure 8.1: Heat leak derived from measurements during single-phase cooling for different mass flow rates.

8.3 Single-phase

Single-phase experiments were performed in order to validate and prove the feasibility of the set-up prior to the two-phase experiments. The pressure drop measured and heat transfer coefficients derived from the measurements were compared with theoretical/empirical equations. The T_{ah} gives the temperature of the fluid when no heat load is applied (no ΔT between outer wall temperature and fluid temperature). Therefore, it is assumed that we are in the single-phase region when the T_{ah} is below the saturated temperature, which is the temperature calculated from the pressure at the inlet and outlet of the test section. Later, when heating is applied, the temperatures increase with the heating, indicating still the presence of single-phase flow.

8.3.1 Pressure drop

In this section, the pressure drop (dP_{meas}) measured in the test section is compared with the pressure drop (dP_{pred}) derived from the theoretical/empirical models. The pressure at the inlet (P_{in}) and the total pressure drop (dP_{tot}) is measured between the inlet of the preheater and the exit of the return line as shown in figure (8.2).

The total pressure drop is a function of several pressure gradients. From inlet

Figure 8.2: Illustrative drawing of the test section with the temperature and pressure sensors. P_{in} and dP_{tot} are measured, whereas P_{out} is evaluated. The total pressure drop is a combination of friction and pressure drop due to contraction, expansion and bend.

to outlet, there is frictional pressure drop along the preheated section, flow contraction, frictional flow in the micro-channel and glass section, flow expansion, bend and finally frictional flow in the return line. Empirical and theoretical correlations for single-phase flows given in eqs. (2.11) - (2.20) are used to estimate the dP_{pred} . The total dP_{pred} derived from the theory is compared with the measurements dP_{meas} in figure (8.3). The dP in the heater section only is a small fraction of the total dP , and the value is close to the accuracy of the sensor. Therefore, the dP in the heater section is not presented separately, but instead the comparison is made between the total pressure drop derived from measurements and predicted. Due to the low pressure drop in single-phase flow the error shown is comparable to the measured value. Nevertheless, it can be concluded that the derived from measurements dP_{meas} is predicted from the theory within the error range.

Figure 8.3: Comparison total pressure drop measured and predicted for single-phase flow.

8.3.2 Heat transfer coefficient

Temperature measurements were taken at single-phase flow in order to determine the heat transfer coefficient and compare it ultimately with prediction models. Several single-phase measurements were performed with the heater on. A limiting factor in this respect was the inaccuracy with which the power supply and the DAQ could measure very low voltages and currents for determining very low heating powers in addition to the presence of heat leak in the section before the heater.

Assuming a heat flow $\dot{q}$ (in W/m) is uniformly distributed over the length of the heater (no heat flow along the tube), the following equation gives the temperature of the fluid (T_{fluid}) at each location.

$$T_{fluid} = T_{start} + \frac{\dot{q}z}{\dot{m}c_p} \quad (8.2)$$

where z is the distance in [m] of each thermocouple from the start of the heater, T_{fluid} is the fluid temperature at the location z , T_{start} is the temperature when no heating is applied as described in section 8.1, $\dot{m}$ is the mass flow rate [kg/s], and c_p is the specific heat at constant pressure [J/kg K]. Also, we can write that at each position along the heater section the heat flow $\dot{q}$ enters the fluid

with section dz .

$$\dot{q}dz = hA(T_{wall-inner} - T_{fluid}) \quad (8.3)$$

where A is the area [m^2]

$$A = \pi D dz \quad (8.4)$$

and q is

$$\dot{q} = \frac{\dot{Q}}{L} \quad (8.5)$$

with $\dot{Q}$ the rate of heat transfer [W], and L the total length [m] of the heater on the evaporator. $T_{wall-inner}$ is the tube inner-wall temperature estimated from the outer-wall temperature following eq. (8.6).

$$T_{wall-inner}(z) = T_{wallout}(z) - \frac{\dot{Q} \ln\left(\frac{D_{out}}{D_{in}}\right)}{2\pi k L} \quad (8.6)$$

where $\dot{Q}$ is the rate of heat transfer [W] applied with the heater, D_{out} and D_{in} are the outer and inner diameter in m, k is the thermal conductivity of the wall in W/mK , and L is the total distance in length [m]. Using eq. (8.5) and solving eq. (8.3) for the heat transfer coefficient h we have:

$$h(z) = \frac{\dot{Q}}{\pi D L (T_{wall-inner}(z) - T_{fluid}(z))} \quad (8.7)$$

and substituting eq. (8.2) in terms of T_{fluid} and using eq. (8.5)

$$h(z) = \frac{\dot{Q}}{\pi D L (T_{wall-inner}(z) - T_{start}) - \frac{\pi D \dot{Q} z}{\dot{m} c_p}} \quad (8.8)$$

where z is the location at each point. The first and last thermocouples were affected by heat-flow out of the heater along the tube and, therefore, we neglect those in evaluating the heat transfer coefficient. The heat transfer coefficient is calculated for the remaining 4 thermocouples. We then have the following comparison of the average heat transfer coefficient with the prediction shown in figure (8.4). The blue lines indicate the error range for the measured values, which is due to the error in the temperature measurements. The error range for the predicted values of the heat transfer coefficient is assumed to be 40%. Most of the data are predicted within the error range. Everything within the red dashed lines also indicate an agreement within 40% between the measured heat transfer coefficient and the predicted. The flow is laminar and the average heat transfer coefficient is expected to be fairly constant, as described

in section 2.2.5. Taking into account the increasing error range for the lowest temperatures measured, figure (8.4) shows a satisfying comparison with the predicted heat transfer coefficient.

Figure 8.4: Comparison of heat transfer coefficient h derived from measurement and predicted for single-phase flow. The blue lines indicate the error range of the measured values, whereas the dashed red lines indicate the predicted value plus/minus an error of 40%.

8.4 Two-phase

During two-phase flow, the pressure and temperature were recorded and the flow pattern was observed, as described in section 7.8. The goal is to derive the heat transfer coefficient and pressure drop and compare with theoretical predictions. The two-phase measurements matrix for the single straight micro-channel is two-dimensional. Mass flow and heating power are varied separately. In the two-phase region, the local fluid temperature is directly linked to the saturation pressure. It is assumed that the section after the heater where the temperature T_{ah} is measured is adiabatic (no heat exchange between the tube and environment) and it shows the temperature of the fluid at the exit of the

evaporator. In figure (7.8) a jump in temperature with increasing power indicates we enter the two-phase region, and from there then the temperature is stable. Any decrease in fluid temperature from inlet to outlet in two-phase flow is due to the pressure drop. Therefore, data are selected when the temperature T_{ah} is between the saturated temperature at the inlet T_{satin} (derived from the inlet pressure) and the saturated temperature at the outlet T_{satout} (derived from the outlet pressure), indicating we are in two-phase region. Particularly, steady state data were selected after the ‘jump’ in the temperature was seen (as it was discussed in section 7.8), indicating that at the entrance of the evaporator there is two-phase flow and assuring saturated conditions. No heat outflow is considered along the tube. In the next discussions, the only correction applied on the T_{ah} and the evaporator temperatures is the offset in the temperatures when no heat is applied since before any heating is applied, all the sensors should show the same temperature.

8.4.1 Flow Regimes

In single-phase flow the type of the flow (laminar, transient or turbulent) is the starting parameter to be defined. In two-phase flow this information is still useful but more important is the knowledge of the liquid-vapor distribution in the flow. The flow regimes in the single micro-channel were observed and are compared to the theoretical predictions. Particularly, visualization of the flow was possible via the glass section at the exit of the evaporator as described in chapter 7. Effort was taken to improve the image by adjusting the illumination and shuttered light but it was not always possible due to limitations of the camera. The images taken were black and white, where white indicates the presence of vapor and black or darker colors indicate the presence of liquid. The circular geometry of the channel and the thickness of the channel enlarge and distort the image (outer diameter 1.6 mm and inner diameter only 0.25 mm). Moreover, at high flows, it was quite difficult to capture details like liquid film thickness, therefore, the images were used only as a qualitative indication of the flow pattern regime. No detailed parameters could be obtained. In the experiments here, different flow patterns were recorded that can be interpreted and categorized in 3 main regimes. Initially, small bubbles are seen of diameter smaller than that of the inner diameter of the micro-channel. These bubbles were random and sometimes a couple of bubbles are seen together but never more. This is the first regime known as isolated bubbles (IB) regime. In figure (2.1) this is shown as bubbly flow. The isolated bubbles were all spherical as a result of the strong interfacial pressure difference. As the heating power is increased, more spherical bubbles appear, that increase in diameter. As soon as the bubbles are constrained by the micro-channel size, they start to grow in length rather than in diameter. Slugs of liquid and vapor are created where their length is larger than the inner diameter of the micro-channel. The flow changes from bubbly to slug flow, as shown in figure

(2.1), and is known as the elongated bubbles regime (EB). Here, the velocity of the long vapor bubbles/slugs is not high enough to overcome the surface tension force of the liquid slugs between them. Then, as the heating flux increases, the liquid slugs get smaller and smaller, reaching a point where they disappear. This is because the velocity of the vapor slugs increases compared to the velocity of the liquid slugs, therefore, the liquid slugs are forced and are being dispersed as film. A uniform wavy or smooth liquid film is present around the channel and the vapor occupies most of the area in the middle. This is the third regime known as annular flow. With further increase of the heating flux, dryout is seen. The observation showed that the liquid film was completely evaporated and as a result the temperature readings started to increase rapidly one by one starting from the last one. Partial evaporation was also observed locally. No stratification of the flow was observed, as the gravity forces are negligible at the micro-channel level.

Figure 8.5: Top: snap shot of EB flow video; bottom: same snap shot with background removed.

Visualizing the bubbles in the images taken from the recorded videos appeared difficult. Due to the geometry of the channel, and as a result of the illumination angle, all images did not show bubbles or slugs, but only showed white lines at the bottom side of the channel when a vapor bubble or slug was present. Looking at the videos, it was clearly seen that the vapor bubbles and vapor slugs occupy the full cross section of the channel. Therefore, an image in which the channel was completely filled with liquid was taken as a reference. And images could be processed by taking the difference compared to that reference, while inverting white and black in the images. As an illustration, the top image in figure (8.5) shows a snap shot of a video with a long white line, a shorter one and a very short one. The first two indicate slugs whereas the latter indicates the presence of a bubble. By processing this image as discussed above, the bottom picture in figure (8.5) is obtained, clearly showing a bubble and two elongated bubbles. In the further discussion in this section, the unprocessed images are considered knowing that the white lines at the bottom indicate the presence of vapor bubbles and slugs. Figure (8.6) shows the images taken at 0.02 g/s constant mass flow and 0.20 W of heating power at different time

frames. Here, the flow is laminar (low mass flux). Isolated bubbles can be seen as well as vapor slugs of various lengths as a result of bubbles coalescing. The vapor slugs are bullet shaped and their trail is wavy many times. This is a result of the surface tension forces that are dominant here. Over the different time frames, the vapor slugs have varying lengths but overall the flow is fairly constant. The flow pattern seen here is indicated in figure (2.1) as bubbly and plug flow.

Figure (8.7) shows the images taken at 0.02 g/s constant mass flow with increasing heating power. Initially isolated bubbles are seen but as the heating flux increases, the bubbles coalesce forming long vapor slugs. This flow pattern is indicated in figure (2.1) as bubbly/plug flow. A further increase in heating power makes the bubbles to coalesce more frequent, forming very long vapor slugs with very small liquid slugs in between. This flow pattern seen here is indicated in figure (2.1) as plug/slug flow. The increasing heating power results in a transition from the surface tension dominant region to the shear dominant region and forced convection. The increasing velocity of the vapor forces and disperses the liquid to form a thin film on the wall and the vapor occupies the core area. The stratified flow shown in figure (2.1) has not been observed due to the negligible effect of gravity forces at this micro-channel level. The flow is distributed uniformly over the cross sectional area of the micro-channel.

Figure (8.8) shows the images taken at 0.05 g/s constant mass flow with

Figure 8.6: Images taken at the exit of the evaporator at different times. White lines indicate the presence of vapor. The mass flow is constant at 0.02 g/s and the heating power at 0.20 W. The width of an image is 5-6 mm.

increasing heating power. The starting heating power is higher than in figure (8.7), and therefore no isolated bubbles are observed here. Instead, initially, very long elongated bubbles are seen, that are interrupted by shorter liquid slugs. This flow pattern seen here is indicated in figure (2.1) as plug/slug flow.

In figure (8.8) as the heating power increases the liquid slugs disappear due to the higher velocity of the vapor slugs. Some liquid droplets may be dispersed in the bulk of the vapor. A further increase in heating power creates a uniform vapor section in the middle and a thin liquid film at the channel's wall. This flow pattern seen here is indicated in figure (2.1) as annular flow. The inertia forces are predominant here compared to the surface tension forces, which makes the liquid slugs to break and vapor slug occupy most of the area.

Figure 8.7: Images taken in the glass section after the evaporator at increasing heating power. The mass flow is constant at 0.02 g/s. The heating power is increased in steps 0.20 W to 0.50 W and 0.70 W. The width of the image is 5-6 mm.

Figure 8.8: Images taken at the exit of the evaporator at increasing heating power. The mass flow is constant at 0.05 g/s. The heating power is increased from top to bottom. The width of the image is 5-6 mm.

Figure (8.9) summarizes the flow regimes captured at different mass fluxes and quality and shows the comparison with the theory of refs. [30] and [32]. The lines represent the predictions and the markers the observed flow regimes. Regarding the observed flow regimes, at low vapor quality and low mass fluxes the IB regime is seen. Here, the surface tension forces are dominating the inertia forces of the vapor flow. The EB regime is observed at low mass flux,

and at higher heat fluxes, with vapor quality between 0.2 and 0.4. As the mass flow increases, the EB regime shrinks to a smaller range of vapor quality. This is expected because at increased velocities more frequent interaction occurs between the vapor and liquid slugs and an annular flow is formed sooner. At higher vapor quality, the annular regime prevails, due to the increased vapor velocity, that forces the liquid to be dispersed on the wall forming a film.

Figure 8.9: Flow regimes observed during the measurements compared to the theoretical predictions. The observed data are represented as points, whereas the theoretical predictions of the transition from one regime to the other are represented as lines.

In figure (8.9) some overlap can be seen between the EB and annular regime at higher mass flows. The reason for this is that the elongated bubbles were quite long, even longer than the length that camera was focused on, but in reality further upstream liquid slugs were present between. In a single recorded image, this resembles annular flow. The lines shown in figure (8.9) delimit the transition from one regime to the other. For example, the black line indicates that left of the line the regime is IB and anything right of the black line is EB (but not exceeding the blue line, which indicates the transition to the Annular regime). As it can be seen, there is no just one single line because the theoretical models predicted slightly different results for each set of mass flux and heat flux. Here, two lines are shown for each regime, indicating the range at which the transition lines were predicted. In total, 86% of the observed data points are in the flow regime corresponding to the theory. Some of the points falling outside the predicted region can be a result of poor visualization, where for instance a distinction between EB and annular regime was difficult. The conclusion is that the theoretical models can be used to predict the flow

regime in micro-channels with an acceptable error margin.

8.4.2 Pressure drop

In this section, the two-phase pressure drop across the evaporator is derived from the measurements and compared with predicted values, based on section 2.5.3. The two-phase pressure drop dP_{2ph} , being directly related to the fluid temperature, can be calculated from the temperature difference measured in the adiabatic regions before T_{ref} and after the evaporator T_{ah} . However, since in the experiments T_{ref} does not provide accurately the fluid temperature due to poor thermal contact, this method is not used here. Alternatively, a similar approach is followed as in the single-phase case discussed in section 8.3.1; dP_{2ph} in the evaporator is estimated using the two-phase measured data (i.e. total pressure drop dP_{total} and inlet pressure P_{in}), and calculated pressure drops over parts of the total test section (see figure (8.10)). Particularly, the

Figure 8.10: Drawing of the test section indicating the various pressure positions and pressure drops (only P_{in} and dP_{total} are measured, others are calculated).

pressure drop in the evaporator dP_{2ph} is given in eq. (8.9) and is the pressure difference between estimated values of P_{inevap} and $P_{outevap}$.

$$dP_{2ph} = P_{inevap} - P_{outevap} \quad (8.9)$$

The pressure P_{inevap} at the inlet of the evaporator is estimated as :

$$P_{inevap} = P_{in} - (dP_{preheater} + dP_{contraction} + dP_{startmicrochannel}) \quad (8.10)$$

Here, $dP_{preheater}$ is the pressure drop across the preheater which is evaluated using eq. (2.11) and $dP_{contraction}$ is the pressure drop due to the contraction when moving from D_1 to D_2 which is evaluated using eq. (2.16) with $D_1=0.88$ mm and $D_2=0.25$ mm. Eq. (2.11) is also used to find $dP_{startmicrochannel}$ which is the pressure drop in the first 25 mm of the micro-channel before the heater assuming single-phase flow.

The pressure at the exit of the evaporator $P_{outevap}$ can be evaluated from the pressure P_{ah} and $P_{outmicro}$ which we then first need to estimate. The pressure P_{ah} is derived from the measured temperature T_{ah} during two-phase flow and adiabatic flow. Eq. 3.13 of ref. [35] gives the vapor pressure curve of CO₂ and relates P_{ah} to T_{ah} assuming saturated conditions. The pressure $P_{outmicro}$ is found using eq. (8.11)

$$P_{outmicro} = P_{in} - dP_{total} + dP_{return} + dP_{bend} \quad (8.11)$$

where the P_{in} and dP_{total} are measured and dP_{bend} evaluated using eq. (2.33) whereas dP_{return} is the pressure drop in the return line evaluated using the relevant equation for the mini-channels depending on the flow regime. Next, $P_{outevap}$ can be evaluated from P_{ah} and $P_{outmicro}$ assuming a linear pressure profile in the tube under adiabatic conditions (which is supported by model evaluations). Then the pressure drop between P_{ah} and $P_{outmicro}$ can be scaled to the total length between the exit of the evaporator and the end of the micro-channel, and use this to find $P_{outevap}$.

$$P_{outevap} = P_{outmicro} + \left(P_{ah} - P_{outmicro} \right) \left(\frac{L_2}{L_1} \right) \quad (8.12)$$

The uncertainties in the pressure estimates are calculated using the square root of sum of squares of each of the components. The uncertainty in the calculation of P_{ah} from T_{ah} is 1% , 1.5% for P_{inevap} and $P_{outmicro}$, and 2% for $P_{outevap}$. The uncertainty in the pressure drop dP_{2ph} is ± 0.5 bar.

The predicted pressure drop across the evaporator dP_{pred} is found using equations from section 2.5.3. The predicted pressure drop across the evaporator is now compared to the results obtained from measurements and pressure drops evaluated in the adiabatic sections. The strategy followed is summarized below:

1. P_{inevap} is evaluated using eq. (8.10).
2. $P_{outevap}$ is evaluated using eqs. (8.11) and (8.12).

3. The ‘measured’ pressure drop across the evaporator then results from eq. (8.9).
4. The pressure dP_{2ph} is also evaluated using equations in section 2.5.3.
5. Compare resulting pressure drops: measured (step 3) with predicted (step 4).

The uncertainty in dP_{pred} is estimated by using P_{inevap} (as found in step 2) within the error range. The uncertainty in dP_{pred} due to the inlet pressure is less than 1%.

Figure 8.11: Two-phase pressure drop in the evaporator dP_{2ph} and dP_{pred} versus heat flux applied on the outer wall.

Figure (8.11) shows the pressure drop in the evaporator as predicted from the theory and as derived from the measurements. The pressure drop in the evaporator increases with the heat flux applied and therefore, with the vapor quality. This is expected since at higher vapor quality the vapor velocity increases and, therefore, the friction losses increase as discussed in section 2.5.3. Comparison of all two-phase data shows that the dP_{2ph} values as derived from measurements fall within the error range of the prediction dP_{pred} , taken into account the uncertainty in the measurements. The model thus can be used

within error margin to predict the two-phase pressure drop in micro-channels.

8.4.3 Heat transfer coefficient

In this section, the two-phase heat transfer coefficient is derived from the measurements and compared with the predicted values. In order to estimate the two-phase heat transfer coefficient h_{2ph} using eq. (8.7), the inner wall temperature along the evaporator has to be known. The inner wall temperature can be derived from the outer wall temperature (which is measured) by using eq. (8.6). In the evaluation of the inner wall temperature distribution and the resulting heat transfer coefficient, the measured outer wall temperatures are corrected for the offset along the length of the tube, as discussed in section 8.1. In the two-phase region the temperature profile of the fluid along the tube can be estimated based on the pressure profile along the tube. The pressure at the exit of the heater, as derived from the measured temperature, falls within the error range of the predicted pressure profile as discussed in section 8.4.2. For this reason, the temperature profile along the channel's length, predicted by the pressure profile, can be used in eq. (8.7) as the fluid temperature in the estimation of h_{2ph} .

Figure 8.12: Heat transfer coefficient derived from measurements at an average mass flux 1550 kg/m²s for different heat fluxes over the various positions along the tube length.

Figure (8.12) shows the evaluated heat transfer coefficients h_{2ph} at the positions T_2 (0.025 m) to T_5 (0.06 m) along the heater where the wall temperature is measured. T_1 and T_6 are not considered here because these have too high thermal contact with the outer parts. Here, the mass flux is on average $1550 \text{ kg/m}^2\text{s}$ and the heat flux is varied. Each set of data is at a constant heat flux and mass flux. The two-phase heat transfer coefficient h_{2ph} appears to increase significantly as the fluid moves further into the evaporator, with a peak at 0.055 m from the start of the evaporator. Then, the h_{2ph} drops. An explanation can be that locally the liquid film on the wall can be such that the heat transfer improves. For example, if the liquid film is thin enough, it can be quickly moved with the vapor flow, but it should not be too thin such that it can be easily fully evaporated. Indeed, for all the heat flux cases shown here, the flow at the first point (0.025 m) is in the IB regime, and then switches to the EB regime due to the high vapor velocity. The elongated vapor slug occupies the whole length, and the liquid film decreases along the evaporator length. The h_{2ph} in the EB regime is derived as the sum of the heat transfer due to the liquid slug, the liquid film and the local wall dryout region, as it was shown in eq. (2.64). The h_{2ph} drops due to the partial dryout on the wall or the liquid slug, before the next vapor slug arrives.

Figure (8.13) shows h_{2ph} for the case of a lower mass flux of $490 \text{ kg/m}^2\text{s}$. As the heat flux increases the value of h_{2ph} appears to increase significantly at T_2 (0.025 m). The IB regime is predicted at the low heat flux case (8997 W/m^2) whereas the regime at the higher heat flux cases ($>8997 \text{ W/m}^2$) is EB due the low mass flux. As the heat flux increases at low mass flow rate, the bubbles quickly collide forming elongated bubbles. At the EB regime the heat transfer is better due to the increased vapor velocity compared to the IB regime, and heat is removed faster from the wall. In the EB regime (higher heat fluxes), the value of h_{2ph} seems fairly constant with the exception of some peaks at T_3 (0.045 m) for the 38285 W/m^2 heat flux case and at T_4 (0.055 m) for the 81999 W/m^2 case. The peaks in the value of h_{2ph} can be due to the very thin liquid film locally on the vapor slug (further discussion follows). After the peaks, the heat transfer coefficient values drop and return to the values seen at the start of the heater. For the low heat flux case (8997 W/m^2), h_{2ph} increases along the tube length. The value of h_{2ph} at T_4 (0.055 m) and at T_5 (0.06 m) is similar for all the cases. Generally, close to the transition of EB to annular flow, peaks and instabilities are seen in the heat transfer coefficient, corresponding to the rapid switching of one regime to the other (see also section 9.4.6). The heat transfer in the EB regime is due to nucleate boiling, whereas the heat transfer in the annular regime is due to forced convection as described in section 2.3.2. A change in the heat transfer mechanics due to the transition from one regime to the other could lead locally to an increased heat transfer performance (peak in h_{2ph} at certain locations). Locally, the liquid film on the wall can be such that the heat transfer improves. For example, if the liquid film is thin enough, it can quickly be moved with the vapor flow, but it should not be too thin such that it can be easily fully evaporated, as discussed earlier.

Figure 8.13: Heat transfer coefficient derived from measurements at an average mass flux $490 \text{ kg/m}^2\text{s}$ for increasing heat flux over the various positions along the tube length.

In the case of the EB, as a vapor slug passes the channel, the liquid film evaporates reaching an optimum thickness, where the heat transfer improves locally. At the event of a bubble collision (to make an even larger elongated bubble or transition to annular regime), the liquid slug between will be dispersed on the wall, making the liquid film slightly thicker. The heat transfer will deteriorate locally, compared to the previous case where the liquid film was optimum. In the videos recorded it was difficult to distinguish whether the liquid film was indeed decreasing with increasing heat flux. Depending on the length of the vapor slugs, the peaks can be seen at different locations. In the case of the annular regime, as long as there is a balance between the heating input and the cooling effect, the heat transfer shall stay constant. A further evaporation though leads to a drop in heat transfer locally, as the liquid is not removing sufficiently the heat added, due to the smaller liquid film thickness. From this point on, dryout is expected.

Figure (8.14) shows the heat transfer coefficient h_{2ph} as a function of heat flux for the IB, EB and annular flow regime. The flow regime was observed at the glass section after the micro-channel and 86% of the observed data points were in the flow regime corresponding to the theory, as it is indicated in figure (8.9). Therefore, the predicted regime is used in the next figures. Moreover, the sharp peaks in the heat transfer coefficient predicted, as seen

in figures (8.12) and (8.13), are not included in the next figures, as these values are not considered representative of the average heat transfer performance.

Figure 8.14: Two-phase heat transfer coefficient as a function of heat flux for the different regimes observed.

Figure (8.14) shows that there is an increase in the two-phase heat transfer coefficient h_{2ph} with increasing heat flux. There is a large positive slope in the heat transfer up to 80000 W/m^2 and at higher heat fluxes the heat transfer further increases but with a lower slope. All three regimes are seen at high heat fluxes, but it shall be noted that the IB regime was not observed at the highest heat fluxes. The reason for this is that at the highest heat fluxes the isolated bubbles could not be sustained, and they were growing in length, forming elongated bubbles. The annular regime was not observed at the low heat fluxes, because the vapor velocity and vapor quality is low and the liquid occupies most of the area. Particularly, the IB regime was observed at heat fluxes from 10000 up to 130000 W/m^2 , the EB regime is seen from 20000 up to 160000 W/m^2 heat flux, whereas the annular regime covers from 60000 up to 160000 W/m^2 heat flux.

Overall, the annular regime showed higher heat transfer coefficients than the IB regime. At higher heat fluxes, the vapor quality and the vapor velocity increase, and so does the vapor Reynolds number. Indeed, the vapor flow is

Figure 8.15: Two-phase heat transfer coefficient as a function of quality for the different regimes.

turbulent at increasing heating power. As a result, the heat transfer mechanism is forced convection, as discussed in section 2.3.2, which shows better heat transfer coefficients compared to the nucleate boiling. In the forced convective region, due to the reducing thickness of the liquid film, the heat transfer coefficient increases for increasing vapor quality. The heat transfer was also high in the EB regime. Above 80000 W/m^2 the EB showed similar performance as the annular regime. At high heat fluxes, the elongated bubbles were very long, and sometimes were occupying the whole visualization length, thus resembling the annular regime.

Figure (8.15) shows the heat transfer coefficient h_{2ph} as a function of vapor quality for the IB, EB and annular flow regimes. The vapor quality is determined at the exit based on the total heating power and the mass flow rate assuming certain inlet conditions. It can be seen that there is an increase in the h_{2ph} with increasing vapor quality, up to a certain vapor quality. At around 0.25 vapor quality, the heat transfer slightly drops and then stays constant, indicating that the optimum vapor quality has been overpassed. A further decrease in the liquid film on the wall due to increased vapor quality (and heating power) does not improve the heat transfer but instead a decrease is expected as soon as the liquid starts to evaporate partially on the wall. Dry-out data are not included here. Finally, no clear trend on the heat transfer coefficient is seen with respect to the mass flux.

Comparison of h_{2ph} with the predicted heat transfer coefficient

The two-phase heat transfer coefficient h_{2ph} as derived from measurements is compared with the predicted heat transfer coefficient h_{pred} based on the empirical models described in chapter 2 in section 2.5.4. Figures (8.16) - (8.18) show the comparison of the measured h_{2ph} with the predicted value h_{pred} for each regime separately including error range as a function of the quality. The uncertainty in h_{2ph} is calculated using the square root of sum of squares with the temperature difference between the fluid and the inner wall being the main contributor. The estimated errors in h_{2ph} are quite high, up to 40%. The uncertainty in h_{pred} is estimated by using P_{inevap} (as found in step 1, see section 8.4.2) within the error range. The uncertainty in h_{pred} due to the inlet pressure is less than 10%.

In figure (8.16), h_{2ph} and h_{pred} are plotted for varying vapor quality, in the

Figure 8.16: Comparison of h_{2ph} and h_{pred} for the IB regime.

IB regime. Here, most of the data in the IB regime fall within the error range. The IB regime observed covers up to 0.07 vapor quality. Overall, the h_{2ph} and h_{pred} values in the IB regime do not vary with increasing vapor quality. This is expected, as in eq.(2.63) the heat transfer is not related to the vapor quality. The heat transfer in the IB regime is mainly affected by the fluid flow and the velocity of the liquid. In figure (8.17), h_{2ph} and h_{pred} are plotted for varying vapor quality, in the EB regime. Here, most of the measured h_{2ph} data in the EB regime fall on the upper limit of the h_{pred} error range and in general the

Figure 8.17: Comparison of h_{2ph} and h_{pred} for the EB regime.

theory under-predicts the heat transfer coefficient. Although the error ranges are relatively large, the measured as well as the predicted values of h_{2ph} increases up to 0.15 and drop a bit and stay constant at higher vapor quality. An explanation of this behavior could be that initially the EB regime has vapor and liquid slugs of a certain length, but as the vapor quality increases, the liquid slugs get shorter and the vapor slugs longer due to the increased vapor velocity. Closer to the transition to the annular regime, very long vapor slugs can be present, and the heat transfer is fairly constant, with some peaks due to a thinner film. Figure (8.18) shows h_{2ph} and h_{pred} for varying vapor quality, in the annular regime. The observed flow regime covers a vapor quality range between 0.15 and 0.55. The value of h_{pred} increases initially up to 0.3 vapor quality and then drops to lower values. Between 0.35 and 0.55 vapor quality, h_{pred} stays on average constant. The measured h_{2ph} on average seems to stay constant over the whole vapor range within measuring error. However, the same trend as in the predicted values could be present but obscured by the relatively large measuring errors. Nevertheless, it can be stated that the theory under- predicts the heat transfer coefficient by roughly a factor of 5. The proposed model for the annular flow described in ref. [10] estimates parameters like void fraction and entrained liquid fraction on basis of experimental data of flows in much larger higher diameters than the one we have examined here (1 mm up to 95 mm as compared to 0.25 mm). The authors in ref. [10] suggest that the proposed model should include measurements of lower diameters (<1 mm) to support their findings. They also note that a variation of the

Figure 8.18: Comparison of h_{2ph} and h_{pred} for the annular regime.

void and entrained liquid fraction of 20% had little effect on the heat transfer coefficient. Running the model and varying a single parameter each time (void fraction and entrained liquid fraction), showed that indeed the void and entrained liquid fraction do not significantly affect the predicted heat transfer coefficient. On the other hand, the Nusselt number as estimated in ref. [10], does affect the h_{pred} significantly. A higher dependency on the average liquid film thickness introduces higher predicted heat transfer values.

8.5 Empirical Parameters

The theoretical models, examined here, include empirical parameters for the calculation of h_{pred} . These empirical parameters usually depend at least to some extent on the experimental set-up. Since the heat transfer coefficient, as derived from the measurements, is higher than the one predicted in the theory, this section examines the empirical parameters in more detail. Since the discrepancy specifically shows up in EB and annular flow, only these are considered. Goal is to investigate whether and which empirical parameters play an important role in the predicted heat transfer coefficient.

The EB regime model includes 3 adjustable parameters that are experimentally determined. In order to identify these parameters, ref. [9] uses a least-squared error method on a large set of data of different fluids and geometries. These

parameters were optimized for one fluid (R-113), whereas for the other fluids a correction is applied. The first parameter is the minimum film thickness (δ_{min}) at dryout, which according to ref. [8] is linked to the roughness of the wall and the properties of the fluid. The second parameter (C_{δ_o}) is a correction factor for the initial film thickness (δ_o) which is optimized for R-113 taking into account the fluid properties and geometry. The last parameter (f) deals with the liquid-vapor pair frequency, and is a complicated parameter with a dependency on the channel diameter, surface roughness, bubble dynamics, subcooling etc [8]. The different parameters for each study that was included in the derivation of the EB model are summarized in table 2 in ref. [9]. From this table, it can be concluded that despite the variation in the data, for none of these parameters a clear relation is found with the channel diameter. For example, in two-phase studies on R-134a with diameters of 2.00 mm and 0.71 mm the values of δ_{min} were similar whereas in another study with a diameter of 2.01 mm a difference compared to these by a factor of 1.8 was found [9].

To determine the parameter, that in this study most critically influences the heat transfer coefficient, all three parameters are varied by a factor of two. The critical parameter is considered the one that has the largest effect on h_{pred} . It was found that only the C_{δ_o} and δ_{min} parameters affected h_{pred} . Their effect is illustrated in figure (8.19). Here, the parameters were increased by a factor of 2 and also reduced by that same factor. As before, the measurement points T1 and T6 are not considered. Furthermore, the measurement point T1 is not included since the flow at that point is in IB regime whereas the model considers EB. Modification of the parameter δ_{min} did not have an effect on the predicted heat transfer and in figure (8.19) the h_{pred} is overlapping with the h_{pred} δ_{min} modified parameters.

On the other hand, it can be seen that the parameter C_{δ_o} used for the correction of the initial film thickness affects more the h_{pred} . A decrease by a factor of 2 of the C_{δ_o} did show a significant effect, bringing it closer to the values derived from measurements. Higher factors did not further increase the effect. A higher value of the C_{δ_o} parameter affects less the heat transfer coefficient.

The proposed model for the An flow described in ref. [10] estimates parameters like void fraction and entrained liquid fraction on basis of experimental data of flows in much larger higher diameters than the one we have examined here (1 mm up to 95 mm as compared to 0.25 mm). The authors in ref. [10] suggest that the proposed model should include measurements of lower diameters (<1 mm) to support their findings. They also note that a variation of the void and entrained liquid fraction of 20% had little effect on the heat transfer coefficient. Running the model and varying a single parameter each time (void fraction and entrained liquid fraction), showed that indeed the void and entrained liquid fraction do not significantly affect the predicted heat transfer coefficient. On the other hand, the Nusselt number as estimated in ref. [10], does affect the h_{pred} significantly. A higher dependency on the average liquid film thickness introduces higher predicted heat transfer values.

Figure 8.19: Heat transfer coefficient predicted with varying parameters in the EB regime model at mass flux $485 \text{ kg/m}^2 \text{ s}$ and heat flux 38000 W/m^2 .

8.6 Conclusions

Tests on a single straight micro-channel were conducted in order to validate the empirical models for the two-phase flow presented in chapter 2. The parameters that could be derived from measurements are the pressure drop and the heat transfer coefficient, whereas the flow regime was directly visualized with a high speed camera right after the test section.

Single-phase validation of the set-up was performed prior to the two-phase tests. Pressure drops and heat transfer coefficients based on measurements match with theoretical predictions within measuring errors.

The recorded flow regimes in two-phase flow are in good agreement with the Ong model that correctly predicts the regime in about 86% of the total data. The pressure drop is compared with the relevant model depending on the flow regime. The estimation of the pressure drop is also satisfying using the models discussed in chapter 2. The heat transfer coefficient during two-phase flow is calculated from the measured wall temperatures. A peak in heat transfer coefficient is seen at different locations, and further downstream a drop indicating possibly the presence of a local dryout. The effect of the heating power on the heat transfer coefficient was clearly seen in all three regimes. The heat transfer coefficient increases with increasing heat flux, as we move from the nucleate boiling to the forced convection regime.

The heat transfer coefficient is compared with model predictions depending on the flow regime. As a general remark, the heat transfer coefficient is under-predicted by the models. In the IB regime the difference is smallest and predictions match with the measuring results within their error ranges. In the EB regime the difference is roughly a factor of 2 although the error ranges are ‘touching’. The difference is highest in the annular regime, about a factor of 5. The models are examined further on different empirical parameters that need to be adjusted depending on the specific measuring set-up. To define the most critical parameter, these parameters are modified by the same factor. The C_{δ_o} parameter (i.e. a correction factor for the initial film thickness) for the EB regime proves to be the one that has the largest effect on the resulting heat transfer coefficient. In the analysis, it is shown that the heat-transfer prediction of the model can be improved by fine tuning the relevant empirical parameters. A similar conclusion can be made for the annular regime, where a higher dependency of the Nusselt number on the average liquid film thickness affects the heat transfer. The average liquid film thickness depends on the geometry and the fluid thermodynamic parameters.

Generally, the models examined here in terms of heat transfer coefficient are useful for qualitative trend studies. However, if quantitative heat transfer studies are needed, then the experimental parameters need to be well known and may need to be determined in a separate set of test experiments. This, in particular, holds for the EB model that shows a significant sensitivity to these empirical parameters. For the annular regime, flow data on lower diameters could be included in the database used in ref. [10] to extend the annular regime model for the prediction of the heat transfer.

Chapter 9

Studies on silicon-glass multi-channels

Chapters 7 and 8 presented the work on CO₂ evaporative cooling in a single circular horizontal micro-channel. In HEP experiments though multiple micro-channels are preferred for cooling effectively high power density particle detectors. A common practice is to use multiple micro-channels etched in a silicon substrate. In this chapter, the study of a silicon-glass multi-channel device is presented. The device could not be tested under the same conditions as the single micro-channel described in chapters 7 and 8, and therefore, direct comparison is impossible. The tests show that the device performs not as it was designed for, because one capillary channel was clogged and the flow was not distributed uniformly in all the channels. Nevertheless, a few conclusions can be drawn on the flow regimes seen and the heat transfer parameters based on averaged flow rates.

9.1 Silicon-glass multi-channel device

The silicon-glass device is built at CERN in collaboration with EPFL to study flow and thermal properties of CO₂ in micro-channels. The device depicted in figure (9.1) was intended for testing the cooling performance of prototype silicon detectors. It consists of a silicon substrate in which the channels are etched (via plasma anisotropic etching). The silicon substrate is sealed with a 2 mm thick Borofloat glass layer by anodic bonding to enable flow visualization. A silicon heater is realized at the back side by uniformly depositing a layer of metal on silicon. Electrical contacts are soldered to power the heater, all along the width on the opposite sides, providing a uniform power distribution. Furthermore, the heat is transferred to the underlying silicon substrate via the 525 micron thick silicon that also acts as heat spreader.

The CO₂ entering from the inlet hole of the device, passes then 3 different sections, before exiting the outlet hole. The first section consists of capillaries, the second part is the section with rectangular cross-section micro-channels

and the last section consists of a pillar structure. Figure (9.1) depicts the front side of the silicon-glass device where the 3 different sections can be seen. The capillaries and the micro-channels have the same depth ($200\text{ }\mu\text{m}$) but the micro-channels are a factor ten wider than the capillaries, ($200\text{ }\mu\text{m}$ and $20\text{ }\mu\text{m}$, respectively).

Figure 9.1: Top of the silicon-glass device. The device has micro-channels etched in the silicon substrate [65].

There are in total 16 micro-channels. The CO_2 enters the inlet tube sub-cooled, and from there enters the capillaries. The capillary channels should distribute the flow uniformly in all the micro-channels since the pressure drop is determined by the capillary channels. The capillaries all have the same cross sectional area and as much as possible the same length. The capillaries are designed to induce a large pressure drop in order to avoid vapor back-flow from the channels producing instabilities and ensure boiling immediately at the channel entrance. Directly after the capillaries, the micro-channels are extended. The micro-channels end up in a pillar structure of around 40 circular pillars in a staggered pattern. The pillar structure at the exit was selected to maximize the bonded surface and mechanical strength of the device.

Figure (9.2) depicts the back side of the silicon-glass device where the heater is positioned. The heater is a silicon substrate that covers only the area where the micro-channels are. A thin film metal layer (20 nm of titanium and 200 nm of gold) is deposited on the silicon to realize the heating. Electrical wires are soldered at the back of the heater that can be connected to the power supply. Finally, metallic connectors are soldered at the inlet and outlet holes of the device to interface between the silicon device and the cooling inlet/outlet tubes [66].

Figure 9.2: Back view of the silicon-glass device. A silicon heater is placed at the back of the channels and metallic connectors are welded.

Table (9.1) presents the geometrical information of the silicon-glass device. The micro-channels occupy an area of 198 mm^2 in total of the silicon.

Table 9.1: The material and geometrical information of the different sections in the device.

Layer	Material	Thickness (mm)	
Top	Borofloat glass	2.00	
Mid 1	Si	0.380	
Mid 2	Metal layer	0.220	
Bottom	Si heater	0.525	
Section	Type	Width, Depth (mm)	Length (mm)
Inlet	Capillaries	0.020, 0.20	5-7
Evaporator	Micro-channels	0.20, 0.20	62
Exit	Pillar structure	0.35-0.40 pillar diameter	-

Figure (9.3) shows a cross section of the silicon-glass device. The silicon heater is glued on the silicon substrate.

Figure 9.3: Drawing of the silicon-glass device cross section.

9.2 Set-up

The modified blow system presented in section 7.2 is used for testing the silicon-glass device. Figure (9.4) presents the set-up for the silicon-glass tests. The objectives of the testing were to visualize the flow distribution and flow patterns (regimes) in the micro-channels and to study the overall thermal performance of the silicon device. The fact that the visualization is possible directly on the silicon-glass device, provides an advantage over the single micro-channel study in chapter 8, as the effect of the heating on the flow is immediately visible along the micro-channels. Another difference is that the HX (figure (7.3)), used to sub-cool the incoming coolant in chapter 7, is not used, as the expected parasitic heat leak is negligible compared to the heating input. For this reason, no vacuum insulation is used. Instead, two transparent compartments are built around the device to minimize the freezing/condensation process on the top side of the silicon-glass device. N_2 gas is purged in the inner compartment to prevent moisture being deposited on the cold test sample.

Figure 9.4: Schematic drawing of adapted blow system for the silicon-glass tests.

9.3 Test section

Figure (9.5) shows the front, back and side views of the silicon-glass sample. It is placed vertically with respect to the ground with the inlet at the top and outlet at the bottom. Three PT100 temperature sensors (PT_1 , PT_2 , PT_3) are placed at the back side of the silicon-glass device away from the heater. One PT100 temperature sensor is placed on the heater (T_{heater}). Two PT100 temperature sensors are placed at the same distance before the inlet ($PT_{wall-in}$) and after the outlet ($PT_{wall-out}$) on the tube wall. Two thermocouples are placed in direct contact with the fluid and measure the inlet and outlet temperature (T_{CO2in} , T_{CO2out}).

Figure 9.5: Front, back and side views of the silicon-glass device mounted in the set-up.

Figure 9.6: Drawing of the side view of the test section with the temperature and pressure drop measurement points (dimensions are not to scale).

Figure (9.6) shows the measurement points for the fluid and wall temperatures as well as for the pressure drop. The components used in the silicon-glass set-up

are the same as in the single micro-channel set-up described in chapter 7. The main difference is that PT100 sensors are used to measure the temperature on the silicon-glass device. The PT100 readings have an uncertainty of $\pm 0.40^\circ\text{C}$. The uncertainty of the other components can be found in table (7.4).

9.4 Experimental results

9.4.1 Measurements

The two-phase measurements of the silicon-glass heat exchanger examine a two-dimensional matrix in which mass flow and power are varied at a constant saturation pressure. Goal of these measurements is to examine the thermal performance of the device during two-phase flow at different heating and flow conditions. The procedure can be summarized as follows: The device is cooled with CO_2 , and then heated with the heater located at the back side of the device. The incoming CO_2 is sub-cooled only by 1-2 $^\circ\text{C}$, which is in the error range of the measuring temperature sensor. Due to the parasitic heat leaks towards the inlet tube, boiling starts already at the inlet tube. Then the pressure along the capillaries drops significantly before entering the micro-channels. Temperature, pressure and flow are recorded and the flow pattern is monitored through the glass. The following section summarizes the steps followed during the test and presents the measurements taken during this time. The measurements are taken at a total mass flow rate up to 0.05 g/s and power up to 29 W. The mass flux for each micro-channel is calculated from eq. (9.1) for 15 channels since one of the 16 channels appeared to be clogged as is discussed in section 9.4.3.

$$G_{si} = \frac{\dot{m}}{NWH} \quad (9.1)$$

where $\dot{m}$ is the total mass flow rate through the test sample, N is the number of micro-channels, W is the width of the micro-channel, and H is the height. The pressure at the outlet is fixed at 20 bar by adjusting the pressure regulator and initially the mass flow rate is set at 0.04 g/s. Figure (9.7) shows the temperatures as a function of time following the changes in mass flow rate and in heater power. Figure (9.8) shows the corresponding total mass flow at the inlet pipe and the total pressure drop in the device, that is indicated in figure (9.6) as dP_{meas} . The horizontal axis is the time in seconds and is set to 0 at the start of the recording. Measurements were taken every 2 seconds. The measurements shown in figures (9.7) and (9.8) after the first 40 minutes. In figure (9.7), TCO2in and TCO2out are fluid temperatures at the inlet and outlet, whereas the PTwall-in and PTwall-out temperatures are measured at the inlet and outlet wall tubes connected on the device. The temperatures (PT1, PT2, PT3) are measurements on the silicon, whereas the T_{heater} is

measured on the heater. At 4500 s the total mass flow is adjusted to 0.03 g/s and at 4900 s the heater is switched on and the power is increased in steps up to 5 W. The total pressure drop at this mass flow rate is around 5.2 bar. The temperatures on the silicon (PT1, PT2, PT3) and on the heater (T_{heater}) increased also in steps due to the increasing heating power. Particularly, the T_{heater} responds immediately to the heating power which is expected since it is placed on the heater, whereas the temperatures on the silicon (PT1, PT2, PT3) react after the power is increased to 1 W. It shall be noted that the fluid temperature TCO2out also increases, indicating that the fluid is almost fully evaporated. A sharp peak in temperatures is seen at 5 W where complete dryout occurred. The power is then switched off at 6000 s and the total mass flow is adjusted to 0.02 g/s. The pressure drop reduces at this mass flow to about 4.2 bar. Due to the very low mass flow level, parasitic heat affects the temperature readings for temperatures on the silicon (PT1, PT2, PT3) and the TCO2out. It seems that the parasitic heating introduces some thermal instability and the flow at the exit is almost fully evaporated (TCO2out increases due to parasitic leaks). At this time, evaporation was observed already half-way through the micro-channels without switching on the heater and is mainly attributed to the parasitic heating. At 10k s, the total mass flow is adjusted to 0.055 g/s. The pressure drop reaches 8 bar. At 11k s, the power is increased in steps. At 12.5k s, the power reaches up to 15 W, where dryout was observed. The temperatures (PT1, PT2, PT3, T_{heater}) react respectively and a sharp peak in temperatures is seen, due to the deteriorating heat transfer, as a result of the dryout. Immediately after, the power is decreased in steps. The temperatures (PT1, PT2, PT3, T_{heater}) react respectively with the PT1 and PT3 temperatures showing some small hysteresis.

Looking at the measurements, the following can be concluded. During this test, no temperature jumps were seen for the temperatures (PT1, PT2, PT3) measured on the silicon device, as it was seen in figure (7.8) in chapter 7. This is expected since two-phase flow was present already at the inlet tube (therefore, no boiling delay). Overall, it can be seen that switching from a high heating power to zero gives only a small change in pressure drop, whereas the pressure drop follows the mass flow rate. This indicates that most of the pressure drop occurs in the capillaries due to friction and less due to the heating of the micro-channel during evaporation, as expected in two-phase flow. Further discussion follows in section 9.4.5. Moreover, the TCO2in temperature readings follow the pressure at the inlet (P_{in}), as shown in figure (9.6). Due to the fact that the pressure at the exit is fixed by the pressure controller, any change in the pressure drop (due to a change in the mass flow, i.e.) will cause the pressure at the inlet tube to change accordingly. Due to the small sub-cooling margin of 1-2 °C at the inlet, and considering that the measuring error of the temperature sensor is 1.7 °C, and taking into account the parasitic heat leaks towards the inlet section, the flow at the inlet tube is two-phase, and therefore will follow the pressure.

Figure 9.7: Recording of the wall temperatures and the heater power levels during two-phase flow.

Figure 9.8: Recording of the total mass flow rate settings at the inlet tube and total pressure drop evolution during the two-phase flow experiments presented in figure (9.7).

9.4.2 Mass flow fluctuations

Figure (9.8) shows that the mass-flow rate fluctuates at specific settings of mass-flow rate and heater power. Fluctuations appear to mainly occur in the EB regime and are larger as the bubbles get longer. In order to illustrate this effect, figure (9.9) shows the measurement data for the power, pressure drop, mass flow rate, and TCO_{2out} in the time interval 13300-14400 sec. At a mass flow rate of around 0.055 g/s, the mass flow rate fluctuates at about 30 % when the heating power is 5 W. At higher heating power, the mass flow rate is fairly stable, whereas at a heating power less than 5 W it fluctuates at about 10%. The differences are due to the flow regime, and the transitions do reproduce. At 5 W, where the highest fluctuations in mass flow occur, the vapor flow is turbulent and the regime is EB (on average for the device) with very long bubbles, whereas at lower heating power (<5 W), the regime is still EB but the bubbles are shorter. The mass flow is fairly stable at a heating power of 7.7 W where the regime is annular. Flow pattern transition instabilities seen on the micro-channels may cause fluctuation in the total mass flow rate. More discussion on the flow regimes follows in section 9.4.4. Moreover, as can be seen in figure (9.9), the frequency and amplitude of the temperature fluctuation of the fluid measured at the return tube (TCO_{2out}) change with the different power levels at 0.055 g/s. At a heating power more than 5 W, the temperature measured TCO_{2out} is fairly stable, whereas at a heating power of 5 W the TCO_{2out} fluctuates at about $0.04^{\circ}C$. At even lower heating power (<5 W), the frequency increases and the fluctuations increase to about $0.06^{\circ}C$. During bubbly flow, the wall conditions constantly change depending on the length of the bubble passing by. The temperature TCO_{2out} is affected whether a vapor bubble or liquid slug passes by from the outlet tube due to the locally different heat transfer coefficient on the wall. During annular flow, on the other hand, a fairly stable behavior is seen, due to the liquid film that uniformly covers the whole length of the channel.

As shown in figures (9.7) and (9.8), at mass flow rates less than 0.04 g/s, the mass flow fluctuation is less than 5%, at all heating steps. It shall be noted also, that the temperatures on the silicon (PT1-PT3) are not affected by the mass flow fluctuations due to the high thermal conductivity of silicon, which smooths out the temperature profile. On the other hand, the temperature of the fluid measured at the return tube (TCO_{2out}) seems to fluctuate at the low mass flow rates (0.02 g/s) when no heating power is added from the heater (6500-9800 s). At this low mass flow rate, even without any heating added from the heater, heat leaks can affect the thermal stability. A mass flow fluctuation is not observed at this flow rate, neither pressure drop fluctuations.

Figure 9.9: Fluctuations during the two-phase tests.

9.4.3 Flow distribution

The silicon-glass device includes capillaries prior to the micro-channels to induce a uniform flow distribution along all the micro-channels and two-phase

flow at the entrance of the micro-channels. Since the silicon substrate is sealed with a Borofloat glass layer flow visualization was possible. Examining one by one the micro-channels during the tests, it was seen that one of the micro-channels had no flow at all times. Moreover, images of the silicon-glass taken initially without flow showed particles deposited on the micro-channels most likely after flushing the device. These deposits are seen as white marks in the micro-channels and on the back of each pillar in the typical re-circulation zone behind a pillar, as it is shown in figure (9.10). In figure (9.10) the image on the left shows the micro-channels close to the capillaries and the image on the right shows the pillar section without flow. In the right image of figure (9.10), the exit of the micro-channels, and therefore the flow direction, is from the right bottom corner towards the left top corner.

Figure 9.10: Particles deposition in the micro-channels (on the left) and on the pillar section (on the right) right at the back of each pillar (size of pillar is $40\mu\text{m}$).

These deposits can be a result of a contaminated test stand since the device was fully verified with circulating CO_2 at CERN and then it was stored for some time before it was flushed again. Another reason can be that before the anodic bonding of the device and during the Piranha cleaning, as explained in ref. [66], dust particles could have been introduced. Further flushing of the device was not successful and it is very likely that these deposits were also present in the capillaries and also due to their small width ($20\mu\text{m}$) were very sensitive to clogging. Unfortunately, it was not possible to capture the capillaries in images in such detail as in figure (9.10).

Figure 9.11: Mass flux per micro-channel as predicted from the videos captured during the test with total mass flow of 0.04 g/s and 3 W heating power.

As a next step, recorded videos were examined in detail to predict the flow distribution among the micro-channels. Not all data were examined, instead some samples were used to get a qualitative impression of the flow distribution. The velocity of a bubble flowing in each of the micro-channels was evaluated by noting the time it takes in the video for the bubble to move along the micro-channel over a certain distance. Figure (9.11) shows the mass flux evaluated in that way per micro-channel. The horizontal axis refers to each micro-channel, starting with micro-channel 1 as the leftmost one when the flow direction is from top (from inlet) to bottom (to outlet) as shown in figure (9.12). Micro-channel 13 is without any flow, whereas micro-channel 14 has a relatively low flow. Apparently, the capillary before the micro-channel 13 is fully clogged, and therefore no flow is possible. Similarly, the low flow in micro-channel 14 is resulting from a partly clogged capillary. Obviously, this method of measuring the mass flux is not very accurate but it clearly shows the channels that are clogged or partly clogged. The other micro-channels have an average mass flux of 35-45 kg/m^2s .

9.4.4 Flow Regime

Based on the video images a qualitative analysis was made of the flow regimes. The elongated bubbles (EB) regime and annular (An) regime were mainly observed during the testing of the silicon-glass device. Following the procedure discussed in section 8.4.1, figure (9.12) shows the original image and the processed image. The processed image was created by subtracting the background and inverting the white and black color to enable clearer pattern distinction. An image without flow in the channels was used as reference for the background. The images are taken half-way the micro-channels. The silicon is seen as completely black in the processed image (right image of figure (9.12)). The vapor (either from an elongated bubble or an isolated bubble) is seen as white whereas the gray areas are liquid. Due to the illumination angle being mainly focused on the central and right side of the device, the vapor in the central and left area is shown as white with a black line in the middle. In the further discussion in this section, the unprocessed images are considered.

Figure 9.12: Left: is a snap shot of the EB flow video; right: same snap shot with background removed.

Due to the presence of a clogged channel an averaged flow rate is considered for the remaining 15 unclogged channels. Also, the 14th micro-channel from the left, appeared to have a partially clogged capillary therefore, the flow pattern of this micro-channel was different than the rest at a specific time frame. More specifically, the 14th micro-channel reached the different regimes at an earlier stage than the others and also dryout and full evaporation was seen earlier than in the other micro-channels, due to the significantly lower mass flux, as seen in figure (9.11).

Generally, the flow in the micro-channels appeared to repetitively switch from EB to An and even reaching dryout. For example, this cyclic flow pattern

transition sequence (EB to An and dryout) was repeated every 25 to 125 milliseconds when the mass flow rate was set to 0.05g/s and the power was set to 5 W (note that the largest mass flow fluctuations were seen at this mass flow rate and heating power as it is indicated in figure (9.9)). The dryout was lasting for 10-40 milliseconds before the next cycle started. The mechanism behind this flow pattern transition instability is that the bubbly flow has less void but higher pressure drop than that of the annular flow. As explained in ref. [67], close to the point of transition between EB and An regime, a potential increase in the size and number (due to a decrease in the mass flow) of the bubbles will lead to switching to An flow.

Figure 9.13: Flow pattern of the rightmost three micro-channels. The flow pattern is at zero heating power and the mass flow is on average 0.001 g/s per micro-channel (assuming uniform flow distribution). The length of the micro-channel visible is around 1 mm.

This excess pressure drop in the An regime will cause the mass flow to rise. As this happens, the vapor generation will decrease, and as the flow cannot stay in the annular pattern, it switches to bubbly flow. The effect of this cyclic pattern in the flow was not visible in the temperature measurements (PT1, PT2, PT3). The silicon substrate acted as a heat spreader, smoothing out any peaks in temperature that may have been caused by the dryout at this short period of time. Therefore, the heat transfer estimated in section 9.4.6, can be considered representative of the overall behavior. Moreover, it was observed that even without heating, the flow was two-phase mainly due to the presence of the capillaries and the parasitic heating (up to 0.3 W on the silicon device). Elongated bubbles of various lengths are mainly seen half way through the length of the micro-channels when no heating power is added.

Figure (9.13) shows a series of images taken of the exit of the capillaries and the entrance of the micro-channels to show the bubble formation (for illustration purposes mainly). The images are taken with a few milliseconds difference apart and depict always the same channels. Particularly the images are zoomed in at the rightmost three micro-channels on the right edge right (micro-channels 14 to 16) next to the clogged channel (channel 13). The first channel from the left in figure (9.13) has the partly clogged capillary and shows clearly the bubble formation after the capillary. The newly arrived isolated bubble collides with one or two bubbles downstream creating an elongated bubble. The rightmost two micro-channels have a significantly higher flow and show turbulence at the entrance of the micro-channel. The vapor enters the micro-channel at high velocity as the diameter expands. The first channel from the left in figure (9.13) further down in the micro-channel (not shown here) reaches dryout, whereas the other micro-channels are in EB/An regime. The capillary design affects the development of the flow and the bubble formation. The partly clogged capillary with the high flow resistance, enables smooth bubble formation right at the entrance of the micro-channel, due to the low mass flow rate. However, the device is not intended for such low flows and heating power, and therefore, further down stream the flow evaporates fully. On the other hand, the expansion from the unobstructed capillaries leads to a turbulent flow at the entrance of the micro-channels, and bubbles mix quickly forming elongated bubbles further down the channel.

Figure (9.14) shows the observed flow at the exit of the micro-channels and the pillar area at increasing heating power and at constant mass flow. Initially, at low heating power the EB regime is seen and as the heating power is increased the flow changes to wavy-annular. At even higher heating power (5 W) the flow is An and partially dry-out at the exit of the micro-channels. At 12 W it looks completely dry-out (no liquid slugs) which is also observed in the temperature measurements in figure (9.7), where the temperatures rise rapidly at 12 W heating power. The 14th micro-channel from the right has reached dryout in all these images. The black spots clearly visible in the pillar section, at dryout in all micro-channels, are the particles deposited as discussed in section 9.4.3. The deposits have been there from the beginning (result of production process).

Figure 9.14: Observed flow patterns at the end of the micro-channels close to the pillar section. The heating power is increased from 0.5 W to 12 W. The averaged mass flow is 0.003 g/s per micro-channel. (EB: Elongated Bubbles; W-A: Wavy-Annular; A-D: Annular-Dryout; D: Dryout)

Figure (9.15) gives a summary of all observed regimes, half-way the length of the micro-channel, at the different combinations of mass flux per micro-channel and vapor quality. The mass flux is calculated in eq. (9.1) with $N=15$. The vapor quality is the predicted vapor quality half-way through the micro-channel. The vapor quality at the entrance of the micro-channels was estimated using eq. (2.30), where $\dot{h}$ is the enthalpy at the inlet section calculated from the measured pressure (P_{in}) and fluid temperature (T_{CO2in}) using REFPROP. The change in enthalpy due to the parasitic heat leaks at the inlet section was found negligible (heat leaks <0.05 W at the inlet). All the heat from the heater and the parasitic leaks on the silicon device contribute to the evaporation of the flow (Δh_{vap}). Δh_{vap} and $\dot{h}_L$ are based on the pressure of interest. The different colors in the markers indicate a different regime and

Figure 9.15: Flow patterns observed during the measurements half-way the length of the micro-channels.

it was selected based on the prevailing regime in terms of timing during a repetitive regime cycle.

The purpose of the “capillaries” in the device is that they dominate the total pressure drop, ensuring uniform fluid distribution among the different channels, and avoiding vapor back-flow from the micro-channels. However, the small size of the capillaries appeared to be problematic. During the manufacturing or testing processes one of them got clogged due to dust particles of up to micrometer size. Moreover, the small size of the capillaries introduces large pressure drops making it difficult to preserve two-phase flow along the micro-channels at higher flow rates. Sub-cooling of the inlet section was also not possible. Therefore, the measurements include only relatively low mass flow levels. At these low flow rates it appeared impossible to establish flow in the isolated bubble regime. Due to the parasitic heat load at zero heating power and the pressure drop in the capillaries, the flow already was in the EB regime. Therefore, the IB was not observed along the length of the micro-channels. Indeed, figure (9.15) shows that most of the measurements were at the EB regime, whereas a few An flow regimes were observed and dryout. As the vapor quality increases, the An regime dominates, which is expected, as more liquid is evaporated, forming an annular flow. Many times it was difficult to assess the prevailing regime due to the above-mentioned repetition seen in the flow (switching from one regime to the other), and due to the fact that only a small fraction of the micro-channel length was visible, confusing sometimes long elongated bubbles with annular flow. At these low mass flux levels the

Ong model predicts the IB regime at all vapor quality levels. However, the Ong model is based on experiments at mass fluxes above $200 \text{ kg/m}^2\text{s}$ [30], which is typically a factor of 5 above the mass fluxes in our experiments. Extrapolation of the Ong model to our low mass fluxes appears not to be valid. Therefore, this model should be extended with low mass-flux experimental data and adapted.

9.4.5 Pressure drop

The pressure drop in the silicon-glass device, as shown in figure (9.16), is the sum of the two-phase frictional pressure drop in the capillaries, the two-phase pressure drop in the micro-channels, the pressure drop in the pillar structure, and the pressure drops due to expansion from one section to the other.

Figure 9.16: Schematic drawing of the test section for the pressure drop calculation.

The predicted pressure drop (dP_{si-p}) is calculated using the models described in chapter 2. It is seen that the pressure drop in the pillar structure is significantly smaller than that in the capillaries, as calculated in eq. (2.21) (assuming gas flow as a worst case). The overall pressure drop is measured between the sections before and after the silicon-glass device as shown in figure (9.6). The pressure drop in the silicon-glass device dP_{si-m} is estimated as the difference between the measured overall pressure drop dP_{meas} and the calculated pressure drop in the sections before and after the silicon-glass device. The two-phase flow equations were used for calculating the pressure drop in the capillaries, in the micro-channels and up to the exit of the test section. The pressure drop due to the two 90 degrees bends are neglected as they are significantly smaller. As a result the pressure drop derived from measurements is evaluated as:

$$dP_{si-m} = dP_{meas} - (dP_{con} + dP_{frict-inlet} + dP_{exp} + dP_{frict-outlet}) \quad (9.2)$$

The pressure drop in the section before the silicon-glass consists of pressure drop due to the friction ($dP_{frict-inlet}$) and due to the contraction to the smaller diameter of the capillaries (dP_{con}). The pressure drop in the section after the silicon-glass consists of pressure drop due to friction ($dP_{frict-outlet}$) and due to the expansion to the larger diameter (dP_{exp}). Both sections have two-phase flow and, therefore, the two-phase equations for the mini-channels (as presented in ref. [1] and [2]) and eqs. (2.31) and (2.32) are used. The pressure drop due to contraction (dp_{con}), and due to friction in the inlet ($dP_{frict-inlet}$) and outlet ($dP_{frict-outlet}$) tube were found to be negligible. Figure (9.17) shows the pressure drop derived from measurements dP_{si-m} and the predicted pressure drop dP_{si-p} based on eq. (9.2) at different mass fluxes in the micro-channel. The uncertainty in the pressure drop dP_{si-m} is ± 0.5 bar and ± 0.3 bar for the dP_{si-p} . The pressure drop increases with increasing mass flux. The dominant contribution in the pressure drop is due friction in the capillaries as it is predicted by the theory. Considering the mass fluxes from 0 to 100 kg/m²s, dP_{si-p} and dP_{si-m} both show an increasing trend, but the slope is quite different. Further discussion follows below.

Figure (9.18) shows the pressure drop in the silicon-glass device as derived from measurements and as predicted from the theory at different heat fluxes applied to the micro-channels. In two-phase flow, as the heat flux increases for a constant mass flow, the vapor quality increases. Higher vapor quality gives higher pressure drop due to the increased vapor velocity, and therefore higher frictional losses, as it was seen in figure (8.11) in chapter 8. However, based on figure (9.17) and figure (9.18) where the measuring data are with different heat fluxes, it can be concluded that the pressure drop in the silicon-glass device does not depend on the heat flux. This is expected since the dominant factor in the pressure drop is that of the capillaries and not that of the parallel micro-channels, where most of the evaporation takes place. Moreover, the heat

Figure 9.17: Pressure drop based on theory (dP_{si-p}) and derived from measurements (dP_{si-m}) in the silicon-glass device versus average mass flux per micro-channel.

Figure 9.18: Pressure drop predicted (dP_{si-p}) and derived from measurements (dP_{si-m}) versus heat flux applied on the device.

leak towards the capillaries is quite small, and thus the effect of the heating is not expected to be seen in the pressure drop in the capillaries. Therefore, the two-phase relationship between the heat flux and the pressure drop as seen in the single circular micro-channel measurements is not valid here.

Overall, there is a significant discrepancy between the predicted and the measured values, which is visible in both figures (9.17) and (9.18). The two-phase flow equations were used for calculating the pressure drop in the capillaries and in the micro-channels and up to the exit of the test section. In the predictions, the dimensions of the channels (inlet/outlet, capillary and micro-channel) were taken from table (9.1) and an averaged mass flow rate was used as calculated from eq. (9.1) for 15 channels. However, in figure (9.11) it is shown that the mass flow rate is not uniform across all the micro-channels. Channel 13 is completely blocked and channel 14 has a relatively small flow. But also the other channels show a large spread in the flow among the channels. It is considered that this is due to production errors and/or contamination. As can be seen in eqs. (2.11), (2.45), and (2.48), the pressure drop per unit length in a channel scales with $\frac{fG^2}{D_h}$ in which f is a friction factor that for small Reynolds numbers ($Re < 2000$) scales with Re^{-1} (eqs. (2.12), (2.49), and (2.50)). The Reynolds number is given by eq. (2.13). Knowing that the mass flux G equals $\frac{\dot{m}}{A}$, with $\dot{m}$ the mass flow rate and A the cross-sectional area, it can be derived that the pressure drop per unit of length is proportional to $\frac{\dot{m}}{(D_h^2 A)}$. If the mass-flow rate is constant then the pressure drop scales as $(AD_h^2)^{-1}$. If the channel has rectangular cross section with width w and height h then $A = wh$ and $D_h = \frac{2wh}{(w+h)}$. Now, if because of production errors or contamination the channel dimensions reduce by a factor β , then the pressure drop increases by a factor of β^4 ! The difference shown in figure (9.17) is roughly one order of magnitude. This is already established if the capillaries are about $11 \times 110 \mu m$ instead of $20 \times 200 \mu m$.

For the heat transfer coefficient section, the predicted two-phase pressure drop in the micro-channels is used, taking into account the conclusion of chapter 8 that the two-phase pressure drop predicted by the models can be used with confidence, based on the comparison shown in figure (8.11).

9.4.6 Heat transfer coefficient

In the silicon-glass device, the heat is transferred from the heater to the coolant via heat conduction in the silicon and via heat convection from the silicon to the coolant. The predicted heat transfer coefficient (h_{si-p}) is calculated using the models described in chapter 2 assuming certain entrance thermodynamic conditions and that all the heat added from the heater is transferred to the coolant. The thermodynamic conditions (pressure, enthalpy, fluid temperature) at the entrance of the silicon-glass device are derived from the measurements. The pressure at the entrance of the silicon-glass device is estimated by subtracting

the predicted values (pressure drop in the silicon-glass device and up to the outlet tube) from the outlet pressure P_{out} , (where $P_{out}=P_{in}-dP_{meas}$, see figure (9.16)). The enthalpy at the entrance of the silicon-glass device (H_{si}) is estimated, from eq. (2.29) (where $\Delta h_{vap}=H_{si}-H_{in}$). The enthalpy at the inlet section (H_{in}) is found from REFPROP based on the measured pressure (P_{in}) at the inlet section (see figure (9.16)) and the fluid temperature measured ($TCO2_{in}$) and taking into account the parasitic heat in the inlet section as the $\dot{H}$ (<0.05 W). Since the flow was saturated liquid or even two-phase, the temperature at the entrance of the silicon-glass device is estimated from the pressure. Convective parasitic heat leak towards the silicon-glass device is also included in the heating input. The parasitic heat on the silicon device surface due to convection is in the order of 0.3 W and is calculated based on section 2.1.2 assuming a vertical plate for the calculation of the characteristic dimension in eq. (2.8). The temperature difference used in eqs. (2.7) and (2.5) is between the average temperature measured on the silicon (PT1, PT2, PT3) and the environmental temperature. The convective heat leaks towards the inlet and outlet tube sections are in the order of 0.05 W and 0.1 W and were calculated based on section 2.1.2 assuming a vertical cylinder and using the length of the inlet/outlet tube as the characteristic dimension, and the temperature difference between the temperatures measured ($TCO2_{in}$ and $TCO2_{out}$, respectively) and the environmental temperature.

It should be noted, as already discussed in section 9.4.4, that the model proposed by [30] and [32] for predicting the flow-pattern map is based on experiments at mass fluxes above 200 kg/m²s whereas in the experiments described here the mass fluxes are a factor 5 lower. At these low mass flux levels (<100 kg/m²s) the model proposed by [30] and [32] predicts the IB regime at all vapor quality levels whereas the observed regimes were EB or Annular. For this reason, already it is expected to see a discrepancy between the predicted heat transfer values and those derived from measurements.

The heat transfer coefficient derived from measurements (h_{si-m}) is estimated assuming that all the heat added from the heater and due to parasitic leaks is transferred to the coolant. Moreover, the heating in the silicon is considered uniform, due to the high thermal conductivity of silicon compared to the thermal conductivity of the coolant (Biot number $\ll 1$). Therefore, with a uniform heating on the silicon, eq. (2.5) yields the h_{si-m} ; heat transfer by convection, with A the total area where CO_2 is in contact.

The temperature difference in eq. (2.5) is the difference of the average of the measured temperatures on the back side of the device; PT1, PT2, PT3, (uncertainty of 0.3°C), and the fluid temperature (uncertainty of 0.2°C) as derived from the pressure. Since the heat applied at the back of the micro-channels is known, as well as the heat-exchange area A of the micro-channels and the temperature differences, eq. (2.5) yields the heat transfer coefficient h_{si-m} . The contribution of the capillaries in the heat transfer is neglected because of the much higher heat resistance mainly as a result of the smaller area compared

to the micro-channels. The uncertainty in the heat transfer coefficient h_{si-m} is calculated using the square root of sum of squares. The main contributor in the heat transfer coefficient uncertainty is the temperature difference (with an uncertainty of 0.4°C) between the measured temperature on the back side of the device and the fluid temperature as derived from pressure. The estimated errors in the heat transfer coefficient h_{si-m} and h_{si-p} for the silicon-glass device are up to 40% due to the low temperature differences measured.

Figure (9.19) shows the heat transfer coefficient, derived from measurements (h_{si-m}) and predicted (h_{si-p}), at different heat fluxes for 3 mass flux cases (51, 68 and $89\text{ kg/m}^2\text{s}$). It can be seen that the heat transfer coefficients as derived from measurements do not match with the predicted values. This was expected since the models do not accurately predict the low regimes at these flow mass fluxes, therefore different formulas are used. However, it can be seen that at the highest vapor quality/heat flux/mass flux levels, the model predictions get closer to the error range of the measured heat transfer coefficient; h_{si-p} and h_{si-m} come closer. The reason for this can be that the physics behind the heat transfer shares a common ground and, especially near the transition of one flow regime to another, the estimation of the heat transfer coefficient shall represent the same physics in both flow regimes. Therefore, even if the flow regime is not predicted correctly by the prediction models, the physics expected shall be the same.

Figure (9.19) shows that, within measuring accuracy, all three cases presented show similar heat transfer coefficients h_{si-m} for heat flux values higher than 2.5 kW/m^2 whereas at a lower heat flux the $89\text{ kg/m}^2\text{s}$ mass flux case shows better heat transfer performance. Furthermore at heat-flux values below 2.5 kW/m^2 heat flux values, the $89\text{ kg/m}^2\text{s}$ mass flux case shows a fairly constant heat transfer coefficient h_{si-m} . The lack of extensive data in terms of heat flux variation in small steps makes it difficult to assess if this is a random event or whether it is repeated on the other mass flux cases. According to the model predictions, no effect of the mass flux is expected on the heat transfer coefficient.

Figure (9.19) shows that for average mass fluxes of 51, 68 and $89\text{ kg/m}^2\text{s}$, the heat transfer coefficient h_{si-m} increases with increasing heat flux up to about $40\text{ kW/m}^2\text{K}$ at 9 kW/m^2 . Note that in the mass-flux case of $51\text{ kg/m}^2\text{s}$ the highest heat flux applied was about 2.5 kW/m^2 with a measured heat transfer coefficient of about $6\text{ kW/m}^2\text{K}$. At a higher heat flux ($>9\text{ kW/m}^2$) for the $89\text{ kg/m}^2\text{s}$ mass flux case, as the heat flux increases, the heat transfer coefficient drops to about $12\text{ kW/m}^2\text{K}$. This drop in the heat transfer coefficient is not seen in the predicted values, but instead the heat transfer h_{si-p} keeps increasing with increasing heat flux.

According to the theory, at increasing heat flux, the liquid film on the tube wall becomes thinner, and is moved quickly with the vapor flow, removing at the same time faster the heat from the wall; meaning locally, the heat transfer

Figure 9.19: Two-phase heat transfer coefficient predicted (h_{si-p}) and derived from measurements (h_{si-m}) versus heat flux applied on the device for three mass-flux cases.

improves. In the transition from EB to An, when bubbles collide, part of the liquid between the bubbles evaporates, but also a part will add to the film thickness at the wall, thus reducing locally the heat transfer. Indeed, the flow observation indicates that the decrease in the heat transfer coefficient coincides with the flow transition from EB to An flow. The flow regime observed for a heat flux $< 9 \text{ kW}/m^2$ was EB and for a heat flux $> 9 \text{ kW}/m^2$ was An. It should be noted that the rapid switching from one regime to the other within a very short time frame, as discussed in section 9.4.4, made it difficult to assess the flow regime, and a distinction between EB and An was not always straightforward. However, this does not seem to influence the temperature reading and, therefore, the average heat transfer coefficient h_{si-m} .

Comparing figure (9.19) with figure (8.14) (single micro-channel measurements) it can be seen that the silicon-glass measurements covered low heat fluxes (up to about $15 \text{ kW}/m^2$), whereas the single micro-channel measurements covered higher heat fluxes ($10\text{--}160 \text{ kW}/m^2$). In both cases, the heat transfer coefficient increases with increasing heat flux up to the transition point of EB to An. Furthermore, at around $10 \text{ kW}/m^2$ the heat transfer coefficients are of the same order of magnitude. Figure (9.20) shows the heat transfer coefficient, derived from measurements (h_{si-m}) and predicted (h_{si-p}), versus vapor quality. In this figure the dry-out data are not included. Figure (9.20)

agrees with the discussion above on figure (9.19). Again, similar to figure (9.19), although the magnitude is different, the general tendency in measured and predicted heat-transfer coefficients is the same: an increase up to a certain quality level and then a leveling off, and in the case of the measured data a reduction at the highest quality level. Although the amount of data is limited, the leveling off starts at around a quality of 0.3 to 0.4. The increase of the heat transfer with increasing vapor quality is because the flow more and more shifts from EB to An, and in relative sense more and more of the fluid is in close contact with the heated wall. Once in fully developed annular flow, the heat transfer coefficient levels off to a constant value. A similar trend is seen in the case of the single micro-channel in figure (8.15). Near the transition point EB-An the rapid switching of the one regime to the other, as discussed above, may result in incidental higher heat-transfer coefficients. Indeed as shown in figure (9.15) the transition of EB to An flow is at a quality of around 0.4. Overall, the silicon-glass device, despite the fact that the flow is not uniformly distributed in all the channels as it was designed, due to a clogged and a partially clogged channel, the overall thermal behavior of the device is uniform, due to the high silicon thermal conductivity, that smooths out any differences in the flow patterns.

Figure 9.20: Two-phase heat transfer coefficient predicted (h_{si-p}) and derived from measurements (h_{si-m}) versus vapor quality.

9.5 Conclusions

The silicon-glass device was intended as a prototype to study two-phase flow in multiple micro-channels etched in silicon. The very small and narrow capillaries though made it difficult to study the device under the same conditions as the single micro-channel in chapter 7. Nevertheless, a few interesting conclusions can be drawn.

The models, derived by Thome and Ong [30], for predicting the flow regimes in two-phase flow and presented in section 2.5, do not correctly predict the flow regime for mass fluxes $>100 \text{ kg/m}^2\text{s}$. The models predict mainly the IB regime, whereas the flow patterns observed for all the micro-channels on average were the EB and the An, as is shown in figure (9.15). The reason for this is that authors in ref. [30] used experimental data covering mass fluxes $>200 \text{ kg/m}^2\text{s}$, and apparently the model is less accurate at much lower mass fluxes. Therefore, a comparison would be inaccurate.

Furthermore, flow pattern instabilities were observed in the parallel micro-channels. The flow pattern changes back and forth from one regime to the other, while the test parameters (mass flux and heat flux) were not modified. This fluctuation is present at each micro-channel and it is seen in figure (9.11). This flow regime instability is caused by the fact that the bubbly flow has less void but higher pressure drop than the annular flow as is explained in ref. [67]. Particularly, the two regimes have locally different pressure drops, therefore at the transition point, the curve of pressure drop versus mass flow rate will have a negative slope and periodic oscillations can occur according to [68]. And indeed the total mass flow was fluctuating reaching even a 20% fluctuation at certain heating power. This was a result of a flow pattern transition from bubbly flow to annular flow for the micro-channels on average, as well as due to the non-uniform distribution of the flow among the micro-channels, as a result of different flow resistances of the corresponding capillaries. The cyclic pattern of the flow regime was repeated every 125 milliseconds (worst case), however the mass flow was registered every 2 seconds which makes it difficult to get the complete picture and conclusion. The pressure drop for the silicon-glass device, derived from the measurements, was mainly due to the friction in the capillaries. The pressure drop was increasing with increasing mass flux, whereas no clear trend was seen with the heat flux. This is expected since the dominant contributor during two-phase flow for this small diameter capillary was the frictional pressure drop, and due to the fact that the heating of the capillaries due to parasitic heat was less important (most of the evaporation takes place in the micro-channels). The heat transfer coefficient derived from the measurements in the micro-channels increases with increasing heat flux, whereas peaks in heat transfer coefficient are seen before switching from one regime to the other, similarly as in the single circular micro-channel presented in chapter 8.

Overall, the thermal behavior of the device is uniform, despite the non-uniformity

seen in the flow and the flow patterns. The high silicon thermal conductivity played an important role here, as it smooths out any differences in the flow patterns. Further studies are highly advised to conclude on whether the existing two-phase models can be used reliably to estimate thermal and flow characteristics in multiple micro-channels and in micro-channels etched in silicon. It is clear that the models used to estimate the flow regimes need to be expanded to lower mass fluxes.

Outlook

An abundance of studies on CO_2 evaporative cooling in mini-channels already exists; consequently, in previous years, prediction models have been confidently used by the European Organization for Nuclear Research (CERN) for cooling applications. Cooling with micro-channels has become another valuable method of cooling particle detectors with the main focus on high density particle detectors. Despite the successful use of micro-channel cooling devices in LCH-b and AMS, uncertainty remains in the theoretical predictions because of limited studies at lower temperatures with small diameters. In this thesis, studies on a single micro-channel examined the flow and thermal behavior at low temperatures at a diameter of 0.2 mm. Even though the qualitative flow patterns and pressure drop predictions seem to agree with the measurements, the heat transfer coefficient, which is the most important parameter in the cooling device, appears to depend on the experimental set-up. Efforts need to be made to determine the dependency on experimental parameters with separate testing, with a best case scenario of prediction models having no experimental dependency. This effort will require the inclusion of models based on non-dimensional numbers. Ideally, a wide range of diameters and temperatures will be studied to create a map covering. In order to extend the validity of micro-channel models, additional data are needed from independent experimental set-ups.

Further studies are recommended for multiple micro-channels configurations, such as those in chapter (9). Careful design of the device is crucial for studying the flow and thermal characteristics of the CO_2 in micro-channels. The comparison of a multiple micro-channel configuration to a single micro-channel configuration is then the first step to identify whether existing single-channel models can be used for multiple micro-channels as well. Quite important is the study how the parallel channels effect each other with respect to flow and thermal instabilities.

CO_2 evaporative cooling is a promising technique for industry. Its application in high-tech industries could not only reduce the use of material but also provide high-efficiency cooling systems for high density ASICs. Currently, CO_2 evaporative cooling as presented here is based on an open system, which requires bulky components to condition CO_2 at both the inlet and outlet of the cooling tube. To make CO_2 evaporative cooling feasible and attractive

for high-volume production and autonomous systems, a closed system shall be utilized. CERN created a closed system that can be more easily adapted to industrial environments. An additional step will include the construction of a miniaturized closed CO_2 cooling system, which will be more applicable to smaller-scale customers such as PC processors.

Finally, due to lack of available studies, a lesser level of confidence exists about prediction models for vertical oriented mini-channels. At the mini-scale level, gravity contributes to the distribution of the flow in vertical tubes. Testing vertical oriented petals would provide greater insight and increased confidence for all petal configurations.

Appendix A

Heat Analysis I

Goal of this section is to analyze the heating pattern and show the uniform temperature and heating at the inner tube wall of the micro-channel. The analysis of the heating pattern in chapter 7 is quite complicated since the heat conduction in the tube wall is also dependent on the heat transfer coefficient which is unknown and varying over the length. The problem is simplified by considering a small section of the tube with length half of which is heated and half is not. In the small section the heat transfer coefficient is considered constant. The section examined includes two heated and an unheated section in between having the same length. The problem can be assumed symmetric with respect to the center of the unheated section along the z direction. The problem is described next.

A.0.1 Non-uniform heating

A hollow cylinder is examined with r_1 and r_2 the inner and outer radius. The cylinder is heated in turns leaving unheated parts.

The temperature is considered independent in the ϕ direction due to the following assumptions:

1. Constant temperature of the fluid and constant heat transfer coefficient along the length.
2. Uniformly distributed heat (q) on the tube.

To see the temperature distribution in the thickness wall of the hollow cylinder, a finite area is examined that includes an unheated part with heated parts on each side. This is the smallest section that is repeated along the length and the solution of this domain is representative of the T distribution for the whole cylinder.

Symmetry exists on the mid plane of unheated part. Therefore, half of the domain can be solved instead of the total adding symmetry conditions. The

Figure A.1: Heat conduction problem with non-uniform heating and heat convection in 2D

Laplace equation describes the problem and will be used to find the temperature distribution and heat flux over the r - z domain.

The Laplace equation for the cylindrical coordinates on the r - z is given by the following equation.

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = 0 \quad (\text{A.1})$$

The boundary conditions are summarized.

$$\frac{\partial T_{(r,0)}}{\partial z} = 0 \quad (\text{A.2})$$

$$\frac{\partial T_{(r,L)}}{\partial z} = 0 \quad (\text{A.3})$$

$$\frac{\partial T_{(r_1,z)}}{\partial r} = -\frac{h}{k} \frac{A_h}{A_k} (T_{(r_1,z)} - T_{CO2}) \quad (\text{A.4})$$

$$\frac{\partial T_{(r_2,z)}}{\partial r} = q(z) \quad (\text{A.5})$$

where

$$q(z) = \left\{ \begin{array}{ll} 0, & \text{for } 0 \leq z \leq \frac{L}{2} \\ -\frac{q''}{k A_k}, & \text{for } \frac{L}{2} \leq z \leq L \end{array} \right\} \quad (\text{A.6})$$

Figure A.2: Heat conduction problem with non-uniform heating and heat convection in 2D

To solve the second order homogeneous partial differential equation with the method of the Separation of Variables, the variable $T(r,z)$ is changed in order to keep only one non-homogeneous boundary condition (A.5).

$$\Theta(r, z) = T_{(r,z)} - T_{CO2} \quad (\text{A.7})$$

The boundary conditions stay the same but include the Θ variable while (A.4) becomes

$$\frac{\partial \Theta_{(r_1,z)}}{\partial r} = -\frac{h}{k} \frac{A_h}{A_k} (\Theta_{(r_1,z)}) \quad (\text{A.8})$$

The problem has a homogeneous partial differential equation (A.1), three homogeneous boundary conditions (A.2),(A.3), (A.8), and one non homogeneous (A.5). The method of Separation of Variables is used to find a solution. First, we assume the solution is a product of the following form:

$$\Theta(r, z) = R(r)Z(z) \quad (\text{A.9})$$

The solution should satisfy the partial differential equation and the boundary conditions. Therefore, it is inserted in the PDE which is separated to get a function of z and r introducing also a separation constant. As it can be seen, the z direction is the homogeneous one with two homogeneous boundary conditions. The selection of the sign for the separation constant is chosen below respectively. It is expected to obtain the eigenfunctions in the z direction in terms of sines and cosines and the solutions in the r direction will be modified Bessel functions. Two ordinary differential equations are concluded at the end as follows:

$$\frac{d^2 R}{dr^2} + \frac{1}{r} \frac{dR}{dr} - \lambda^2 R = 0 \quad (\text{A.10})$$

$$\frac{d^2 Z}{dz^2} + \lambda^2 Z = 0 \quad (\text{A.11})$$

A.0.1.1 Solutions for the ODE

The partial differential equation has been transformed to two ordinary differential eqs. (B.8) and (A.11) with solutions:

$$R_n = A_1 I_0(\lambda_n r) + A_2 K_0(\lambda_n r) \quad (\text{A.12})$$

$$Z_n = B_1 \sin(\lambda_n z) + B_2 \cos(\lambda_n z) \quad (\text{A.13})$$

$$R_0 = A_0 \ln(r) + B_0 \quad (\text{A.14})$$

$$Z_0 = C_0 z + D_0 \quad (\text{A.15})$$

So each product of $R(r)Z(z)$ is a solution of equation (A.1) That is:

$$\Theta(r, z) = R_0(r)Z_0(z) + \sum_{n=1}^{\infty} R_n(r)Z_n(z) \quad (\text{A.16})$$

A.0.1.2 Application of boundary conditions

Application of boundary condition (A.2) in (A.16) gives $B_1=0$ and $C_0=0$ while the application of boundary condition (A.3) gives the eigenfunction.

$$\sin(\lambda_n L) = 0 \quad (\text{A.17})$$

The positive roots of the eigenfunction (A.17) give the eigenvalues λ_n .

$$\lambda_n = \frac{n\pi}{L} \quad \text{for } n = 0, 1, 2, \dots \quad (\text{A.18})$$

Application of boundary condition (A.8) gives

$$B_0 = A_0 \left(\frac{1}{r_1} + \frac{h}{k} \frac{A_h}{A_k} \ln r_1 \right) \quad (\text{A.19})$$

$$A_1 = A_2 N \quad (\text{A.20})$$

where

$$N = \frac{-\frac{h}{k} \frac{A_h}{A_k} K_0(\lambda_n r_1) + \lambda_n K_1(\lambda_n r_1)}{-\lambda_n I_1(\lambda_n r_1) + \frac{h}{k} \frac{A_h}{A_k} I_0(\lambda_n r_1)} \quad (\text{A.21})$$

The solution so far is

$$\Theta(r, z) = A_2 [N I_0(\lambda_n r) + K_0(\lambda_n r)] [A_4 \cos(\lambda_n z)] \quad (\text{A.22})$$

The coefficient A_2 and A_4 can be combined to one A_{24}

$$\Theta(r, z) = A_{24}[NI_0(\lambda_n r) + K_0(\lambda_n r)][\cos(\lambda_n z)] \quad (\text{A.23})$$

Application of boundary condition (A.5) gives

$$A_0 = \frac{q}{k} \frac{r_2}{A_k} \quad (\text{A.24})$$

So the solution at $n=0$ with $R_0(r)=0$

and for the positive n

$$A_{24}[-NI_1(\lambda_n r_2)\lambda_n - K_1(\lambda_n r_2)\lambda_n][\cos(\lambda_n z)] = q(z) \quad (\text{A.25})$$

This can't be solved directly because of the variable z and the summation sign. At this point we invoke orthogonality. The z Eq. is a solution to Eq. (30). Orthogonality can be applied if the above Eq. is a Sturm-Liouville Eq. and has homogeneous boundary conditions. Comparing with a Sturm-Liouville Eq., we get a $w(z)=1$. Multiplying both sides with the Eq. above and integrating from $L/2$ to L where we get:

$$A_{24} = - \frac{\frac{q''}{k A_k} \int_{\frac{L}{2}}^L \cos(z \lambda_n) dz}{\lambda_n (-NI_1(\lambda_n r_2) - K_1(\lambda_n r_2)) \int_{\frac{L}{2}}^L \cos(z \lambda_n)^2 dz} \quad (\text{A.26})$$

The final solution is summarized;

$$\begin{aligned} \Theta(r, z) = & \frac{q}{k} \frac{r_2}{A_k} \left(\ln(r) + \frac{1}{r_1} + \frac{h}{k} \frac{A_h}{A_k} \ln(r_1) \right) \\ & + \sum_{n=1}^{\infty} A_{24}[NI_0(\lambda_n r) + K_0(\lambda_n r)][\cos(\lambda_n z)] \end{aligned} \quad (\text{A.27})$$

The analytical solution (A.27) gives the temperature difference distribution $\Theta(r, z)$ along the radial and longitudinal plane. Figure (A.3) gives the solution of the equation (A.27) for inner radius $r_1=0.125$ mm, outer radius $r_2=0.8$ mm, length $L=0.5$ mm, thermal conductivity $k=17$ W/mK, heat transfer coefficient $h=80000$ W/m²K constant along the length and heating power $q=15$ W. Figure (A.3) shows that with a heating space of 0.5 mm adequate uniformity of the

Figure A.3: Temperature difference distribution of the heat conduction problem with heat convection and non-uniform heating in 2D for $r_1=0.125$ mm, $r_2=0.8$ mm, $L=0.5$ mm, $k=17$ W/mK, $h=80000$ W/m²K constant along the length and $q=15$ W.

temperature can be achieved at the inner wall of the tube despite the non-uniform heating pattern at the outer wall.

Appendix B

Heat Analysis II

B.0.1 Uniform heating

A hollow cylinder of total length L is heated uniformly peripherically only at a length L_2 while the rest length is treated adiabatically. The inner surface of the cylinder is subject to heat convection with a heat transfer coefficient h and temperature T_c . At a certain distance L from the heated part, the temperature is considered equal to T_c . The problem is considered one dimensional, z direction due to the small thickness compared to the total length.

Figure B.1: Heat conduction problem with uniform heating and heat convection in 1D

Setting the problem

Neglecting the r -direction and assuming a z -direction problem we get the problem in figure (B.1). We split the problem in two domains. The first domain (A) is for $0 < z < L$ and the second domain (B) for $L < z < L_1$.

Approximate solutions: The integral method

In the integral method the basic laws are satisfied in an average sense unlike the differential formulation which are satisfied at every point. Starting with the principle of conservation of energy, we result in a heat balance integral which is the governing equation. For this example, I used the control volume formulation. The formulation for the domain A is explained below.

Figure B.2: Heat conduction problem with uniform heating and heat convection in 1D-domain A

$$E_{in} - E_{out} = 0 \quad (\text{B.1})$$

$$E_{in} = kA_c \frac{dT(L)}{dz} \quad (\text{B.2})$$

$$E_{out} = -hc \int_0^L (T - T_c) dz \quad (\text{B.3})$$

$$kA_c \frac{dT(L)}{dz} + hc \int_0^L (T - T_c) dz = 0 \quad (\text{B.4})$$

and the boundary conditions with c the perimeter

$$T(0) = T_c \quad (\text{B.5})$$

$$kA_c \int_0^L \frac{d^2T(L)}{dz^2} = 0 \quad (\text{B.6})$$

$$T_A(L) = T_B(L), \quad \frac{dT_A(L)}{dz} = \frac{dT_B(L)}{dz} \quad (\text{B.7})$$

The boundary condition (B.7) is the natural boundary condition, ie the equality of temperatures and heat fluxes at the common surface.

Assumed temperature profile for domain A

Assume a polynomial of the second degree which should satisfy the boundary conditions and governing equation.

$$T(z) = a_0 + a_1z + a_2z^2 \quad (\text{B.8})$$

Determination of unknown coefficients

Replacing (B.8) to (B.6) we get

$$a_2 = 0 \quad (\text{B.9})$$

$$a_0 = T_c \quad (\text{B.10})$$

Here we stop and continue with the domain B, since the common boundary conditions should be applied.

The formulation for the domain B is explained below.

$$E_{in} = q'' \quad (\text{B.11})$$

$$E_{out} = -kA_c \frac{dT(L)}{dz} - hc \int_L^{L_1} (T - T_c) dz \quad (\text{B.12})$$

Figure B.3: Heat conduction problem with uniform heating and heat convection in 1D-domain B

$$q'' + kA_c \frac{dT(L)}{dz} + hc \int_L^{L_1} (T - T_c) dz = 0 \quad (\text{B.13})$$

and the boundary conditions

$$kA_c \frac{dT(L_1)}{dz} = 0 \quad (\text{B.14})$$

$$T_A(L) = T_B(L), \quad \frac{dT_A(L)}{dz} = \frac{dT_B(L)}{dz} \quad (\text{B.15})$$

Assumed temperature profile for domain B

Again here a polynomial of the second degree is assumed which should satisfy the boundary conditions and governing equation.

$$T(z) = a_0 + a_1 z + a_2 z^2 \quad (\text{B.16})$$

Determination of unknown coefficients

Replacing (B.16) to (B.14) we get the a_1 coefficient for the second problem.

$$a_1 = -2a_2L_1 \quad (\text{B.17})$$

So far we have

$$T_a(z) = T_c + a_1z \quad (\text{B.18})$$

$$T_b(z) = a_0 - a_2(2L_1z - z^2) \quad (\text{B.19})$$

Application of the common boundary conditions give the a_1 coefficient for the first and a_0 for the second problem.

$$a_1 = -2a_2(L_1 - L) \quad (\text{B.20})$$

$$a_0 = T_c + a_23L^2 \quad (\text{B.21})$$

So we get

$$T_a(z) = T_c + 2a_2(L - L_1)z \quad (\text{B.22})$$

$$T_b(z) = T_c + a_2(3L^2 - 2L_1z + z^2) \quad (\text{B.23})$$

Replacing (B.23) to the equation (B.13) gives

$$a_2 = \frac{q''}{kA_c2(-L_1 + L) - hc(\frac{7}{3}L^3 + \frac{2}{3}L_1 - 4L_1L^2)} \quad (\text{B.24})$$

Replacing (B.22) to the equation (B.4) gives quadratic polynomial. Finding the positive roots gives the L , the entrance length at which we have no effect of the heating.

$$(h2pir_2)L^4 - (h2pir_2L_1)L^3 - (hcL_1)L^2 + (kA_c2)L - kA_c2L_1 = 0 \quad (\text{B.25})$$

Appendix C

Error analysis

C.1 Error analysis

Temperature measurements

The accuracy of the temperature measurement is a combination of the accuracy of the sensor, the reading device and the cold junction temperature when a thermocouple is used. The accuracy of the thermocouple sensor is determined mainly by its materials and the accuracy of the emf conversion to temperature. These errors are considered systematic errors and not included in the analysis.

The random errors taken into account is the error introduced when averaging the measurement over a period of time at steady state conditions. The accuracy of the absolute temperature derived by adding the T_{ref} is given by this random error and the T_{ref} error.

Pressure measurements

The uncertainties in the pressure estimation are calculated using the square root of sum of squares of each of the components.

$$\sigma P_{inevap} = \sqrt{\sigma P_{in}^2 + \sigma dp_{preheater}^2} \quad (C.1)$$

$$\sigma P_{outevap} = \sqrt{\sigma P_{ah}^2 + \sigma P_{outmicro}^2} \quad (C.2)$$

The uncertainty in the calculation of the P_{ah} from T_{ah} is 1% , 1.5% for the P_{inevap} and 2% for the $P_{outevap}$.

To estimate the uncertainty in the prediction, the model is run with input the P_{inevap} (as found in step 2) over its error range. This results in a predicted $P_{outevap}$ with its error range.

Prediction

To estimate the uncertainty in the prediction, the model is run with input the P_{inevap} (as found in step 2) over its error range. This results in a predicted $P_{outevap}$ with its error range. As well as the $T_{wallout}$, h_{pred} and T_{fluid} .

The uncertainty in the prediction is found by giving as input the P_{in} over an offset range.

Appendix D

Petal Information

D.1 Heater Information

Heater	Circuit	Resistance [Ohm]
1	Sensor	3.12
	Hybrid-1	1.00
	Hybrid-2	1.13
2	Sensor	2.80
	Hybrid-1	1.25
	Hybrid-2	1.37
3	Sensor	2.11
	Hybrid-1	1.73
4	Sensor	1.73
	Hybrid-1	1.00
5	Sensor	1.67
	Hybrid-1	1.16
6	Sensor	1.49
	Hybrid-1	1.45

Table D.1: Resistance specifications for each heater circuit on each heater. Starting from the innermost ring sensor (heater 1) to the outermost ones (heater 6) the sensor-heaters present a decreasing resistance indicating the highest expected heat load in the outermost sensors. The resistance in the hybrid-heaters also varies due to the different number of chips on each hybrid.

D.2 Petal model

Figure D.1: View of honeycomb core material close-outs and carbon foam. Cooling tubes are embedded in the carbon foam.

Figure D.2: View of facesheets that cover all the petal core area.

Figure D.3: View of petal with silicon wafers (blue) and hybrids (green).

Figure D.4: View of petal with ASICs placed on the hybrids and DC/DC converters placed on the silicon wafers. The number of ASICs depends on the size of the wafer.

Appendix E

Radiation damage at TID bump

We would like to find the level of radiation damage in the silicon sensor at the time of TID bump. Assuming the TID bump peak ($\widehat{TID}$) occurs at the same dose of 2.5×10^4 Gy, we look now at the position dependency $k(r, z)$; where k in $n_{eq} \text{ cm}^{-2}/\text{Gy}$. $k(r, z)$ is defined by:

$$k(r, z) = \frac{n_{eq}(3000 \text{ fb}^{-1}; r, z)}{\widehat{TID}(3000 \text{ fb}^{-1}; r, z)} \quad (\text{E.1})$$

The non-ionizing dose at the TID bump peak is found by multiplying with the ionizing dose at TID bump.

$$n_{eq}(\widehat{TID}; r, z) = k(r, z) \times \widehat{TID} \quad (\text{E.2})$$

Table (E.1) gives the position dependent radiation dose at the time of the TID bump.

Ring	Non-ionizing dose at 3000 fb^{-1} $n_{eq} \text{ cm}^{-2}$	Ionizing dose at 3000 fb^{-1} Gy	$k(r, z)$ $n_{eq} \text{ cm}^{-2}/\text{Gy}$	Non-ionizing dose at TID bump $n_{eq} \text{ cm}^{-2}$
0	$8.17 \cdot 10^{14}$	$0.36 \cdot 10^6$	$22.3 \cdot 10^{14}$	$0.56 \cdot 10^{14}$
1	$6.74 \cdot 10^{14}$	$0.22 \cdot 10^6$	$31.0 \cdot 10^{14}$	$0.76 \cdot 10^{14}$
2	$5.81 \cdot 10^{14}$	$0.15 \cdot 10^6$	$37.5 \cdot 10^{14}$	$0.95 \cdot 10^{14}$
3	$5.34 \cdot 10^{14}$	$0.13 \cdot 10^6$	$41.4 \cdot 10^{14}$	$1.05 \cdot 10^{14}$
4	$4.53 \cdot 10^{14}$	$0.09 \cdot 10^6$	$48.2 \cdot 10^{14}$	$1.28 \cdot 10^{14}$
5	$4.00 \cdot 10^{14}$	$0.08 \cdot 10^6$	$50.6 \cdot 10^{14}$	$1.37 \cdot 10^{14}$

Table E.1: Estimated non-ionizing dose in each ring at the TID bump peak for disk 5.

The integrated luminosity received by the sensors is quite low compared to the end of life case therefore can be neglected.

Appendix F

Silicon electrical conductivity

For each sensor we calculate Q_{Tref} from equation (3.6) with j_{Tref} calculated as

$$j_{Tref} = I_{Tref} t \quad (F.1)$$

and I_{Tref} is taken from eq. (3.1) neglecting I_o . Q_{Tref} is converted to a conductivity σ_{Tref} using eq. (3.7). Then σ_T is obtained at 5 K intervals using the scale factor $R(T)$ given in eq. (3.4).

Table (F.1) gives the temperature dependent electrical conductivity for sensor R0 only which is given as input in the thermal model in order to estimate the power produced in the sensor. In the model, σ_T is scaled linearly between the nearest two values in this table.

Table F.1: Temperature dependent electrical conductivity for the sensor in R0.

Temperature	Electrical conductivity
$^{\circ} C$	$\text{Ohm}^{-1} \text{ mm}$
-40	1.29E-11
-35	2.45E-11
-30	4.54E-11
-25	8.19E-11
-20	1.44E-10
-15	2.49E-10
-10	4.20E-10
-5	6.96E-10
0	1.13E-09
5	1.807E-09
10	2.83E-09
15	4.38E-09
20	6.67E-09
28	1.27E-08
30	1.48E-08
35	2.16E-08
40	3.12E-08
45	4.46E-08

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Summary

The first goal of this thesis was to show that CO₂ evaporative cooling can enable the ATLAS detector to meet the stringent thermal and flow requirements in the upcoming upgrade. The next goal was to extend the validity of prediction models for CO₂ evaporation to the micro-channel level as those are useful for high-power density detectors that will be used in the ATLAS experiment in the future.

Petal detector

At the outset of this thesis work, the decision was made to adopt evaporative CO₂ cooling for the inner tracker (ITk). Optimization of the cooling design and proving that the design cools well enough remained to be done. Nikhef had a prototype petal and CO₂ cooler available for this investigation. The petal was equipped with heaters and dummy modules to represent the detector heating. A thermal finite element analysis (FEA) model of the petal was built and verified against measurements performed on the prototype. The overall results allow good verification of an FEA model, which can then be used to reliably improve the petal design without the need to build new prototypes at every change in the design. The validated model was applied to study the behavior of the detector in new designs. The optimized parameters are the fractional radiation length of the petal, minimizing hot spots such as electrical chips and maximizing the thermal safety margins such as distance from thermal runaway.

An FEA thermal model of a real petal was built and the current design was examined under two cases during the operation: the Total Ionization Dose (TID) bump and the end of life. The results revealed some important issues in the current design that need to be tackled to run safely for 10 years of operation. Firstly, the DC/DC converter ASIC receives more current than it can handle and, therefore, a considerable drop in the efficiency takes place, leading to excess heat generated at the ASIC. We proposed a design with two DC/DC converters instead of one for R1 and R3 (the second and fourth ring from the inside radius of the petal). The FEA results showed that the presence of two DC/DC converters in R1 and R3 significantly reduces the hot spots on the silicon by 10 °C. Secondly, the study of the CO₂ in terms of flow and heat

transfer revealed that in the current design the expected flow regime is not optimum. The optimum flow regime should have high heat transfer between the coolant and the wall. Different inner diameters were examined in terms of heat transfer and thermal runaway to find the optimum regime, and it was shown that a decrease in the inner diameter will give similar results as a larger one. A smaller diameter is preferred even when the thermal performance is similar because of the better radiation length. Moreover, according to the studies in ref. [61] a smaller diameter will not deteriorate the mechanical performance of the petal, it is easier to build, and it creates less stress during cool down. For these reasons, a reduction of the inner diameter down to 1.3 mm is recommended. Furthermore, the facesheet material K13C2U 45 gsm is preferred over the K13D2U 80 gsm. Despite the slightly warmer silicon, it is preferred because it contributes less to the material budget and the mechanical performance of the petal remains adequate [61]. Other optimization steps such as change of the tube configuration, change in the ASIC-hybrid glue, and the addition of carbon foam in thermally critical locations can improve the thermal performance, but depend on the positioning of all the power dissipating components, which is still unclear. Finally, it is suggested to use the CoBra model with non-uniform heat flux input, as it can improve the precision. It was shown that the uniform assumption is optimistic by 1 to 2 °C.

The ATLAS detector will be installed in 2022. Overall, the petal design taking into account the above suggestions will reliably work safely during the 10 years of operation. CO₂ evaporative cooling is highly advised for this design.

Evaporative CO₂ in micro-channels

To go beyond the current standard practice in thermal engineering, a second goal of the thesis is to evaluate the performance of CO₂ cooling in micro-channels. Cooling in micro-channels is favorable for high power-density detectors. The models for the mini-channel (diameter > 1 mm) cooling need adaptation for this goal; prediction models exist but only for low-pressure evaporative cooling and temperatures higher than -10°C and in channels larger than 0.5 mm diameter. The goal of this research is to extend the range of application to below 0.3 mm diameter and to -20°C. Experimental validation of the models is a major goal of this research.

Tests on a single straight micro-channel were conducted to validate the empirical models for the two-phase flow presented in chapter 2. A single-phase validation of the set-up was performed prior to the two-phase tests. Pressure drops and heat transfer coefficients based on measurements match with theoretical predictions within measuring errors.

The two-phase parameters that were derived from the two-phase measurements are the pressure drop and the heat transfer coefficient, whereas the flow regime is directly visualized with a high speed camera, right after the test section. The

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recorded flow regimes in two-phase flow are in good agreement with the Ong model that correctly predicts the regime in about 86% of the total data. The two-phase pressure drop is predicted well with the relevant model depending on the flow regime. The heat transfer coefficient during two-phase flow was calculated from the measured wall temperatures. Along the tube, the heat transfer coefficient increased until after a maximum it dropped indicating the presence of a local dry-out. There was no clear relationship seen with the mass flux and heat flux for the EB regime, whereas for the annular regime a trend was seen in the heat-transfer coefficient indicating a linear dependency with the heat flux.

The heat transfer coefficient was compared with the relevant model depending on the flow regime. As a general remark, the heat transfer coefficient was under-predicted by the models, the differences ranging up to one order of magnitude. The models were then further examined on different empirical parameters that were adjusted depending on the set-up. To define the most critical parameter, we modified all parameters by the same factor. The c_{do} (which is a correction factor for the initial film thickness) for the EB regime proves to be the one that had the largest effect on the results. In the analysis, it is shown that the heat-transfer prediction of the model can be improved by fine tuning the relevant empirical parameters.

Generally, the models examined here in terms of heat transfer coefficient are useful for qualitative trend studies. However, if quantitative heat transfer studies are needed, then the experimental parameters need to be well known and shall be determined in a separate set of test experiments. This, in particular, holds for the EB model that shows high sensitivity to these empirical parameters.

Finally, because in HEP experiments multiple micro-channels are preferred for effectively cooling high-power density particle detectors, a silicon-glass multi-channel device was studied. The very small and narrow capillaries, however, made it difficult to study the device with the same conditions as the single micro-channel in chapter (7). Moreover, the model of ref. [30] for predicting the flow regimes cannot be used at low flows as the model is valid for mass fluxes $>200 \text{ kg/m}^2\text{s}$. So, a comparison would be inaccurate. Nevertheless, some interesting conclusions can be drawn. Generally, the flow patterns observed were the elongated bubbles and the annular flow. Sometimes instabilities were seen, as the flow changed back and forth from one regime to the other, whereas the test parameters (mass flux and heat flux) were constant. The pressure drop in the device as derived from the measurements increases with increasing mass flux as expected due to the significant contribution from the two-phase frictional pressure drop in the capillaries. The heat transfer coefficient showed a linear relationship with the heat flux. The heat transfer values derived from measurements for the parallel micro-channel showed similar behavior as in the single micro-channel presented in chapter (8). Further studies at higher mass fluxes are needed to conclude on the behavior in multiple micro-channels and in micro-channels etched in silicon.

To conclude, CO₂ evaporative cooling has proven to be a promising method for cooling detectors in HEP. In this thesis, it was demonstrated that the evaporative cooling successfully keeps the petal detector in the ATLAS experiment cold and safe from thermal runaway for the 10 years of operation. CO₂ evaporative cooling in micro-channels is also promising for high power-density detectors. In this thesis, it was demonstrated that the models correctly predict the flow regime and pressure drops so far. Poor prediction was seen for the estimation of the heat transfer coefficient using the prediction models described in chapter (2). And finally, because it was not possible to validate the prediction models with the measurements in the multiple micro-channels, only qualitative conclusions could be drawn.

Samenvatting

Het eerste doel van dit proefschrift was aan te tonen dat koeling door middel van verdamping van CO₂ de ATLAS detector in staat zal stellen te voldoen aan de thermodynamische eisen die gesteld worden aan de toekomstige upgrade. Het tweede doel was de geldigheid van voorspellende modellen voor CO₂ koeling uit te breiden naar micro-kanalen, die geschikt worden geacht voor de hoge-vermogensdichtheid detectoren die in het ATLAS experiment van de toekomst gebruikt zullen worden.

Petal detector

Bij de start van dit onderzoek was al beslist dat de binnenste subdetector in de toekomstige ATLAS (de “inner tracker”, ITk) gekoeld zou worden door CO₂ verdampings-koeling. De optimalisatie van het koeler ontwerp en het aantonen dat het ontwerp goed functioneerde moest nog gedaan worden. Nikhef had hiervoor een “petal” (schijfsegment van de ATLAS detector) met CO₂ koeler beschikbaar. Voor dit onderzoek was de petal uitgerust met heaters en dummy modules waarmee de te verwachten warmtebelasting op de petal nagebootst kon worden. Een thermisch eindige-elementen model (FEA) van de petal is gemaakt en gevalideerd met metingen aan het prototype. Overall bleek het model goed te voldoen en kan dus gebruikt worden om betrouwbaar het petal-ontwerp te verbeteren zonder telkens nieuwe prototypes te moeten maken bij elke voorgestelde ontwerpverandering. Het gevalideerde model is in dit proefschrift gebruikt om het gedrag van de detector in enkele nieuwe ontwerpen te bestuderen. Optimalisatie was gericht op het minimaal beïnvloeden van straling die gedetecteerd moest worden: maximaliseren van de “radiation length” (de gemiddelde afstand waarover de energie van een electron met een factor 1/e afneemt), en op het minimaliseren van hot spots ten gevolge van bijvoorbeeld elektronische componenten, en het maximaliseren van thermische veiligheidsmarges.

Een FEA thermisch model is gemaakt van de huidige petal en het ontwerp van die petal is beschouwd voor twee cases: de “TID bump” en “end of life”. In de “TID bump” wordt nagegaan wat het effect is van een maximale totale stralingsdosis die verwacht kan worden (de “Total Ionizing Dose”); De “end-of-life” case beschouwt het thermisch gedrag aan het eind van de beoogde levensduur (10 jaar). Uit dit onderzoek kwamen enkele zaken naar voren die

in het huidig ontwerp aangepast moeten worden om de volledige 10 jaar goed en veilig met de detector te kunnen werken. Ten eerste, is de stroom door de DC/DC converter ASIC te groot waardoor de efficiency afneemt hetgeen leidt tot een grotere dissipatie-warmte in de ASIC. Als oplossing wordt voorgesteld met twee DC/DC converters te werken in plaats van met een enkele, en die te plaatsen op de ringen R1 en R3 van de petal. De FEA resultaten laten zien dat hierdoor de “hot spots” op het silicium van de petal met 10 °C teruggebracht worden. Ten tweede toonde de studie van de warmte-overdracht in de CO₂ flow aan dat in het huidig ontwerp het flow regime niet optimaal is. Met een optimaal flow regime zou een hogere warmte-overdracht tussen CO₂ en wand te realiseren moeten zijn. Verschillende binnendiameters zijn onderzocht in de zin van thermisch gedrag om een optimaal regime te vinden, maar het bleek dat een kleinere of grotere diameter nauwelijks verschil maakt. Dan verdient een kleinere diameter de voorkeur vanwege de grotere “radiation length”. Bovendien, volgens studies gepubliceerd in [61], beïnvloedt een kleinere diameter het mechanisch gedrag van de petal niet nadelig, is een kleinere diameter makkelijker te realiseren en geeft het minder spanning in het materiaal tijdens afkoeling. Vanwege deze redenen wordt een diameter van 1.3 mm aanbevolen. Verder wordt aanbevolen als materiaal voor de facesheets (dunne lagen die de warmte moeten spreiden over de petal) K13C2U 45 gsm te gebruiken in plaats van K13D2U 80 gsm. Ondanks dat het silicium iets warmer wordt, heeft dit materiaal de voorkeur vanwege de lagere massa terwijl het mechanisch gedrag van de petal afdoende blijft. Andere suggesties voor verbetering zoals andere configuraties van de koelkanalen, andere ASIC-hybrid lijm, en het gebruik van carbon-schuim in thermisch kritische zones, hangen alle af van de posities van de verschillende vermogen-dissiperende componenten, en die zijn nog niet duidelijk. Tenslotte is voorgesteld om CoBra te gebruiken met niet-uniforme warmtestroom als input, omdat dit de precisie kan verbeteren. Aangetoond is dat de aanname van uniforme warmtestroom te optimistische resultaten geeft (1 á 2 °C te laag). Het is de bedoeling de ATLAS detector in 2022 te gaan installeren. Overall, zal het petal ontwerp met inachtneming van de bovenvermelde verbeteringen betrouwbaar en veilig werken voor de duur van 10 jaar. Koeling door verdamping van CO₂ wordt nadrukkelijk aanbevolen.

CO₂ koeling door verdamping in micro-kanalen

Een tweede aspect van dit proefschrift betreft het onderzoek aan de prestaties van CO₂ koeling in micro-kanalen om daarmee een volgende stap te kunnen zetten in thermisch ontwerpen. Zoals boven al aangegeven heeft het de voorkeur gebruik te maken van kleine kanalen, maar de beschikbare modellen zijn opgezet en gevalideerd voor mini-kanalen (diameters typisch 1 mm of meer). Voorspellende modellen bestonden al wel maar alleen voor koeling door

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verdamping bij lage druk, temperaturen boven $-10\text{ }^{\circ}\text{C}$, en in kanalen met diameters boven 0.5 mm . Doel van het onderzoek is toepassing van de modellen te verkennen bij kleinere diameters en lagere temperaturen ($-35\text{ }^{\circ}\text{C}$ tot $-20\text{ }^{\circ}\text{C}$). Hierbij is de experimentele validatie van de modellen het belangrijkste doel van dit onderzoek. Testmetingen zijn uitgevoerd aan een recht microkanaal om de empirische modellen voor twee-fasen flow, zoals gepresenteerd in hoofdstuk 2, te valideren. Voorafgaand is een-fase flow bestudeerd, waarbij de gemeten drukval en warmte-overdrachtscoëfficiënten binnen de meetnauwkeurigheden overeenstemden met de theoretische voorspellingen.

Ook van de twee-fasen flow zijn drukval en warmte-overdrachtscoëfficiënten gemeten, maar bovendien werd het flow regime zichtbaar gemaakt middels een hoge-snelheids camera, direct na de testsectie. De waargenomen flow regimes zijn in goede overeenstemming met het Ong model dat in ongeveer 86% van de totale data het juiste regime voorspelt. Ook de twee-fasen drukval wordt goed voorspeld door het betrokken model afhankelijk van het flow regime. De warmte-overdrachtscoëfficiënten in twee-fasen flow werd bepaald op basis van de gemeten wandtemperaturen.

Stroomafwaarts langs het kanaal nam de warmte-overdrachtscoëfficiënten toe tot die na het bereiken van een maximum waarde instortte, hetgeen wijst op plaatselijk “dry-out”. In de warmte-overdrachtscoëfficiënten is geen duidelijke relatie gezien met de massastroom en de warmtestroom in het EB-regime (Elongated Bubbles), terwijl voor het “annular flow”- regime een trend zichtbaar was duidend op een lineaire afhankelijkheid van de warmtestroom. De gemeten warmte-overdrachtscoëfficiënten zijn vergeleken met de waarden van voorspellende modellen afhankelijk van de flow regimes. In zijn algemeenheid leveren de modellen een te lage waarde op, oplopend tot een orde van grootte. Aansluitend zijn de modellen nader onderzocht op het effect van bepalende empirische parameters, die aangepast werden afhankelijk van de experimentele set-up. Daartoe werden de relevante parameters aangepast met eenzelfde factor. Het grootste effect in het EB-regime bleek de c_{do} parameter te hebben (een correctiefactor voor de initiële vloeistof-filmdikte). In de analyse is aangetoond dat de voorspellende waarde van het model verbeterd kan worden door de relevante empirische parameters kritisch af te regelen. De modellen die beschouwd zijn blijken geschikt voor kwalitatieve trend studies. Wanneer echter kwantitatieve warmte-overdrachtanalyses vereist zijn, dan moeten de betrokken empirische parameters goed bekend zijn en bepaald worden in een afzonderlijk serie test-experimenten. Dit geldt in het bijzonder voor het EB-regime waarin de warmte-overdracht zeer gevoelig blijkt voor deze empirische parameters.

Tenslotte, omdat in HEF experimenten meerdere micro-kanalen gebruikt zullen worden om effectief deeltjesdetectoren met hoge-vermogens dichtheid te kunnen koelen, is een silicium-glas multichannel warmtewisselaar onderzocht. De zeer kleine en nauwe capillairen maakten het echter moeilijk om deze warmtewisselaar onder dezelfde omstandigheden te bestuderen als het enkele microkanaal. Bovendien was de flow in de micro-warmtewisselaar veel te laag

voor het beschikbare model [30] dat gevalideerd is voor massastromen boven $>200 \text{ kg/m}^2\text{s}$. Niettemin konden een aantal interessante conclusies getrokken worden. In het algemeen is het flow regime EB of annular flow. In sommige gevallen traden flow-instabiliteiten op waarin heen-en-weer gewisseld werd tussen twee regimes terwijl de testparameters (massastroom en warmtestroom) constant gehouden werden. De gemeten drukval neemt toe met toenemende massastroom, zoals te verwachten is door de significante bijdrage van de twee-fasen viskeuze drukval in de capillairen. De warmte-overdrachtscoëfficiënten bleek lineair afhankelijk van de warmtestroom. De warmte-overdracht waarden afgeleid uit de metingen aan de multichannel warmtewisselaar toonden eenzelfde gedrag als in het enkele rechte micro-kanaal. Om sluitende conclusies te kunnen trekken omtrent het gedrag in multichannel warmtewisselaars en in micro-kanalen geëst in silicium, zijn vervolgmetingen nodig bij hogere massastromen.

Concluderend kan gesteld worden dat CO_2 koeling door verdamping een veelbelovende methode blijkt te zijn voor de koeling van detectoren in Hoge Energie Fysica. In dit proefschrift hebben we gedemonstreerd dat koeling door verdamping de petal detector in het ATLAS experiment succesvol koud kan houden voor een periode van 10 jaar. Verder hebben we laten zien dat flow regime en drukval goed voorspeld kunnen worden. De warmte-overdrachtscoëfficiënten bleek evenwel matig te voorspellen met de modellen beschreven in hoofdstuk (2). Tenslotte, omdat het niet mogelijk bleek de modellen te valideren met metingen aan de multichannel micro-warmtewisselaar, kunnen voor multichannel micro-kanalen alleen kwalitatieve conclusies getrokken worden.

Περίληψη

Το πρώτο μέρος της παρούσας διδακτορικής διατριβής έχει ως στόχο τον σχεδιασμό και την μοντελοποίηση ενός επικυρωμένου συστήματος ψύξης με χρήση διφασικού διοξειδίου του άνθρακα για τον ανιχνευτή μικροσωματιδίων του πειράματος *ATLAS*. Το εν λόγω πείραμα για να πετύχει τους ερευνητικούς στόχους του απαιτεί την αναβάθμιση τόσο του λογισμικού όσο και των μηχανημάτων γύρω από αυτό, ενώ παράλληλα θέτει αυστηρές προδιαγραφές όσο αφορά τα συστήματα ψύξης και τις μηχανολογικές εγκαταστάσεις. Το δεύτερο μέρος της παρούσας διδακτορικής διατριβής έχει ως στόχο την πειραματική επικύρωση μοντέλων πρόβλεψης για τον χαρακτηρισμό της μεταφοράς της θερμότητας και της ροής διφασικού διοξειδίου του άνθρακα σε κανάλια (μικρο-σωλήνες) διαστάσεων μερικών μικρομέτρων. Ψύξη με μικρο-σωλήνες έχουν παρατηρηθεί ότι είναι ιδιαίτερα αποτελεσματική σε ανιχνευτές πυριτίου σε ένα περιβάλλον με υψηλά επίπεδα ραδιενέργειας.

Ο ανιχνευτής μικροσωματιδίων *petal*

Με την έναρξη της παρούσας διδακτορικής διατριβής, είχε ήδη παρθεί η απόφαση της υιοθέτησης διφασικού διοξειδίου του άνθρακα για την ψύξη του Εσωτερικού Ιχνηλάτη (*Inner Tracker - ITk*). Επόμενο μέλημα ήταν η βελτιστοποίηση του συστήματος ψύξης και η πειραματική απόδειξη ότι το σύστημα ψύξης θα μπορέσει να καλύψει επαρκώς τις απαιτήσεις του ανιχνευτή. Μια θερμικά πρωτότυπη ελαφριά κατασκευή από ίνες άνθρακα με ενσωματωμένη ψύξη ήταν διαθέσιμη ως δομή για τον ανιχνευτή πυριτίου καθώς επίσης και μια άρτια και ολοκληρωμένη εγκατάσταση ψύξης με διοξείδιο του άνθρακα στο *Nikhef*. Για την πειραματική μελέτη της πρωτότυπης κατασκευής χρειάστηκε να σχεδιαστούν και να τοποθετηθούν ειδικά θερμικά στοιχεία και μέρη πυριτίου ώστε να προσομοιωθούν επαρκώς οι πηγές θερμότητας και ο αισθητήρας πυριτίου. Έπειτα, κατασκευάστηκε ένα πειραματικά επικυρωμένο θερμικό μοντέλο ενώ η προσομοίωση της πρωτότυπης κατασκευής πραγματοποιήθηκε με χρήση πεπερασμένων στοιχείων (*Finite Element Analysis*). Η χρήση του επικυρωμένου θερμικού μοντέλου συνετέλεσε στην αξιόπιστη προσομοίωση της δομής διαφορετικών σχεδίων ανιχνευτή με απώτερο στόχο ένα βέλτιστο σχεδιασμό. Παράμετροι που λήφθηκαν υπόψη στην διαδικασία της βελτιστοποίησης είναι το μήκος της ακτινοβολίας (*radiation length*), η ελαχιστοποίηση συσσωρευμένων σημείων θερμότητας, και η μεγιστοποίηση του θερμικού ορίου ασφαλείας, όπως η απόσταση από την θερμική

διαφυγή.

Η προσομοίωση συμπεριέλαβε δυο κρίσιμες και συνάμα αναμενόμενες στιγμές κατά την διάρκεια ζωής του ανιχνευτή και των ολοκληρωμένων κυκλωμάτων (ASIC) , όπως την στιγμή της μέγιστης ολικής ιονίζουσας δόσης και κατά το πέρας της λειτουργίας του ανιχνευτή. Συνοπτικά, η μελέτη ανέδειξε κρίσιμα ζητήματα που χρίζουν την επίλυση τους για την απρόσκοπτη λειτουργία του ανιχνευτή για δέκα συνεχή έτη. Τα κρίσιμα ζητήματα αναφέρονται επιγραμματικά ακολούθως. Πρώτο ζήτημα είναι η παροχή αυξημένου φορτίου ρεύματος σε έναν μετατροπέα συνεχούς/ συνεχούς ρεύματος, το οποίο έχει ως αποτέλεσμα την σημαντική μείωση της αποδοτικότητας του μετατροπέα και ακολούθως την μετατροπή σε θερμότητας τοπικά. Η χρήση δυο μετατροπέων συνεχούς ρεύματος συνιστάται καθώς η θερμοκρασία τοπικά μπορεί να ελλατωθεί μέχρι και 10 βαθμούς Κελσίου. Δεύτερον, η μελέτη της ροής και της μεταφοράς θερμότητας του διοξειδίου του άνθρακα αποκάλυψε ότι στον τρέχοντα σχεδιασμό το αναμενόμενη μορφή της ροής δεν είναι βέλτιστη. Η βέλτιστη μορφή ροής προϋποθέτει υψηλή μεταφορά θερμότητας μεταξύ του ψυκτικού και του τοιχώματος του σωλήνα. Δι-αφορετικές εσωτερικές διάμετροι εξετάστηκαν όσον αφορά τη μεταφορά θερμότητας και τη θερμική διαφυγή για να βρεθεί η βέλτιστη μορφή ροής. Αποδείχθηκε ότι μια μείωση στην εσωτερική διάμετρο μπορεί να αποδώσει παρόμοια αποτελέσματα με μια μεγαλύτερη. Μια μικρότερη διάμετρος ωστόσο προτιμάται ακόμη και όταν η θερμική απόδοση είναι παρόμοια λόγω του καλύτερου μήκους ακτινοβολίας. Επιπλέον, σύμφωνα με τις μελέτες στην αναφορά [61] μια μικρότερη διάμετρος δεν θα επιδεινώσει τη μηχανική απόδοση του petal , είναι πιο εύκολο να κατασκευαστεί και δημιουργεί λιγότερη τάση κατά την ψύξη. Για τους λόγους αυτούς, συνιστάται μείωση της εσωτερικής διαμέτρου έως 1,3 mm.

Επιπλέον, το υλικό K13C2U 45 gsm προτιμάται από το K13D2U 80 gsm. Παρά τις ελαφρώς θερμότερες μετρήσεις πάνω στο πυρίτιο, προτιμάται επειδή συμβάλλει λιγότερο στον προϋπολογισμό υλικών και η μηχανική απόδοση του petal παραμένει επαρκής [61]. Άλλα βήματα βελτιστοποίησης όπως αλλαγή διαμόρφωσης σωλήνα, αλλαγή υβριδικής κόλλας ASIC και προσθήκη αφρού άνθρακα σε θερμικά κρίσιμες θέσεις μπορούν να βελτιώσουν τη θερμική απόδοση, αλλά εξαρτώνται από τη θέση όλων των εξαρτημάτων διασποράς ισχύος, η οποία είναι ακόμα ασαφής. Τέλος, προτείνεται η χρήση του μοντέλου *CoBra* με μη ομοιόμορφη είσοδο ροής θερμότητας, καθώς μπορεί να βελτιώσει την υπολογιστική ακρίβεια (κατά 1 έως 2 βαθμούς Κελσίου).

Ο ανιχνευτής ATLAS θα εγκατασταθεί το 2022. Συνολικά, ο σχεδιασμός των πετάλων λαμβάνοντας υπόψη τις παραπάνω προτάσεις θα λειτουργήσει αξιόπιστα με ασφάλεια κατά τη διάρκεια των δέκα ετών λειτουργίας. Συνιστάται ιδιαίτερα η ψύξη με εξάτμιση διοξειδίου του άνθρακα για αυτό το σχέδιο.

Ψύξη με διφασικό διοξείδιο του άνθρακα σε μικρο-κανάλια

Ένας δεύτερος στόχος της διατριβής είναι η αξιολόγηση της απόδοσης της ψύξης με διοξείδιο του άνθρακα σε μικρο-κανάλια. Η ψύξη σε μικρο-κανάλια είναι ενδείκνυται για ανιχνευτές υψηλής πυκνότητας ισχύος. Τα μοντέλα ψύξης μίνι καναλιών (διαμέτρου μεγαλύτερα του 1 mm) χρειάζονται προσαρμογή για αυτόν τον στόχο. Μοντέλα πρόβλεψης είναι διαθέσιμα στην βιβλιογραφία, αλλά αφορούν μόνο για ψύξη με εξαέρωση ψυκτικού σε χαμηλή πίεση και θερμοκρασίες υψηλότερες των -10 βαθμούς Κελσίου και σε κανάλια μεγαλύτερης διαμέτρου 0,5 mm. Ο στόχος αυτής της έρευνας είναι να επεκταθεί το εύρος εφαρμογής των μοντέλων πρόβλεψης σε διαμέτρους κάτω από 0,3 mm και σε θερμοκρασίες -20 βαθμούς Κελσίου. Η πειραματική επικύρωση των μοντέλων αποτελεί βασικό στόχο αυτής της έρευνας.

Κατά την διάρκεια της διδακτορικής διατριβής πραγματοποιήθηκαν δοκιμές σε ένα οριζόντιο μικροκανάλι για την επικύρωση των εμπειρικών μοντέλων διφασικής ροής όπως παρουσιάζονται στο κεφάλαιο 2. Πραγματοποιήθηκε αρχικά επικύρωση της εγκατάστασης σε μονοφασική ροή πριν από τις δοκιμές διφασικών ροών. Οι πτώσεις πίεσης και οι συντελεστές μεταφοράς θερμότητας σε μονοφασική ροή βάσει μετρήσεων αποδεικνύεται ότι ταιριάζουν με θεωρητικές προβλέψεις λαμβάνοντας υπόψη τα σφάλματα μέτρησης.

Οι παράμετροι της διφασικής ροής που προέκυψαν από τις μετρήσεις είναι η πτώση πίεσης και ο συντελεστής μεταφοράς θερμότητας, ενώ η μορφή της ροής απεικονίζεται άμεσα με κάμερα υψηλής ταχύτητας, αμέσως μετά το τμήμα δοκιμής. Οι καταγεγραμμένες μορφές ροής δύο φάσεων συμφωνούν καλά με το μοντέλο *Onq* που προβλέπει σωστά την μορφή ροής σε περίπου 86% του συνόλου των δεδομένων. Η πτώση πίεσης σε μικρο-σωλήνα δύο φάσεων προβλέπεται εξίσου καλά με το σχετικό μοντέλο ανάλογα με την μορφή της ροής. Ο συντελεστής μεταφοράς θερμότητας κατά τη διάρκεια της ροής δύο φάσεων υπολογίστηκε από τις μετρούμενες θερμοκρασίες τοιχώματος. Κατά μήκος του σωλήνα, ο συντελεστής μεταφοράς θερμότητας αυξάνεται έως ότου πέσει μετά από ένα μέγιστο υποδεικνύοντας την παρουσία τοπικής ξήρανσης. Δεν υπήρχε σαφής σχέση μεταξύ της ροής μάζας και της ροής θερμότητας για την μορφή ροής με επιμήκεις φουσαλίδες, ενώ για την ροή μορφής με δακτυλιοειδής γεωμετρία παρατηρήθηκε μια γραμμική εξάρτηση του συντελεστή μεταφοράς θερμότητας από τη ροή θερμότητας.

Ο συντελεστής μεταφοράς θερμότητας συσχετίστηκε με το σχετικό μοντέλο ανάλογα με την μορφή της ροής. Ως γενική παρατήρηση, ο συντελεστής μεταφοράς θερμότητας υπο-προβλεπόταν από τα μοντέλα, οι διαφορές κυμαινόνταν έως μία τάξη μεγέθους. Τα μοντέλα στη συνέχεια εξετάστηκαν περαιτέρω αναφορικά με τις εμπειρικούς τους παραμέτρους. Για να οριστεί η πιο κρίσιμη παράμετρο, όλες οι παράμετροι τροποποιήθηκαν με τον ίδιο παράγοντα. Το c_{do} (που είναι ένας συντελεστής διόρθωσης για το αρχικό πάχος της υγρής μεμβράνης σε επαφή με τον σωλήνα) για την μορφή ροής με επιμήκεις φουσαλίδες αποδεικνύεται ότι

ήταν εκείνο που είχε τη μεγαλύτερη επίδραση στα αποτελέσματα. Στην ανάλυση, αποδεικνύεται ότι η πρόβλεψη μεταφοράς θερμότητας του μοντέλου μπορεί να βελτιωθεί με λεπτό συντονισμό των σχετικών εμπειρικών παραμέτρων.

Γενικά, τα μοντέλα που εξετάζονται εδώ ως προς τον συντελεστή μεταφοράς θερμότητας είναι χρήσιμα για ποιοτικές μελέτες. Ωστόσο, εάν απαιτούνται ποσοτικές μελέτες μεταφοράς θερμότητας, τότε οι πειραματικές παράμετροι πρέπει να είναι καλά γνωστές και να προσδιορίζονται σε ξεχωριστό σύνολο πειραμάτων δοκιμής. Αυτό ισχύει ιδιαίτερα για το μοντέλο ροής με επιμήκεις φυσαλίδες που δείχνει υψηλή ευαισθησία σε αυτές τις εμπειρικές παραμέτρους.

Τέλος, επειδή στα πειράματα φυσικής υψηλής ενέργειας προτιμώνται πολλαπλά μικροκάναλα για την αποτελεσματική ψύξη ανιχνευτών σωματιδίων υψηλής πυκνότητας, μελετήθηκε ένας εναλλάκτης θερμότητας πολλαπλών καναλιών από πυρίτιο και γυαλί. Τα πολύ μικρά και στενά τριχοειδή αγγεία, ωστόσο, δυσκόλεψαν τη μελέτη του εναλλάκτη θερμότητας υπό παρόμοιες συνθήκες όπως με την μελέτη του μικρο-κανάλιου στο κεφάλαιο 7. Επιπλέον, το μοντέλο [30] για την πρόβλεψη των μορφών ροής δεν μπορούν να χρησιμοποιηθούν σε χαμηλές ροές μάζας καθώς το μοντέλο ισχύει για ροές μάζας μεγαλύτερες από $200 \text{ kg/m}^2\text{s}$. Έτσι, μια σύγκριση θα ήταν ανακριβής.

Ωστόσο, μπορούν να εξαχθούν ορισμένα ενδιαφέροντα συμπεράσματα. Γενικά, τα πρότυπα ροής που παρατηρήθηκαν ήταν οι επιμήκεις φυσαλίδες και η δακτυλιοειδής ροή. Μερικές φορές παρατηρήθηκαν αστάθειες, καθώς η ροή άλλαζε από το ένα καθεστώς στο άλλο, ενώ οι πειραματικοί παράμετροι (ροή μάζας και ροή θερμότητας) ήταν σταθερές. Η πτώση πίεσης στον εναλλάκτη θερμότητας όπως προκύπτει από τις μετρήσεις αυξάνεται με την αυξανόμενη ροή μάζας όπως αναμενόταν λόγω της σημαντικής συμβολής από την πτώση πίεσης στα τριχοειδή αγγεία. Ο συντελεστής μεταφοράς θερμότητας παρουσίασε γραμμική σχέση με τη ροή θερμότητας. Οι τιμές μεταφοράς θερμότητας που προέρχονται από μετρήσεις για τα παράλληλα μικρο-κάναλα στον εναλλάκτη θερμότητας παρουσίασαν παρόμοια συμπεριφορά όπως στο μεμονωμένο μικρο-κάναλι όπως παρουσιάζεται στο κεφάλαιο 8. Απαιτούνται περαιτέρω μελέτες σε ροές μεγαλύτερης μάζας σχετικά με τη συμπεριφορά σε πολλαπλά μικρο-κάναλα και σε μικρο-κάναλα χαραγμένα σε πυρίτιο.

Εν κατακλείδι, η ψύξη με διφασικό διοξείδιο του άνθρακα αποδείχθηκε ότι είναι μια πολλά υποσχόμενη μέθοδος για τους ανιχνευτές ψύξης στην φυσική υψηλών ενεργειών. Σε αυτή τη διατριβή, αποδείχθηκε ότι η ψύξη με εξάτμιση διατηρεί επιτυχώς τον ανιχνευτή *petal* στο πείραμα ATLAS κρύο και ασφαλές από την θερμική φυγή για τα 10 χρόνια λειτουργίας. Η ψύξη με διφασικό διοξείδιο του άνθρακα σε μικρο-κάναλα είναι επίσης ελπιδοφόρα για ανιχνευτές υψηλής πυκνότητας ισχύος. Σε αυτή τη διατριβή, αποδείχθηκε ότι τα μοντέλα προβλέπουν σωστά την μορφή ροής και τις πτώσεις πίεσης μέχρι τώρα. Η πρόβλεψη για την εκτίμηση του συντελεστή μεταφοράς θερμότητας χρησιμοποιώντας τα μοντέλα πρόβλεψης που περιγράφονται στο κεφάλαιο 2 δεν ήταν ικανοποιητική. Και τέλος, επειδή δεν ήταν δυνατό να επικυρωθούν τα μοντέλα πρόβλεψης με τις μετρήσεις στα πολλαπλά μικρο-κάναλα, μπορούσαν να εξαχθούν μόνο ποιοτικά συμπεράσματα.

Acknowledgements

Constantine Cavafy in his poem ‘Ithaka’ narrates :

‘ As you set out for Ithaka
hope your road is a long one
full of adventure, full of discovery
Laistrygonians, Cyclops, angry Poseidon—don’t be afraid of them:
you’ll never find things like that on your way
as long as you keep your thoughts raised high,
as long as a rare excitement
stirs your spirit and your body...’

This PhD thesis has been a journey to my own Ithaka. It has been a journey full of excitement, dreams, challenges, failures, and new experiences. It definitely has been a journey where I learnt about myself and explored my limits. It took me some time to complete it, but I am very happy I went through the end.

I do surely acknowledge that I would have not reached my journey if it was not for all those people around me that helped me and supported me throughout these years, with patience and belief.

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The next person I would like to greatly thank is my promoter Marcel ter Brake. Marcel, I remember, a year after starting my PhD, I came to your desk with Bob van Eijk, where I introduced myself and the research proposal I had in mind for the second part of my PhD. Soon after, you accepted me as your PhD student and became part of your group, even though my study topic was not about cryogenic temperatures. That meant a lot for me, and I immediately I felt we can talk the ‘same language’. From the very first moment, I admired your engineering but always scientific interest and views you had. Till the end, you were a great supporter and guidance for me, despite the distance most of

the times. I am certain, the EMS group will keep growing even further in numbers (both quality and quantity), and I wish you to send more advanced EMS coolers in space!

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I would like to thank my family. Thank you for being always next to me and

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