IN FACULTY OF ENGINEERING

Assessment of Nucleate Pool Boiling Heat Transfer and Critical Heat Flux for Power Electronics Cooling with a Low-GWP Refrigerant

Ilya T'Jollyn

Doctoral dissertation submitted to obtain the academic degree of Doctor of Electromechanical Engineering

Supervisor

Prof. Michel De Paepe, PhD

Department of Electromechanical, Systems and Metal Engineering Faculty of Engineering and Architecture, Ghent University

December 2021





FACULTY OF ENGINEERING

Assessment of Nucleate Pool Boiling Heat Transfer and Critical Heat Flux for Power Electronics Cooling with a Low-GWP Refrigerant

Ilya T'Jollyn

Doctoral dissertation submitted to obtain the academic degree of Doctor of Electromechanical Engineering

Supervisor Prof. Michel De Paepe, PhD

Department of Electromechanical, Systems and Metal Engineering Faculty of Engineering and Architecture, Ghent University

December 2021



ISBN 978-94-6355-551-7 NUR 959, 926 Wettelijk depot: D/2021/10.500/99

Members of the Examination Board

Chair

Prof. Patrick De Baets, PhD, Ghent University

Other members entitled to vote

Prof. Jocelyn Bonjour, PhD, Institut National des Sciences Appliquées de Lyon, France Prof. Joris Degroote, PhD, Ghent University Prof. Toni Pujol Sagaró, PhD, Universitat de Girona, Spain Prof. Hendrik Vansompel, PhD, Ghent University

Supervisor

Prof. Michel De Paepe, PhD, Ghent University

Dankwoord

Het werken aan dit doctoraat en schrijven van deze thesis was een eigenaardige ervaring. Hoewel ik ontzettend blij en zeer trots ben om dit werk af te leveren, is er geen haar op mijn hoofd dat eraan denkt om nog eens een doctoraatsthesis te schrijven. Dit werk had niet tot stand kunnen komen zonder de hulp van meerdere personen, die ik met dit schrijven graag wil bedanken.

Michel, bedankt om mij de mogelijkheid te geven om de weg tot dit doctoraat af te leggen op mijn eigenwijze manier, met enkele inbegrepen pitstops. Jouw inzichten en begeleiding hebben me tot hier gebracht.

Jasper, bedankt om een geweldige thesisstudent te zijn en me zo een mooi startpunt te geven voor dit doctoraat. Ook later was je hulp, advies en inzicht in dit onderwerp onbetaalbaar. Wim, dank om alles met een kritische blik na te lezen. Je snelle doorzicht heeft me meermaals geholpen tijdens dit doctoraat. Zaaq, bedankt om altijd klaar te staan voor een babbel, over technische of andere zaken.

Annie, Annick en Griet, dank voor alle administratieve ondersteuning. Patrick, Thomas en Bart, bedankt voor alle technische ondersteuning en hulp in het labo. Frederik, jij krijgt een aparte vermelding: dank voor het helpen ontwerpen en bouwen van de setup. Zonder jouw begrip van tweefasige systemen had ik hem nooit aan de praat gekregen.

Mam en Pap, bedankt voor het nalezen en de wijze raad, zowel op wetenschappelijke als andere vlakken. Jullie hebben al sinds ik jong was mijn wetenschappelijke nieuwsgierigheid aangewakkerd, waardoor ik nu dit doctoraat heb kunnen afwerken. Zus, bedankt om altijd een goede zus en vriendin voor me te zijn.

Bedankt aan Luke en Leia, jullie waren de (bijna) ideale schrijfpartners.

Tot slot wil ik graag Jolien bedanken. Jouw steun en aanmoediging tijdens dit doctoraat en alle andere zaken die ik doe, zijn onbeschrijfbaar belangrijk voor mij. Je bent de liefde van mijn leven.

> Antwerpen, augustus 2021 Ilya T'Jollyn

Table of Contents

Da	ankwo	oord		i
Li	st of l	Figures		vii
Li	st of '	Fables		xiii
No	omen	clature		XV
Ne	ederla	indse sa	menvatting	xix
Er	ıglish	summa	ıry	xxiii
1	Intr	oductio	n	1
	1.1	Electri	fication and energy efficiency	1
	1.2	Boiling	g heat transfer	2
	1.3	Object	ives of the study	3
	1.4	Outline	e	3
2	The	rmal ma	anagement of electric drivetrains	5
	2.1	Introdu	uction	5
	2.2	Electri	c drivetrains	5
	2.3	Therm	al management strategies	7
		2.3.1	Air cooling	7
		2.3.2	Water-glycol cooling	9
		2.3.3	Oil cooling	9
		2.3.4	Two-phase cooling	9
	2.4	Power	electronics cooling	10
		2.4.1	Inverter	10
		2.4.2	Air cooling	11
		2.4.3	Water-glycol cooling	12
		2.4.4	Oil cooling	13
		2.4.5	Two-phase cooling	14
		2.4.6	Comparison	16
	2.5	Coolar	nt selection	16
		2.5.1	Requirements	16
		2.5.2	Types of fluids	19

		2.5.3	Comparison
	2.6	Conclus	sions
•			
3	Nuc	leate poo	l boiling heat transfer 23
	3.1	Introduc E	2001
	3.2	Fundam	Dettingen and the second
		3.2.1	Bolling curve
		3.2.2	Heat transfer mechanisms
		3.2.3	Non-dimensional groups
	2.2	3.2.4	Bubble nucleation, growth and departure diameter 38
	3.3	Heat tra	nster correlations $\dots \dots \dots$
		3.3.1	Kruzhilin correlation (1947) $\dots \dots \dots$
		3.3.2	Rohsenow correlation (1952) $\ldots \ldots \ldots \ldots \ldots 43$
		3.3.3	Forster-Zuber correlation (1955)
		3.3.4	Kutateladze-Borishanskii correlation (1958) 47
		3.3.5	Borishanskii-Mostinski correlation (1963) 47
		3.3.6	Shekriladze-Ratiani correlation (1966)
		3.3.7	Labuntsov correlation (1972)
		3.3.8	Gorenflo correlation (1973)
		3.3.9	Stephan-Abdelsalam correlation (1978)
		3.3.10	Cooper correlation (1984)
		3.3.11	Kutateladze correlation (1990)
		3.3.12	Leiner correlation (1994)
		3.3.13	Pioro correlation (2004)
		3.3.14	Summary
	3.4	Paramet	ter sensitivity
		3.4.1	Surface heat flux
		3.4.2	Fluid properties
		3.4.3	Geometric parameters
		3.4.4	Surface material properties
		3.4.5	Other factors
		3.4.6	Conclusions
	3.5	Critical	heat flux
		3.5.1	Mechanisms
		3.5.2	Parameter sensitivity
		3.5.3	CHF correlations
	3.6	Nucleat	e pool boiling heat transfer with FK-649
	3.7	Researc	h questions \ldots \ldots 22
			1
4	Boil	ing and 1	nucleation measurement device 95
	4.1	Introduc	xtion
	4.2	Reservo	vir
	4.3	Refrige	rant
	4.4	Coolant	circuit
	4.5	Sensors	and data acquisition

		4.5.1	Sensors	105
		4.5.2	Data acquisition	105
		4.5.3	Summary	107
	4.6	Contro	ls	107
	4.7	Measu	rement procedure	108
	4.8	Measu	rement analysis	111
		4.8.1	Boiling curve	111
		4.8.2	Energy balance	114
		4.8.3	Uncertainty analysis	115
5	Hor	izontal r	plate nool boiling	110
5	5 1	Introdu	nate poor bonning	110
	5.1	Hantar		120
	5.2	Satur		120
	5.5	Setup C		124
	5.4	Deput		12/
	5.5	Results	9	129
		5.5.1		129
		5.5.2		132
		5.5.5		134
		5.5.4		130
		5.5.5		136
	5.6	Compa	rison with previous study	138
	5.7	Conclu	S10ns	139
				157
6	Pow	er modu	ıle baseplate pool boiling	141
6	Pow 6.1	er modu Introdu	Ile baseplate pool boiling	141 141
6	Pow 6.1 6.2	er modu Introdu Inverter	Ile baseplate pool boiling Iction r module	141 141 141
6	Pow 6.1 6.2 6.3	er modu Introdu Inverte Selectio	Ile baseplate pool boiling Iction r module on of experimental parameter range	141 141 141 141 149
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverte Selectio Results	Ile baseplate pool boiling Iction r module on of experimental parameter range	141 141 141 149 150
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverte Selectio Results 6.4.1	ule baseplate pool boiling inction r module on of experimental parameter range s. Surface temperature	141 141 141 149 150 150
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverter Selectio Results 6.4.1 6.4.2	ule baseplate pool boiling iction r module on of experimental parameter range s Surface temperature Module temperature	141 141 141 149 150 150 152
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverter Selectio Results 6.4.1 6.4.2 6.4.3	ale baseplate pool boiling iction r module on of experimental parameter range s Surface temperature Module temperature Energy balance	141 141 141 149 150 150 152 155
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverter Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4	ale baseplate pool boiling iction r module on of experimental parameter range s Surface temperature Module temperature Energy balance Boiling curve	141 141 141 149 150 150 152 155 156
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverte: Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4	ale baseplate pool boiling iction r module on of experimental parameter range s Surface temperature Module temperature Energy balance Boiling curve 6.4.4.1	141 141 141 149 150 150 152 155 156 158
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverter Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4	ale baseplate pool boiling iction r module on of experimental parameter range s Surface temperature Module temperature Energy balance Boiling curve 6.4.4.1 Effect of liquid height 6.4.4.2 Effect of saturation temperature	141 141 141 149 150 150 152 155 156 158 159
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverter Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4	ale baseplate pool boiling action r module on of experimental parameter range s Surface temperature Module temperature Energy balance 6.4.4.1 Effect of liquid height 6.4.4.2 Effect of saturation temperature 6.4.4.3 Effect of fouling	141 141 141 149 150 150 152 155 156 158 159 160
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Invertes Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4	ale baseplate pool boiling action r module on of experimental parameter range Surface temperature Surface temperature Module temperature Energy balance Boiling curve 6.4.4.1 Effect of liquid height 6.4.4.2 Effect of fouling 6.4.4.3 Effect of fouling	141 141 141 150 150 152 155 156 158 159 160 161
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Invertes Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4	ale baseplate pool boiling iction r module on of experimental parameter range Surface temperature Surface temperature Module temperature Energy balance Boiling curve 6.4.4.1 Effect of liquid height 6.4.4.3 Effect of fouling 6.4.4.4 Nucleate boiling regimes Critical heat flux	141 141 141 149 150 150 152 155 156 158 159 160 161 165
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverter Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4	ale baseplate pool boiling r module on of experimental parameter range Surface temperature Module temperature Boiling curve 6.4.4.1 Effect of liquid height 6.4.4.3 Effect of fouling 6.4.4.4 Nucleate boiling regimes Critical heat flux 6.4.5.1 Effect of saturation temperature	141 141 141 149 150 150 155 155 155 155 158 159 160 161 165 165
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverte: Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4	ule baseplate pool boiling iction r module on of experimental parameter range Surface temperature Surface temperature Module temperature Energy balance 6.4.4.1 Effect of liquid height 6.4.4.2 Effect of saturation temperature 6.4.4.3 Effect of fouling 6.4.4.4 Nucleate boiling regimes Critical heat flux 6.4.5.1 Effect of saturation temperature 6.4.5.2 Effect of liquid height	141 141 141 149 150 150 152 155 156 158 159 160 161 165 165
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Inverte: Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4	ule baseplate pool boiling inction r module on of experimental parameter range Surface temperature Module temperature Energy balance Boiling curve 6.4.4.1 Effect of liquid height 6.4.4.2 Effect of fouling 6.4.4.3 Effect of fouling regimes Critical heat flux 6.4.5.1 Effect of liquid height 6.4.5.2 Effect of liquid height	141 141 141 149 150 150 152 155 156 158 159 160 161 165 165 165
6	Pow 6.1 6.2 6.3 6.4	er modu Introdu Invertes Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4 6.4.5	Ale baseplate pool boiling iction r module on of experimental parameter range Surface temperature Surface temperature Module temperature Boiling curve 6.4.4.1 Effect of liquid height 6.4.4.2 Effect of saturation temperature 6.4.4.3 Effect of fouling 6.4.4.4 Nucleate boiling regimes Critical heat flux 6.4.5.1 Effect of liquid height 6.4.5.3 Effect of fouling 6.4.5.3 Effect of fouling	141 141 141 149 150 150 152 155 156 158 159 160 161 165 165 165
6	Pow 6.1 6.2 6.3 6.4 6.5 6.5	er modu Introdu Invertes Selectio Results 6.4.1 6.4.2 6.4.3 6.4.4 6.4.5	Ale baseplate pool boiling iction r module on of experimental parameter range Surface temperature Module temperature Module temperature Boiling curve 6.4.4.1 Effect of liquid height 6.4.4.2 Effect of saturation temperature 6.4.4.3 Effect of fouling 6.4.4.4 Nucleate boiling regimes Critical heat flux 6.4.5.1 Effect of liquid height 6.4.5.2 Effect of fouling critical heat flux 6.4.5.3 Effect of fouling critical heat flux float float	141 141 141 149 150 150 152 155 156 158 159 160 161 165 165 165 166 168

7.1 Introduction .	
7.2 Methodology	17
	17
7.2.1 Correlations	17
7.2.2 Inputs	17
7.2.3 Performance parameters	17
7.3 Nucleate boiling heat transfer	17
7.3.1 Predictive performance	17
7.3.2 Effect of heat flux	17
7.3.3 Effect of saturation temperature	
7.3.4 Effect of boiling surface	
7.4 Critical heat flux	
7.4.1 Predictive performance	
7.4.2 Effect of saturation temperature	
7.5 Assessment of other working fluids	
7.6 Conclusions	
8 Conclusions	18
8 Conclusions References	18 19
8 Conclusions References A Publications	18 19 21
8 Conclusions References A Publications B Measurement uncertainty analysis	18 19 21 21
 8 Conclusions 8 References A Publications B Measurement uncertainty analysis B.1 Boiling curve	18 19 21 21
 8 Conclusions 8 References A Publications B Measurement uncertainty analysis B.1 Boiling curve B.2 Energy balance 	18 19 21 21
 8 Conclusions 8 Conclusions 8 References A Publications B Measurement uncertainty analysis B.1 Boiling curve B.2 Energy balance C Future setup design recommendations 	18 19 21 21 21 21 21 21 21
 8 Conclusions 8 Conclusions References A Publications B Measurement uncertainty analysis B.1 Boiling curve B.2 Energy balance C Future setup design recommendations D Datasheets 	18 19 21 21 21 21 21 21 21 21 22
 8 Conclusions 8 Conclusions References A Publications B Measurement uncertainty analysis B.1 Boiling curve B.2 Energy balance C Future setup design recommendations D Datasheets D.1 3M Novec 649 (FK-649) 	18 19 21 21 21 21 21 21 21 21 22 22
 8 Conclusions 8 Conclusions References A Publications B Measurement uncertainty analysis B.1 Boiling curve B.2 Energy balance B.2 Energy balance C Future setup design recommendations D Datasheets D.1 3M Novec 649 (FK-649) D.2 NESLAB Merlin Series Recirculating Chillers 	18 19 21 21 21 21 21 21 21 22 22 22 22
 8 Conclusions References A Publications B Measurement uncertainty analysis B.1 Boiling curve B.2 Energy balance B.2 Energy balance C Future setup design recommendations D Datasheets D.1 3M Novec 649 (FK-649) D.2 NESLAB Merlin Series Recirculating Chillers D.3 GEMS 2200 Series Pressure Transducers 	18 19 21 21 21 21 21 21 21 21 21 21

List of Figures

2.1	Electric drivetrain of the Audi e-tron 55 quattro	6
2.2	Schematic of the typical components of an electric drivetrain	7
2.3	Thermal management of the electric drivetrain of the Audi Q4 50	
	e-tron quattro.	8
2.4	Heat fluxes that can be attained at specified temperature differ-	
	ences with various heat transfer mechanisms and fluids	8
2.5	Electrical diagram of a three-phase full bridge inverter.	10
2.6	Cross-section of a typical power module	11
2.7	Power module with air-cooled heat sink.	11
2.8	Air-cooled heat sink with integrated heat pipes	12
2.9	Air-cooled heat sink with integrated vapour chamber	12
2.10	Power module with cold plate.	13
2.11	Power module with pin fins integrated in the baseplate	13
2.12	Power module cooled by pool boiling	15
2.13	Safety group classifications according to ASHRAE Standard 34	18
3.1	Experimental boiling curve for a platinum wire submerged in wa-	
	ter (surface heat flux as a function of surface superheat temperature).	24
3.2	Typical boiling curve (surface heat flux as a function of surface	
	superheat temperature)	25
3.3	Typical boiling curve (surface heat flux as a function of surface	
	superheat temperature)	25
3.4	Nucleate boiling heat transfer mechanisms.	28
3.5	Predicted heat partitioning (y-axis) with the Judd and Hwang model	
	compared to the measured heat flux (x-axis)	29
3.6	Relative contributions of different heat fluxes during nucleate boil-	
	ing according to Paul and Abdel-Khalik	30
3.7	(a) Bubble shape, (b) IR temperature field (°C), (c) Temperature	
	line profile, (d) Heat flux distribution (W/m^2)	32
3.8	Bubble volume (V_{bub}) , latent heat flux (Q_{bub}) and microlayer evap-	
	oration heat flux (Q_{mic}) during bubble growth	32
3.9	Measured bubble radius and equivalent bubble radius calculated	
	from heat flux during bubble growth	33
3.10	Measured and predicted boiling curves, showing different heat flux	
	contributions.	33

3.11 3.12	Measured heat flux partitioning for flow boiling heat transfer Measured and modelled heat flux partitioning for flow boiling heat	35
	transfer.	35
3.13	A vapour bubble in superheated liquid (left) and close-up of the vapour-liquid interface and the forces (in the plane) acting upon it	
	(right)	39
3.14	Mind map of all the parameters used in the cited nucleate boiling	40
2 15	Completion of heiling heat transfer data using the much demonsion size	42
5.15	Correlation of boiling heat transfer data using thermodynamic sim- ilarity (function $E(u)$) in equation 2.40	10
2.16	finally (function $F_1(p_r)$ in equation 5.40	48
3.16	Idealized nucleation site.	50
3.17	Nucleate pool boiling curve of various fluids. Left: water at 20	
	kPa (1) and 3.6 kPa (2). Middle: water (1) and ethanol (2) at 6	-
	kPa. Right: 13% NaCl-water solution (1) and ethanol (2) at 3.6 kPa.	59
3.18	Overview of the cited correlations and division into three groups.	60
3.19	Comparison of the effect of reduced pressure for the Borishanskii-	
	Mostinski, Gorenflo, Cooper and Leiner correlation	63
3.20	Comparison of the effect of reduced pressure for the correlations based on fluid properties of water.	64
3.21	Comparison of the effect of reduced pressure for the correlations	
	based on fluid properties of R134a.	65
3.22	Boiling surface inclination angles	67
3 23	Visualization of the actual surface (grey) envelope surface (or-	0,
0.20	ange) and the largest sphere fitting in between.	69
3.24	Contact angle of a vapour bubble.	72
3.25	Nucleate pool boiling pattern in microgravity.	74
3.26	Hydrodynamic instability model (a) Vapour jets proir to CHF (b)	, ,
5.20	Top and side views of the vapour jets. (c) Helmholtz instability	
	and vapour blanket formation.	80
3.27	Macrolayer dryout model.	81
3.28	Irreversible dry spot model	81
3.29	Interfacial lift-off model.	82
3.30	Force balance on a growing bubble.	83
3.31	Effect of surface orientation on CHF	85
3.32	Effect of contact angle on CHF.	86
3.33	Effect of liquid height on CHF	88
4.1	Boiling and nucleation measurement device. 1: reservoir, 2: chiller,	
	3: power supply, 4: data acquisition system, 5: reference junction .	97
4.2	3D CAD of the reservoir without insulation and front plate	98
4.3	Picture of the reservoir without insulation	98
4.4	Cross-sectional view of the reservoir.	99
4.5	Critical heat flux for FK-649 calculated with the correlation of	
	Lienhard and Dhir as a function of saturation temperature	101

4.6	Coolant circuit. $m = mixer$, $T = temperature measurement$, $V = volumetric flow measurement$	102
17	Disture of the coolent circuit, 1 – inlet thermoscouples 2 – outlet	102
4.7	thermocouples, $3 =$ flow meter, $4 =$ three-way valve, $5 =$ actuator.	102
4.8	Cooling (top) and pumping (bottom) capacity of the Neslab M100	
	chiller	103
4.9	Mixers to achieve a uniform temperature distribution in the coolant.	104
4.10	Measured temperature of the power module as a function of the measured NTC thermistor resistance and the corresponding cali-	
	bration curve.	106
4.11	Flowchart of the measurement procedure.	110
4.12	Saturation temperature as a function of pressure for FK-649	113
4.13	Control volume (blue) around the reservoir, indicating mass flows	
	(green) and electrical power (red) across the boundary	115
4.14	Uncertainty on simulated boiling curves for different saturation	
	temperatures, liquid levels and configurations	116
4.15	Uncertainty on the heat loss estimation as a function of electrical	
	power for the heater and power module configuration	118
4.16	Uncertainty on the energy balance as a function of electrical power	
	for the heater and power module configuration.	118
5.1	Internal structure of Infineon HybridPACK inverter module: square	
	components are IGBTs, rectangular components are diodes,	120
5.2	Ceramic resistive heater.	121
5.3	Aluminium heat contractor.	121
5.4	Boiling plate with five thermocouple grooves.	122
5 5	Guard heater assembly	123
5.6	Heater configuration assembly Red dots indicate thermocouple	125
5.0	locations	124
57	Variation of the boiling plate temperature with a PL controlled power	127
5.7	The temperature variation is fitted using a transfer function	125
58	Simulated step response equal to 1 °C for the boiling plate temper	123
5.0	sture with PL control	125
5.0	Variation of the temperature difference between the guard and main	125
5.9	bester for a step in guard bester power. The temperature difference	
	variation is fitted using a transfer function	126
5 10	Simulated star response equal to 1 °C for the guard bester temper	120
5.10	sture difference with PL control	127
5 1 1	Maggured temperature by thermoscounles of five locations clong	121
5.11	the long side of the boiling plate as a function of surface best flux	
	with a saturation temperature of $36 ^{\circ}\text{C}$	120
5 1 2	Deviation of thermocouple measurements compared to everage	129
5.12	temperature as a function of surface best flux with a seturation	
	temperature of $36 ^{\circ}\text{C}$	120
		100

5.13	Deviation of thermocouple measurements compared to average temperature as a function of thermocouple location.	131
5.14	Measured average boiling plate temperature and derived surface temperature as a function of heat flux for a saturation temperature	
	of 36 °C	132
5.15	Net efflux as a function of electrical power.	133
5.16	Estimated heat losses as a function of electrical power.	133
5.17	Relative deviation of the energy balance as a function of heat flux.	134
5.18	Measured boiling curve for a saturation temperature of 36 °C. Dif- ferent colours indicate different measurement dates	135
5.19	Measured boiling curve for a saturation temperature of 36 °C with	100
0.17	logarithmic-scale axes.	136
5.20	Measured boiling curve for two different saturation temperatures.	137
5.21	Fouling on the boiling surface after boiling for several days in pres-	137
5.22	Boiling curve for a clean ($T_{sat} = 41$ °C) and a fouled ($T_{sat} = 40$ °C) surface	120
5 23	Measured boiling curve for five different saturation temperatures	150
5.25	Boiling curves of 49 °C. 71 °C and 96 °C are reproduced from	
	Forrest et al	139
6.1	Infineon six-pack power module.	142
6.2	Infineon HybridPACK electrical circuit diagram	142
6.3	Internal structure of Infineon HybridPACK inverter module: square components are IGBTs, rectangular components are diodes. Green	
	dashed rectangle indicates cooled surface	143
6.4	IGBT output characteristic	145
6.5	Electrical circuit of the power module and power control, blue lines indicate the flow of current.	145
6.6	Picture of the power module (only baseplate is visible) and sup-	
	porting metal plate mounted on the bottom plate	146
6.7	Schematic of the thermocouples positioned between baseplate and sealing mat.	147
6.8	Variation of the temperature difference between module and fluid	
	for a step change of the module voltage. The temperature differ- ence variation is fitted using a transfer function	148
6.9	Simulated step response equal to 1 °C for the module temperature with PI-control.	148
6.10	Measured temperature by thermocouples at five locations along	
	the long side of the baseplate as a function of the average baseplate temperature.	151
6.11	Deviation of thermocouple measurements compared to average	
	temperature as a function of surface heat flux	151

6.12	Deviation of thermocouple measurements compared to average temperature as a function of thermocouple location. Blue dots are	
	the original measurements, orange crosses are the measurements	
	after remounting.	152
6.13	Module temperature measured by NTC thermistor as a function of	
	the average surface temperature.	153
6.14	Internal thermal resistance as a function of the boiling surface heat	
011 .	transfer rate	153
615	Measured module temperature and estimated IGBT die tempera-	100
0110	ture as a function of the average surface temperature	154
6 16	Net efflux as a function of electrical power	155
6.17	Estimated heat losses as a function of electrical power	156
6.18	Relative deviation of the energy balance as a function of heat flux	150
6.19	Boiling curve for a saturation temperature of 36 °C and a liquid	157
0.17	height of 1 cm. The cross marker indicates critical heat flux	157
6 20	Boiling curve for a saturation temperature of $36 ^{\circ}\text{C}$ and a liquid	157
0.20	beight of 1 cm on a log log plot	158
6 21	Boiling curves for a saturation temperature of 36 °C and varying	150
0.21	liquid height	150
6 22	Poiling ourses for verying seturation temperatures	159
6.22	Earling on the neuron module becometer	160
6.23	Pouling on the power module baseptate	101
0.24	Boiling curves for a saturation temperature of 56°C on clean and	160
6.05	Still formers of multiple to heiling without for different heat formers	162
0.23	Suil frames of nucleate boiling videos for different near fluxes.	103
0.20	Boiling curve for a saturation temperature of 50 °C with an indi-	164
6 27	Critical bast flow as a function of actuation townshow for some	104
0.27	in a liquid height	165
6 20	Ing inquia neight.	105
0.28	Critical heat flux as a function of saturation temperature on a clean	100
(20	and a fouled surface.	100
0.29	Boiling curves for the neater and power module configuration with	1(7
	a saturation temperature of 36 °C.	10/
71	Predicted surface superheat temperature as a function of the mea-	
/.1	sured surface superheat temperature for fifteen correlations	176
72	Measured boiling curve and boiling curves according to correla-	170
1.2	tions (1/3)	178
73	Measured boiling curve and boiling curves according to correla	170
1.5	tions (2/3)	170
71	Measured boiling curve and boiling curves according to correla	1/9
/.4	tions (2/2)	170
75	Magurad hast flux as a function of the product of the magurad	1/9
1.5	interstited near flux as a function of the product of the measured surface superbasit temperature and $E(\alpha)$ for the measurements of	
	surface superinear temperature and $F(p_r)$ for the measurements on the power module configuration. The functions $E(r_r)$ are based	
	the power module configuration. The functions $F(p_r)$ are based	101
		101

7.6	Measured heat flux as a function of the product of the measured surface superheat temperature and factors based on the boiling sur-	
	face according to the heat transfer correlations.	183
7.7	Predicted CHF by the correlations as a function of the measured	
	CHF	184
7.8	CHF as a function of saturation temperature for the measurements	
	and correlations	185
7.9	CHF as a function of saturation temperature for the measurements	
	and correlations with extrapolation to higher temperatures	186
B .1	Flow rate range for which the uncertainty on the energy carried by	
	the coolant is below 3% as a function of the electrical power	218

xii

List of Tables

2.1	Overview of heat transfer coefficients (wall to coolant) reported in literature, based on baseplate area, for different cooling methods.	16
2.2	Overview of possible two-phase coolants and their properties	21
3.1	Heat transfer mechanisms included in different models. NC = nat- ural convection, ML = microlayer evaporation, QC = quenching, MC = microconvection, DS = dry spot, FC = forced convection. Y = mechanism included, Y- = mechanism included but accounts for lass than 10% of total heat transfer N = mechanisms not included	26
32	Overview of the cited correlations $(N - mechanisms not metuded.)$	56
33	Value of n in equation 3.66 for different correlations	58
3.4	Exponents of fluid properties in different correlations.	62
3.5	Selected properties of water and R134a.	64
3.6	Dependence of heat transfer rates of surface roughness	70
3.7	Exponent of gravitational acceleration in different correlations	76
3.8	List of studies on pool boiling of FK-649.	91
4.1 4.2	Selected fluid properties of FK-649	100
	surement device.	107
6.1	Maximal heat transfer coefficients and heat fluxes, based on base- plate area, for different cooling methods	169
7.1	$NMAE$ and NAE_{max} of the heat transfer correlations with regard to the measurements.	175
7.2	$NMAE$ and NAE_{max} of the CHF correlations with regard to the	
	measurements.	185
7.3	Comparison of different working fluids	187

Nomenclature

Latin symbols

A	area	m^2
NAE_{max}	maximal normalized absolute error	-
Ar	Archimedes number	-
C	constant	-
C_{sf}	surface-fluid parameter in equation 3.31	-
c_p	specific heat capacity	J/kgK
\hat{D}	diameter	m
d	thickness	m
E	energy	J
e	relative deviation of energy balance	-
f	bubble departure frequency	s^{-1}
G	mass flux	kg/m ² s
g	gravitational acceleration	m/s^2
h	specific enthalpy	J/kg
h_{lv}	latent heat of vaporization	J/kg
Ι	current	А
Ja	Jakob number	-
Ja'	modified Jakob number	-
K	dimensionless critical heat flux	-
k	thermal conductivity	W/mK
L	characteristic length	m
L_c	capillary length	m
M	molar mass	kg/mol
\dot{m}	mass flow	kg/s
NMAE	normalized mean absolute error	-
N	nucleation site density	m^{-2}
N_a	Avogadro constant	mol^{-1}
N_m	number of measurements	-
Nu	Nusselt number	-
P	power, rate of energy transfer	W
p	pressure	Pa
Pr	Prandtl number	-
\dot{Q}	heat flow	W

ġ	heat flux	W/m^2
\overline{R}	surface roughness	m
R_a	arithmetic mean surface roughness	m
$R_{p,DIN}$	surface roughness according to DIN 4762	m
R_{th}	thermal resistance	K/W
r	radius	m
r_c	radius of the critical nucleus	m
R	specific gas constant	J/kgK
$\bar{\Re}$	universal gas constant	J/molK
Re	Reynolds number	-
T	temperature	Κ
ΔT_0	minimal superheat temperature for nucleation	Κ
t	time	S
U	heat transfer coefficient	W/m ² K
u	velocity	m/s
V	voltage	V
\dot{V}	volume flow rate	m ³ /s
\dot{W}	power	W
We	Weber number	-
z	height	m

Greek symbols

α	thermal diffusivity	m ² /s
β	static contact angle	rad
Δ	difference	-
δ	uncertainty	-
θ	surface inclination	rad
μ	dynamic viscosity	Pa s
ν	kinematic viscosity	m ² /s
ho	density	kg/m ³
σ	surface tension	N/m ³

Subscripts

0	reference
Al	aluminium
b	bubble departure
bg	bubble growth
C	collector
c	critical
CHF	critical heat flux

E	emitter
e	electrical
ef	efflux
F	Fritz
G	gate
i	in
int	internal
l	liquid
loss	losses
m	measured
max	maximal
0	out
p	predicted
r	reduced (relative to critical value)
ref	at reference pressure
rel	relative
s	surface superheat
sat	saturation
sub	subcooling
TC	thermocouple
v	vapour
w	wall (boiling surface)

Acronyms

AC	alternating current
ASHRAE	American Society of Heating, Refrigeration
	and Air-Conditioning Engineers
CAD	computer-aided design
CFC	chlorofluorocarbon
CHF	critical heat flux
CVD	chemical vapour deposition
DC	direct current
DIN	Deutsches Institut für Normung
DNB	departure of nucleate boiling
DS	dry spot
EM	electric motor
FC	forced convection
FK	fluoroketone
FTIR	Fourier transform infrared
GWP	global warming potential
HCFC	hydrochlorofluorocarbon
HCFO	hydrochlorofluoroolefin
HFC	hydrofluorocarbon

HFE	hydrofluoroether
HFO	hydrofluoroolefin
IGBT	insulated-gate bipolar transistor
NMAE	normalized mean absolute error
MC	microconvection
ML	microlayer
MOSFET	metal-oxide-semiconductor field-effect transistor
NC	natural convection
NREL	National Renewable Energy Laboratory
NTC	negative temperature coefficient
ODP	ozone depleting potential
OEL	occupational exposure limit
ONB	onset of nucleate boiling
PE	power electronics
PFC	perfluorocarbon
PID	proportional-integral-derivative
PMMA	polymethyl methacrylate
POM	polyoxymethylene
PUR	polyurethane
QC	quenching
TC	thermocouple
TIM	thermal interface material
WG	water-glycol

Nederlandse samenvatting -Summary in Dutch-

Om de klimaatverandering tegen te gaan, moet de uitstoot van broeikasgassen sterk verminderd worden in alle sectoren. In de transportsector worden verschillende mogelijkheden bekeken om de uitstoot van CO_2 te verminderen, zoals verbrandingsmotoren die werken op waterstof of biobrandstoffen. Voor voertuigen is de meest toegepaste strategie de omschakeling naar elektrische aandrijving met batterijen die kunnen worden opgeladen met hernieuwbare elektrische energie. Elektrische wagens hebben momenteel verschillende nadelen in vergelijking met standaard voertuigen met verbrandingsmotor, zoals een lagere actieradius en een hogere kostprijs. Deze nadelen kunnen worden tegengegaan door de elektrische aandrijflijn (motor, vermogenselektronica en batterij) compacter te maken en dus de vermogensdichtheid te verhogen. Om een verdere stijging in vermogensdichtheid te verkrijgen, zijn betere koelsystemen een vereiste. De hoogste warmtefluxen vinden plaats in de halfgeleidermodules in de vermogenselektronica van de aandrijflijn waardoor deze bij de meest uitdagende componenten zijn om te koelen.

Moderne koelsystemen van elektrische aandrijflijnen gebruiken vloeistofkoeling met mengsels van water en glycol. Tweefasige koeling, waarbij de koelvloeistof begint te koken wanneer het in contact komt met componenten die warmte afgeven, heeft het potentieel voor betere warmteoverdracht en dus performantere koeling. Tweefasige koeling brengt echter ook een groot risico met zich mee. Als de warmteflux van een component van de aandrijflijn naar de koelvloeistof stijgt boven een maximale waarde, de kritische warmteflux, dan zal een laag van gasvormige koelvloeistof het gehele oppervlak bedekken. Deze laag gedraagt zich als thermische isolatie waardoor de warmteoverdracht drastisch daalt met als resultaat oververhitting en falen van de component. FK-649, een fluorketon, is gekozen als koelvloeistof in het tweefasige koelsysteem dat wordt bestudeerd in dit werk. Deze koelvloeistof heeft een aardopwarmingspotentieel gelijk aan 1, wat ervoor zorgt dat sporadische lekken van de vloeistof verwaarloosbaar zullen bijdragen aan de klimaatopwarming.

Warmteoverdracht tijdens koken is een complex fenomeen met een sterke interactie tussen warmteoverdracht en massatransport. Hoewel onderzoek naar warmteoverdracht tijdens koken al bijna een eeuw aan de gang is, is er nog geen consensus over welke warmteoverdrachtsmechanismen dominant zijn. Dit gebrek aan fundamentele kennis maakt het moeilijk om warmteoverdracht te voorspellen en tweefasige koelsystemen te ontwerpen. Meerdere correlaties voor warmteoverdracht zijn opgesteld door verschillende onderzoekers. Deze correlaties houden in diverse mate rekening met de effecten van warmteflux, vloeistofeigenschappen en eigenschappen van het kookoppervlak. Ook voor de kritische warmteflux zijn er meerdere correlaties opgesteld, hoewel deze niet zo verschillend zijn van elkaar als deze voor warmteoverdracht. De beschikbare meetgegevens van warmteoverdracht door koken van FK-649 zijn beperkt. Het is ook niet geweten welke van de bestaande correlaties het meest geschikt zijn om warmteoverdracht en de kritische warmteflux te voorspellen voor FK-649.

Een meetopstelling is ontworpen en gebouwd om de warmteoverdracht te bepalen tijdens het koken van FK-649. De opstelling bestaat uit een afgesloten reservoir met daarin FK-649. Het koken vindt plaats op een horizontaal oppervlak onderaan in het reservoir, terwijl het gas dat ontstaat tijdens het koken condenseert op een spiraalvormige condensor bovenaan in het reservoir. De zijkanten van de opstelling zijn voorzien van transparante platen uit polycarbonaat om de kookverschijnselen te bekijken. Het soort kookoppervlak kan worden aangepast en de saturatietemperatuur van het FK-649 kan worden ingesteld op de gewenste waarde. De saturatietemperatuur werd tijdens de testen gevarieerd tussen drie waarden: 36 °C, 41 °C en 46 °C.

In de halfgeleidermodule die wordt gebruikt tijdens de experimenten wordt de warmte voornamelijk gedissipeerd door IGBTs. Deze componenten zijn geplaatst op een basisplaat met een grotere oppervlakte dan die van de componenten. Dit laat toe dat de in de component ontstane warmte kan worden verspreid zodat warmteoverdracht kan plaatsvinden op een groter oppervlak. Aangezien de warmte dus niet gelijkmatig verdeeld is over de basisplaat intern in de module, is het mogelijk dat er op het kookoppervlak ook een ongelijkmatige verdeling van de warmteflux optreedt. Om dit effect te analyseren, worden de metingen van vrij koken van de halfgeleidermodule vergeleken met metingen op een kookoppervlak dat gelijkmatig wordt verwarmd door een elektrische weerstand. De opgemeten warmteoverdracht voor beide configuraties verschilt minder dan 10% van elkaar. Ook visueel is er geen bewijs waargenomen voor een ongelijkmatig verdeelde warmteflux. Bellen ontstaan gelijkmatig over het kookoppervlak en vertonen geen hogere concentratie op de locaties het dichtst bij de IGBTs. Daaruit wordt geconcludeerd dat de basisplaat ervoor zorgt dat de warmte goed wordt verspreid en dat aan het kookoppervlak een zo goed als gelijkmatige warmteflux wordt bereikt. Deze resultaten geven aan dat de bestaande correlaties voor gelijkmatig verdeelde warmteflux zonder aanpassing kunnen worden toegepast voor de koeling van de basisplaat van een halfgeleidermodule.

Dertien correlaties die de warmteoverdracht tijdens bellenkoken voorspellen worden beoordeeld. Volgens chronologische volgorde van publicatie gaat het over de correlaties van Kruzhilin, Rohsenow, Forster-Zuber, Kutateladze-Borishanskii, Borishanskii-Mostinksi, Shekriladze-Ratiani, Labuntsov, Gorenflo, Stephan-Abdelsalam, Cooper, Kutateladze, Leiner en Pioro. De correlaties van Rohsenow en Pioro vereisen het bepalen van een of meerdere constanten gebaseerd op meetdata voor elke combinatie van oppervlak en vloeistof. Een vergelijking is gemaakt tussen de voorspelde thermische prestatie en de meetdata. Zoals verwacht geven de correlaties met constanten gebaseerd op de meetdata de beste voorspelling. De correlatie van Rohsenow geniet van deze de voorkeur. De gemiddelde afwijking tussen voorspelling en meting is 7.5% en alle meetpunten worden voorspeld met een afwijking lager dan 25.6%. De correlatie van Labuntsov is de beste correlatie om de meetdata met FK-649 te voorspellen zonder gebruik te maken van de meetdata. De gemiddelde afwijking tussen voorspelling en meting is 7.5% en alle meetpunten worden voorspeld met een afwijking lager dan 29.1%. Opmerkelijk is dat deze gemiddelde afwijking van de correlatie van Labuntsov even groot is als de afwijking van de correlatie van Rohsenow die is geoptimaliseerd naar de meetdata. Beide correlaties voorspellen nauwkeurig het effect van de saturatietemperatuur op de warmteoverdracht. De correlatie van Labuntsov is de te verkiezen correlatie om warmteoverdracht te voorspellen als er geen meetdata voorhanden is. Deze correlatie wordt echter best niet gebruikt voor niet-metallische kookoppervlakken of kookoppervlakken met een ruwheid die ver afwijkt van standaard waarden, aangezien de correlatie deze invloeden niet in rekening brengt.

De meetdata toont aan dat de invloed van de warmteflux op warmteoverdracht tijdens bellenkoken kan worden opgedeeld in drie afzonderlijke regimes. Deze regimes zijn ook visueel waargenomen. De drie regimes van lage naar hoge warmteflux zijn gedeeltelijk bellenkoken, ontwikkeld bellenkoken en gedeeltelijk uitgedroogd koken. Deze regimes werden ook beschreven in de wetenschappelijke literatuur door verschillende onderzoekers. Geen enkele van de correlaties houdt echter rekening met de verschillende regimes tijdens bellenkoken en deze maken daardoor geen nauwkeurige voorspellingen van de verandering in warmteoverdracht met wijzigende warmteflux. De meetdata in deze studie werden gecorreleerd volgens een specifieke machtsfunctie voor elk van de drie regimes. Het is aangeraden om de verschillen tussen de regimes in rekening te brengen bij het opstellen van nieuwe correlaties voor warmteoverdracht.

Vier correlaties voor de kritische warmteflux zijn vergeleken met de meetdata, namelijk deze voorgesteld door Kutateladze, Zuber, Lienhard-Dhir en Mudawar et al. Zowel de correlatie van Zuber als Lienhard-Dhir voorspellen alle meetdata met een maximale afwijking van 10%. Deze resultaten tonen aan dat de correlaties voor de kritische warmteflux veel betere voorspellingen geven dan de correlaties voor warmteoverdracht.

Om de haalbaarheid van tweefasige koeling van vermogenselektronica met FK-649 te beoordelen, wordt de gemeten thermische prestatie vergeleken met die van koeling met water-glycol mengsels. De hoogste warmteoverdrachtscoëfficiënt gemeten in deze studie is 4965 W/m²K en de hoogste kritische warmteflux gemeten is 146 kW/m². Deze warmteoverdrachtscoëfficiënt is gelijkaardig aan die van water-glycol koeling op een vlakke basisplaat. De hoogst haalbare warmteflux voor water-glycol koeling is echter meer dan twee maal hoger dan de kritische warmteflux van FK-649. Om de tweefasige koeling van halfgeleidermodules met FK-649 te verbeteren ten opzichte van water-glycol koeling, zal de kritische warmteflux verhoogd moeten worden. De mogelijkheden om dit te verwezenlijken zoals de verhoging van de saturatietemperatuur, gebruik maken van onderkoeling van de vloeistof of stromingskoken moeten verder worden bestudeerd.

English summary

To combat climate change, greenhouse gas emissions have to decrease across all sectors. In the transport sector, several options to reduce the emission of CO_2 are considered, such as hydrogen and bio-fuel powered engines. For vehicles, the most adopted strategy is to convert to battery electric vehicles, which can be charged using renewable electrical energy. Currently, electric vehicles have several drawbacks compared to conventional internal combustion engine vehicles, such as lower driving range and higher cost. These drawbacks can be combated by making the electric drivetrain (motor, power electronics and battery) more compact and therefore more power-dense. Further increasing the power-density requires more effective cooling systems. The power modules in the power electronics of the drivetrain feature the highest heat fluxes and are therefore one of the most challenging components to cool.

State-of-the-art thermal management systems of electric drivetrains utilize liquid cooling with water-glycol mixtures. Two-phase cooling, with a coolant that starts to boil as it comes into contact with heat generating components, has the potential of acquiring higher heat transfer rates and thus more effective cooling. Two-phase cooling does come with a major risk. If the heat flux from a drivetrain component to the coolant increases above a maximal value, the critical heat flux, a continuous vapour film will be formed on that surface. This layer acts as a thermal insulator and thereby reduces the heat transfer rates dramatically, which results in overheating and failure of the component. In this study, the fluid chosen for the two-phase cooling system is FK-649, a fluoroketone. It has a low global warming potential equal to 1, which means that sporadic leaks of the fluid will not contribute significantly to global warming.

Boiling heat transfer is a complex phenomenon with a strong interaction of heat and mass transfer. Although research on boiling heat transfer has been conducted for almost a century, there is still no consensus on which heat transfer mechanisms are dominant. This lack of fundamental understanding renders it difficult to predict heat transfer rates. Several heat transfer correlations have been proposed by different authors, taking into account to various degrees the effects of heat flux, fluid properties and boiling surface properties. Several correlations have also been proposed for the critical heat flux and these show a larger agreement with each other than the heat transfer correlations. Limited data is available for boiling heat transfer of FK-649. It is also unclear which correlations are best suited to predict heat transfer rates and critical heat flux for FK-649.

To determine the heat transfer rates associated with pool boiling of FK-649,

an experimental setup is designed and constructed. The setup consists of a sealed reservoir containing FK-649. Boiling occurs on a horizontal boiling surface at the bottom of the reservoir, while the generated vapour is condensed by a spiral condenser at the top. Polycarbonate windows are added for visual access to the boiling phenomenon. The type of boiling surface can be altered and the saturation temperature of FK-649 can be controlled to the desired value. The saturation temperature during the experiments was varied in three levels: 36 °C, 41 °C and 46 °C.

In the power module under test, IGBTs are the highest heat dissipating components. These components are placed on a baseplate which has a larger area than those of the components. This allows to spread the heat generated in the components with the goal of having a larger heat transfer area. As the heat is not generated uniformly over the entire internal baseplate area, this may result in a non-uniform heat flux at the boiling surface. To analyse this effect, measurements of pool boiling on the power module are compared to measurements with a boiling surface that is uniformly heated by an electrical resistive heater. The measured heat transfer rates matched within 10%. Also visually, no evidence of non-uniform heat fluxes were perceived. Bubble nucleation occurred uniformly across the boiling surface and was not correlated with the location of the IGBTs. It is concluded that the baseplate effectively spreads the heat and that a quasi-uniform heat flux is achieved at the boiling surface. These results indicate that the correlations developed for surfaces with uniform heat fluxes can be directly applied to power module baseplate cooling.

Thirteen correlations that predict nucleate pool boiling heat transfer are assessed. In order of the date of publication, these are the correlations of Kruzhilin, Rohsenow, Forster-Zuber, Kutateladze-Borishanskii, Borishanskii-Mostinksi, Shekriladze-Ratiani, Labuntsov, Gorenflo, Stephan-Abdelsalam, Cooper, Kutateladze, Leiner and Pioro. Of these correlations, the Rohsenow and Pioro correlations require fitting one or more constants to experimental data for each specific surfacefluid combination. The predicted thermal performance is compared to the experimental results. As expected, the correlations with fitting constant perform best. The Rohsenow correlation is the preferred correlation, it has an average deviation of 7.5% and predicts all measurement points within 25.6%. Of the non-fitted correlations, the Labuntsov correlation is best at predicting heat transfer rates with FK-649. The average deviation of the prediction from the measurements is equal to 7.5% and all data is predicted within 29.1%. Remarkably, the average error of the Labuntsov correlation is equal to that of the fitted Rohsenow correlation. Both correlations take into account accurately the effect of saturation temperature on heat transfer. The Labuntsov correlation is thus preferred for predicting heat transfer rates without available measurement data for fitting. Caution should however be exercised if the correlation is used for non-metallic surfaces or surfaces with a non-standard surface roughness, as the Labuntsov correlation does not take into account influences of the boiling surface.

The experimental results showed that the effect of the heat flux on the nucleate boiling heat transfer rates can be divided into three separate regimes. These regimes were also perceived visually. From low to high heat flux, the three regimes are the partial nucleate boiling, fully developed nucleate boiling and partial dryout boiling regimes. These regimes are also reported by several authors in literature. However, none of the heat transfer correlations take into account the different nucleate boiling regimes and they therefore do not predict adequately the variation of the heat transfer rates with heat flux. The measurements in this study were correlated using a specific power-law relation for each of the three regimes. For more accurate predictions, future heat transfer correlations should take into account the differences between the three regimes.

Four critical heat flux correlations are compared to the experimental data: those of Kutateladze, Zuber, Lienhard-Dhir and Mudawar et al. Both the Zuber as the Lienhard-Dhir correlations predict all data within 10%. It is concluded that the critical heat flux correlations perform much better than the heat transfer correlations.

To assess the feasibility of two-phase power electronics cooling with FK-649, the measured thermal performance is compared to that of water-glycol cooling. The maximal heat transfer coefficient measured in this study was 4965 W/m²K and the highest critical heat flux was 146 kW/m². The heat transfer coefficients are similar to those of water-glycol cooling of a flat baseplate. However, the maximal attainable heat fluxes with water-glycol are more than two times higher than the critical heat flux of FK-649 boiling. For two-phase cooling of power modules with FK-649 to outperform water-glycol cooling, strategies for increasing the critical heat flux such as increasing the saturation temperature, using subcooled boiling or using flow boiling should be investigated.

Introduction

1.1 Electrification and energy efficiency

In December 2019, The European Commission presented the European Green Deal [1]. To combat climate change, all EU member states committed to turning the EU into the first climate neutral continent by 2050. In July 2021, the European Commission presented several proposals for the transformation of the economy to reach these targets [2]. These proposals are focused on several key areas, among which are sustainable transportation and a green transition in industry.

For the transport sector, one of the goals is for all new cars starting from 2035 to have no net greenhouse gas emissions [3]. There are several potential candidates for zero-emission vehicles such as bio-fuel or hydrogen powered cars, but at this time most progress is made with electric vehicles utilizing renewable electrical energy as power source. Currently, electric vehicles are still underperforming compared to conventional internal combustion engine vehicles with regard to cost and driving range [4]. To improve competitiveness, drivetrains of electric vehicles need to become more efficient, power-dense and smaller, to increase driving range and reduce cost.

In the industrial sector, the green transition is powered by electrification and greater use of renewable energy [5]. Another key point is to increase energy efficiency of industrial systems. Electrical drivetrains can be found in every industrial branch, from manufacturing equipment to generators in power plants. Increasing the energy efficiency of these drivetrains will result in more sustainable products

and activities.

In general, the goal is to strive for more compact, power-dense and efficient drivetrains. However, although energy efficiency can be increased, some power losses are unavoidable. These losses are transformed into heat, which results in an increase in the temperature of the drivetrain components [6]. When designing more compact systems, this heat will be generated in smaller volumes and can result in overheating and failure of the powertrain component. The thermal management or cooling of the drivetrain therefore becomes increasingly important when moving to more power-dense systems.

Increasing the effectiveness of the thermal management system can have two beneficial effects. As heat is transferred from the component more effectively, the system can be made more compact as the same amount of heat can be transferred from a smaller surface. Advanced cooling methods can thus lead to more compact and power-dense drivetrains. Otherwise, if the size of the drivetrain is not changed but more effective cooling is used, the component temperatures will decrease. Generally, this results in a more energy-efficient system [7, 8] and a longer lifetime of the components [9].

One of the crucial components of electrical drivetrains are the power electronics [10]. These components control the flow of electrical energy from the energy source to the actuator. The highest heat fluxes of the entire drivetrain are found in the power modules of the power electronics. This makes them one of the most challenging components to cool.

1.2 Boiling heat transfer

A potentially very effective thermal management strategy is to utilize two-phase cooling [11]. A liquid coolant is used, which starts to boil when it comes into contact with the heat-generating components. This boiling phenomenon results in highly effective heat transfer. Although boiling heat transfer has been studied for many decades, there is no consensus yet on the physical phenomena resulting in the high heat transfer rates. As such, boiling heat transfer rates remain hard to predict and the correlations to do so have high uncertainties.

A two-phase cooling system requires a fluid that boils at the desired temperature and pressure. Furthermore, the fluid should have satisfactory properties, both from a thermal as a practical viewpoint. Many fluids used currently in twophase systems are potent greenhouse gases. A commonly used fluid in mobile airconditioning systems is R134a. When released into the atmosphere, one molecule of R134a contributes 1430 times more to global warming than a molecule of carbon dioxide (this value is called the global warming potential or GWP) [12]. For the two-phase system to have an overall positive impact on climate change, a low-GWP potential fluid should be chosen. Boiling heat transfer does not only have an application in drivetrain cooling. Two-phase cooling of other components such as electronics in telecom and computing is also a possibility. Boiling also occurs in heat exchangers in many applications (some examples are refrigeration, heat pumps and steam generation) and in a multitude of industries (for example petrochemical, automotive and nuclear). This shows that a better understanding of the fundamental aspects of boiling heat transfer can be beneficial for numerous applications and sectors.

1.3 Objectives of the study

The objective of this study is to analyse two-phase cooling of power electronic devices. An experimental setup will be made to determine the heat transfer performance. A low-GWP fluid will be selected as coolant.

This study has both a fundamental as a technical aspect. The fundamental aspect consists of studying pool boiling heat transfer for a low-GWP fluid. The influences of different boundary conditions on heat transfer will be determined. Different prediction methods (correlations) for heat transfer will be assessed by comparing predictions to the experimental data. The technical aspect concerns the application of pool boiling heat transfer to power electronics. The influence of the specific geometry of the power module on the boiling heat transfer will be investigated. The heat transfer performance of two-phase cooling will be compared with other state-of-the-art cooling methods.

Additionally, the curious and/or impatient reader is referred to section 3.7, where an extensive list of all the research questions to be answered in this book is given, based upon the literature study in the next chapters.

1.4 Outline

Chapter 2 starts of by introducing electric drivetrains and their components. An overview is given of potential thermal management strategies. The selection criteria for coolants and the different potential coolants for two-phase cooling are described. Based on this analysis, FK-649 is chosen as coolant.

Chapter 3 discusses into detail the current knowledge of pool boiling heat transfer. The fundamental heat transfer mechanisms are described for a better understanding of the heat transfer phenomenon. Many authors have proposed correlations which translate the fundamental heat transfer mechanisms to equations for the prediction of heat transfer rates during pool boiling. The most frequentlyused correlations are reported and summarized. The sensitivity of pool boiling heat transfer to different boundary conditions and parameters is analysed. Subsequently, the mechanisms, parameter sensitivity and correlations of the critical heat flux are reported on. The critical heat flux is a limiting value for cooling and is as such crucial for the design of two-phase cooling systems. Finally, the limited available studies of pool boiling cooling with FK-649 are discussed. Based on the review of drivetrain cooling in chapter 2 and pool boiling heat transfer in chapter 3, a set of fundamental and technical research questions is formulated which need to be answered to reach the research objective.

The experimental setup used for measuring pool boiling heat transfer is described in chapter 4. The goal of this setup is to have accurate and repeatable measurements. To this end, the different components, sensors and controls are discussed. The measurement procedure and analysis are established. Finally, an uncertainty analysis is made to quantify the effect of sensor inaccuracies.

In the first set of measurements, pool boiling from a horizontal plate is studied which is heated by an electrical heater. The results are given in chapter 5. The electrical heater supplies a uniform heat flux to the boiling surface. These measurements give a baseline performance to compare to the measurement on the power module. The effects of saturation temperature and liquid height on heat transfer are discussed.

Chapter 6 reports on the measurements of two-phase cooling of a power module. The influences of liquid height, saturation temperature and fouling on heat transfer are discussed. It is shown that nucleate pool boiling heat transfer occurs in three different regimes with different behaviours. The critical heat flux is also determined in the experiments. A comparison with the measurements with electrical heater shows that both configurations have similar heat transfer rates. Finally, the performance of two-phase cooling is compared to other state-of-the-art cooling methods.

In chapter 7, the experimental data from the two previous chapters are compared to the predictions of the correlations summarized in chapter 3. This is done for both heat transfer and critical heat flux correlations.

Chapter 8 repeats the main conclusions of this study. Additionally, an overview of lessons learned is given as well as recommendations for future studies.
Thermal management of electric drivetrains

2.1 Introduction

This chapter gives an overview of the state of the art regarding thermal management of electric drivetrains. Section 2.2 introduces the electric drivetrain and its components. The next section lists the different options for thermal management strategies of electric drivetrains. These different strategies are subsequently applied to the cooling of the power electronics of the drivetrain in section 2.4. The thermal management system of the drivetrain also requires a coolant. The requirement for and the selection of this coolant is discussed in section 2.5. In the last chapter, the main conclusions of the chapter are summarized.

2.2 Electric drivetrains

An electric drivetrain or powertrain always consists of at least an electric motor (EM) and an electrical energy source. Most drivetrains use AC motors, which typically require three-phase electrical power. The energy source for stationary applications is typically the electrical energy grid, while for mobile applications usually batteries are used. An example of an electric vehicle drivetrain is given in figure 2.1.

To be able to control the rotation of the motor, a power electronics (PE) unit is



Figure 2.1: Electric drivetrain of the Audi e-tron 55 quattro [13].

required. In the case of a battery as energy source, the power electronics unit will consist of a DC/AC unit or inverter, to transform the DC voltage of the battery to an AC waveform for the motor. A DC/DC converter is also used between to convert the high voltage of the battery to a low voltage for powering auxiliary systems (headlights, fans, pumps, electronics, ...). To charge the battery, an additional AC/DC converter (on-board charger) can be included. If electrical power from the grid is used, the power electronics unit consists of an AC/DC rectifier and an inverter. A schematic of a typical electrical drivetrain for an electric vehicle is given in figure 2.2.

State-of-the-art drivetrains are becoming ever more power dense, as this has many advantages [14]. More compact drivetrains have lower component costs for the same power output. Smaller electric motors can operate more dynamically and react faster. For mobile applications, a reduction in volume and weight results in an increase of the driving range.

All components of the electrical drivetrain inherently have some power losses and therefore heat up. To avoid overheating and damage to these components, this heat has to be transferred effectively from the component to the environment. With the drive towards more power dense systems, heat losses occur in smaller volumes and need to be transferred from smaller areas, putting extra strain on the cooling system. For drivetrains to keep increasing in power density, new highly effective



Figure 2.2: Schematic of the typical components of an electric drivetrain.

thermal management (cooling) systems are a necessity.

2.3 Thermal management strategies

Figure 2.3 shows an example of a thermal management system for an electric vehicle drivetrain. The battery is cooled by a different cooling circuit (low-temperature circuit) than the EM and PE (medium-temperature circuit). This is common for all drivetrains, as the batteries require lower operating temperatures (< 50 °C) than the EM and PE (< 150 °C).

Different heat transfer fluids can be used for the EM and PE cooling. A general classification of utilizing different fluids as heat transfer media is given in figure 2.4. This graph shows the expected temperature difference between component and coolant as a function of the heat flux from the component. The goal of the cooling system is to have a low temperature difference for a high heat flux. The most simple solution, air cooling, performs worst in this respect. The most adopted strategy, using a water(-glycol) cooling circuit, improves the cooling performance by almost two orders of magnitude. The figure also shows that immersion boiling might be able to perform even better. The following sections go more into detail into the different heat transfer fluids.

2.3.1 Air cooling

Air cooling is typically used in small and low-power systems where efficiency is not of high importance and adding a coolant circuit would make the drivetrain too bulky. Air cooling can be passive (through natural convection) or active (with the use of fans). For stationary systems of higher power, air cooling is sometimes preferred for the lower maintenance costs. For larger electric vehicles (excluding



Figure 2.3: Thermal management of the electric drivetrain of the Audi Q4 50 e-tron quattro [13].



Figure 2.4: Heat fluxes that can be attained at specified temperature differences with various heat transfer mechanisms and fluids (reproduced by Cengel [9] from Kraus and Bar-Cohen [11]).

bikes and scooters), liquid cooling is almost always preferred over air cooling, as this has the advantage of higher power densities.

2.3.2 Water-glycol cooling

The most common liquid cooling system utilizes a water-glycol mixture as coolant in an actively circulated system. Glycol is added to avoid freezing of the liquid. The main advantages of using a water-glycol cooling circuit are the higher heat transfer coefficients when compared to air cooling. This allows for a significant increase in power density for both the EM and the PE, without risking overheating. A downside of water-glycol cooling is that the coolant is electrically conductive. Care has to be taken that the coolant does not come into contact with any active components of the EM and PE.

2.3.3 Oil cooling

The full drivetrain typically consists of bearings, gears and/or gearboxes next to the electrical components. These components need to be lubricated (and possibly cooled) and as such require a pumped oil circuit [15]. If this oil circuit is also used for cooling of the electrical components, an additional circuit is avoided which results in a higher overall power-density of the drivetrain. Oils have the additional benefit of being dielectric, which enables the use of direct cooling of the active components of the EM and PE. The downside of using oils is the inferior heat transfer properties when comparing to water-glycol mixtures. The SingleOilCnL project further investigates this strategy of combining the lubrication and coolant circuit [16].

2.3.4 Two-phase cooling

As indicated in figure 2.4, using a boiling fluid as coolant can possibly reach even higher heat transfer rates than forced water-glycol cooling. Due to the density difference between liquid and vapour, natural circulation of the fluid is possible. In this case, the cooling loop is typically called a thermosyphon. The downside of this configuration is that the cooling performance is very dependent on the orientation of the coolant circuit: the components to be cooled should be lower than the radiator which condenses the coolant vapour. An alternative is a pumped two-phase system, which can operate in any orientation. Next to the higher heat transfer rates, most fluids used in two-phase systems are dielectric (more information can be found in section 2.5). This indicates that direct contact cooling of the active components is a possibility.

2.4 Power electronics cooling

2.4.1 Inverter

An electric drivetrain can have several power electronic units. This section is focused on cooling of the inverter as this is a common component in most drivetrains. The results can easily be translated to the thermal management of other power electronic components.

The cores of the three-phase full bridge inverter are the three half-bridge inverter legs, one for each phase of the three-phase electric motor. Each of these legs consists of four power switches: two diodes and two actively controlled switches. The electrical circuit is given in figure 2.5. The controlled switches are commonly IGBTs (insulated-gate bipolar transistors) but can also be MOSFETs (metal-oxide-semiconductor field-effect transistor). Other components typically found in an inverter are capacitors, filters, sensors and a driver board. Most heat losses occur in the semiconductor power switches, where also the highest heat fluxes can be found. These components are thus the most challenging to cool and are the focus of this study.



Figure 2.5: Electrical diagram of a three-phase full bridge inverter.

The switches are typically arranged in a power module. The three-phase full bridge can be implemented in one power module or can be made up out of three half-bridge power modules. A cross-sectional view of a typical power module is given in figure 2.6. The semiconductor dies are soldered to a substrate. This substrate consists of a ceramic layer which provides electrical insulation and a layer of copper on either side. The bottom copper layer is subsequently soldered to the baseplate, which can be copper or aluminium-based. The module is designed to effectively transfer heat from the semiconductor dies to the baseplate, while at the same time guaranteeing electrical insulation.

The baseplate is larger than the semiconductor dies. This allows to spread the heat generated in the dies throughout the baseplate, such that a larger heat sink can



Figure 2.6: Cross-section of a typical power module.

be used and improved cooling can be achieved. Several methods of cooling the baseplate are covered in the next sections.

2.4.2 Air cooling

Air-cooled power electronics typically utilize large heat sinks with numerous and large fins. An example is given in figure 2.7. This is done to increase the heat transfer area which counteracts the low convective heat transfer coefficients of air flows to achieve a lower thermal resistance.



Figure 2.7: Power module with air-cooled heat sink [17].

To be able to accommodate as much fins as possible, the base of the heat sink is larger than the power module baseplate. Heat has to be spread from the module baseplate throughout the entire heat sink, which also has a relatively large thermal resistance (up to 30% of the total thermal resistance from semiconductor die to air) [17]. Options to reduce this spreading resistance are integrating heat pipes in the heat sink base (figure 2.8) [17] or by using a vapour chamber (figure 2.9) [18].



Figure 2.8: Air-cooled heat sink with integrated heat pipes [17].

Figure 2.9: Air-cooled heat sink with integrated vapour chamber [18].

2.4.3 Water-glycol cooling

Even with heat pipes or vapour chambers, air cooling still has a significantly higher thermal resistance than liquid cooling. Liquid cooling enables the use of smaller power electronics and heat sinks. The heated liquid can subsequently be transported to another location where it can be effectively cooled by the ambient air. As such, liquid cooling can be regarded as a more effective way of heat spreading, allowing for both a lower local as lower total thermal resistance.

The most common liquid cooling method is to attach a cold plate to the baseplate of the power module, through which a water-glycol (WG) mixture flows. The interface layer between the baseplate and cold plate causes a thermal interface resistance due to the air left in between the two solid faces. A thermal interface material (TIM) is typically used to reduce this thermal interface resistance. Some examples of TIMs are thermal grease, thermal pads and metallic foils.

In the HIPERCOOL project, a cold plate (figure 2.10) was designed and its thermal performance was experimentally characterised [19]. When using the baseplate area as reference area, the cold plate has a maximal effective heat transfer coefficient equal to 7036 W/m²K. However, the TIM material in between the cold plate and baseplate has a significant effect on the total thermal resistance, resulting in a total effective heat transfer coefficient of 2814 W/m²K.

To eliminate the TIM thermal resistance, direct contact cooling of the baseplate with the coolant is tested. The most straight-forward method is to pump the coolant directly across the baseplate of the power module. Nonneman et al. [20] experimentally tested this approach and determined a maximal effective heat transfer coefficient equal to 7880 W/m²K. A drawback of this method is that the cooling is non-uniform: heat transfer coefficients are higher at the inlet than at the outlet. Turbulators inserts (such as the 'ShowerPower' concept [21]) can help alleviate this problem.



Figure 2.10: Power module with cold plate [19].

In order to further reduce the thermal resistance, the baseplate of the module can be manufactured containing fins. Fins increase the heat transfer area and break up thermal boundary layers which increases the heat transfer coefficient. An example of a power module with pin fins integrated in the baseplate is shown in figure 2.11. When recalculated to the rectangular baseplate area, this cooling method results in very high effective heat transfer coefficients. In the HIPERCOOL project [19], effective heat transfer coefficients up to 46000 W/m²K were measured for the pin fin heat sink. Reeves et al. [22] tested a baseplate with integrated straight fins and measured a maximal effective heat transfer coefficient of 24000 W/m²K.



Figure 2.11: Power module with pin fins integrated in the baseplate [23].

2.4.4 Oil cooling

Oil cooling can be used with similar heat sinks and techniques as for water-glycol cooling. However, the properties of oil are inferior to those of water-glycol mixtures when regarding thermal performance (lower thermal conductivity, lower specific heat capacity, higher viscosity). In the HIPERCOOL project [19], a pin fin heat sink was tested with automatic transmission fluid as coolant. This resulted in a maximal effective heat transfer coefficient of 6355 W/m²K. This is about seven times lower than the heat transfer coefficient of the same geometry with water-glycol, while a higher pressure drop requires more pumping power.

Most oils have the advantage of being dielectric, allowing direct contact cooling with the current carrying components of the power module. This removes the thermal resistance of the insulating ceramic substrate and the substrate to baseplate bonding layer. However, this concept has not been tested, so no indications of the thermal performance are as yet available.

2.4.5 Two-phase cooling

Two-phase cooling has the potential for higher heat fluxes at lower temperature differences when comparing to single-phase cooling. Flow of the two-phase fluid at the heated surface can occur naturally by the density difference of liquid and vapour or flow can be induced by a pump. These mechanisms are typically referred to as pool boiling and flow boiling respectively.

Although two-phase cooling entails high heat transfer rates, a major risk is associated to it as well. When the heat flux increases to a certain limiting value, a vapour film is formed over the entire cooled surface [24]. This vapour film acts as an insulator, drastically reducing the heat transfer rates and thus causing overheating of the power module. The heat transfer region of interest is thus below the critical heat flux value.

Similarly as for water-glycol and oil cooling, two-phase cooling can occur in cold plates or in direct contact to the baseplate. Aranzabal et al. [25] experimentally compared single-phase and two-phase cold plate cooling using a water-glycol mixture and refrigerant R134a respectively. The heat transfer coefficients determined for two-phase cooling were up to 60% higher than those for single-phase cooling.

The National Renewable Energy Laboratory (NREL) tested direct baseplate pool boiling cooling of a power module using refrigerants HFE-7100 and R134a (figure 2.12) [26]. Heat transfer coefficients up to 13000 W/m²K for HFE-7100 and 50000 W/m²K for R134a were measured for boiling on a plain copper baseplate. Higher values were achieved by using engineered microporous and nanostructured surfaces. The critical heat flux values were 210 kW/m² for HFE-7100 and 600 kW/m² for R134a.

Barnes and Tuma [27] tested pool boiling cooling of an IGBT module, where both the IGBT and baseplate were in direct contact with the fluid. The fluid under test was HFE-7000. The module was tested with a standard copper baseplate and with a microporous coated baseplate. Heat transfer coefficients and critical heat



Figure 2.12: Power module cooled by pool boiling [26].

fluxes were reported based on the IGBT die area and not on the baseplate area, so these cannot be directly compared to those cited above. Barnes and Tuma concluded that the total thermal resistance of a directly cooled IGBT module could be 50% to 70% lower than that of a traditional liquid cooled module.

For designing two-phase cooling systems, the component temperature should be predicted from the component heat flux and coolant saturation temperature. For pool boiling heat transfer, the heat flux (component heat dissipation per area) is typically correlated to the surface superheat temperature, which is the difference of the surface (component) temperature and the saturation (coolant) temperature. This relation is typically referred to as the boiling curve and is thus crucial to be able to predict boiling heat transfer. A detailed analysis of pool boiling heat transfer is given in chapter 3.

When the boiling curve is known, boiling heat transfer can be implemented in thermal models of power modules to assess the cooling potential. This can be done in several ways, ranging from least to most accurate and least to most complex. The simplest model is to derive the boiling thermal resistance (which is dependent on the heat flux) and to add this to the internal power module thermal resistance, which is typically given in datasheets. A more complex and accurate lumped parameter can be made, where the conduction problem in the power module can be described in more detail. In this case, the boiling thermal resistance can be distributed in several separate thermal resistances based on a discretization of the baseplate area. If even more detail is needed, a three-dimensional finite element simulation of conduction in the module can be made. At the interface of the baseplate and fluid, a boundary condition should be imposed such that the heat flux and temperature are related as the boiling curve describes.

2.4.6 Comparison

Nonneman et al. [28] made a comparison of seven techniques for cooling of power modules: air-cooled heat sink, air-cooled heat sink with integrated heat pipes, aircooled heat sink with integrated vapour chamber, water-glycol cold plate, direct baseplate water-glycol cooling, integrated pin fin baseplate water-glycol cooling and pool boiling baseplate cooling. The maximal power dissipation (without overheating of the power module) was calculated based on analytical models. The air-cooled heat sink performed the worst, followed by the integrated vapour chamber air-cooled heat sink and the integrated heat pipes air-cooled heat sink. Next in line was the cold plate, which had almost equal performance to the pool boiling baseplate cooling. In this analysis, the two-phase cooling method was limited by the critical heat flux, not by the thermal resistance. With direct liquid baseplate cooling, almost double the amount of heat could be dissipated and for integrated pin fin baseplate cooling, the maximal heat dissipation was almost thrice as high.

Table 2.1 gives an overview of heat transfer coefficients of the water-glycol and two-phase cooling methods described in the previous sections. The direct baseplate cooling methods clearly outperform cold plate cooling. Two-phase pool boiling cooling of the baseplate seems to have potential, although there is a big difference between the two reported fluids. For the two-phase methods, the critical heat flux is also an important limiting factor, next to the potentially achievable heat transfer coefficients. This study aims to further investigate the potential of pool boiling cooling.

Coolant	Method	Heat transfer coefficient
Water-glycol	Cold plate [19]	2814 W/m ² K
Water-glycol	Flat baseplate [20]	7880 W/m ² K
Water-glycol	Pin fin baseplate [19]	46000 W/m ² K
Water-glycol	Straight fin baseplate [22]	24000 W/m ² K
HFE-7100	Flat baseplate (pool boiling) [26]	13000 W/m ² K
R134a	Flat baseplate (pool boiling) [26]	$50000 \text{ W/m}^2\text{K}$

Table 2.1: Overview of heat transfer coefficients (wall to coolant) reported in literature, based on baseplate area, for different cooling methods.

2.5 Coolant selection

2.5.1 Requirements

Any fluid can act as a pool boiling fluid under the right conditions. However, even when not taking into account the thermal performance of the fluid, several requirements need to be met for safety and environmental concerns. The safety concerns of most importance are the flammability, toxicity, pressure and electrical resistivity of the fluid. Ozone depleting potential and global warming potential are environmental concerns to be taken into account.

Flammability

Although there are no strict regulations regarding flammable substances in mobile applications, in practice flammable coolants are not used as possible failures in the electronics would lead to large fires. ASHRAE (American Society of Heating, Refrigeration and Air-Conditioning Engineers) Standard 34 gives a classification of refrigerants according to their flammability based on three classes: no flame propagation (1), lower flammability (2) and higher flammability (3) [12]. In a recent update the subclass 2L was added for refrigerants that burn very slowly (burning velocity lower than 10 cm/s). The fluids under consideration as coolant in mobile application are preferably in class 1, although subclass 2L is also acceptable.

Toxicity

ASHRAE Standard 34 also provides a classification of toxicity for refrigerants in two classes, based on the occupational exposure limit (OEL) [12]. This is the time-weighted average concentration to which nearly all workers can be repeatedly exposed without adverse effect during a normal eight-hour workday and a fortyhour work week. If OEL is higher than or equal to 400 ppm, a refrigerant is classified in class A, otherwise it is categorized in class B. Obviously class A is preferred for coolants.

A schematic overview of the flammability and toxicity classification is shown in figure 2.13. Applicable coolants should fall under class A1 or A2L.

Pressure

It is preferred to have a working pressure close to atmospheric pressure. This will reduce the risk of fluid leakage and requires less heavy tubing. When assuming the fluid is at its saturation state, the working pressure will be linked to the working temperature of the fluid. Typical coolant temperatures for the cooling of electric drivetrains (excluding the battery) are around 60 °C. During longer idle times of the drivetrain, the temperature of the coolant will drop to atmospheric temperature. The saturation temperature of the fluid (T_{sat}) at atmospheric pressure is thus preferably in the range between 20 °C and 60 °C.

Electrical resistivity

Advanced cooling strategies for drivetrains are moving towards direct contact cooling of the current carrying parts. This removes several thermal resistances due to

- 1		SAFETY GROUP			
	Higher Flammability	A3	B3		
R M E A B S I I L G T Y	Lower	A2	B2		
	Flammability	A2_L*			
	No Flame Propagation	A1	B1		
		Lower Toxicity	Higher Toxicity		
* 401		INCREASIN	IG TOXICITY		

' A2L and B2L are lower flammability refrigerants with a maximum burning velocity of \leq 3.9 in./s (10 cm/s).

Figure 2.13: Safety group classifications according to ASHRAE Standard 34 [12].

insulating materials, but requires the use of a dielectric coolant. Although the twophase cooling of a power module baseplate does not require a dielectric fluid, it is an advantage to have a dielectric fluid to remove the risk of leak currents or short circuits resulting from coolant leaks.

Ozone depleting potential

The ozone layer in the stratosphere protects the Earth from harmful UV radiation. Some substances, when released in the environment, migrate to the stratosphere and react with ozone, causing a chain reaction which reduces the ozone concentration. An index called the ozone depleting potential (ODP) was introduced to quantify this effect. The European Union implemented a total ban on fluids which deplete ozone (ODP larger than 0) [29].

Global warming potential

Emissions of greenhouse gasses cause global warming effects and climate change. The global warming potential (GWP) of a fluid is determined as the 100 year warming potential of one kilogram of that fluid relative to one kilogram of carbon dioxide. According to the Kigali amendment to the Montreal protocol, the use of high-GWP fluids should be gradually phased out in the coming decades [30]. The European Union further decided that as of 2017, the GWP of fluids used in new vehicles cannot be higher than 150 [31].

2.5.2 Types of fluids

Over time, different types of fluids have been used in two-phase systems. This section gives an overview of these fluids. In combination with the criteria described above, a selection can be made for fluids in two-phase cooling systems.

Natural refrigerants

The first fluids to be used were the natural refrigerants, fluids that can be found in nature. Water is the most widely used fluid in pool boiling research and has excellent heat transfer properties. However its boiling point at atmospheric pressure is out of the ideal range, and as it freezes at 0 °C, additives have to be used to avoid damage due to freezing. Furthermore, water is not a dielectric and can thus not be used for direct-contact cooling. Ammonia is another possible natural refrigerant, but it is toxic and very corrosive. Hydrocarbons are used for two-phase systems, however their high flammability makes them inapplicable as coolant. Carbon dioxide requires high pressures to be in a liquid state, as do all gases that are used as cryogens (f.e. nitrogen, helium), which is not desirable in an electronics cooling system. In conclusion, none of the natural refrigerants are preferable for two-phase electronics cooling.

CFC and HCFC

Chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) were the first synthetic fluids widely used as refrigerants. They were particularly interesting fluids due to their low toxicity, flammability and reactivity. CFC-11, CFC-12, HCFC-21, HCFC-22 and CFC-113 are examples of previously common fluids. Due to their contribution to the ozone layer depletion, these fluids have been phased out under the Montreal protocol [32].

HFC and PFC

Hydrofluorocarbons (HFCs) and perfluorocarbons (PFC) do not contain any chlorine atoms and therefore (mostly) do not contribute to ozone depletion. Widely used fluids are HFC-134a, HFC-404a, HFC-410a and PFC-72. The main drawback of these chemicals is their high global warming potential. The Kyoto protocol was an international treaty committing the agreeing parties to reduce the emissions of greenhouse gasses, among which HFCs and PFCs [33]. The Kigali amendment to the Montreal protocol is a recent international agreement which determines the time-line to gradually phase out the use of these fluids [30]. As these fluids all have a GWP higher than 150, they will not be considered for this study.

HFO and HCFO

Hydrofluoroolefins (HFOs) and hydrochlorofluoroolefins (HCFOs) were recently introduced as low-GWP alternative to HFCs. Due to the double carbon bond in the molecule, these fluids are more reactive and therefore have shorter atmospheric lifetimes. HFO-1234yf is the most commonly studied fluid and already adopted for air-conditioning in certain mobile applications.

HFE and FK

Mostly focused at immersion electronics cooling, hydrofluoroethers (HFEs) and fluoroketones (FKs) have been introduced in the market to replace PFCs in order to reduce global warming effects. HFEs and FKs have larger molecular weights than the previously mentioned fluids and as a result they do not remain in the atmosphere but are absorbed into the ground. Among these fluids, HFE-7100 and FK-649 have been most commonly featured in studies.

2.5.3 Comparison

Only HFOs, HCFOs, HFEs and FKs are able to meet all requirements for use as a mobile drivetrain coolant. Table 2.2 gives an overview of the fluids in these categories which are available on the market. The names are based on the abbreviations of the type of fluid and the numbering given by ASHRAE [12] for HFOs and HCFOs and the numbering of 3M [34] for HFEs and FKs. The HCFOs still have some ozone depleting potential and are therefore not chosen. Of the HFOs, only HFO-1336mzz(Z) has a boiling point at atmospheric pressure in the range that is desired, the others require high pressures in the coolant circuit. Most of the HFEs have too high GWP (> 150) to be used in mobile applications. HFE-7200 and HFE-7500 have low GWP values, but they are slightly toxic and their boiling points are too high when comparing to the desired range. Overall, HFO-1336mzz(Z) and FK-649 are the most suited candidates. In this study, FK-649 is chosen as coolant.

2.6 Conclusions

Effective cooling of electric drivetrains is key for improving power density. Pool boiling cooling has the potential to be a more effective cooling method than waterglycol cooling, due to the higher heat transfer coefficients and potential for direct contact cooling of active components. A coolant for testing two-phase cooling of

Fluid	ODP	GWP	Safety ^a	Dielectric	$\mathbf{T_{sat}}^{b}$
HCFO-1224yd(Z)	pprox 0	<1	A1	Yes	15 °C
HCFO-1233zd(E)	pprox 0	1	A1	$N.A.^{c}$	18 °C
HFO-1234yf	0	<1	A2L	Yes	-30 °C
HFO-1234ze(E)	0	<1	A2L	$N.A.^{c}$	-19 °C
HFO-1336mzz(E)	0	18	A1	$N.A.^{c}$	8 °C
HFO-1336mzz(Z)	0	2	A1	Yes	33 °C
HFE-7000	0	530	A1	Yes	34 °C
HFE-7100	0	320	A1	Yes	61 °C
HFE-7200	0	55	B1	Yes	76 °C
HFE-7300	0	200	A1	Yes	98 °C
HFE-7500	0	90	B1	Yes	128 °C
HFE-7700	0	420	A1	Yes	167 °C
FK-649	0	<1	A1	Yes	49 °C

Table 2.2: Overview of possible two-phase coolants and their properties[12, 34].

^a according to ASHRAE Standard 34 [12]

^b evaluated at 1.013 bar

 c no data available

power electronics, FK-649, was chosen based on several practical requirements. The following chapter will elaborate on the studies on pool boiling heat transfer available in scientific literature.

3

Nucleate pool boiling heat transfer

3.1 Introduction

This chapter gives an overview of the present understanding of nucleate pool boiling heat transfer. Section 3.2 describes the fundamental processes related to nucleate pool boiling. Next, an overview of the commonly used correlations to predict nucleate pool boiling heat transfer is given. All parameters affecting heat transfer rates are discussed in section 3.4. The phenomenon of critical heat flux is examined in the next section. Research on the fluid chosen for this study, FK-649, is summarized in section 3.6. To end this chapter, a list of research questions which will be answered in this work is given.

3.2 Fundamentals

3.2.1 Boiling curve

The boiling curve is the relation between surface heat flux and surface superheat temperature. The surface superheat temperature is defined as the difference between the surface temperature and the fluid saturation temperature. In 1934, Nukiyama [35] was the first to theorize and experimentally determine the shape of the boiling curve. He conducted experiments with metal wires immersed in water. Current was sent through the wire, thereby causing the water to boil. Both the heat dissipation and wire temperature were measured. The results of these experiments are shown in figure 3.1.



Figure 3.1: Experimental boiling curve for a platinum wire submerged in water (surface heat flux as a function of surface superheat temperature) [35].

Figure 3.2 shows the theoretical boiling curve as originally proposed by Nukiyama. A similar but more descriptive figure of the boiling curve is shown on figure 3.3. In the following paragraphs references will be made to points on this graph (figure 3.3) with letters A to F. Several regimes can be distinguished with increasing heat flux: single-phase natural convection, nucleate boiling, transition boiling and film boiling. The transition heat fluxes between the four regimes are called the onset of nucleate boiling (A), the critical heat flux (D) and the Leidenfrost point (E).

Single-phase natural convection

When heat is transferred from a solid surface to a liquid at saturation temperature and at low heat fluxes, this will occur by single-phase natural convection. The liquid will overheat but will not yet start to boil, as some superheating is required to overcome surface tension forces and bubble nucleation to start. As this regime is characterised by low heat transfer rates, it is not of particular interest for power electronics cooling.



Figure 3.2: Typical boiling curve (surface heat flux as a function of surface superheat temperature) [35].



Figure 3.3: Typical boiling curve (surface heat flux as a function of surface superheat temperature) [36].

Nucleate boiling

From a certain heat flux, bubbles will start to form (nucleation) and the surface temperature decreases due to the increased heat transfer rate related to boiling (point A' to A"). This point is called the onset of nucleate boiling (ONB). When increasing the heat flux further, more nucleation sites become activated and the temperature of the surface slightly increases. This region is sometimes referred to as partial nucleate boiling. From a certain heat flux (point B in figure 3.3), all possible nucleation sites are activated and bubbles start merging and forming jets. This regime is usually referred to as fully developed nucleate boiling. By further increasing the heat flux, more of the surface is covered in vapour and the surface temperature starts to increase at a higher rate with the increase of the heat flux (point C). The region between point C and D is sometimes referred to as the partial dryout region. At even higher heat fluxes, the boiling curve reaches a maximum (point D). This is caused by the formed vapour not being able to evacuate from the heated surface, thereby forming an insulating vapour layer over the surface. The maximal heat flux has several terms which are used to describe it: peak heat flux, boiling crisis, burnout, departure from nucleate boiling (DNB) and critical heat flux (CHF). In this work it is chosen to use the term critical heat flux or CHF as the standard term.

Transition boiling

When the heat flux is increased beyond the critical heat flux, the transition boiling regime starts. In a standard boiling heat transfer experiment, the surface is heat flux controlled, for example the wires in the experiments of Nukiyama [35]. Due to the drop in heat flux after the critical heat flux, heat will be transferred mostly to heating up the surface, where the transitional regime is practically bypassed (to point F) and film boiling occurs. This bypassing is accompanied by a temperature increases of several 100 °C, which melted most of the wires. With a temperature controlled surface, the transitional regime can be studied and it is characterised by alternating vapour and liquid contact with the heated surface.

Film boiling

The film boiling regime starts from the Leidenfrost point, which is also called the minimal heat flux. In 1756, Leidenfrost observed that from a certain surface temperature, liquid water droplets are supported by a stable water vapour film and do not come into contact with the surface. This phenomenon also happens in pool boiling (which was not studied by Leidenfrost but identified later), where a vapour film is formed which covers the entire heated surface. When the heat flux is increased even further, radiation starts to become the dominant heat transfer mechanism due to the high surface temperatures.

3.2.2 Heat transfer mechanisms

In the previous section it is shown that the nucleate boiling regime is that of interest for electronics cooling. This section goes more into detail on the mechanisms of heat transfer corresponding to this regime. A schematic of the different heat flux mechanisms is given in figure 3.4. Although the relative importance of the different mechanisms is still under discussion, it is generally agreed that nucleate boiling heat transfer consists of a combination of the following processes [37]:

Natural convection

At locations with no active nucleation site, natural convection will occur induced by buoyancy of the heated liquid (q_{nc} in figure 3.4). This mechanism is typically only important at low heat fluxes.

Microlayer evaporation

When a bubble is formed and growing, a three-phase contact line and liquid microlayer are present. Conduction through this microlayer and consequent evaporation at the liquid-vapour interface $(q_{ml} \text{ in figure 3.4})$ is an important factor in the total heat transfer, of which the contribution to the total heat flux increases with increasing heat flux. Heat transfer by evaporation at the three phase contact line $(q_{cl} \text{ in figure 3.4})$ is sometimes regarded as a different mechanism, here it is included in the microlayer evaporation mechanism.

Quenching or transient conduction

After departure of a bubble, relatively cooler bulk liquid moves to the location where the bubble departed. This causes quenching of the superheated surface and thus transient conduction through the liquid (q_{tc} in figure 3.4).

Microconvection or enhanced convection

The movement of the bubbles (growing and departing) causes perturbations of the liquid adjacent to the bubbles, thereby increasing the convective heat transfer (q_{mc} in figure 3.4). The effect of perturbations of the liquid layer by the bubble is sometimes included in correlations for the quenching mechanism.

Dry spot heat transfer

Heat transfer can occur directly from the surface to the vapour in contact with the surface. The contribution of this mechanism of heat transfer to the total heat transfer is usually insignificant for nucleate boiling and is therefore mostly neglected in models.

Heat transfer can also occur between superheated liquid and the bubble cap at the liquid-vapour contact surface (q_{sl} in figure 3.4). This will cause evaporation



a. Heat transfer mechanisms during bubble growth.



b. Heat transfer mechanisms during bubble departure.

Figure 3.4: Nucleate boiling heat transfer mechanisms [37].

and further growth of the bubble and can thus also influence the other heat transfer mechanisms. However, since the heat first needs to be transferred from the wall to the liquid (through natural convection, quenching or microconvection) and is therefore not directly transferred from the heated surface, this mechanism is not included in the above list.

More information on the contributions of the different heat transfer mechanisms to the total heat transfer would give more insights into the fundamental basics upon which correlations can be based. The contributions of the different mechanisms or heat flux partitioning have gathered plenty of debate [37, 38]. Many different authors proposed models taking into account empirically measured parameters to determine the different heat fluxes. Although these models are thus not fully predictive, they can give additional insights in the relative importance of the different mechanisms.

Mikic and Rohsenow [39] first suggested a model which only takes into account natural convection and transient conduction. This analysis was done based on the partial nucleate boiling regime.

Judd and Hwang [40] continued on this approach but added a term for micro-

layer evaporation. A plot of the predicted heat flux as a function of the measured heat flux is shown in figure 3.5. The three contributions to the total heat transfer are shown in this plot: below the microlayer evaporation, in the middle natural convection and on the top transient conduction (which is labelled nucleate boiling in the figure). In this model, microlayer evaporation has a significant contribution, making up about 60% of the total heat flux at the highest heat flux shown on the figure (100 kW/m²).



Figure 3.5: Predicted heat partitioning (y-axis) with the Judd and Hwang model compared to the measured heat flux (x-axis) [40].

Paul and Abdel-Khalik [41] developed a similar model by partitioning the heat flux in three parts: latent heat transport (evaporation), natural convection and enhanced convection. The contributions of the three parts are shown graphically in figure 3.6. Although modelled differently, it is clear these parts follow the same trend as their counterparts in the Judd and Hwang model.



Figure 3.6: Relative contributions of different heat fluxes during nucleate boiling according to Paul and Abdel-Khalik [41].

Stephan and Hammer [42] propose a model assuming heat transfer is dominated by evaporation of the thin liquid meniscus at the three-phase contact line. As this three-phase contact line covers a rather small fraction of the boiling surface, a very high local heat flux is required for this assumption to be true. Indeed, the model predicts local heat fluxes up to 100 times the critical heat flux.

When comparing the measured boiling curves to the predicted ones, all of the previously mentioned models fare reasonably well, even though they are based on different assumptions. To gain more insights into which assumptions may be correct, experimental techniques directly measuring the heat flux partitioning are required.

In the last two decades, measurement techniques were significantly improved. This allowed for spatially and temporally distributed temperature measurements of a boiling surface. From this measured temperature field, the surface heat flux can be determined by locally solving the energy balance of the heated surface, taking into account sensible heating of the surface, conduction in the lateral directions, the heat dissipated within the heater and possible heat losses to the back of the heater. Wagner and Stephan [43] used this approach with measurements of an infrared camera at the rear side of a heating foil. Figure 3.7 shows the analysis for one timestep in the early growth period of the bubble. The high heat flux related to microlayer evaporation is clearly visible from the band on the heat flux distribution plot. With a high-speed camera, also the bubble growth was monitored. They analysed the bubble volume (V_{bub}) and the corresponding heat transfer required for the bubble growth (Q_{bub}) . They compared this to the heat transfer integrated over the microlayer evaporation region (Q_{mic}) . The results are shown in figure 3.8. It is clear that microlayer evaporation accounts for less than 50% of the total instantaneous heat transfer and for about 25% of the total integrated heat transfer. This indicates that the majority of the heat transfer to the bubble must have come from the superheated liquid layer, as this is the only other source (neglecting dry spot heat transfer, which is a valid assumption when considering the heat flux distributions). An important side note to make, is that these measurements are done at low heat fluxes (partial nucleate boiling). This was done so that only one nucleation site was active, which enabled more reliable and reproducible analyses. However, the contributions of the different heat transfer mechanisms can be different at higher heat fluxes.

Gerardi et al. [44] used a similar setup with infrared thermometry and highspeed video for pool boiling of water. They determined an equivalent bubble radius by integrating the surface heat flux, assuming a spherical bubble shape and that all heat is used for evaporation. In their analysis, the measured bubble radius for the high-speed camera and the calculated equivalent radius from the instantaneously measured heat flux correspond very well (figure 3.9), indicating that bubble growth is directly fueled by wall to bubble heat transfer. As these results differ from those of amongst others Wagner and Stephan [43], they conclude that the relative importance of microlayer evaporation to heat transfer from the superheated liquid presumably depends on the fluid, surface and heat flux. Based on the measured bubble departure diameters and frequency, the heat flux is partitioned in evaporation (q_e) , quenching (q_q) and natural convection (q_c) as shown in figure 3.10. In this modelling, the quenching heat flux is dominant for the entire nucleate boiling region. As bubble departure diameter and frequency become harder to measure for higher heat fluxes due to large vapour fractions near the surface, bubble coalescence and irregular bubble shapes, determining evaporation heat flux with this method is not very accurate at high heat fluxes. Furthermore, possible condensa-



Figure 3.7: (a) Bubble shape, (b) IR temperature field (°C), (c) Temperature line profile, (d) Heat flux distribution (W/m^2) [43].



Figure 3.8: Bubble volume (V_{bub}) , latent heat flux (Q_{bub}) and microlayer evaporation heat flux (Q_{mic}) during bubble growth [43].

tion at the bubble cap is not taken into account. Quenching heat transfer is based on the model of Mikic and Rohsenow [39], which has been shown to overestimate quenching heat transfer rates by taking a too large area of influence [45]. It is therefore questionable if the low contribution of microlayer evaporation at high heat fluxes is valid.



Figure 3.9: Measured bubble radius and equivalent bubble radius calculated from heat flux during bubble growth [44].



Figure 3.10: Measured and predicted boiling curves, showing different heat flux contributions [44].

In his review on nucleate pool boiling heat transfer mechanisms, Kim [37] concludes that transient conduction and/or microconvection are the dominant mechanisms by which heat is transferred. Based on a multitude of experimental researches, he concludes that microlayer evaporation accounts for less than 25% of the total heat transfer. Although the model of Mikic and Rohsenow uses transient conduction as the main mode of heat transfer, experimental results show that this effect is not limited to the time after bubble departure. Movement of the bubble during growth also causes large heat transfer rates by perturbing the liquid layers adjacent to it. The effect of heat flux on heat flux partitioning is not discussed. It is therefore unclear if these conclusions are valid for all nucleate boiling regimes.

Researchers for the Massachusetts Institute of Technology have developed an experimental setup to directly measure heat flux partitioning in boiling heat transfer [46, 47]. It consists of high-speed video at the two lateral sides and a high-speed infrared camera to analyse the temperature of the boiling surface. Surface heat fluxes are determined through a coupled radiation-conduction calibration model of the heater and boiling surface. Richenderfer et al. [46] present measurements of heat flux partitioning of flow boiling of water. Although flow boiling is distinctly different from pool boiling due to the forced convective heat transfer, it has been shown that nucleate heat transfer behaves similarly in both situations. The total heat flux is partitioned in dry spot heat transfer (q_{dry}) , evaporation (q_{evap}) , forced convection (q_{fc}) and quenching (q_q) . This has been done by analysing the dry, microlayer and liquid covered areas. The total heat flux to the liquid covered areas (q_{sp}) is separated in forced convection and quenching heat flux by analysing forced convective heat transfer at low heat fluxes without bubble nucleation. The results are shown in figure 3.11. Dry spot heat transfer contributes very little over the entire heat flux range. Forced convection is dominant at low heat fluxes, while quenching and evaporation become more important at intermediate heat fluxes. At the highest heat fluxes, heat transfer through evaporation is larger than singlephase heat transfer. About half of the heat is transferred by evaporation at this point. On the same measurement device, Su et al. [47] report heat flux partitioning measurements for flow boiling on a different surface. Figure 3.12 shows the measured single-phase heat flux (SP) and the total heat flux (HF). A stochastic model is also shown, dividing the heat flux in evaporation (EV), quenching (QC) and forced convection (FC). Similar partitioning as in figure 3.11 is found, with high evaporative heat flux contributions in the high heat flux regions.

An overview of the different models and the heat transfer mechanisms each of them takes into account is given in table 3.1. Some general conclusions can be drawn about the nucleate pool boiling heat transfer mechanisms. Dry spot heat transfer is negligible for all heat fluxes. Natural convective heat transfer is only significant for low heat fluxes (partial nucleate boiling). Single-phase heat transfer (transient conduction and/or enhanced convection) is the dominant heat transfer



Figure 3.11: Measured heat flux partitioning for flow boiling heat transfer [46].



Figure 3.12: Measured and modelled heat flux partitioning for flow boiling heat transfer [47].

mechanism in the intermediate heat flux region (fully developed nucleate boiling). The importance of microlayer evaporation is still under debate. This mechanism seems to become more important with increasing heat flux, however it is still unclear whether this mechanism is dominant in the partial dryout region. More measurements on spatial and temporal distribution of temperature and heat flux for nucleate pool boiling in the different regimes might give more insights on the heat flux partitioning. For deriving nucleate pool boiling heat transfer correlations, both (microlayer) evaporation as single-phase heat transfer are possible valid bases.

Table 3.1: Heat transfer mechanisms included in different models. NC =natural convection, ML = microlayer evaporation, QC = quenching, MC =microconvection, DS = dry spot, FC = forced convection. Y = mechanism included, Y- = mechanism included but accounts for less than 10% of total heat transfer, N = mechanisms not included.

Author(s)	NC	ML	QC	MC	DS	FC
Mikic and Rohsenow [39]	Y	Ν	Y	Ν	Ν	/
Judd and Hwang [40]	Y	Y	Y	Ν	Ν	/
Paul and Abdel-Khalik [41]	Y	Y	Ν	Y	Ν	/
Stephan and Hammer [42]	Ν	Y	Y	Ν	Ν	/
Gerardi et al. [44]	Y-	Y-	Y	Ν	Ν	/
Richenderfer et al.* [46]	/	Y	Y	Ν	Y-	Y
Su et al.* [47]	/	Y	Y	Ν	Ν	Y

* Models are for flow boiling and thus include forced convection.

3.2.3 Non-dimensional groups

Non-dimensional groups or dimensionless numbers have been proven to be effective in characterizing transport phenomena in fluid mechanics and heat transfer. These groups or numbers are usually the ratio of different kinds of diffusivities (momentum, thermal), forces (inertia, viscosity, surface tension, gravity) and flows (heat, mass). Also for boiling heat transfer, many correlations are proposed based on non-dimensional groups. The most important and commonly used ones are therefore defined here.

Many authors draw a parallel between single-phase heat transfer and nucleate boiling heat transfer. For forced single-phase heat transfer, it is well-known that three non-dimensional groups determine the heat transfer rate: the Nusselt number Nu, the Reynolds number Re and the Prandtl number Pr.

The Nusselt number is the ratio of the convective heat transfer to the conductive heat transfer, defined as:

$$Nu = \frac{\dot{q}L}{k\Delta T} \tag{3.1}$$

where \dot{q} is the surface heat flux, L is a characteristic length, k is the fluid thermal conductivity and ΔT is the temperature difference between the wall and the fluid.

The Reynolds number is the ratio of the inertial force to the viscous force:

$$Re = \frac{uL}{\nu} = \frac{GL}{\mu} \tag{3.2}$$

with u the fluid velocity, ν the fluid kinematic viscosity, G the mass flux and μ the fluid dynamic viscosity.

The Prandtl number is the ratio of the momentum diffusivity to the thermal diffusivity:

$$Pr = \frac{\nu}{\alpha} = \frac{\nu\rho c_p}{k} \tag{3.3}$$

with α the fluid thermal diffusivity, ρ the fluid density and c_p the fluid specific heat capacity.

In boiling heat transfer, properties related to the phase change will also effect the heat transfer. A commonly used dimensionless group is the Jakob number Ja, which is the ratio of the sensible heat to the latent heat:

$$Ja = \frac{c_{p,l}\Delta T}{h_{lv}} \tag{3.4}$$

where h_{lv} is the enthalpy of vaporization and the index l indicates the liquid phase.

In some cases, the Jakob number is defined on a volumetric basis taking into account the ratio of the density of the liquid and vapour phase. Here this number is called the modified Jakob number Ja':

$$Ja' = \frac{\rho_l c_{p,l} \Delta T}{\rho_v h_{lv}} \tag{3.5}$$

with the index v indicating the vapour phase.

For the nucleate boiling phenomena, the density difference between liquid and vapour will cause gravitational effects to play a role. The Archimedes number Ar is the ratio of the gravitational forces to the viscous forces:

$$Ar = \frac{gL^3(\rho_l - \rho_v)}{\rho_l \nu^2} \tag{3.6}$$

where g is the gravitational acceleration.

Besides gravity, surface tension will be an additional factor to take into account for boiling phenomena. The Weber number We is the ratio of the inertia to the force due to surface tension:

$$We = \frac{\rho u^2 L}{\sigma} = \frac{G^2 L}{\rho \sigma}$$
(3.7)

with σ the surface tension.

Many other dimensionless numbers can be defined and could be relevant to nucleate boiling heat transfer. As will be seen in section 3.3, there is not yet a consensus on which groups or properties play a role in the heat transfer phenomena.

3.2.4 Bubble nucleation, growth and departure diameter

A lot of interesting work has been done on the theory of bubble dynamics. This includes many topics such as bubble nucleation, growth, deformation, merger, departure diameter, departure frequency, formation waiting times and nucleation site densities. Only three of these topics will be briefly touched upon here: bubble nucleation, bubble growth and bubble diameter. The derivations surrounding these three phenomena will give some more insights in the bases of some correlations. For more information on the other topics, which are also interesting but not required to understand the following sections, the author refers the reader to the works of Stephan [48], Hewitt [38] and Brennen [49].

Bubble nucleation

It can be shown theoretically that a certain liquid superheat (temperature above saturation temperature) is required for the nucleation of bubbles. The simplified outline of that analysis is given here, for the detailed analysis the reader is referred to [48].

To analyse bubble nucleation, a spherical vapour bubble with radius r in equilibrium with the surrounding liquid is considered. The force balance on the vapourliquid interface (see also figure 3.13) results in:

$$p_{v,r} - p_{sat} = \frac{2\sigma}{r} \tag{3.8}$$

Due to the surface tension effects, the pressure in the vapour bubble $p_{v,r}$ is larger than the overall system pressure p_{sat} . Due to the curvature of the vapourliquid interface, the saturated vapour pressure $(p_{v,r})$ is slightly lower than that which it would be at planar interface (p_v) . This effect can be approximated by the following equation:

$$p_{v,r} = p_v - \frac{\rho_v}{\rho_l - \rho_v} \frac{2\sigma}{r}$$
(3.9)

Combining equations 3.8 and 3.9 results in:

$$p_v - p_{sat} = \frac{\rho_l}{\rho_l - \rho_v} \frac{2\sigma}{r}$$
(3.10)



Figure 3.13: A vapour bubble in superheated liquid (left) and close-up of the vapour-liquid interface and the forces (in the plane) acting upon it (right).

The derivation of the saturation pressure to saturation temperature T_{sat} is given by the Clausius-Clapeyron equation:

$$\frac{dp_{sat}}{dT_{sat}} = \frac{h_{lv}}{T_{sat}(1/\rho_v - 1/\rho_l)}$$
(3.11)

Using linear approximation for small pressure differences, the following equation can be derived:

$$\frac{p_v - p_{sat}}{\Delta T_0} = \frac{h_{lv}}{T_{sat}(1/\rho_v - 1/\rho_l)}$$
(3.12)

where ΔT_0 is the temperature difference between the saturation temperature and superheated liquid temperature. The combination of equations 3.10 and 3.12 results in:

$$\Delta T_0 = \frac{2\sigma T_{sat}}{r\rho_v h_{lv}} \tag{3.13}$$

The value ΔT_0 is the minimal liquid superheat required for bubble nucleation in a liquid. This value goes to infinity for very small radius, indicating that homogeneous nucleation is very uncommon. Instead, nucleation occurs at so-called nucleation sites on the surface. On the surface, microcavities are present which always retain a small amount of vapour. These cavities provide locations where the equivalent radius is larger, therefore allowing bubbles to nucleate and grow at lower surface superheats. Although the nucleation normally occurs at the surfacefluid interface and thus this derivation is not appropriate for that situation, this value remains an interesting property which is frequently used in describing and correlating nucleate boiling heat transfer.

Bubble growth

Again for a vapour bubble surrounded by liquid, the growth of a spherical bubble is analysed. A detailed analysis is given by Brennen [49], the results are summarized here. Two limiting cases of bubble growth are observed: inertia-controlled growth and heat-transfer-controlled growth.

In the early stages, bubble growth is limited by inertia or how rapidly the growing bubble can push back the surrounding liquid. The radius of the bubble increases linearly with time t:

$$r(t) = \left(\frac{2}{3} \frac{\Delta T_s h_{lv} \rho_v}{T_{sat} \rho_l}\right)^{1/2} t \tag{3.14}$$

In this equation, ΔT_s is the superheat temperature, which is the difference between the superheated liquid (or surface) temperature and the saturation temperature.

In the later growth stage, bubble growth is limited by the heat transfer across the liquid-vapour interface and the radius increases with the square root of time:

$$r(t) = \frac{2\Delta T_s k_l}{h_{lv} \rho_v} \left(\frac{3t}{\pi \alpha_l}\right)^{1/2}$$
(3.15)

Departure diameter

The bubble departure diameter is the diameter of a vapour bubble when it detaches from the heated surface. The first reported correlation for the bubble departure diameter was by Fritz [50]. From analysing the forces of surface tension and gravity on a bubble attached to a surface, the following relation was derived for the departure bubble diameter D_b :

$$D_b = f(\beta) \sqrt{\frac{2\sigma}{g(\rho_l - \rho_v)}} \tag{3.16}$$

In this equation, β is the bubble contact angle and the values of $f(\beta)$ where tabulated by Fritz. It was later shown that this function had a simple linear approximation:

$$D_F = 0.851\beta \sqrt{\frac{2\sigma}{g(\rho_l - \rho_v)}} \tag{3.17}$$
This correlation for the bubble departure diameter D_F is commonly referred to as the Fritz correlation. β has to be evaluated in radians. Besides the Fritz bubble departure diameter, another commonly used characteristic length removes the influence of the contact angle and is named the capillary length or the Laplace constant L_c :

$$L_c = \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \tag{3.18}$$

More recent studies have shown that for larger surface superheat, other influences such as those of inertia and flow resistance can no longer be neglected. For correlations taking these effects into account, the reader is referred to the works of Kutateladze [51] and Stephan [48].

3.3 Heat transfer correlations

This section gives an overview of nucleate boiling heat transfer correlations that have been proposed in scientific literature. The goal of these correlations is to describe the nucleate pool boiling part of the boiling curve (which is the relation between the heat flux and the surface superheat temperature). The correlation should further take into account any parameters (fluid properties, surface properties, ...) which can influence the boiling curve (more information on the parameters can be found in section 3.4). As there are a lot of parameters used in the different correlations, figure 3.14 gives an overview of these parameters in the form of a mind map. This schematic can be consulted for the definition of the different symbols used.

The overview is by no means meant to be comprehensive. The correlations reported here are chosen on the basis of their use by other researchers. The correlations are covered chronologically with regard to the first publication that was found describing it. As it is possible that not all relevant publications were found with regard to the discussed correlations, the dates that are given between brackets serve as an indication of the publication of the correlation.

To make this overview, the author did not start from scratch. Many review papers have been published on nucleate boiling heat transfer and correlations. The works of Pioro [52], Forrest [53] and Bartle [54, 55] have served as thorough starting points, along with the correlations reported in the Heat Exchanger Design Handbook [56].

Many of these correlations are reported by the nucleate boiling heat transfer coefficient h_{nb} . It is the choice of the author to not use this notation in this doctoral dissertation. A heat transfer coefficient has little meaning when describing nucleate boiling heat transfer. The use of the heat transfer coefficient derives from



Figure 3.14: Mind map of all the parameters used in the cited nucleate boiling heat transfer correlations.

Newton's law of cooling, which states that the heat flux from a body is proportional to the temperature difference between the body and its surroundings. This holds true for forced single-phase convection (neglecting effects of temperature on fluid properties) and fairly true for natural single-phase convection (heat flux proportional to the temperature difference to the power 1.25 to 1.33). For nucleate boiling heat transfer, the heat flux varies with the temperature difference to the power 2.5 to 5. Since the heat transfer coefficient is therefore very much dependent on the heat flux or temperature difference, it loses its added value. It becomes an additional parameter which is redundant in correlations, as the parameters of significance are the heat flux and difference between the surface temperature and fluid saturation temperature. Occasionally it can be useful to determine a numerical value of the nucleate boiling heat transfer coefficient at a certain heat flux for comparison with heat transfer coefficients of other heat transfer mechanisms. If a correlation is described which originally featured the heat transfer coefficient, it is simply transformed to the ratio of the heat flux and surface superheat temperature.

3.3.1 Kruzhilin correlation (1947)

The first correlation to predict nucleate boiling heat transfer that was found, dates back to 1947 and was proposed by Kruzhilin [57]:

$$\frac{\dot{q}L_c}{k_l\Delta T_s} = 0.082 \left(\frac{\dot{q}h_{lv}}{gT_{sat}k_l}\frac{\rho_v}{\rho_l - \rho_v}\right)^{0.7} \left(\frac{T_{sat}c_{p,l}\rho_l\sigma}{h_{lv}^2\rho_v^2L_c}\right)^{1/3} Pr_l^{-0.45}$$
(3.19)

The characteristic length L_c is given by equation 3.18. When introducing the minimal superheat of equation 3.13 with the radius equal to twice the characteristic length, equation 3.19 reverts to:

$$\frac{\dot{q}L_c}{k_l\Delta T_s} = 0.082 \left(\frac{\dot{q}L_c}{k_l\Delta T_0}\right)^{0.7} \left(\frac{\rho_l c_{p,l}\Delta T_0}{\rho_v h_{lv}}\right)^{1/3} Pr_l^{-0.45}$$
(3.20)

The four dimensionless groups used are thus the Nusselt number, the minimal superheat rendered non-dimensional in a similar manner to the Nusselt number, the modified Jakob number using the minimal superheat and the liquid Prandtl number.

The author was not able to find the original paper in Russian describing the correlation. So no analysis can be made about the basis of the derivation of the correlations, the manner of determining the coefficients and the experimental data used to verify it.

3.3.2 Rohsenow correlation (1952)

One of the most well-known and earliest correlations for nucleate boiling heat transfer was proposed by Rohsenow [58] in 1952. Rohsenow argued that heat transfer in boiling is dominated by heat transfer directly from the surface to the liquid. The motion of the bubble causes agitation of the liquid phase which results in an increase of the surface-to-liquid heat transfer. With these assumptions in mind, he proposed a correlation based on a Nusselt and Reynolds number using a characteristic length related to the bubble size and the liquid's Prandtl number. The following is a summary of the derivations and findings of Rohsenow.

The bubble Reynolds number Re_b is defined as:

$$Re_b = \frac{G_b D_F}{\mu_l} \tag{3.21}$$

with G_b the mass flux of bubbles leaving the heated surface and D_F the bubble departure diameter according to Fritz. The mass flux of bubbles is determined as the product of the volume of a departing bubble, the vapour density, the number of nucleation sites per surface or nucleation site density N and the bubble departure frequency f:

$$G_b = \frac{\pi}{6} D_F^3 \rho_v N f \tag{3.22}$$

The mass flux of bubbles can be related to the heat flux through the bubbles \dot{q}_b with the latent heat of vaporization:

$$\dot{q}_b = h_{lv}G_b \tag{3.23}$$

The heat transfer to the bubbles is linked to the total heat transfer to the fluid based on two more observations. First, based on a study by Jakob [59], it is proposed that the frequency of bubble formation and the bubble departure diameter are inversely correlated (equation 3.24). Secondly, based on another study by Jakob [60], the number of nucleation sites per surface is directly proportional to the total heat transfer rate (equation 3.25).

$$f \sim \frac{1}{D_F} \tag{3.24}$$

$$N \sim \dot{q}$$
 (3.25)

By combining equations 3.22, 3.23 and 3.24, it can be shown that for a certain saturation pressure, the heat transfer to the bubbles is directly proportional to the nucleation site density. Combining these results with equation 3.25 leads to the conclusion that the total surface heat flux is proportional to the heat flux to the vapour bubbles leaving the surface. The surface heat flux can then be written as:

$$\dot{q} = C_q h_{lv} G_b \tag{3.26}$$

where C_q is a constant. Implementing equations 3.16 and 3.26 in equation 3.21 results in:

$$Re_b = C_R \frac{\dot{q}}{\mu_l h_{lv}} \beta \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}$$
(3.27)

with the new constant $C_R = \sqrt{2} 0.851/C_q$. The bubble Nusselt number Nu_b can be written as:

$$Nu_b = \frac{\dot{q}D_F}{\Delta T_s k_l} = C_n \frac{\dot{q}}{\Delta T_s k_l} \beta \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}$$
(3.28)

where another new constant is defined as $C_n = 0.851\sqrt{2}$.

Since both bubble Reynolds and bubble Nusselt number contain a term with the surface heat flux, a new dimensionless group is formed:

$$\frac{Re_b Pr_l C_q}{Nu_b} = \frac{c_{p,l} \Delta T_s}{h_{lv}} = Ja$$
(3.29)

This new dimensionless group boils down to the ratio of the sensible heat related to the surface superheat temperature and the latent heat of vaporization, also referred to as the Jakob number Ja. Subsequently, Rohsenow established a relation between Jakob number, bubble Reynolds number and Prandtl number based on experimental data with the fluids water, benzene, ethyl alcohol, n-pentane and the surfaces platinum, chromium and brass. Rohsenow removes the contact angle from the correlations, thereby neglecting effects of saturation pressure on the contact angle and lumping the effect of fluid and surface on the contact angle in the proportional constant. The best fit found and thus the proposed resulting correlation is:

$$\frac{c_{p,l}\Delta T_s}{h_{lv}} = C_{sf} \left[\frac{\dot{q}}{\mu_l h_{lv}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^{0.33} Pr_l^{1.7}$$
(3.30)

with C_{sf} a parameter dependent on the surface-fluid combination. The author suggested that the powers 0.33 and 1.7 might not be the best fit for all cases and as such the correlation was generalized to:

$$\frac{c_{p,l}\Delta T_s}{h_{lv}} = C_{sf} \left[\frac{\dot{q}}{\mu_l h_{lv}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \right]^r Pr_l^s$$
(3.31)

This correlation adds two new parameters r and s. Together with C_{sf} , these three parameters are dependent on the surface-fluid combination and have to be known to use the correlation to predict the boiling curve. These three parameters have been documented for a wide variety of surface-fluid combinations by Vachan et al. [61] and Pioro [62].

Vachan et al. [61] checked the assumption that r is equal to 0.33 for all surfacefluid combinations. Based on communication with Rohsenow, it was decided that s is equal to 1 for water and 1.7 for all other fluids. The results indicated that rshould be a function of the surface-fluid combination and varies between 0.09 and 0.71.

Pioro [62] took a different approach. He suggests that r should be equal to 0.33 for all surface-fluid combinations as is supported by experimental data in numerous papers. Most correlations do indeed take into account the effect of heat flux with a similar power. The parameter s is determined for different surface-fluid combinations, with values ranging from 0.45 to 3.47.

Some critical notes can be made about the Rohsenow correlation and its derivation. Firstly, the basis of the analysis is that heat transfer in nucleate pool boiling is dominated by heat transfer to the liquid phase. However, multiple studies have shown that microlayer evaporation at the very least contributes significantly to the overall heat transferred to the fluid, as shown in section 3.2.2. Secondly, as discussed by Jakob in the discussion section of the original paper, the proportionality between the heat flux and the nucleation site density (equation 3.25) is not necessarily valid. This proportionality was only perceived in the lower heat flux range investigated in the experiments [58]. Later research shows that the nucleation site density varies with the surface heat flux with a power ranging from 1 to 2 [63]. Furthermore, the observation that bubble departure frequency is inversely correlated to bubble departure diameter (3.24) is not widely accepted. Malenkov [64] showed that the bubble release frequency varies according to the diameter to a variable power. For very small bubbles, the release frequency varies according to the departure diameter to the power -3/2, while for very large bubble this power is equal to -1/2. Lastly, the correlation only has predictive qualities if at least one test with the envisaged surface-fluid combination has been done and the required parameters have been reported.

3.3.3 Forster-Zuber correlation (1955)

Forster and Zuber [65] take a similar approach to Rohsenow [58] in drafting a correlation for nucleate boiling heat transfer by defining Nusselt and Reynolds numbers relevant for the boiling phenomenon. This approach is also based on the assumption that single-phase heat transfer to the liquid is the dominant heat transfer mode. In contrast to Rohsenow, they take a different approach in defining the representative lengths and velocities. In the analysis, it is assumed that heat transfer to the microlayer below a growing bubble is the main heat transfer mode. Therefore, theoretical analyses of bubble dynamics are done to obtain the characteristic lengths and velocities related to the growth of the bubbles on the heated surface, rather than on those of bubbles leaving the surface. The authors propose the following Reynolds and Nusselt number for bubble growth, Re_{bg} and Nu_{bg} respectively:

$$Re_{bg} = \frac{\rho_l}{\mu_l} \left(\frac{\Delta T_s c_{p,l} \rho_l \sqrt{\pi \alpha_l}}{h_{lv} \rho_v}\right)^2$$
(3.32)

$$Nu_{bg} = \frac{\dot{q}L_{bg}}{\Delta T_s k_l} \tag{3.33}$$

 L_{bg} is the characteristic length for bubble growth defined as:

$$L_{bg} = \frac{\Delta T_s c_{p,l} \rho_l \sqrt{\pi \alpha_l}}{h_{lv} \rho_v} \sqrt{\frac{2\sigma}{\Delta p_s}} \sqrt[4]{\frac{\rho_l}{\Delta p_s}}$$
(3.34)

In this equation, Δp_s is the difference between the saturation pressure at the surface temperature and the fluid saturation temperature. With the definition of the Reynolds and Nusselt numbers, a power-law correlation is fitted on experimental critical heat flux data by Cichelli and Bonilla [66]. The resulting correlation is proposed as:

$$Nu_{bg} = 0.0015 Re_{bg}^{0.62} Pr_l^{0.33} \tag{3.35}$$

Some remarks can be made about the derivation of the correlation by Forster and Zuber. Firstly, the authors start from the assumption that heat transfer is dominated by single-phase heat transfer. It has been argued in the previous section that this assumption does not appear to be true for all nucleate boiling regimes. Secondly, it is unclear how the characteristic length is fully derived. The roots in equation 3.34 represent a time constant, of which it is not clear how they were derived or chosen. The authors themselves state that "this time constant may not be the only one that is significant, other possibilities are at present under consideration" [65]. It is not further expanded on which other time constants these may be. Lastly, the fit with experimental data is done on critical heat flux data and not on the full boiling curve. It has been shown that at high heat fluxes, the boiling curve behaves differently than at low and moderate heat fluxes, for example in [67] and in the experiments shown in section 6.4. Therefore it is doubtful that a correlation fitted on only CHF data will adequately represent the full boiling curve.

3.3.4 Kutateladze-Borishanskii correlation (1958)

Kutateladze and Borishanskii [68] propose the following correlation:

$$\frac{\dot{q}L_c}{k_l\Delta T_s} = 0.44Pr_l^{0.35} \left(\frac{1}{9.81\,10^3} \frac{\dot{q}p}{\rho_v h_{lv}g\mu_l} \frac{\rho_l}{\rho_l - \rho_v}\right)^{0.7} \tag{3.36}$$

~ -

The factor $9.81 \, 10^3$ is required for the conversion of the units used by Kutateladze and Borishanskii to Pa. Merging this factor with the constant 0.44 results in:

$$\frac{\dot{q}L_c}{k_l \Delta T_s} = 0.00071 P r_l^{0.35} \left(\frac{\dot{q}p}{\rho_v h_{lv} g \mu_l} \frac{\rho_l}{\rho_l - \rho_v} \right)^{0.7}$$
(3.37)

The correlation was determined by fitting to experimental data of water (in the pressure range 0.1 bar to 22.1 bar) and some other fluids. It is unclear how the dimensionless number in between the rightmost brackets is determined.

3.3.5 Borishanskii-Mostinski correlation (1963)

In later work, Borishanskii and Mostinski [69, 70] used the theory of thermodynamic similarity to derive a correlation for heat transfer in nucleate pool boiling. This theory assumes that the variation of fluid properties with reduced pressure (and reduced temperature) is similar for all different fluids. This can be written as:

$$\frac{a(p_r)}{a(p_{r*})} = f_a(p_r)$$
(3.38)

with a a fluid property, p_r the reduced pressure or the ratio of the pressure to the critical pressure, p_{r*} a certain reference reduced pressure and f_a a function for the

specific fluid property, but independent of the type of fluid. Several figures to show the validity of this assumption are shown in [69] for the fluids pentane, heptane, water, ethanol and benzene. Another assumption is made, namely that a correlation can be made out of a power-law relation of dimensionless groups depending on heat flux, surface superheat and the fluid properties (and thus neglecting surface material effects). This results in the following relation:

$$\left. \frac{\dot{q}^{1-n}}{\Delta T_s} \right|_{p_r} = \left. \frac{\dot{q}^{1-n}}{\Delta T_s} \right|_{p_{r*}} F_1(p_r) \tag{3.39}$$

where *n* is a constant and F_1 is a function taking into account the variation in pool boiling heat transfer with reduced pressure, determined from the graph in figure 3.15. With some further assumptions and by comparing to experimental data, a general equation for nucleate pool boiling heat transfer is given by Borishanskii [69] as:

$$\frac{\dot{q}^{0.3}}{\Delta T_s} = A_1 \Phi(p_c) F_1(p_r)$$
(3.40)

where A_1 is a constant and $\Phi(p_c)$ is a function of the fluid critical pressure p_c , taking into account the variation of fluid properties between different fluids. The assumptions made to get to this equation is that n in equation 3.39 is equal to 0.7 (based on experimental data) and that the fluid property variation between different fluids can be correlated to the critical pressure of these fluids.



Figure 3.15: Correlation of boiling heat transfer data using thermodynamic similarity (function $F_1(p_r)$ in equation 3.40)[70].

Mostinski [71] provided a value for the parameter A_1 and function for $\Phi(p_c)$ and $F_1(p_r)$, resulting in the combined Borishanskii-Mostinski correlation:

$$\frac{\dot{q}^{0.3}}{\Delta T_s} = 0.1011 p_c^{0.69} (1.8 p_r^{0.17} + 4 p_r^{1.2} + 10 p_r^{10})$$
(3.41)

In this equation, \dot{q} is evaluated in W/m², ΔT_s in K and p_c in bar.

Using the theory of thermodynamic similarity is very useful to predict the pool boiling performance of fluids of which not all the properties are known or of which the properties are not accurately known. Furthermore, as is shown by Cooper [72] and covered more in detail in section 3.3.10, it removes the ambiguity in determining which of the fluid properties are actually relevant to the nucleate boiling heat transfer. However, it does inherently introduce inaccuracies, as the similarity is not perfect and thus deviations of the relevant properties from the overall curve are not taken into account. Furthermore, it is also doubtful that only the critical pressure can be used to take into account difference in fluid properties amongst different fluids.

3.3.6 Shekriladze-Ratiani correlation (1966)

Shekriladze and Ratiani [73, 74] stress the importance of the microgeometry of the boiling surface, which is neglected by the previous correlations. They use an idealized model of the boiling surface consisting of a large number of identical conical recesses as shown in figure 3.16. The radius at the mouth of the recess r_c , also called the effective radius of the nucleation site or radius of the critical nucleus, is chosen as the characteristic length, as it directly affects bubble nucleation. This is clear from the derivation made in section 3.2.4. The superheat temperature from equation 3.13 can be made dimensionless in a similar manner such as the Nusselt number, to result in the dimensionless parameter K, with r_c as half the characteristic length and bubble radius:

$$K = \frac{2r_c \dot{q}}{k_l \Delta T_0} = \frac{\dot{q} r_c^2 \rho_v h_{lv}}{\sigma k_l T_{sat}}$$
(3.42)

Next to the dimensionless parameter K, a Reynolds number Re_* is determined with the inertial forces based on the expansion work of a growing bubble:

$$Re_{*} = \frac{\sqrt{p\rho_{l}(\rho_{l}/\rho_{v}-1)}\sigma T_{sat}c_{p,l}}{\nu_{l}(\rho_{v}h_{lv})^{2}}$$
(3.43)

Based on experimental data of boiling heat transfer on stainless steel surfaces with highly different fluids (water, ethyl alcohol, diphenyl, R12, R22, cryogenics and liquid metals), the following correlation was proposed:

$$\frac{\dot{q}r_c}{k_l\Delta T_s} = 0.0122K^{0.7}Re_*^{0.25} \tag{3.44}$$



Figure 3.16: Idealized nucleation site [75].

It is clear from numerous studies (see section 3.4.3) that the surface microstructure plays a big role in nucleate boiling heat transfer. Therefore the addition of a parameter taking this effect into account is very beneficial. However, it is not straightforward to determine this parameter for a boiling surface, as the random surface structure containing many types of crevices has to be translated to a single radius of an equivalent conical recess. It is not described how this value should be measured, nor is it indicated how this value relates to more common surface structure parameters such as the various types of surface roughness parameters. Shekriladze [74] proposes to use a value of 5 μ m for common commercial surfaces.

3.3.7 Labuntsov correlation (1972)

Labuntsov [76] proposed the following correlation, based on a theoretical analysis of heat transfer in a thin liquid layer under the vapour region:

$$\frac{\dot{q}^{1/3}}{\Delta T_s} = 0.075 \left[1 + 10 \left(\frac{\rho_v}{\rho_l - \rho_v} \right)^{2/3} \right] \left(\frac{k_l^2}{\nu_l \sigma T_{sat}} \right)^{1/3}$$
(3.45)

It is not clear how this correlation was derived, if the correlation is partly fitted to experiments or which experiments were used for this fitting. However, it was shown to fit experimental data quite well [54].

3.3.8 Gorenflo correlation (1973)

Gorenflo et al. [77] proposed a correlation treating the influence of heat flux, type of fluid, pressure and surface properties separately:

$$\frac{\dot{q}}{\Delta T_s} = \left. \frac{\dot{q}}{\Delta T_s} \right|_0 F_{\dot{q}} F_f F_p F_w \tag{3.46}$$

where this index 0 indicates the reference heat flux, fluid, pressure and surface, and $F_{\dot{q}}$, F_f , F_p and F_w represent the influences of heat flux, type of fluid, pressure and surface properties respectively. The latest updates on the influences are based on new reported measurement data and were published in 2010 by Gorenflo and Kenning [78].

The ratio of heat flux to surface superheat at reference configuration $\dot{q}/\Delta T_s|_0$ is equal to 3580 W/m²K. The influence of the surface heat flux is determined as:

$$F_q = (q/q_0)^n (3.47)$$

with q_0 equal to 20 kW/m² and the exponent n a function of the reduced pressure:

$$n = 0.95 - 0.3p_r^{0.3} \tag{3.48}$$

The influence of the type of fluid is captured by:

$$F_f = \left(\frac{\frac{dp_s}{dT_s}\frac{1}{\sigma}}{P_{f,ref}}\right)^{0.6}$$
(3.49)

with $P_{f,ref}$ equal to 1 (μ m K)⁻¹ and the fluid properties evaluated at p_r equal to 0.1. This parameter is derived from the minimal superheat (equation 3.13), by implementing the Clausius-Clapeyron equation (equation 3.11) with the assumption that the vapour density is negligible compared to the liquid density.

The effect of pressure on the fluid properties and thus on the boiling heat transfer is captured by a function of reduced pressure, similar to the one proposed by Mostinski (last part of equation 3.41):

$$F_p = 0.7p_r^{0.2} + 4p_r + \frac{1.4p_r}{1 - p_r}$$
(3.50)

Lastly, the effect of the surface is determined based on two parts: the effect of the surface roughness and the effect of the surface material properties. This correlation is the first to explicitly link a surface roughness parameter to the influence of the surface microgeometry (more information on this effect can be found in section 3.4.3). The surface effect is characterised as:

$$F_{w} = \left(\frac{R_{a}}{R_{a0}}\right)^{2/15} \left(\frac{k_{w}\rho_{w}c_{p,w}}{k_{0}\rho_{0}c_{p,0}}\right)^{0.25}$$
(3.51)

where R_a is the arithmetic average roughness, R_{a0} is equal to 0.4 μ m and k_w , ρ_w and $c_{p,w}$ are the surface material thermal conductivity, density and specific heat capacity respectively. The reference surface material is copper, with $k_0\rho_0c_{p,0}$ equal to 1250 kJ²/(m⁴K²s).

This correlation is similar to that of Borishanksi and Mostinski, but also includes parameters of the boiling surface and utilizes another property to determine the influence of the type of fluid used. The same comments hold as for the Borishanskii-Mostinski correlation: utilizing reduced pressure instead of fluid properties inherently results in inaccuracies and utilizing one fluid property to account for the differences between all fluids may not be sufficient.

3.3.9 Stephan-Abdelsalam correlation (1978)

Stephan and Abdelsalam [79] made a regression analysis to develop a new nucleate boiling heat transfer correlation. The analysis was done on a dataset of nearly 5000 data points, consisting of a wide range of fluids, surfaces and operating conditions. They determined thirteen dimensionless groups from variables, fluid properties and surface properties. The surface roughness $R_{p,DIN}$ is used in the regression analysis. This is the German DIN measure "Glättungstiefe", which will be discussed in section 3.4.3. A general correlation was derived, but also specific correlations for four groups of fluids were determined: water, hydrocarbons, cryogenics and refrigerants. They chose the Fritz bubble departure diameter (equation 3.17) as characteristic length, which includes the contact angle as a parameter. As mostly no data were available for the contact angle, it was assumed 45° for water, 35° for hydrocarbons and refrigerants and 1° for cryogens in the regression analysis. The resulting general correlation is:

$$\frac{\dot{q}D_F}{k_l\Delta T_s} = 0.23 \left(\frac{\dot{q}D_F}{k_lT_{sat}}\right)^{0.674} \left(\frac{\rho_v}{\rho_l}\right)^{0.297} \left(\frac{h_{lv}D_F^2}{\alpha_l^2}\right)^{0.371} \\ \left(\frac{\rho_l}{\rho_l - \rho_v}\right)^{1.73} \left(\frac{\alpha_l^2\rho_l}{\sigma D_F}\right)^{0.35} \left(\frac{R_{p,DIN}}{R_{p,DIN0}}\right)^{0.133}$$
(3.52)

with $R_{p,DIN0}$ equal to 1 μ m.

A few remarks can be made about the results. Firstly, the dimensionless groups used do not necessarily reflect the physics behind the boiling heat transfer phenomenon. Therefore, certain fluid properties may be introduced which may not influence the heat transfer and are negated by other properties in the correlation. For example the liquid thermal diffusivity appears squared in the correlation in the third (nominator) and fifth (denominator) group. However, as both groups are raised to approximately the same power (0.371 and 0.35), the overall effect on the thermal diffusivity is almost negligible. Lastly, it is not clear which criterion is used by Stephan and Abdelsalam to stop the addition of dimensionless groups. The example is given for water, where the first two dimensionless groups clearly are necessary for a proper correlation. Afterwards two additional groups are added, of which the utility is questionable based on both the graphical representation of the data and the values of the average error (adding the third and fourth term decreases the average error from 13.96% to 12.2% and 11.3%).

3.3.10 Cooper correlation (1984)

Similarly to the analysis of thermodynamic similarity by Borishanskii [69], Cooper [72] argues that all fluid properties relevant to pool boiling heat transfer can be written as a function of reduced pressure (p_r) , reduced temperature (T_r) and one minus the reduced temperature $(1 - T_r)$. He argues further that the term T_r can be removed and that the term $(1 - T_r)$ can be replaced by $log(p_r)$ without losing a lot of accuracy. The resulting correlation is thus simplified and uses only the reduced pressure. Cooper makes the case that several fluid properties vary in a very similar manner with the reduced pressure, thereby making it impossible to determine which properties are actually influencing boiling heat transfer by statistical analysis of experimental data. The variation in fluid properties at equal reduced pressure for different fluids can be broadly correlated with the molar mass M. Based on the work of Nishikawa et al. [80], the effect of surface roughness is taken into account with an interaction effect with the reduced pressure. A factor $log(R_{p,DIN})$ is therefore included in the power of the reduced pressure in the correlation.

The experimental data used to determine the correlation includes mostly water, refrigerants and cryogenics as fluid and copper and stainless steel as surfaces. The full correlation based on the fitting of experimental data is:

$$\frac{\dot{q}^{1/3}}{\Delta T_s} = 55 \, p_r^{0.12 - 0.2 \log R_{p,DIN}} \left(-\log p_r \right)^{-0.55} \, M^{-0.5} \tag{3.53}$$

In this equation, SI units are used for all parameters except for $R_{p,DIN}$ and M, which should be evaluated in μ m and g/mol respectively. For heat transfer from copper cylinders, the factor 55 has to be multiplied by 1.7. The author states that this is not logically consistent (for stainless steel cylinders this is not required) and might be superseded when more data is available.

Due to its simplicity, this correlation is frequently used to predict nucleate boiling heat transfer. Several comments can be given on the correlation, most of which have been covered by the author himself. Utilizing the molar mass to account for the difference between fluids entails significant inaccuracies. More (accurate) data is needed to decide which fluid properties are actually the ones contributing to the heat transfer rates. Cooper himself states that he hopes that with more experimental data, the correlation he proposed will be rapidly outdated. However, it is still widely used to this day.

3.3.11 Kutateladze correlation (1990)

Kutateladze [68] first proposed a correlation in collaboration with Borishanskii around 1958. Years later, around 1990, he proposed a different correlation based on newer insights [81]:

$$\frac{\dot{q}L_c}{k_l \Delta T_s} = 3.37 \cdot 10^{-9} \left(\frac{c_{p,l} \Delta T_s}{h_{lv}}\right)^2 \left[\frac{(p/\rho_v)^2}{\sigma g/(\rho_l - \rho_g)}\right]$$
(3.54)

In this equation, two different dimensionless numbers are used compared to the earlier correlation of Kutateladze-Borishanskii [68]. The first dimensionless number is the Jakob number, while the second number is based on capillary and gravitational interactions. The correlation is based on the similarity of heat transfer in nucleate boiling to that in bubbling gas flows. It is unclear to which data the correlation has been fitted.

3.3.12 Leiner correlation (1994)

Continuing on the work of Gorenflo [77], Leiner [82] proposes a general correlation based on thermodynamic similarity. Before the latest update by Gorenflo and Kenning [78] which includes the factor F_f (equation 3.49) to take into account the effect of fluid specific parameters, a table of values was given with a reference heat transfer coefficient per fluid. Leiner uses these values to fit an equation based on the critical data (p_c and T_c), the molar mass and two fluid-specific parameters, arguing that these specific features cannot be characterised by the previously stated parameters. These parameters are the caloric parameter C_{cal} and the vapour pressure parameter K_{vap} :

$$C_{cal} = \frac{c_{p,l}(p_r = 0.1)}{\Re}$$
(3.55)

$$K_{vap} = \frac{h_{lv}(p_r = 0.1)}{\Re T_c}$$
(3.56)

with \Re the specific gas constant, equal to:

$$\Re = \frac{\Re}{M} \tag{3.57}$$

where \Re is the universal gas constant equal to 8.314 J/(mol K). Leiner further uses non-dimensional groups for the surface superheat temperature, heat flux and

roughness based on the critical temperature and pressure, the specific gas constant and the Avogadro number N_a . From all the above considerations, the following correlation is derived:

$$\frac{\dot{q}/\dot{q}_{00}}{\Delta T_s/T_{00}} = 0.6161 C_{cal}^{0.1512} K_{vap}^{0.4894} F'(p_r) \left(\frac{\dot{q}}{\dot{q}_{00}}\right)^{0.9 - 0.3 p_r^{0.3}} \left(\frac{R_a}{L_{00}}\right)^{0.133}$$
(3.58)

The subscript 00 indicates the scale factors given by:

$$\dot{q}_{00} = p_c (\Re T_c)^{1/2} \tag{3.59}$$

$$T_{00} = T_c$$
 (3.60)

$$L_{00} = \left(\frac{\Re MT_c}{N_a p_c}\right)^{1/3} \tag{3.61}$$

The function $F'(p_r)$ is defined by:

$$F'(p_r) = 43000^{0.15 - 0.3p_r^{0.3}} F(p_r)$$
(3.62)

where an older version of equation 3.50 from Gorenflo is used:

$$F(p_r) = 1.2p_r^{0.27} + \left(2.5 + \frac{1}{1 - p_r}\right)p_r$$
(3.63)

Leiner also assessed the influence of a non-dimensional gravitational acceleration term and concluded that this influence is negligible.

This correlation combines the theory of thermodynamic similarity with several fluid-specific properties $(p_c, T_c, M, c_{p,l} \text{ and } h_{lv})$. This allows for a better prediction of the influence of different fluids. However, it remains unclear if the chosen fluid properties are those that actually influence heat transfer, due to the correlations existing between the different properties. The influence of reduced pressure on the fluid properties (equation 3.63) is also based on an outdated correlation.

3.3.13 Pioro correlation (2004)

Similarly to Rohsenow [58], Pioro [52] proposes a correlation containing a constant C_{sf}^* and exponent *m* depending on the surface-fluid combination:

$$\frac{\dot{q}L_c}{k_l\Delta T_s} = C_{sf}^* \left[\frac{\dot{q}}{h_{lv}\rho_v^{1/2} [\sigma g(\rho_l - \rho_v)]^{1/4}} \right]^{2/3} Pr_l^m$$
(3.64)

Pioro named the parameter between the square brackets the Kutateladze number, although it is not used by Kutateladze in either of the two correlations proposed by him. Several different ratios are linked to the name Kutateladze number, so it is not used further to avoid confusion. By introducing the capillary length L_c and utilizing the assumptions by Rohsenow which result in equation 3.26, it can be shown that this dimensionless group is proportional to the square root of the Weber number We:

$$We = \frac{G_b^2 L_c}{\rho_v \sigma} \tag{3.65}$$

which indicates the ratio of the vapour's inertia to fluid's surface tension. This is in contrast to the Rohsenow correlation, which is based on the Reynolds number (ratio of the inertial forces to the viscous forces).

This correlation has not yet been widely used. It is hampered in its predictive capabilities by the required values for C_{sf}^* and m for the specific surface-fluid combination, thereby requiring experimental data to be able to predict the boiling curve for every new surface-fluid combination.

3.3.14 Summary

Table 3.2 gives an overview of all the cited correlations and some of their important parameters. The following chapter will go more into detail on the different parameters that the different correlations take into account.

	Thermodynamic	Fitting	Surface
Author(s)	similarity	coefficients	effects
Kruzhilin [57]	No	No	-
Rohsenow [58]	No	Yes	C_{sf}
Forster-Zuber [65]	No	No	-
Kutateladze-Borishanskii [68]	No	No	-
Borishanskii [69]	Yes	No	-
Shekriladze-Ratiani [73]	No	No	r_c
Labuntsov [76]	No	No	-
Gorenflo [77]	Yes	No	$R_a, k\rho c_p$
Stephan-Abdelsalam [79]	No	No	$R_{p,DIN}$
Cooper [72]	Yes	No	$R_{p,DIN}$
Kutateladze [81]	No	No	-
Leiner [82]	Yes	No	R_a
Pioro [83]	No	Yes	C_{sf}^*

Table 3.2: Overview of the cited correlations.

All of the described correlations have been used by several authors to predict nucleate pool boiling heat transfer. Evaluation and comparisons have also been done by several authors. Pioro et al. compared the Kruzhilin, Rohsenow, Kutateladze-Borishanskii, Labuntsov, Kutateladze and Pioro correlation to experimental data [52]. No correlations using the theory of thermodynamic similarity are included in the comparison. The authors conclude that the Rohsenow and Pioro correlations are most accurate. This is to be expected, as both these correlations were fitted with two parameters to the experimental data to which they are compared. They can therefore not be directly used for untested surface-fluid combinations. Of the other correlations, the Kutateladze-Borishanskii correlation renders the best prediction of the experimental data. Forrest et al. [53] made an analysis of the Rohsenow, Borishanskii-Mostinski, Stephan-Abdelsalam, Cooper and Leiner correlations by evaluating the predictive power with regard to their experimental results. Also here, the Rohsenow correlation performed best as a result of the fitting coefficients used. Of the other correlations, Stephan-Abdelsalam performed best, followed by Leiner and Cooper, while Borishanskii-Mostinski underpredicted heat transfer rates by about 50%. Bartle et al. [54] performed a similar analysis with the Kruzhilin, Rohsenow, Borishanskii-Mostinski, Labuntsov, Gorenflo, Stephan-Abdelsalam, Kutateladze and Leiner correlations. The Cooper correlation was not included in the analysis as the molar mass of the fluid under test was higher than the range for which the Cooper correlation was fitted. Also in this analysis, using the fitting parameters in the Rohsenow correlation resulted in the best prediction. However, the Rohsenow correlation was also compared to the experimental data when using the coefficients originally suggested. For this case, the predictive accuracy was less than the correlation of Labuntsov, which performed best of the correlations without fitting parameters. The Gorenflo correlation also provided adequate predictions while in addition taking into account the effect of surface roughness. All other correlations had significantly larger deviations from the experimental results.

It is clear that there is no consensus on which a priori correlation (without fitting constants) is most suited for predicting nucleate pool boiling heat transfer. Large deviations from measured heat transfer rates are common for all correlations, indicating that not all effects are properly taken into account and that the heat transfer mechanisms are not yet fully understood. The next section provides more details on the different parameters which can affect nucleate pool boiling heat transfer.

3.4 Parameter sensitivity

3.4.1 Surface heat flux

The relationship between surface heat flux and surface superheat temperature is usually presented as a simple power relation:

$$\dot{q} \sim \Delta T_s^n$$
 (3.66)

with *n* somewhere between 2.5 and 5 depending on the fluid and surface properties. This is also evident from the described correlations in section 3.3. When reworking these correlations to the form of equation 3.66, the values of table 3.3 are determined. The values for the Rohsenow correlation are based on the determined constants by Vachon et al. [61]. The Forster-Zuber correlation uses the pressure difference Δp_s , which is related to the surface superheat using the Clausius-Clapeyron equation 3.11.

Correlation	n	n_{min}	n_{max}
Kruzhilin [57]	3.33		
Rohsenow [58]	3	1.41	11.1
Forster-Zuber [65]	1.99		
Kutateladze-Borishanskii [68]	3.33		
Borishanskii- [69]	3.33		
Shekriladze-Ratiani [73]	3.33		
Labuntsov [76]	3		
Gorenflo [77]	$(0.05 + 0.3 p_r^{0.3})^{-1}$	2.86	20
Stephan-Abdelsalam [79]	3.07		
Cooper [72]	3		
Kutateladze [81]	3		
Leiner [82]	$(0.1 + 0.3 p_r^{0.3})^{-1}$	2.5	10
Pioro [83]	3		

Table 3.3: Value of n in equation 3.66 for different correlations.

All correlations except for those of Rohsenow, Gorenflo and Leiner use a fixed value for n. This value is typically between 3 and 3.33, with the exception of Forster-Zuber which is equal to 1.99. This is presumably the result of using critical heat flux data to fit their correlation. As stated before, this correlation will probably not work well for nucleate boiling heat transfer. For the correlation of Rohsenow, the standard value is 3, but when taking into account the values determined by Vachon for different surface-fluid combinations it varies from 1.41 to 11.1. The Gorenflo correlation (and also the Leiner correlation which is based on the Gorenflo correlation) take into account the effect of reduced pressure on the exponent. This takes into account the decrease of the slope of the boiling curve with increasing pressure. At very low reduced pressures, the exponent seems to become very large (n goes to 20 and 10 for Gorenflo and Leiner respectively).

On a logarithmic scale, a boiling curve according to equation 3.66 would become a straight line. When regarding the nucleate boiling regime on figure 3.3 (from point A or A" to point D), it is clear that the entire regime cannot be represented by a single power law, but that there are actually three zones: partial nucleate boiling (A to B) with a lower slope, fully developed nucleate boiling (B to C) with a steeper slope and partial dryout nucleate boiling (C to D) where the slope gradually goes to zero at the critical heat flux. The representation according to equation 3.66 with a single slope can only work for one of these regimes. Typically the correlations are only valid for the fully developed nucleate boiling regime, and not for the low and high heat flux regions. Since all correlations use this kind of approximation, the correlations will only be accurate for the moderate heat flux nucleate boiling region.

The deviation from the power law in low and high heat flux nucleate boiling regions has been reported by several researchers. Kandlikar and Chung [36] describe the three regions similarly to the previous paragraph when explaining the general boiling curve. El-Genk and Bostanci [84] performed measurements on pool boiling heat transfer of refrigerant HFE 7100. Based on the measurement results, they describe the three nucleate boiling regimes at low, intermediate and high surface superheat with different slopes. The measurements shown by Kutateladze [81] also indicate three regions with distinctly different slopes on a logarithmic plot, as shown in figure 3.17. Dhir [45] mentions that a small change in the slope of the boiling curve can occur upon transition from partial to fully developed nucleate boiling. Forrest et al. [53] did measurements on refrigerants FK-649 and R134a. The measurements of the heat transfer coefficient show a drop at the highest heat fluxes, indicating the transition to critical heat flux. Ji et al. [67] performed measurements with R134a in the intermediate and high heat flux nucleate boiling regions. They compared the measurements with the prediction by the Cooper correlation and concluded that above 250 kW/m², the slope of the boiling curve decreases and the correlation no longer adequately predicts the heat transfer rate.



Figure 3.17: Nucleate pool boiling curve of various fluids [81]. Left: water at 20 kPa (1) and 3.6 kPa (2). Middle: water (1) and ethanol (2) at 6 kPa. Right: 13% NaCl-water solution (1) and ethanol (2) at 3.6 kPa.

From these different sources, it is clear that the entire region of nucleate pool boiling cannot be described by a single power law. Since all correlations to this date use this assumption, they will either only be valid for one of the nucleate boiling regimes or they are fitted on the entire region and thereby do not capture the changes in slope between the three regions.

3.4.2 Fluid properties

When regarding the influence of fluid properties in the correlations described in literature, three distinct groups can be determined. The first group uses the reduced pressure to take into account variations of fluid properties of a single fluid and uses limited fluid data to take into account the variation of fluid properties amongst fluids. This group encompasses the Borishanskii-Mostinski, Gorenflo, Cooper and Leiner correlations. The second group relies fully on a multitude of fluid properties to determine the influence of the fluid on nucleate boiling heat transfer. This group is composed of the Kruzhilin, Forster-Zuber, Kutateladze-Borishanskii, Shekriladze-Ratiani, Labuntsov, Stephan-Abdelsalam and Kutate-ladze correlations. The last group uses a combination of fluid properties and constants which have to be determined for each surface-fluid combination. The Rohsenow and Pioro correlation make up this group. This division into the three groups is also described visually in figure 3.18.



Figure 3.18: Overview of the cited correlations and division into three groups.

For the correlations of groups 2 and 3, the powers to which the different fluid properties are raised can be compared to evaluate which properties might be influencing heat transfer most. To be able to compare the different correlations, they have to be in the same format. It was chosen to evaluate the correlations according to equation 3.67, as most correlations are presented in such a form:

$$\frac{\dot{q}^{1/n}}{\Delta T_s} = g' \prod_i a_i^{x_i} \tag{3.67}$$

with a_i representing the different fluid properties, x_i the exponents and g' a function taking any non-fluid properties or influences into account. The fluid properties that are used in the correlations are: $c_{p,l}$, h_{lv} , k_l , p_{sat} , T_{sat} , ν_l , ρ_l , ρ_v and σ . The term $(\rho_l - \rho_v)$ is also added, as this is also a commonly used parameter. The liquid dynamic viscosity, liquid thermal diffusivity and liquid Prandtl number are recalculated to the equivalent terms expressed in $c_{p,l}$, ρ_l , ν_l and k_l . The correlation of Labuntsov has a term which cannot be transformed to fit equation 3.67, therefore two exponents have a range of possible values. For the Rohsenow correlation, the originally proposed correlation is used. For the Pioro correlation, a value of -1.2 is used for m, which is valid for water as fluid.

Table 3.4 shows the results of the analysis. The first observation that can be made is that there is no agreement between the different correlations on the contributions of the different properties. Some consensus between the correlations can be found. A high liquid thermal conductivity has a positive effect on heat transfer. If the saturation pressure is taken into account, heat transfer rates will increase with pressure. Increasing surface tension deteriorates heat transfer. For both the saturation temperature and the liquid kinematic viscosity an increase will result in lower heat transfer rates or there is no effect. For the liquid specific heat capacity, latent heat of vaporization, liquid density, vapour density and density difference, there is no consensus on the effect on heat transfer. Another observation is that most outliers in exponents originate from the Rohsenow and Pioro correlations, which is related to the high or low values for the exponents given to the liquid Prandtl number in the respective correlations. It seems the effect of these high powers is negated by the constants which also depend on the fluid.

For the correlations of group 1, changes of fluid properties with pressure are determined based on the reduced pressure. A comparison of the functions used in the four relevant correlations is given in figure 3.19. The functions are normalized such that they are equal to 1 at reduced pressure of 0.1. The functions $F(p_r)$ can be determined in a general form as:

$$F(p_r) = \frac{\Delta T_s|_{p_r=0.1}}{\Delta T_s}$$
(3.68)

The functions of Borishanskii-Mostinski and Cooper are very similar. They

Kruzhilin [57] -0.12 0.03 0.75 0 -0.37 -0.45 -0.12 0.03 Rohsenow [58] -0.7 -0.67 1.7 0 0 -1.37 -1.37 0 Forster-Zuber [65] 0.23 0.26 0.40 0 -0.38 -0.15 0.48 0.26 Kutateladze-Borishanskii [68] 0.35 -0.7 0.65 0.7 0 -0.38 0.15 0.48 0.26 Shekriladze-Ratiani [73] 0.25 0.2 0.3 0.13 -0.45 0.25 0.3 0.67 Shekriladze-Ratiani [73] 0.25 0.2 0.3 0.13 -0.45 0.25 0.13 0.67 Shekriladze-Ratiani [73] 0.25 0.2 0.3 0.13 -0.45 -0.25 -0.13 0.33 Labuntsov [76] 0 0 0.67 0 0.67 0 0 0.67 Stephan-Abdelsalam [79] 0.067 0.33 0.67 0 0 0.67 Pioro [83] -1.2 -0.67 0.29 0 -1.2 -0.33 Minimum -1.2 -0.67 0.29 0 -0.67 -1.37 -1.37 -1.37 Minimum -1.2 -0.67 0.29 0 -0.67 -1.37 -1.37 -1.37	Correlation	$c_{p,l}$	h_{lv}	k_l	d	T_{sat}	νı	ы	ρ_v	$\rho_l - \rho_v$	α
Rohsenow [58] -0.7 -0.67 1.7 0 0 -1.37 -1.37 0 Forster-Zuber [65] 0.23 0.26 0.40 0 -0.38 -0.15 0.48 0.26 Kutateladze-Borishanskii [68] 0.35 -0.7 0.65 0.7 0 -0.35 0.35 -0.7 Shekriladze-Ratiani [73] 0.25 0.2 0.3 0.13 -0.45 -0.25 -0.13 0.33 Shekriladze-Ratiani [73] 0.25 0.2 0.3 0.13 -0.45 -0.25 -0.13 0.33 Labuntsov [76] 0 0 0.67 0.29 0 -0.67 0 0 0.67 Stephan-Abdelsalam [79] 0.04 0.37 0.29 0 -0.67 0 0 0.67 Kutateladze [81] -1.2 -0.67 0.33 0.67 0 0 -0.67 Pioro [83] -1.2 -0.67 0.29 0 -1.2 -1.2 -0.33 Minimum -1.2 -0.67 0.29 0 -0.67 -1.37 -1.37 -1.37	Kruzhilin [57]	-0.12	0.03	0.75	0	-0.37	-0.45	-0.12	0.03	-0.03	-0.67
Forster-Zuber [65] 0.23 0.26 0.40 0 -0.38 -0.15 0.48 0.26 Kutateladze-Borishanskii [68] 0.35 -0.7 0.65 0.7 0 -0.35 0.35 -0.7 Shekriladze-Ratiani [73] 0.25 0.2 0.3 0.13 -0.45 -0.25 -0.13 0.35 Labuntsov [76] 0 0 0.67 0 0.67 0 0.67 0.67 0.67 Stephan-Abdelsalam [79] 0.04 0.37 0.29 0 -0.67 0 0.67 Kutateladze [81] 0.67 -0.67 0.33 0.67 0 0 -0.67 Pioro [83] -1.2 -0.67 0.29 0 -0.67 -1.2 -0.33 Minimum -1.2 -0.67 0.29 0 -0.67 -1.37 -1.37 -0.73	Rohsenow [58]	-0.7	-0.67	1.7	0	0	-1.37	-1.37	0	0.17	-0.17
Kutateladze-Borishanskii [68] 0.35 -0.7 0.65 0.7 0 -0.35 0.35 -0.7 Shekriladze-Ratiani [73] 0.25 0.2 0.3 0.13 -0.45 -0.13 0.35 Labuntsov [76] 0 0 0 0.67 0 -0.33 -0.33 0.33 Stephan-Abdelsalam [79] 0.04 0.37 0.29 0 -0.67 0 0.67 Stephan-Abdelsalam [79] 0.04 0.37 0.29 0 -0.67 0 0.67 Kutateladze [81] 0.67 -0.67 0.33 0.67 0 0 -0.67 Pioro [83] -1.2 -0.67 0.29 0 -0.67 -1.2 -0.33 Minimum -1.2 -0.67 0.29 0 -0.67 -1.2 -0.33	Forster-Zuber [65]	0.23	0.26	0.40	0	-0.38	-0.15	0.48	0.26	-0.38	-0.25
Shekriladze-Ratiani [73] 0.25 0.2 0.3 0.13 -0.45 -0.25 -0.13 0.33 Labuntsov [76] 0 0 0.67 0 -0.33 -0.33 0.13 0.33 Labuntsov [76] 0 0 0.67 0 0.67 0 0.33 -0.33 0.33 0.67 0 0.67 $0.0.67$ 0.33 0.67 0 0 0.67 0.67 0.33 0.67 0	Kutateladze-Borishanskii [68]	0.35	-0.7	0.65	0.7	0	-0.35	0.35	-0.7	-0.2	-0.5
Labuntsov [76]000.670-0.33-0.3300067Stephan-Abdelsalam [79]0.040.370.290-0.6701.820.3Kutateladze [81]0.67-0.670.330.67000-0.67Pioro [83]-1.2-0.672.200-1.2-0.33Minimum-1.2-0.670.290-0.67-1.37-1.37-0.7	Shekriladze-Ratiani [73]	0.25	0.2	0.3	0.13	-0.45	-0.25	-0.13	0.33	0.13	-0.45
Stephan-Abdelsalam [79] 0.04 0.37 0.29 0 -0.67 0 1.82 0.3 Kutateladze [81] 0.67 -0.67 0.33 0.67 0 0 -0.67 Pioro [83] -1.2 -0.67 2.2 0 0 -1.2 -0.33 Minimum -1.2 -0.67 0.29 0 -0.67 -1.37 -1.37 -0.7	Labuntsov [76]	0	0	0.67	0	-0.33	-0.33	0	0 0.67	00.67	-0.33
Kutateladze [81] 0.67 -0.67 0.33 0.67 0 0 -0.67 Pioro [83] -1.2 -0.67 2.2 0 0 -1.2 -0.33 Minimum -1.2 -0.67 0.29 0 -0.67 -1.37 -1.37 -1.37 -0.7	Stephan-Abdelsalam [79]	0.04	0.37	0.29	0	-0.67	0	1.82	0.3	-1.76	-0.32
Pioro [83] -1.2 -0.67 2.2 0 0 -1.2 -0.33 Minimum -1.2 -0.67 0.29 0 -0.67 -1.37 -1.37 -0.7	Kutateladze [81]	0.67	-0.67	0.33	0.67	0	0	0	-0.67	0.5	-0.5
Minimum -1.2 -0.67 0.29 0 -0.67 -1.37 -0.7	Pioro [83]	-1.2	-0.67	2.2	0	0	-1.2	-1.2	-0.33	0.33	-0.37
	Minimum	-1.2	-0.67	0.29	0	-0.67	-1.37	-1.37	-0.7	-1.76	-0.67
Maximum 0.6/ 0.3/ 2.2 0./ 0 0 1.82 0.6/	Maximum	0.67	0.37	2.2	0.7	0	0	1.82	0.67	0.5	-0.17

Table 3.4: Exponents of fluid properties in different correlations.

deviate slightly from those of Gorenflo and Leiner, which match very well since the function used by Leiner is an older version of the function proposed by Gorenflo. The small deviation between the two groups might be since the Gorenflo and Leiner correlations also take into account the effect of the reduced pressure on the exponent of the heat flux.



Figure 3.19: Comparison of the effect of reduced pressure for the Borishanskii-Mostinski, Gorenflo, Cooper and Leiner correlation.

The effect of the reduced pressure can also be determined for the correlations of groups 2 and 3 using the definition of equation 3.68. This is done by using the fluid properties of water with variable p_r and otherwise identical inputs. A value of 1 is used for the exponent *s*, while in the Pioro correlation *m* is equal to -1.2. The results are shown in figure 3.20. As expected, a similar curve is produced as the ones which directly use the reduced pressure as parameter. The Kruzhilin, Labuntsov, Kutateladze and Pioro correlations show the largest deviations compared to the general trend.

If the theory of thermodynamic similarity holds, approximately the same curve should be found when using another fluid with the same correlations. R134a is chosen here, as its properties are well known but very dissimilar from those of water. The main characteristics of both fluids are given in table 3.5. The fluid properties which are expected to influence boiling heat transfer show that both fluids are very different. The results of the correlations are again normalized such that the function is equal to 1 at a reduced pressure of 0.1 and are plotted in fig-



Figure 3.20: Comparison of the effect of reduced pressure for the correlations based on fluid properties of water.

ure 3.21. Also for R134a, the same four correlations show diverging behaviour (Kruzhilin, Labuntsov, Kutateladze and Pioro), while the other stay on a very similar curve. These results indicate that even though a very different combination of fluid properties is used by all the correlations of groups 2 and 3, most of them do follow an identical trend as a function of reduced pressure and thus that the theory of thermodynamic similarity holds.

	Water	R134a
Molar mass	18.015 g/mol	102.03 g/mol
Critical pressure	22.064 MPa	4.0593 MPa
Critical temperature	647.10 K	374.21 K
Latent heat $(p_r = 0.1)$	1868.6 J/kg	191.25 J/kg
Surface tension $(p_r = 0.1)$	33.469 mN/m	10.128 mN/m
Liquid thermal conductivity $(p_r = 0.1)$	0.6474 W/mK	0.0879 W/mK

Table 3.5: Selected properties of water and R134a.

If the theory of thermodynamic similarity is used, the question still remains which method is best to incorporate the differences between fluids. The Borishanskii-Mostinski correlation uses a function of the critical pressure. The Gorenflo correlation uses the ratio of the surface tension to derivative of saturation pres-



Figure 3.21: Comparison of the effect of reduced pressure for the correlations based on fluid properties of R134a.

sure to saturation temperature, evaluated at a reduced pressure equal to 0.1. In the Cooper correlation, the molar mass of the fluid is used. Finally, the Leiner correlation uses the critical pressure, critical temperature, molar mass and the liquid specific heat capacity and latent heat of evaporation evaluated at a reduced pressure of 0.1. Only a comparison with a large set of experimental data points can be used to evaluate which of these fluid properties are best suited to predict the difference in nucleate pool boiling heat transfer between fluids.

3.4.3 Geometric parameters

Surface shape and size

It is generally accepted that nucleate boiling heat transfer rates from (horizontal) surfaces are independent of the size or dimensions of that surface. Nucleate pool boiling heat transfer is assumed to be determined by micro-layer evaporation and mechanisms related to the bubble dynamics [45]. As these phenomena occur on small length scales (< 1 mm), there is no influence on pool boiling performance for length scales and sizes important for typical cooling applications (> 1 cm). This independence of surface size is valid for boiling under influence of gravity, but does not hold for boiling in microgravity [85]. This effect will be covered in

section 3.4.5.

For surfaces with a vertical inclination, some additional remarks can be made. Kang [86] studied the effect of the length of a vertical cylinder on the pool boiling heat transfer rate. He concluded that there is an influence on the heat transfer rate, where it is decreased slightly with increasing length. He proposed an empirical correlation where the heat flux is inversely proportional to the height relative to the tube diameter raised to the power 0.072, which signifies a rather small influence. He attributed this effect to the formation of vapour slugs at the top of the cylinder, reducing the number of active nucleation sites and preventing liquid access to the surface. Gupta et al. [87] performed nucleate pool boiling measurements on a vertically positioned tube bundle. They found that the local heat transfer rate increased starting from the bottom to the top of the bundle, contrary to the results of Kang [86]. One of the influencing factors may be the confinement effect of the tube bundle, where due to the limited space in between the tubes, a certain flow boiling effect is created. Furthermore, heat fluxes in these experiments were low ($< 32 \text{ kW/m}^2$) and thus in the partial nucleate boiling region, which indicates that the effect of the height of the surface may be dependent on the nucleate boiling regime. Overall, most authors agree that the influence of surface height on (unconfined) fully developed nucleate boiling heat transfer is negligible.

The shape of the surface can have a small effect on the heat transfer rate. For example, for nucleate pool boiling on a horizontal cylinder, Kang [88] found that the heat transfer rate was highest at the bottom and lowest at the top. This effect can be attributed to vapour bubbles travelling upwards through gravity and combining to form vapour slugs at the top of cylinder, thereby reducing the number of active nucleation sites. This effect is linked to the effect of inclination of the boiling surface, which is covered in section 3.4.5. As the changes in dimensions and shape typically occur on similar length scale, the independence of the heat transfer rate on the size transfers to the independence on the surface shape.

Only the Cooper correlation includes an influence of heater shape, by increasing the heat transfer rate for boiling with a factor 1.7 for (copper) cylinders, although Cooper himself stated that this seems logically inconsistent [72]. All other correlations do not take into account the heater shape, and none take into account the size of the heater. This seems to be consistent with experimental results, as the influence is small.

Boiling surface inclination

Boiling surface inclination has a noticeable effect on the heat transfer rate. In this text, the orientation of the surface θ is defined compared to a horizontal upward facing surface. This results in $\theta = 0^{\circ}$ for horizontal upward facing surfaces, $\theta = 90^{\circ}$ for vertical surfaces and $\theta = 180^{\circ}$ for horizontal downward facing surfaces (see also figure 3.22).



Figure 3.22: Boiling surface inclination angles.

Niskikawa et al. [89] determined the boiling curve for water at atmospheric pressure on a copper plate, with inclinations equal to 0°, 90°, 120°, 150°, 165° and 175°. They observed that the heat transfer rate increased with increasing inclination in the partial nucleate boiling region, while they saw no influence in the fully developed nucleate boiling region. It was proposed that in the low heat flux regions, the sliding motion of the bubble along the surface improves the singlephase heat transfer to the fluid by breaking up the thermal boundary layer. With higher inclination angles, this effect is increased due to the buoyancy forces pushing the liquid against the surface. At high heat fluxes, boiling heat transfer is more dependent on micro-layer evaporation at a nucleation site, which is independent of gravity. As no skewed upward facing orientations ($0^{\circ} < \theta < 90^{\circ}$) were tested, it is not possible to assess if this trend is also valid for these angles. El-Genk and Guo [90] investigated transient boiling of water at atmospheric pressure on a flat copper disc with inclination ranging from 90° to 180°. The results indicated that for the nucleate boiling regime, the heat transfer rates increased when going from a vertical inclination to a horizontal downward inclination with smaller differences at high heat fluxes. This corresponds to the observations of Nishikawa et al. [89]. Kang [88] tested boiling of water at atmospheric pressure on a stainless steel tube at three inclination angles: 0°, 45° and 90°. The heat transfer rate of the horizontal and vertical tube were very similar, but the heat transfer was significantly enhanced on the inclined tube for the entire nucleate boiling region. Kang concludes that the formation of vapour slugs happens near the top of the horizontal and vertical cylinders, while on the inclined cylinder these slugs are more rapidly evacuated from the surface and liquid access to the surface is improved. El-Genk and Bostanci [84] performed measurements of pool boiling heat transfer of HFE-7100 on a flat copper surface at atmospheric pressure. The measurements show little influence of the inclination angle in the fully developed nucleate boiling region. At heat fluxes closer to the critical heat flux, the results indicate deteriorating heat transfer with increasing inclination. This is mainly due to the decrease of the critical heat flux value with inclination angle. For horizontally downward facing surfaces (θ = 180°), The nucleate boiling region is very limited, due to the extremely reduced

critical heat flux.

Generally, the influence of inclination is different for the three different regions of nucleate boiling heat transfer. At partial nucleate boiling, heat transfer rates increase with increasing inclination due to the increased single-phase heat transfer by bubble motion. In the fully developed nucleate boiling region, the heat transfer rates are mainly unaffected by the inclination angle. For partial dryout nucleate boiling, heat transfer deteriorates with increasing inclination, due to difficulties in evacuating the vapour bubbles from the surface. For boiling on cylinders, the results of inclination are not yet conclusive.

None of the described heat transfer correlations take into account the effect of inclination. This may be a good approximation for fully developed nucleate boiling, but will not be valid for partial nucleate boiling and partial dryout nucleate boiling.

Microgeometry

The microgeometry of the surface has a major influence on nucleate boiling heat transfer. Early researches show the influence of the surface finish on the boiling heat transfer [91]. The general conclusion is that a rougher surface reduces the surface superheat temperature for identical heat fluxes.

Shekriladze and Ratiani [74] take this influence into account in their correlation, by determining the minimal superheat for bubble nucleation from the effective radius of the surface nucleation sites, which is a value dependent on the surface microgeometry. It is however not clear how to determine the value of the effective radius of the nucleation sites. In their nucleate pool boiling model, Mikic and Rohsenow [39] also follow a similar approach by relating the cavity radius to the minimal superheat. As they have no information on the (average) values of the cavity radius, the model requires a fitting constant for each separate surface. Although the model is verified with several measurements, no values of the fitting constant are reported, so it is not clear if the fitting constant can be consistently related to the surface roughness or finishing.

Stephan [92] was the first to link the influence of the surface microgeometry on pool boiling heat transfer to a surface roughness parameter. He used a parameter from the discontinued German standard DIN 4762 called "Glättungstiefe" which roughly translates to roughness depth. This roughness parameter was originally denoted by R_p in the DIN standard. However, as the more common ISO 4287 standard also included another surface roughness parameter denoted by R_p , $R_{p,DIN}$ will be used for the DIN standard parameter. As described by Stephan, $R_{p,DIN}$ is determined by rolling a circle with a radius of 25 mm across the measured surface profile. The ratio of the area in between the surface profile and the profile created by the bottom of the rolling circle to the length of the regarded profile is equal to $R_{p,DIN}$. It is therefore a good indication of the size of the cavities present on the surface. This parameter is also used by Nishikawa et al. [80] and Cooper [72].

Gorenflo et al. [77] initially used the relation proposed by Stephan [92] for their heat transfer correlation. Later on, they suggested that the older parameter $R_{p,DIN}$ could be related to the more commonly used arithmetic average roughness R_a through:

$$R_a = 0.4R_{p,DIN} \tag{3.69}$$

The arithmetic average roughness is also used by Stephan and Abdelsalam [79], Leiner [82] and Ribatski and Jabardo [93]. Other types and definitions for surface roughness parameters are defined by ISO 4287, but are not commonly used for boiling heat transfer correlations. Jones et al. [94] remark that although a correlation exists between nucleate boiling heat transfer and surface roughness parameters, the trends are not captured adequately. They therefore suggest to develop new techniques for characterizing boiling surfaces. Luke [95] proposes to use the a parameter similar to $R_{p,DIN}$ but which takes into account the three-dimensional structure. An envelope surface is defined by rolling a sphere of radius 25 mm along the surface. The volume of the largest sphere that can fit in between the surface and the envelope is determined. The third root of this volume is then the new parameter, which is representative for the cavities on the surface. A visualization of this parameter is shown in figure 3.23. Although this parameter is more related to the boiling mechanisms than the arithmetic average roughness, no correlations to this date take it into account.



Figure 3.23: Visualization of the actual surface (grey), envelope surface (orange) and the largest sphere fitting in between [95].

The influence of surface microstructure on nucleate boiling heat transfer rates

as predicted by correlations can be written in the following format:

$$\frac{\dot{q}}{\Delta T_s} \sim f(R) \tag{3.70}$$

f(R) indicates a function of the surface microstructure parameter, which is equal either to r_c , $R_{p,DIN}$ or R_a for the regarded correlations. Table 3.6 shows the different correlation taking into account the surface microgeometry. Typically, the heat transfer rates increase with the roughness parameter to the power ranging from 0.133 to 0.2. Some authors take into account the effect of the reduced pressure on the power, such as that of Cooper [72]. Jabardo et al. [96] indicate that the correlations which use surface roughness as a parameter are not valid for very rough surfaces ($R_a > 4 \mu m$), as in this range heat transfer rates decrease for rougher surfaces.

Author(s)	f(R)	Range exponent
Stephan [92]	$R_{p}^{0.133}$	0.133
Shekriladze-Ratiani [73]	$r_{c}^{0.4}$	0.4
Nishikawa et al. [80]	$R_p^{0.2(1-p_r)}$	00.2
Cooper [72]	$R_p^{-0.2log(p_r)}$	0∞
Gorenflo [78]	$R_{a}^{0.133}$	0.133
Ribatski-Jabardo [93]	$R_a^{\bar{0}.2}$	0.2

Table 3.6: Dependence of heat transfer rates of surface roughness.

3.4.4 Surface material properties

The surface material properties that can have an influence on the heat transfer behaviour are less numerous than those of the fluid. Firstly, the thermal conductivity can play a role in the heat transfer process. As heat transfer rates are higher around nucleation sites, locally temperatures will also be lower. If the thermal conductivity of the surface is high, the heat flux can be more redistributed and a more uniform temperature will occur on the surface. Secondly, as nucleate pool boiling heat transfer is a transient process of bubble nucleation, growth, departure and waiting time, the transient heating and cooling of the surface can also influence the heat transfer. These effects are related to the volumetric heat capacity of the surface material, which is the product of the density and the specific heat capacity.

Cooper [72] performed a regression analysis to determine the influence of the surface thermal conductivity on nucleate boiling heat transfer. The results showed a dependence with a power of -0.02, which has a negligible effect on the predictive accuracy of the correlation. Gorenflo and Kenning [78] take into account the thermal effusivity of the material, which is an important parameter for transient conduction in a solid. It is defined as the square root of the product of thermal conductivity, density and specific heat capacity. Stephan and Abdelsalam [79] also used the effusivity as a parameter in their regression analysis. They found that only for cryogenics this parameter had a significant effect, albeit with a lower power than proposed by Gorenflo and Kenning [78]. For water, refrigerants and hydrocarbons, the effect of the surface effusivity on heat transfer is not significant. Magrini and Nannei [97] did tests on non-metallic heaters with coatings of five different metals: copper, silver, zinc, nickel and tin. They found that the influence on the boiling curve could be correlated best with the change of the material effusivity. Ribatski and Jabardo [93] did experiments on three different boiling surface materials: copper, brass and stainless steel. Their results do not correlate with the surface effusivity or thermal conductivity. Ribatski and Jabardo propose to use a variable surface parameter in their proposed heat transfer correlation which is equal to 100 for copper, 110 for brass and 85 for stainless steel.

Another interesting effect which is linked to the thermal properties of the heated surface is the surface thickness. Magrini and Nannei [97] report on tests of non-metallic rod heaters coated with different metal layers of different thickness (5 μ m to 250 μ m). They noted that from a certain thickness, there is no influence when altering the thickness. At lower thickness, the heat transfer rates increase with decreasing wall thickness. This effect seemed to be related to the transverse heat conduction in the heater and they correlated the effect to the product of the heater thickness and thermal conductivity. The hypothesis was made that the increase in heat transfer rate can be related to more non-uniformity of temperature in the heater surface, thereby increasing the number of active nucleation sites. Chuck and Myers [98] performed similar nucleate boiling heat transfer tests with stainless steel heaters of thickness 25 μ m, 51 μ m and 130 μ m. The experiments showed very small dependence on the wall thickness, with different behaviour at low (around 40 kW/m²) and high (around 80 kW/m²). Compared to the experiments of Magrini and Nannei [97], the tested heat fluxes were in the lower range, which might explain the contradicting results. Zhou and Bier [99, 38] studied heat transfer from a stainless steel tube coated with a copper layer of variable thickness (2 μ m to 2 mm). Experiments were done for heat fluxes equal to 5 kW/m², 10 kW/m² and 20 kW/m² and fluids pentane and R12. The results show an increase in heat transfer rate with increasing coating thickness. The difference in results with those of Magrini and Nannei [97] might be explained by the lower heat flux range. The shape of the heater (cylinder or plate), may also have a combined effect with the thickness. Overall, the effect of wall thickness is not completely understood.

Generally speaking, there is no agreement on the effect of the surface material properties. This is partly due to the low spread of materials used in experiments (mostly copper and stainless steel) and due to the more prominent surface effects of microgeometry and wettability.

3.4.5 Other factors

Wettability and contact angle

Wetting of a surface is the ability of a liquid to maintain contact to that surface. It is determined by the cohesive forces in the liquid and the adhesive forces of the liquid and the surface. For a liquid droplet resting on a solid surface, this results in a contact angle β on the solid-liquid-vapour contact line, as shown in figure 3.24. The wettability of the fluid-surface combination similarly determines the contact angle in nucleate boiling.



Figure 3.24: Contact angle of a vapour bubble.

Contact angles are difficult to measure with reproducibility. One of the main issues is the effect of the microgeometry on the contact angle. This can be represented by the relation determined by Wenzel [100]:

$$rf = \frac{\cos(\beta')}{\cos(\beta)} \tag{3.71}$$

where rf is the roughness factor (not a surface roughness parameter), which is the ratio of the true surface area to the apparent or envelope area, and β' is the measured contact angle. This equation shows that for actual surfaces where rf > 1 and for contact angles smaller than 90°, the measured contact angle will be lower than the true contact angle. Increasing the surface roughness thus results in increased wetting for most typical surface-fluid combinations.

Another difficulty in utilizing and reporting contact angles for boiling is that the static contact angle is not equal to the dynamic contact angle of advancing and receding contact lines. This is also a result of surface roughness and possibly the heterogeneity of the surface.

Wang and Dhir [101] tested the effect of surface wettability on pool boiling of water on a vertical surface. The heated surface was made of copper and different rates of wettability were achieved by varying the degree of oxidation of the surface. Three different static contact angles were obtained: 90°, 35° and 18°. With increasing surface wettability, the heat transfer rate decreased as did the active nucleation site density. Phan et al. [102] investigated the effect of surface wettability on pool boiling heat transfer using different nanocoatings. They were able to apply different nanocoatings to a stainless steel surface to achieve seven different contact angles: 22°, 31°, 67°, 80°, 85°, 104° and 112°. For the hydrophobic surfaces ($\beta >$ 90°) a rapid transition to film boiling was perceived. The highest heat transfer rates were measured for the contact angle of 85°. For increasing wettability, heat transfer deteriorated to reach a minimum between contact angles of 67° and 31°. For the very hydrophilic surface with contact angle 22°, heat transfer rates increased again. It was concluded that heat transfer was optimal for contact angles close to either 90° or 0°. This was attributed to a high bubble departure frequency around 90° and to the large and thin liquid layer enhancing microlayer evaporation around 0°.

Most correlations do not use the contact angle as a parameter. For the correlations of Rohsenow [58] and Pioro [83], the effect of surface wettability is lumped into the surface-fluid combination parameters. Only the Stephan-Abdelsalam correlation uses the contact angle directly as a parameter [79]. It can be derived from the correlation that the predicted heat transfer rate varies according to the contact angle to the power 0.066, which is very low. For their fitting of the correlation, the contact angles were kept constant to 45° for water, 35° for hydrocarbons and refrigerants and 1° for cryogens. It is therefore doubtful that using actual contact angles in this correlation will result in decent predictions of the boiling curve.

Kwok and Neumann [103] show that the contact angle, if correctly measured, can be determined from the liquid-vapour surface tension σ , which is a fluid property, and the solid surface tension or surface free energy σ_s , which is a property of the solid. This indicates that wettability can be described by a combination of a fluid property and a surface property. A fluid-surface specific parameter such as in the correlations of Rohsenow and Pioro is therefore not a requirement for an adequate correlation as surface wettability can be described by parameters of the fluid and solid which are independent of each other. The fluid surface tension is already featured as a parameter in most correlations. The surface free energy could be another important parameter to include in the correlations and might be an alternative parameter to describe the effects of the surface which are now linked to the thermal conductivity or effusivity.

Subcooling

In the partial nucleate boiling regime, convection is an important contributor to the overall heat transfer process. As subcooling increases the driving force of convection (wall to fluid temperature difference), the boiling curve will show increased heat transfer rates in this region [45]. At higher heat fluxes, convection becomes less important and the main driving force is the surface superheat temperature. Therefore, the influence of subcooling becomes negligible in the regions of fully developed nucleate boiling and partial dryout boiling [45].

Gravity

As this research is focused on applications on Earth, investigating the influence of gravity is out of scope and not of importance for the prediction of nucleate pool boiling heat transfer. However, it is interesting to evaluate the studies of boiling in high and microgravity conditions to assess the assumptions made for deriving heat transfer correlations, as these frequently take into account the gravitational acceleration as a parameter.

Di Marco et al. [104] describe the bubble behaviour for nucleate pool boiling in microgravity. In contrast to boiling on Earth, during microgravity boiling bubbles are not removed from the surface due to buoyancy effects. Rather, a large bubble forms which is either attached to the surface or resides near to the surface, as shown in figure 3.25. New bubbles are formed at the surface, which grow and eventually coalesce with the main vapour bubbles. The size of the bubbles is maintained constant in steady-state boiling by condensation at the bubble cap. Liquid renewal to the surface comes from the sides, thereby indicating that in microgravity, pool boiling heat transfer becomes dependent on the heater size.



Figure 3.25: Nucleate pool boiling pattern in microgravity [104].

Straub [105] gathered the results of two decades of pool boiling heat transfer research in microgravity. The results showed no significant difference in nucleate

pool boiling heat transfer for microgravity and Earth gravity. He proposed that the boiling consists of two types of mechanisms: primary and secondary. The primary heat transfer mechanisms are related to the microlayer heat transfer during bubble growth. These mechanisms determine nucleate boiling heat transfer rates and are not affected by gravity due to the small length scales. Secondary mechanisms are responsible for transport of the latent energy of vapour bubbles away from the surface. In high-gravity environments (for example Earth gravity), this transport is dominated by buoyancy effects. In low-gravity systems, surface tension and thermocapillary effects will results in this transport of energy. The secondary mechanisms do not affect nucleate boiling heat transfer but will affect the critical heat flux (see section 3.5). The independence of the nucleate pool boiling curve on gravity was also found by Henry and Kim [106], provided that the heat fluxes were not too close to the critical heat flux. The experiments of Kannengieser et al. [107] also support the conclusions of Straub. They state that this indicates that correlations utilizing the capillary length (equation 3.18) will not properly predict heat transfer in microgravity. As the capillary length is based on the force balance between surface tension and gravity and the primary heat transfer mechanisms are independent of gravity, this parameter is not adequate to describe the nucleate boiling heat transfer phenomena. Raj et al. [85] investigated pool boiling heat transfer for a continuous range of gravitational acceleration from 0 to 1.8 times the gravitational acceleration at the Earth's surface. Contrary to the results of the previously discussed papers, they found that the heat transfer was influenced by the magnitude of the gravitational acceleration. Two regimes were distinguished: a low-qregime (single large stationary bubble) and a high-g regime (separate departing bubbles). In the low-q regime, heat transfer rates were low. In the high-q regime, heat transfer could be correlated with the gravitational acceleration to the power ranging from 0.12 to 0.23. This effect could be related to the influence of lowered critical heat flux at lower gravity levels. It seems that most measurement points are in the partial dryout regime, which will indeed be affected by the gravitational acceleration.

When regarding the correlations discussed in section 3.3, the correlations based on reduced pressure do not take into account the gravitational acceleration (Borishanskii-Mostinski, Gorenflo, Cooper and Leiner). For the other reported correlations, a similar analysis as for the fluid properties can be made in order to check the power to which the gravitational acceleration is featured in the correlation. The results are shown in table 3.7. The value of the exponent ranges from -0.53 to 0.33. This is mainly because most correlations are fitted on nucleate pool boiling data at Earth's gravity, thereby not including the sensitivity of the gravitational acceleration. From the experimental studies described in the previous paragraph, it seems unlikely that a large dependence on the gravitational acceleration has any physical meaning. This would indicate that several correlations (Rohsenow, KutateladzeBorishanskii, Stephan-Abdelsalam, Kutateladze and Pioro) are not grasping the fundamental boiling phenomena related to heat transfer. However, the correlations can still provide adequate results with the condition that they are only used for Earth's gravity.

Correlation	Exponent
Kruzhilin	-0.03
Rohsenow	0.17
Forster-Zuber	0
Kutateladze-Borishanskii	-0.20
Shekriladze-Ratiani	0
Labuntsov	0
Stephan-Abdelsalam	-0.53
Kutateladze	-0.5
Pioro	0.33

Table 3.7: Exponent of gravitational acceleration in different correlations.

Liquid height

The liquid height is the height difference between a horizontal boiling surface and the liquid-vapour interface. This is an important factor for power electronics cooling as lower heights result in lower weight and volume and thus more compact cooling systems. It is generally agreed that from a certain liquid height, there is no influence on the nucleate boiling heat transfer rates. This was experimentally tested by Shukla and Kandlikar [108] and they found that the the limiting value for the liquid height is about eight times the bubble departure diameter (eqaul to 20 mm for water at atmospheric pressure). They found that heat transfer was enhanced at lower liquid height, with a maximum around five times the bubble departure diameter. The effects however are not consistent when regarding different heat fluxes. According to Pioro et al. [83], the effect of liquid height is less pronounced and boiling heat transfer is unaffected for liquid heights above 2 mm.

Fouling

An unwanted but important factor in nucleate pool boiling heat transfer is the fouling of the heated surface. Over time, deposits can occur on the surface. Typically, any organics of which traces are left in the system (for example oils) tend to dissolve in the fluid. At the heated surface, distillation occurs where the liquid refrigerant is evaporated but the dissolved substance remains on the surface [109]. Over time, this layer will grow and can affect the pool boiling heat transfer rates.

Fukada et al. [110] tested nucleate pool boiling heat transfer from wires with different levels of calcium carbonate scale formation. They found that the bub-
ble dynamics were altered by the fouling layer, more bubbles with a higher frequency were formed on fouled wires. This was attributed to the more porous microstructure of the fouled wire. The nucleate boiling curve however was unaffected. Peyghambarzadeh et al. [111] analysed the effect of calcium sulfate fouling on flow boiling heat transfer rates. From experiments, a fouling resistance was determined as a function of time which asymptotically goes to a maximum value. Initially, for low levels of fouling, the fouling resistance is negative, indicating an increase in nucleate pool boiling heat transfer. This effect is attributed to the induced increase in surface roughness. For further fouling, the fouling resistance becomes positive and keeps increasing, indicating that a layer of low-conductivity material is forming on the surface. Sarafraz and Hormozi [112] determined the fouling resistance for nucleate pool boiling of nanofluids with different concentrations. They found a similar asymptotic behaviour of the fouling resistance as Peyghambarzadeh et al. and also concluded that fouling increased with higher concentrations of nanofluid. Overall, although very low levels of fouling can increase the heat transfer rates, care should be taking in systems to avoid excessive fouling of the heated surface.

Non-uniform heat flux

In typical power modules, heat is dissipated in the semiconductor dies which are placed on a baseplate to increase the heat transfer surface by spreading the heat (see section 2.4 and figure 2.6). This can result in non-uniform heat fluxes from the surface to the boiling fluid. As the surface superheat temperature does not vary linearly with heat flux, simple averaging of the fluxes and temperatures will give different boiling curves. This has been experimentally determined by Wang et al. [113] for flow boiling in a non-uniformly heated tube.

3.4.6 Conclusions

When looking at the sensitivity of nucleate pool boiling heat transfer to the parameters discussed in the previous sections, some conclusions can be made about the correlations discussed in section 3.3.

• When data is available for the surface-fluid combination under investigation, using a fitted correlation, such as the ones of Rohsenow and Pioro, will nearly always perform better than an unfitted one. It is therefore advised to use the Rohsenow correlation if this data is available. However, it is also very much of interest to be able to analyse the performance of different surface-fluid combinations a priori, without requiring experiments for each combination. Therefore, the other correlations are discussed in the following points.

- When looking at the variation of the surface superheat temperature with heat flux, none of the correlations properly predict the behaviour in the three regions (partial nucleate boiling, fully developed boiling and partial dryout boiling). The Forster-Zuber correlation predicts a very diverging relation between superheat and heat flux and is therefore not likely to give adequate predictions.
- The influence of fluid properties is analysed and it is shown that the Kruzhilin, Labuntsov and Kutateladze correlations have diverging effects as a function of reduced pressure when compared to the other correlations and measurement data.
- The microgeometry of the surface has a major effect on the nucleate pool boiling curve. The Kutateladze-Borishanskii and Borishanskii-Mostinski correlations do not take this effect into account. The Shekriladze-Ratiani correlation does have a parameter linked to the surface microgeometry, however it is unclear if this parameter can be determined by other means than from boiling heat transfer measurements.
- It has been shown that gravity has a minor effect on nucleate pool boiling heat transfer. This is inconsistent with the correlation provided by Stephan and Abdelsalam.
- After these considerations, the remaining correlations to determine nucleate pool boiling heat transfer a priori are those of Gorenflo, Cooper and Leiner. These three correlations are all based on the theory of thermodynamic similarity, indicating that the underlying physical phenomena governing nucleate pool boiling heat transfer are still not well enough understood to derive a sound, fluid-property based correlation.

3.5 Critical heat flux

It is crucial to be able to predict the critical heat flux, as this gives a practical limit for the applicability of two-phase cooling for power electronics. An excellent recent review on pool boiling critical heat flux mechanisms, sensitivity and correlations is given by Liang and Mudawar [114, 115]. Several pictures of critical heat flux can also be found in the first work. This section is partly based on their work and summarizes the most important aspects of pool boiling CHF.

3.5.1 Mechanisms

The mechanisms governing CHF have been and still are a subject of widespread interest and debate. No consensus is reached on which mechanism best describes CHF. Liang and Mudawar [114] consider five mechanisms proposed by several authors. The mechanism proposed by Kandlikar [116] is discussed as additional mechanism.

Bubble interference

Rohsenow and Griffith [117] proposed that CHF occurs when neighbouring bubbles start to coalesce. Later studies with high-speed video showed that bubbles start to merge well prior to CHF, contradicting the basics of this mechanism.

• Hydrodynamic instability

Zuber [118] analysed CHF based on the hydrodynamic instability theory. He modelled the boiling phenomenon as jets of vapour leaving the surface in a square layout (figure 3.26). As there is a counterflow of vapour (away from the surface) and liquid (towards the surface), Helmholtz instability can occur which results in merging of the jets and the formation of a vapour blanket. This proposed behaviour is not supported by high-speed photography, where instead of large vapour columns, mushroom-like vapour bubbles can be viewed just before CHF. Furthermore, as CHF is initiated away from the surface, surface effects such as wettability do not influence CHF according to this model. This is shown to be incorrect, as will be discussed in section 3.5.2.

Macrolayer dryout

Haramura and Katto [119] proposed a CHF mechanisms where a large number of vapour stems flow up from the surface and merge in a large vapour bubble hovering above the surface (figure 3.27). The layer in between the surface and the large vapour bubble is referred to as the macrolayer. CHF occurs if the macrolayer completely evaporates just before the large vapour bubble departs. Also for this model, there is no direct photographic evidence of a liquid macrolayer containing vapour stems just before CHF occurs.

Irreversible dry spots

Yagov [120] considered a different mechanism related to dry spots at nucleation sites. At a bubble formation site, a local dry spot occurs where there is no liquid at the surface. This results in a locally higher temperature due to the lower heat transfer to the vapour bubble than to the liquid. If the local temperature becomes high enough, liquid is not able to re-wet the surface after bubble departure and CHF occurs (figure 3.28). There is experimental evidence that these local dry and hot spots occur during nucleate boiling at high heat fluxes, however it is still unclear if these hot spots actually initiate CHF.



Figure 3.26: Hydrodynamic instability model. (a) Vapour jets prior to CHF. (b) Top and side views of the vapour jets. (c) Helmholtz instability and vapour blanket formation [114].



Figure 3.27: Macrolayer dryout model [114].



CHF: Irreversible Spot

Figure 3.28: Irreversible dry spot model [114].

• Interfacial lift-off

Mudawar et al. [121] developed the interfacial lift-off model for CHF in flow boiling and realized that it bears close resemblance to the CHF mechanism for vertical and inclined plate pool boiling. At heat fluxes close to CHF, vapour is generated at the surface which forms a vapour flow along the vertical surface. Due to Helmholtz instabilities, waves occur in this layer, allowable for 'wetting fronts' where liquid briefly comes into contact with the surface (figure 3.29). CHF is then reached when these wetting fronts are no longer able to reach the surface. This theory can also be extended to horizontal surfaces, where the vapour velocity is due to momentum conservation. Contrary to the hydrodynamic theory where CHF is initialized by the merging of quasi-stationary vapour jets, the transient vapour release from the surface initializes CHF when this vapour release is strong enough to avoid liquid from reaching the surface. This theory seems to correspond rather well with the photographic evidence of the boiling behaviour.



Figure 3.29: Interfacial lift-off model [114].

Excess vapour momentum

Kandlikar [116] proposed a model based on the force balance of a bubble attached to the heated surface. He suggests that CHF takes place when high evaporation rates cause the force due to change in vapour momentum (F_M) to become larger than the capillary $(F_{S,1} \text{ and } F_{S,2})$ and gravity forces (F_G) , as this will result in vapour covering the entire surface. The relevant forces are shown on figure 3.30. Evaluating this force balance requires the diameter

of the bubble, which assumed to be equal to half the wavelength of the Taylor instability of a vapour film over the heater surface. Therefore this model is based on hydrodynamic instability, but also takes into account effect of surface wettability and orientation.



Figure 3.30: Force balance on a growing bubble [116].

The mechanisms of bubble interference, hydrodynamic instability and macrolayer dryout seem not to correspond to high-speed video data. The irreversible dry spot, interfacial lift-off and excess vapour momentum could all be viable candidates for the actual mechanism. Measurements using novel flow visualisation techniques could shed more light on the CHF mechanism. This is however not the focus of the current work.

3.5.2 Parameter sensitivity

Fluid properties

To assess the influence of the fluid properties, Kutateladze [122], in 1948, made a pioneering dimensional analysis on CHF taking into account vapour inertia, surface tension and buoyancy effects. The velocity of vapour released from the surface (when neglecting the liquid flow to the surface) can be estimated as:

$$u_v = \frac{\dot{q}}{\rho_v h_{lv}} \tag{3.72}$$

At critical heat flux, a vapour film is formed on the surface. At this heat flux, the kinetic energy of the vapour is proportional to the potential energy of liquid suspended above the vapour:

$$\rho_v u_v^2 \sim g(\rho_l - \rho_v) L_c \tag{3.73}$$

The length scale is taken here as the capillary length $(L_c = \sqrt{\sigma/[g(\rho_l - \rho_v)]}]$, equation 3.18). Combining the previous two equations at \dot{q} equal to \dot{q}_{CHF} results in:

$$\frac{\dot{q}_{CHF}}{\rho_v^{1/2} h_{lv} \left[\sigma g(\rho_l - \rho_v)\right]^{1/4}} = K$$
(3.74)

K is a constant, sometimes referred to as the dimensionless critical heat flux. Many experimental studies have shown that this equation properly predicts the global variation of the critical heat flux with fluid properties. Most correlations described in literature feature this kind of relation.

Several correlations (for example Zuber [118], equation 3.82 and Mudawar et al. [121], equation 3.87) also take into account factors including the vapour and liquid density, usually in the form of $(\rho_l + \rho_v)/\rho_l$ and ρ_v/ρ_l . When the pressure of the fluid is not near the critical pressure, these factors are close to 1 and 0 respectively and can thus usually be neglected.

Equation 3.74 does not contain an influence of (liquid) viscosity. Borishanskii [123] and Dhir and Lienhard [124] argue that for very viscous liquids, CHF will be affected. These effects are considered negligible for refrigerants under consideration in this study.

Some authors, such as Wang et al. [125], suggest that K should be dependent on the reduced pressure. However, as analysed by Liang and Mudawar [115], this does not improve the overall accuracy when compared to discarding this correction.

Boiling surface shape and size

The shape of the boiling surface influences the CHF. Lienhard and Dhir [126] showed that K is larger for horizontal plates than for cylinders, for which in turn it is larger than for spheres.

Critcal heat flux is also dependent on the size of the heater. For very small heaters, CHF is higher. This is characterized by the dimensionless length L':

$$L' = \frac{L}{L_c} \tag{3.75}$$

where L is a characteristic dimension of the surface and L_c is the capillary length as defined by equation 3.18. For refrigerants, the capillary length is mostly in the order of magnitude 1 mm. According to Lienhard and Dhir [126], for L' larger than about 3, CHF is equal to that of an infinitely large flat plate. As the surfaces regarded in this research are an order of magnitude larger, the correction for small heaters is insignificant.

Boiling surface inclination

Several authors have illustrated the dependence of CHF on the inclination angle of the boiling surface [114]. The effect for flat surfaces was quantified with a correction by Chang and You [127] (equation 3.86), the results of which are also shown in figure 3.31. CHF monotonically drops with inclination and becomes very low at downward facing horizontal surfaces, as vapour accumulates more easily at the surface.



Figure 3.31: Effect of surface orientation on CHF [127].

Microgeometry

Contradicting conclusions to the effects of microgeometry and surface roughness on CHF can be found in literature. Some authors state that CHF increases with surface roughness, while other researchers find no influence in their measurements [114]. The small influence of surface microgeometry may be related to the change of the apparent contact angle (see equation 3.71). The influence of surface wettability and contact angle is described in one of the following paragraphs.

Surface material properties

CHF is reduced on thin heaters. Several authors have correlated this effect to the parameter S, defined as the product of the heater thickness δ and the surface

thermal effusivity:

$$S = \delta \sqrt{k_w \rho_w c_{p,w}} \tag{3.76}$$

Arik and Bar-Cohen [128] correlated the effect of this parameter on CHF as:

$$K \sim \frac{S}{S+0.1} \tag{3.77}$$

This indicates that if S is larger than 10, CHF is within 1% of the asymptotic CHF value for an infinitely thick heated surface. Baseplate materials of power modules are usually made out of copper or aluminium. For S to be larger than 10, the baseplate thickness should be larger than 0.3 mm and 0.5 mm for copper and aluminium respectively. To allow for heat spreading and mechanical strength, baseplates are always thicker, indicating the effect on CHF of thermal properties and thickness of the baseplate of power modules is negligible.

Wettability and contact angle

CHF increases with increasing wettability and decreasing contact angle. This is illustrated by the data of Kwark et al. [129] in figure 3.32. CHF was measured on surfaces with different nanocoatings resulting in different contact angles. They proposed a linear relation between CHF and contact angle. According to Liang and Mudawar [115], the correlation proposed by Kandlikar [116] (equation 3.88) provides a better overall prediction for the effect of the contact angle on CHF.



Figure 3.32: Effect of contact angle on CHF [129].

Subcooling

Subcooling significantly increases CHF. The effect of subcooling is typically correlated with the subcooling Jakob number Ja_{sub} by the following relation:

$$K \sim 1 + B J a_{sub} \tag{3.78}$$

The definition of the subcooling Jakob number includes the temperature difference between the liquid temperature T_l and the saturation temperature:

$$Ja_{sub} = \frac{c_{p,l}(T_{sat} - T_l)}{h_{lv}}$$
(3.79)

According to Ivey and Morris [130], B is best described by:

$$B = 0.102 (\rho_l / \rho_v)^{0.75} \tag{3.80}$$

Elkassabgi and Lienhard [131] evaluated a larger range of subcooling temperatures. They found a similar linear relation between CHF and subcooling in the low subcooling range ($\Delta T_{sub} < 20^{\circ}$ C ... 40°C). In the highly subcooled region ($\Delta T_{sub} > 40^{\circ}$ C ... 60°C), CHF becomes independent of changes in subcooling and reaches a maximal value, which is about three times larger than CHF at saturated state. This was attributed to the limit set by the effusion of molecules from an interface.

Gravity

Equation 3.74 predicts a variation of CHF with (gravitational) acceleration to the power 0.25. Ivey [132] confirmed this behaviour by measuring CHF in a centrifugal apparatus with acceleration ranging from 1 to 160 times Earth's gravitational acceleration. The best fitting power was 0.27, which is fairly close to 0.25. The influence of gravity on CHF does not conflict with the theory of primary and secondary mechanisms by Straub [105] as described in section 3.4.5. CHF is related to the vapour removal from the surface, which is a secondary mechanism and can therefore be dependent on gravity. At microgravity, the vapour removal from the surface is not determined by gravitational but by other forces, indicating that different correlations are required for CHF at microgravity.

Liquid height

The effect of liquid height on CHF was experimentally determined by Shukla and Kandlikar [108] for pool boiling of water. The results show that from liquid heights above 30 mm, CHF is unaffected by the liquid height (figure 3.33). At lower liquid heights, CHF is severely reduced.



Figure 3.33: Effect of liquid height on CHF [108].

Fouling

Fouling of the heated surface has an effect on the CHF. Bui and Dhir [133] found that CHF was increased by around 25% when comparing a clean and a fouled surface. A similar effect was perceived by Fukada et al. [110] for boiling on wire with and without calcium carbonate scale formation. CHF was about two times higher for the scaled wire. Kim et al. [134] noticed an increase in critical heat flux with pool boiling of nanofluids. This was linked to the deposition of the nanoparticles on the surface, which in turn caused increased wettability of the surface. Overall, all data so far shows that fouling has a positive effect on CHF, which is presumably linked with increased wettability.

Non-uniform heat flux

As the heat dissipated in a power module is not uniformly distributed over the baseplate, local hot spots can occur. These hot spots could cause local premature dryout. This effect has been illustrated by Revellin et al. [135] for flow boiling in microchannels.

3.5.3 CHF correlations

The correlations listed here are those that are cited most (Kutateladze [122], Zuber [118]) and those that have been shown by Liang and Mudawar [115] to fit best to

experimental data (Lienhard-Dhir [126], Chang-You [127], Mudawar et al. [121] and Kandlikar [116]).

Kutateladze (1948)

After the dimensional analysis which resulted in equation 3.74, Kutateladze analysed the available experimental data and proposed the following correlation:

$$\dot{q}_{CHF} = 0.16 \,\rho_v^{1/2} h_{lv} \left[\sigma g(\rho_l - \rho_v)\right]^{1/4} \tag{3.81}$$

. . .

Zuber (1958)

Zuber [118] used the hydrodynamic model for CHF along with the following assumptions: the vapour jets are located with spacing equal to a Taylor wavelength, the jet diameter is equal to half this wavelength and the critical Helmholtz wavelength is equal to the circumference of the jet. The Taylor wavelength is assumed to lie somewhere in between the 'critical' wavelength and the 'most dangerous' wavelength, which differ by a factor $\sqrt{3}$. This results in the following equation:

$$K = C_{Zu} \frac{\left(\frac{\rho_l + \rho_v}{\rho_l}\right)^{1/2}}{1 + \frac{\rho_v}{\rho_l} \frac{\pi}{16 - \pi}}$$
(3.82)

 C_{Zu} is a constant in the following range:

$$C_{Zu} \in \left[\frac{\pi}{24} \frac{3}{\sqrt{2\pi}} \frac{1}{3^{1/4}}, \frac{\pi}{24} \frac{3}{\sqrt{2\pi}}\right]$$
 (3.83)

Zuber chose the intermediate value of $\pi/24$ for the final correlation. A further simplification can be made for pressures well below the critical pressure, resulting in the final correlation:

$$\dot{q}_{CHF} = 0.131 \,\rho_v^{1/2} h_{lv} \left[\sigma g(\rho_l - \rho_v)\right]^{1/4} \tag{3.84}$$

Lienhard-Dhir (1973)

Lienhard and Dhir [126] re-examined the model of Zuber and suggested that a constant of 0.149 fits experimental data better. This corresponds to the assumption that the critical Helmholtz wavelength is equal to the 'most dangerous' Taylor wavelength. The resulting correlation is thus:

$$\dot{q}_{CHF} = 0.149 \,\rho_v^{1/2} h_{lv} \left[\sigma g(\rho_l - \rho_v)\right]^{1/4} \tag{3.85}$$

This correlation renders the best prediction for horizontal upward facing plates with no information of the contact angle [115] and is thus preferred over those of Kutateladze [122] and Zuber [118].

Chang-You (1996)

Chang and You [127] analysed the effect of the inclination of the surface on CHF and proposed the following correction for the inclination angle θ :

$$\frac{\dot{q}_{CHF}}{\dot{q}_{CHF,max}} = 1 - 0.0688\theta \tan(0.414\theta) - 0.122\sin(0.318\theta)$$
(3.86)

This correction can be combined with the Lienhard-Dhir correlation (equation 3.85) for the maximal critical heat flux $\dot{q}_{CHF,max}$. This correlation provides the best results for predicting CHF with varying inclination and no knowledge of the contact angle [115].

Mudawar et al. (1997)

Based on the analysis of the interfacial lift-off model, Mudawar et al. [121] derived the following correlation:

$$K = 0.151 \frac{\rho_l}{\rho_l + \rho_v} (1 + Ja_{sub})$$
(3.87)

For pressures far below critical pressure and saturated boiling, this correlation agrees within 2% to the Lienhard-Dhir correlation (equation 3.85), even though it was based on a different model.

Kandlikar (2001)

The correlation of Kandlikar [116] is based on the force balance of a growing bubble and is given by:

$$K = \frac{1 + \cos\beta}{16} \left[\frac{2}{\pi} + \frac{\pi}{4} (1 + \cos\beta) \cos\theta \right]^{1/2}$$
(3.88)

When the contact angle is known, the Kandlikar correlation provides the most accurate results [115].

3.6 Nucleate pool boiling heat transfer with FK-649

Section 2.5 gives an overview and selection of possible coolants to be used for pool boiling cooling of power electronics. The selected fluid FK-649 is $C_2F_5C(O)CF-(CF_3)_2$ or 1,1,1,2,2,4,5,5,5-nonafluoro-4-(trifluoromethyl)-pentan-3-one according to IUPAC nomeclature. It is also known by its trade name Novec 649 (and also Novec 1230) by the company 3M [136]. Another possible name is FK-5-1-12 according to the ASHRAE numbering system [12].

In 2008, FK-649 was introduced to the market as heat transfer fluid [137]. Table 3.8 gives an overview of the pool boiling studies conducted with FK-649 since then. The overview in the table is limited to standard flat surface, other geometries (wires, tubes) and engineered surfaces are not included, as these are of less interest for this work. Forrest et al. [138] show that pool boiling heat transfer of FK-649 is very similar to that of PFC-72 with the advantage of much lower GWP. In a different study, Forrest et al. [53] report on measurements of FK-649 boiling of an aluminium disc and give a value of 0.0037 for the surface-fluid constant C_{sf} in the Rohsenow correlation (with r equal to 0.33 and s equal to 1.7). Besides the fitted Rohsenow correlation, the Stephan-Abdelsalam correlation gave the best predictions for pool boiling heat transfer rates. Gess et al. [139] investigated two-phase immersion cooling on bare dies for data centres. Kaniowski et al. [140] tested pool boiling on a copper surface for microelectronic chip cooling. Another application, electric motor winding cooling, was envisaged and tested by Bartle et al. [141]. Ghaffari et al. [142] performed experiments on pool boiling of FK-649 and HFE-7100 for microprocessor cooling. Both fluids featured similar heat transfer rates, but CHF was about 20% higher for HFE-7100. Cao et al. [143] experimentally determined the boiling curve of FK-649 on a smooth copper surface and reported a value of 0.0050 for the surface-fluid parameter C_{sf} in the Rohsenow correlation. Furthermore, the boiling curves were compared to those predicted by the Stephan-Abdelsalam, Cooper, Gorenflo and Leiner correlations of which Stephan-Abdelsalam and Cooper gave the best results.

Author(s)	Surface	Α	R_a	$\mathrm{T_{sat}}$	ģ
	material	$[\mathbf{cm^2}]$	$[\mu \mathbf{m}]$	$[^{\circ}\mathbf{C}]$	$[kW/m^2]$
Forrest et al. [53]	Aluminium	5	0.41	49 - 96	5 - 215
Gess et al. [139]	Silicon	6	N.A.	49	5 - 120
Kaniowski et al. [140]	Copper	11	0.13	49	5 - 70
Bartle et al. [141]	Copper	2 - 6	0.83	49	10 - 75
Ghaffari et al. [142]	Copper	5	N.A.	49	40 - 195
Cao et al. [143]	Copper	1	0.09	49	5 - 173

Table 3.8: List of studies on pool boiling of FK-649.

Generally, FK-649 seems to behave similarly to other widely tested pool boiling fluids. Of the two studies that compared correlations for nucleate pool boiling heat transfer with experimental data on FK-649, the Stephan-Abdelsalam correlations provided the best predictions without fitting. For the Rohsenow correlation, two values of the surface-fluid parameter have been reported for aluminium and copper, although this value is expected to also be dependent on surface roughness. The surface areas tested are significantly smaller than those under study in this research for power electronics cooling (51 cm²). It is however expected that the size of the boiling surface will have a negligible effect on the boiling curve. Only one study (Forrest et al. [53]) experimentally determined the influence of the saturation temperature and pressure. Saturation pressures below atmospheric are not yet tested.

Some non-thermal aspects of pool boiling cooling with FK-649 are also relevant to address here. Tuma [109] gave a thorough overview of these considerations for two-phase immersion cooling systems. Although FK-649 is compatible with most common metals and plastics, some caution is required when using elastomers. Usually, traces of oil are left on these elastomers, which can dissolve in the liquid coolant. Due to evaporation occurring at the boiling surface, these oils are deposited and the surface becomes fouled. This can impede nucleation and thus result in reduced heat transfer rates. Thorough pre-cleaning or adding an absorbent such as activated carbon can help to avoid this effect. A drawback of FK-649 is that it reacts with liquid water, forming HFC-227ea and perfluoropropionic acid [137]. The acid is highly corrosive to metals. A system containing FK-649 should thus be properly sealed, avoiding water carried by air to enter the system. Silica gel packets can be added to the system to absorb any water entering.

3.7 Research questions

Chapter 2 and the present chapter give an overview of the state of the art of twophase immersion cooling of power electronics. FK-649, a refrigerant with a low global warming potential, was chosen as working fluid. The overview resulted in a number of research questions which will be answered in this work. As there is both a fundamental and a technical research objective of this study, also the research questions can be divided into these two categories.

Fundamental research questions

- Which correlation is most suited to predict heat transfer rates during pool boiling with FK-649? Can a correlation without fitting constants provide adequate enough predictions? Do the correlations adequately predict the variation of the boiling curve with varying saturation pressure?
- Do the correlations properly describe the variation of the boiling curve with heat flux? Is it necessary to take into account the different regimes of nucleate boiling when using a correlation?
- Which correlation is most suited to predict the critical heat flux for FK-649? Do the correlations adequately predict the variation of the critical heat flux with varying saturation pressure?

Technical research questions

- Is the effect of liquid height on the boiling curve and critical heat flux negligible in the range of interest for power electronics cooling?
- Is there an effect of the non-uniform heat flux from the semiconductor dies on the boiling phenomenon? Will this affect the boiling curve and/or the critical heat flux?
- Is two-phase immersion a feasible method for cooling of power electronics for high-power density drivetrains?

Both the fundamental and technical research questions will be answered by means of an experimental measurement campaign. The experimental setup is described in the next chapter, while chapters 5 and 6 elaborate on the experimental results. In chapter 7, these experimental results are compared to predictions of the reported correlations.

Boiling and nucleation measurement device

4.1 Introduction

The previous chapter defined a set of research questions, all related to the pool boiling curve (the relation between surface heat flux and surface superheat temperature). To answer these questions, a multifunctional setup is designed to measure the heat transfer from a solid surface to a fluid by pool boiling. In this setup, different surfaces can be used, as this can for example be a flat plate (horizontal or vertical) or an electrical component. The fluid to which the heat is transferred is also interchangeable. This setup is the further development of the initial design by Jasper Nonneman, details can be found in [144].

To fully understand the boiling heat transfer phenomenon, the saturation temperature of the fluid during boiling should be varied as this will affect the fluid properties. Since the saturation temperature is directly linked to the pressure of the fluid, a fully sealed reservoir is needed to be able to vary the pressure and saturation temperature.

The surface is heated by dissipating electrical energy which is supplied by a laboratory controllable power source. In order to achieve a steady fluid saturation temperature, the fluid also has to be cooled and thus condensed. A spiral condenser is added in the reservoir through which a water and ethylene glycol mixture is pumped at a temperature below the saturation temperature. This flow of cold water-glycol mixture is provided by a chiller which can control the temperature. The mass flow rate going to the reservoir is controlled by a three-way valve.

To analyse the quality of the measurements, the energy balance of the setup is also determined by adding up all the energy flows to the reservoir. These energy flows are the electrical energy dissipated in the setup, the energy carried by the coolant flowing to and from the reservoir and heat losses from the reservoir to the environment. Accounting for all the energy flows in the system is essential to make sure that the measured heat flux from the solid to the fluid is correctly determined.

Figure 4.1 shows the experimental setup, consisting of the reservoir filled with refrigerant, coolant circuit (not visible, see figure 4.7) with chiller, heating elements and controls and measurement instrumentation. Each of these parts of the setup will be described in detail in the next sections. In the last three sections, an analysis of the measurements is made, including the measurement procedure, processing and uncertainty.

4.2 Reservoir

The sealed reservoir which contains the refrigerant is a rectangular cuboid with a height of 330 mm, a width of 176 mm and a depth of 213 mm. The walls are made out of stainless steel plates with a thickness of 8 mm. All walls of the reservoir are welded together, except for the bottom wall which is bolted to the side walls. Two of the side faces have a rectangular opening through which the boiling phenomena can be observed. Two polycarbonate windows with a thickness of 10 mm are bolted to the reservoir openings. To seal the bolted contact areas, silicone sheets with a thickness of 0.5 mm are placed in the junctions. Valves are added to the top and bottom walls to allow for the evacuation of air and the filling and emptying with refrigerant. All the walls of the reservoir are insulated by adding a layer of polyurethane on the outside with a thickness of 5 cm. A simplified 3D view of the reservoir (without the front plate and the insulation surrounding it) is shown in figure 4.2 and a picture is shown in figure 4.3.

In this work, the heat transfer from a horizontal surface is studied. Both a configuration with a resistive heater as heat source and with a power module are tested. More details on the specifics of both designs are given in chapters 5 and 6 respectively. To incorporate the heat flux from a horizontal surface, the bottom wall is made with a rectangular cut-away of 108 mm by 47 mm, dimensions which are based on those of the studied power module (see section 6.2 for more details). To limit heat transfer between the heat source and the bottom reservoir plate, PMMA (polymethyl methacrylate) or POM (polyoxymethylene) is chosen as plate material instead of stainless steel, for the heater and power module measurements respectively. Both have relatively low thermal conductivities, respectively 0.2 W/mK and 0.35 W/mK.

In the upper part of the reservoir, a spiral wound copper coil is brazed. The



Figure 4.1: Boiling and nucleation measurement device. 1: reservoir, 2: chiller, 3: power supply, 4: data acquisition system, 5: reference junction



Figure 4.2: 3D CAD of the reservoir without insulation and front plate.



Figure 4.3: Picture of the reservoir without insulation.

tubing has an outer diameter of 16 mm, wall thickness of 0.8 mm and the total length of the tube inside the reservoir is 3 m. This results in a total heat transfer area of 0.151 m². Considering a maximal heat dissipation of 1000 W and an overall heat transfer coefficient of 500 W/m²K, which is a conservative estimation for condensation and water-glycol forced convective heat transfer, a temperature difference of 13.2 °C is required between the refrigerant and the coolant. This indicates that for a saturation temperature of 20 °C, the water-glycol mixture should be able to reach temperatures down to 6.8 °C. The cooling rate (1000 W) and inlet temperature (6.8 °C) are the main criteria required for the chiller (see section 4.4).

From the side wall of the reservoir, two thermocouples are inserted through compression fittings. Each thermocouple is positioned to measure the temperature in either the liquid or the vapour phase, which should be equal if the reservoir is in thermodynamic equilibrium and if there is no air present in the reservoir. Additionally, a pressure transducer is added next to the thermocouple measuring the liquid temperature. If the refrigerant is in a two-phase state, the measured pressure can be directly related to the refrigerant temperature by the vaporization curve of the refrigerant (see figure 4.12). A cross-sectional view of the reservoir showing the position of the thermocouples is given in figure 4.4. More information on the sensors and measurements is given in section 4.5.



Figure 4.4: Cross-sectional view of the reservoir.

4.3 Refrigerant

The refrigerant used in the experiments is $C_2F_5C(O)CF(CF_3)_2$, more commonly known by its trade name of the company 3M, Novec 649 (datasheet can be found in appendix D.1). FK-649 will be used to designate the fluid in this work. It is a fluoroketone with a boiling point of 49 °C at atmospheric pressure, which makes it applicable for two-phase immersion electronics cooling. It is dielectric, has no ozone depleting potential (ODP) and has a global warming potential of 1, which is significantly lower than most alternatives. Furthermore, it is non-flammable and not toxic. A more detailed description of the analysis and choice of the refrigerant is given in section 2.5.

An overview of the important parameters for pool boiling heat transfer is given in table 4.1. Using the correlation of Lienhard and Dhir [126] (equation 3.85), the critical heat flux can be estimated as a function of the fluid properties and saturation temperature. The estimation is shown in figure 4.5. The maximal heat flux is 182 kW/m². For the horizontal plate surface of 108 mm by 47 mm, this results in a maximal electrical power dissipation of 925 W. For the design calculations, the maximal electrical power rating is rounded to 1000 W.

316 g/mol
169 °C
18.8 bar
0
1
49 °C
88 kJ/kg
1527 kg/m ³
12.76 kg/m ³
1122 J/kgK
0.054 W/mK
0.3 mm ² /s
10.8 mN/m

Table 4.1: Selected fluid properties of FK-649 [136].

4.4 Coolant circuit

An overview of the coolant circuit is shown in figure 4.6 and a picture (without the chiller) is shown in figure 4.7. The goal of this coolant circuit is control both the mass flow rate and temperature of the coolant delivered to the condenser of the reservoir. By transferring heat from the reservoir to the coolant, the refrigerant in the reservoir condenses and the test setup can operate in steady state. The circuit



Figure 4.5: Critical heat flux for FK-649 calculated with the correlation of Lienhard and Dhir [126] as a function of saturation temperature.

consists of a chiller, mixers and temperature measurements before and after the reservoir, a filter, a volumetric flow meter and a controllable three-way valve.

The fluid used in the coolant circuit is a mixture of ethylene glycol and water. Although adding the ethylene glycol to the coolant results in inferior fluid properties for heat transfer (lower thermal conductivity, lower specific heat capacity and higher viscosity), it is added to avoid freezing at the walls of the chiller evaporator as it decreases the freezing point. To calculate the water-glycol properties with negligible errors, the concentration of ethylene glycol in the mixture has to be accurately measured. This is done by weighing the separate components before mixing with a scale with a measurement uncertainty of ± 1 g. The mass concentration of the ethylene glycol-water mixture used in the test setup is equal to 38.24%.

A Neslab M100 chiller is used to provide cooling (datasheet can be found in appendix D.2). The chiller can control the delivery temperature of the waterglycol mixture from -15 °C to 35 °C with a stability of \pm 0.15 °C. It has a maximal cooling capacity of 3.5 kW, which decreases with decreasing delivery temperature, starting from 15 °C, as shown by curve C in figure 4.8 on the top. At 6.8 °C, it is able to cool about 1.7 kW, which is sufficient for the range of powers which will be investigated in this work. The chiller delivers a maximal pumping head of 4 bar and a maximal flow rate of approximately 12 L/min, according to curve C in figure 4.8 on the bottom. The chiller has an internal three-way valve, which can be adjusted manually. This valve is tuned such that the external three-way can be operated in the flow rate range which is desired for optimal control and measurement accuracy (see sections 4.6 and 4.8.3).



Figure 4.6: Coolant circuit. m = mixer, T = temperature measurement, V = volumetric flow measurement.



Figure 4.7: Picture of the coolant circuit. 1 = inlet thermocouples, 2 = outlet thermocouples, 3 = flow meter, 4 = three-way valve, 5 = actuator.



Figure 4.8: Cooling (top) and pumping (bottom) capacity of the Neslab M100 chiller.

To determine the energy transferred from the reservoir to the coolant, the temperature of the coolant entering and exiting the reservoir has to be measured. Two thermocouples are positioned in both the flow inlet and outlet of the reservoir. Due to the heat transfer in the reservoir and chiller, a temperature profile develops in the coolant flow, which can result in significant differences between the locally measured temperatures and the bulk temperature. The bulk temperature is the relevant tempeature to determine the energy balance. To make sure that the bulk temperature of the coolant is measured, mixers are added in the tubing upstream of the temperature measurements. These elements thoroughly mix the coolant flow, thereby achieving a uniform coolant temperature at the mixer outlet. The design of the mixers is shown in figure 4.9. It consists of four copper twisted tape inserts with a rotation of 180° and a length of 1.5 times the tube diameter. The inserts are brazed together, each time changing the sense of the rotation. The combination of the four inserts is then brazed to the tubing. The design of these mixers is based on the work of Meyer and Everts [145], which in turn is based on the simulations by Bakker et al. [146].

The flow rate to the reservoir has to be controlled and thus also measured. The measurement is done by an oval gear volumetric flow meter (more information on the sensor can be found in section 4.5). To avoid particles entering and clogging the gears, a filter is placed upstream of the meter. To control the flow rate, a Danfoss VRB3 three-way valve with an AME35 actuator is used, which permits a partial bypass of the flow coming from the chiller. More information on the control of the flow rate can be found in section 4.6.



Figure 4.9: Mixers to achieve a uniform temperature distribution in the coolant.

4.5 Sensors and data acquisition

4.5.1 Sensors

The temperature of the boiling plate is measured by type K mineral-insulated thermocouples with a diameter of 0.5 mm. Type T mineral-insulated thermocouples with a diameter of 1 mm are used to measure the temperature of the refrigerant and coolant. The temperature of the reference junction of all thermocouples is determined with a miniature PT100 temperature sensor. The reference junction is insulated from the environment by a box of polyurethane with a wall thickness of 50 mm.

All thermocouples are calibrated using an in-house procedure. The thermocouples are placed in an oven together with an externally calibrated PT100 sensor with a measurement uncertainty of \pm 0.06 °C. The oven is set in the temperature range from 0 °C to 100 °C with steps of 10 °C and the thermocouple reading is compared to that of the PT100. A second degree polynomial is fitted to take into account the deviation between both measurements. This results in an uncertainty of the thermocouple measurements of \pm 0.07 °C.

To monitor the temperature inside the power module, a negative temperature coefficient (NTC) thermistor is provided by the manufacturer. The thermistor is calibrated by placing the module in an oven and slowly heating it up to 80 °C, after which it is cooled down again. The calibrated thermocouples are used as reference temperature. The result of the calibration is shown in figure 4.10.

The pressure is measured using a GEMS series 2200 type pressure transducer (datasheet can be found in appendix D.3). The strain gauge in this sensor is manufactured by the chemical vapour deposition (CVD). The sensor measures the absolute pressure in a range from 0 bar to 1.6 bar with an accuracy of \pm 0.0045 bar. For FK-649 at atmospheric pressure, this uncertainty of 450 Pa relates to an uncertainty of \pm 0.13 °C on the saturation temperature (for the calculation of the uncertainty, see equation B.3). The sensor has a 0 V to 5 V voltage output.

The volumetric flow rate is measured by an oval gear flow meter. This sensor has a measurement range from 0.2 L/min to 8.3 L/min with an accuracy of 1.5% of the reading value. The output of the sensor is a frequency signal, which generates 390 pulses per litre flowing through the sensor.

4.5.2 Data acquisition

The reference junction PT100, all thermocouples and the output signal of the pressure sensor are all measured using a Model 2700 Keithley multimeter. When the power module is used as heat source, additionally the power module voltage and NTC thermistor resistance are measured by the Keithley multimeter. For all derived measurements (temperatures and pressure), the uncertainty of the voltage or



Figure 4.10: Measured temperature of the power module as a function of the measured NTC thermistor resistance and the corresponding calibration curve.

resistance measurement is negligible compared to the inherent uncertainty of the sensor. The power module voltage can go up to 10 V, in this range the uncertainty on the voltage measurement by the multimeter is \pm 50 μ V \pm 0.003% of the reading value.

The current to the heater and power module is supplied and measured by a programmable laboratory DC source. The type of source depends on the device under test (heater or power module), as both require different voltage and current levels to achieve similar heat dissipation rates. For the heater configuration, an Elektro-Automatik PS 9360-30 2U programmable laboratory DC power supply is used. This supply can deliver power up to 360 V, 30 A and 3000 W. For the power module configuration, higher currents are required and an Elektro-Automatik PS 9080-120 2U programmable laboratory DC power supply is used. This supply can deliver power up to 80 V, 120 A and 3000 W. The current is measured by both sources with an uncertainty of $\pm 0.2\%$ of the rated value. For the heater configuration, this results in an uncertainty of ± 0.06 A, while for the power module configuration the uncertainty is ± 0.24 A. For the heater configuration, the voltage is also measured by the power supply, with an uncertainty of 0.1% of the rated value, which results in an absolute uncertainty of ± 0.36 V.

The pulse frequency output of the flow meter is read using an Arduino with

custom coding. The program determines the frequency by counting the time between different pulses. The uncertainty on the measurement of the frequency is negligible compared to the inherent uncertainty of the flow sensor.

All data acquisition units are read using a LabVIEW program. This program reads, processes and visualizes the measurements and writes the data to a text file for further processing.

4.5.3 Summary

Table 4.2 gives a summary of all the quantities measured, used sensors, measurement ranges and uncertainties.

Quantity	Sensor	Range	Uncertainty	Location(s)
Temperature	Thermocouple			
	- Type T	-270	\pm 0.07 °C	Refrigerant,
		400 °C		coolant
	- Type K	-270	\pm 0.07 °C	Boiling
		1372 °C		surface
	Thermistor	0 150 °C	\pm 1 °C	power module
Pressure	CVD	0 1.6 bar	\pm 450 Pa	Reservoir
Flow rate	Oval gear	0.2 8.3	$\pm 1.5\%$	Coolant
		L/min		
Voltage	Keithley	0 10 V	\pm 50 $\mu { m V}$	power module
			$\pm 0.003\%$	
	DC source	0 360 V	\pm 0.36 V	Heater
Current	DC source	0120 A	\pm 0.24 A	power module
	DC source	030 A	\pm 0.06 A	Heater

Table 4.2: Summary of all sensors used in the boiling and nucleation measurement device.

4.6 Controls

The setup has three control variables: the voltage or power going to the power module or heater respectively, the coolant inlet temperature and the coolant flow rate. The goal is to acquire different setpoints of an experimental matrix by controlling two process variables: the refrigerant saturation temperature and the boiling surface temperature. As there is one redundant control variable to achieve the desired setpoints (three control variables and two setpoints), an additional goal of the control setup is to increase the accuracy of the experimentally determined energy balance.

The coolant inlet temperature is manually set through the interface of the chiller with a resolution of 1 °C. The coolant flow rate is set by applying a voltage signal to the actuator of the three-way valve. The 0 V to 10 V signal is supplied by a National Instruments PCI-6722 module, which is set by a LabVIEW program. A digital PI-controller is used to set the output signal, with the P and I coefficient manually optimized. A combination of coolant inlet temperature and flow rate is subsequently manually set, such that the refrigerant saturation temperature achieves a steady-state value which is within \pm 0.5 °C of the setpoint. Next to this, the coolant flow rate is chosen sufficiently low (depending on the heat flux setpoint), such that the temperature difference of the coolant between reservoir inlet and outlet is high enough to reach the desired accuracy of the energy balance determination. A detailed analysis on the uncertainty of the experimental energy balance can be found in section 4.8.3.

The boiling surface temperature is controlled by applying a certain voltage or power to the heat source. This is done by an EA laboratory DC source (see section 4.5). The power supply is controlled remotely by the LabVIEW program. For the initial measurements, a constant electrical power was applied. The results showed that the response of the surface temperature had a time constant of several hours. Furthermore, it was very difficult to assess if steady state was reached, as variations as small as 0.1 °C over an hour could indicate a deviation from the steadystate value in the range of 1 °C, which is a significant measurement error. To reach steady state faster, the voltage setting is actively controlled by a PI-controller to reach a constant surface temperature. Adequate P and I coefficients were estimated by measuring the step response of the surface temperature and fitting a transfer function to this response. Using the transfer function, the PI-controller was tuned using the 'pidtune' function of Matlab. During the measurements, these parameters are also manually adjusted to either react faster to changes in setpoint or to be less sensitive to noise on the measurement signals near the steady-state condition. The controller only influences the transient behaviour in between the steady-state measurement points. When reaching the desired setpoint, the variation in surface temperature is within the measurement accuracy, which results in the controller applying a constant power (within the measurement accuracy).

4.7 Measurement procedure

The measurement procedure consists of several steps. At the start of a measurement sequence, the setup is de-aired. Subsequently, electrical power is supplied to heat up the refrigerant to the desired saturation temperature. When the saturation temperature is reached, the chiller is turned on. The coolant inlet temperature, coolant flow rate and boiling surface temperature are set to achieve the desired conditions. When steady state is reached, the measurements are logged. After this, the steps of setting the controls, waiting for steady state and logging the steady-state condition is repeated for the desired amount of measurement points. Afterwards, the electrical power supply is turned off and the setup is cooled down by the coolant circuit. A flowchart summarizing the measurement procedure is shown in figure 4.11.

Although care is taken in the design and assembly of the reservoir by adding both silicone sheets and silicone mastic, some air will still find its way into the reservoir. At temperatures below 49 °C, the saturation pressure of FK-649 is below atmospheric pressure. Therefore, leakages result in air entering the reservoir. The amount of air in the reservoir can be assessed by comparing the temperature in the reservoir to the saturation temperature determined by the pressure measurement. Saturation temperatures significantly higher than the refrigerant temperature indicate significant quantities of air in the reservoir. As the FK-649 vapour is more dense than air, the air will always accumulate at the top of the reservoir. Two methods are applicable to de-air the setup. The first method consists of heating the reservoir to a temperature above 49 °C, which will result in a pressure above atmospheric pressure. As the air accumulates at the top of the reservoir, the valve at the top can be opened such that the air can evacuate the reservoir. All air is removed when the measured temperature and calculated saturation temperature return to the same value. The second method to remove the air is by connecting a vacuum pump to the valve at the top of the reservoir. The valve is opened slightly, such that a steady flow of air is drawn out of the reservoir. Both methods unavoidably come at a slight loss of refrigerant vapour mixed with the outgoing air. Mostly the second method of de-airing is used, as this method is faster than heating due to the large thermal inertia of the reservoir.

Heating the refrigerant and reservoir is done through the electrical heater or power module, as the chiller can only provide cooling and no additional refrigerant heater is added. This is done by supplying a constant heating power or by setting a boiling surface temperature higher than the desired saturation temperature. When the desired saturation temperature is reached, the chiller is turned on and the coolant starts to flow through the condensing coil.

After the desired saturation temperature is reached, the controls are set to reach the desired measurement point. The desired boiling surface temperature is chosen in a range of values such that the boiling curve can be determined from the onset of nucleate boiling until the critical heat flux. The coolant flow rate is set low enough, such that the temperature difference between inlet and outlet is higher than 3.8 °C. This leads to a high degree of accuracy when determining the energy balance (see section 4.8.3). Lastly the coolant inlet temperature is set to assure that the refrigerant saturation temperature is within ± 0.5 °C of the desired value.



Figure 4.11: Flowchart of the measurement procedure.

To have an accurate measurement, the setup has to operate in steady state (or at least within the measurement uncertainty of the steady-state condition), as a transient measurement can result in an overestimation or underestimation of the heat flux for a certain surface superheat temperature. To assess the steady-state condition, the standard deviations of the last hundred measurements of the saturation temperature, boiling surface temperature and electrical power are determined, which corresponds to a time frame of 8 minutes. For the boiling surface temperature, a standard deviation lower than half the measurement uncertainty (0.035 °C) is considered steady state. This criterion assures that about 95 of the 100 last measurements are within the measurement uncertainty interval. For the saturation temperature, a more strict criterion is used where the standard deviation has to be below 0.03 °C. For the electrical power, the maximal allowable standard deviation was chosen equal to 1 W. When the three steady-state criteria are met, the averages of the last hundred measurements (corresponding to 8 minutes) for each measured quantity are logged.

After logging of the measurement point, the previous steps are repeated: changing the controls for the next setpoint, waiting for steady state and finally logging the measurement. These steps are repeated until the entire desired setpoint range is measured. After logging all the measurement points, the electrical power is turned off and the setup is cooled down by the coolant flow.

To determine the critical heat flux, a slightly different approach is followed. By increasing the boiling surface setpoint temperature with small increments, the heat flux is slowly increased until the critical heat flux is reached, at which moment the temperatures increase dramatically. The PID controller automatically reduces the electrical power to compensate for the high temperature, thereby avoiding damage to the power module. Next to writing the steady-state measurement points to a text file, all transient data are also logged. From this data, the timing of the sudden increase in temperature and the corresponding critical heat flux at that time is determined.

4.8 Measurement analysis

4.8.1 Boiling curve

The main goal of the experimental setup is to determine the heat flux \dot{q} as a function of the surface superheat temperature ΔT_s and additionally the critical heat flux \dot{q}_{CHF} . ΔT_s is the difference of the average wall temperature T_w and the refrigerant saturation temperature T_{sat} :

$$\Delta T_s = T_w - T_{sat} \tag{4.1}$$

For the configuration with the heater, five type K thermocouples are placed on

the bottom of the aluminium boiling surface (more details on the heater configuration can be found in section 5.2). To determine the surface temperature, the conductive heat transfer through the aluminium has to be taken into account, according to equation 4.2. This equation uses the average of the five thermocouple measurements T_{TC} .

$$T_w = T_{TC} - \dot{q} \frac{d_{Al}}{k_{Al}} \tag{4.2}$$

In this equation, d_{Al} is the thickness of the aluminium plate and k_{Al} is the thermal conductivity of the aluminium.

For the configuration with power module, five type K thermocouples are applied directly in contact with the boiling surface. For this configuration, the wall temperature is assumed to be the average of the five thermocouple measurements (more details can be found in section 6.2).

The saturation temperature can be determined in two ways: directly through the thermocouple measurement in the reservoir or indirectly through the pressure measurement. Both methods have their advantages and drawbacks. Using the temperature measurement does not require the use of fluid properties, which can induce additional errors. The method does assume that there is no temperature gradient (stratification) between the boiling surface and the thermocouple location, which is not necessarily correct if not enough mixing occurs in the liquid. The pressure measurement is more robust and less sensitive to noise. To determine the pressure at the boiling surface, the hydrostatic pressure increase due to the liquid refrigerant has to be taken into account. Furthermore, for high heat fluxes and thus intense boiling, local pressure variations are possible. To analyse the measurements, the following approach is used: if the liquid level is high enough such that the lower thermocouple is placed in the liquid, the temperature measurement is used for the saturation temperature. It is assumed that during boiling, the liquid is thoroughly mixed such that the temperature is nearly uniform. If this is not the case, the pressure measurement (with the correction for the hydrostatic pressure increase) is used to determine the saturation temperature. As in this case the thermocouple is in the vapour phase, larger thermal gradients are expected, making the pressure measurement more reliable. For all measurement points analysed in chapters 5 and 6, the maximal deviation between both methods of determining the saturation temperature is 0.8 °C.

To determine the saturation temperature from the pressure measurement p_m , the pressure at the boiling surface p_{sat} has to be determined, taking into account the hydrostatic pressure increase:

$$p_{sat} = p_m + \rho_l \, g \, z_l \tag{4.3}$$

In equation 4.3, ρ_l is the liquid refrigerant density, g is the gravitational ac-
celeration and z_l is the vertical height difference between the boiling surface and the liquid level (if the liquid level is lower than the location of the pressure measurement) or between the boiling surface and pressure sensor (if the liquid level is higher than the location of the pressure measurement). The pressure difference in the vapour column is neglected as the density of the vapour is several orders of magnitude smaller than that of the liquid.

The saturation temperature is determined from the saturation pressure using the open-source thermophysical property library CoolProp [147], which in turn based its data for FK-649 on the study of McLinden et al. [148]. The relation between saturation pressure and temperature for FK-649 is shown in figure 4.12.



Figure 4.12: Saturation temperature as a function of pressure for FK-649.

The heat flux is determined by dividing the electrical power dissipation P_e , which is the product of the measured voltage V and current I, by the boiling surface area A (equation 4.4). For the configuration with power module, a four-wire method is used to measure the voltage, with two wires supplying the current for the power dissipation and the other two wires connected to the voltage measurement. As the current supplied to the power module can go up to 120 A, the voltage drop in the connecting cables is not negligible which requires the use of the four-wire measurement method. For the heater configuration, the maximal current is 4 A, making the four-wire method unnecessary.

$$\dot{q} = \frac{P_e}{A} = \frac{VI}{A} \tag{4.4}$$

4.8.2 Energy balance

Next to the boiling curve, the energy balance of the setup is assessed. Figure 4.13 shows the control volume (in a dashed blue line) used for this analysis. The first law of thermodynamics is applied to this control volume:

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \sum_{i} \dot{m}_{i} (h_{i} + u_{i}^{2}/2 + g z_{i}) - \sum_{o} \dot{m}_{o} (h_{o} + u_{o}^{2}/2 + g z_{o}) \quad (4.5)$$

In this equation, E is the total energy, t is time, \dot{Q} is the heat transfer across the boundary, \dot{W} is the power transferred across the boundary, \dot{m} is the mass flow rate across the boundary, with subscript i indicating the incoming flows and subscript o indicating the outgoing flows, h is the specific enthalpy, u is the velocity and z is the height. Changes in potential and kinetic energy as well as the potential and kinetic energy related to the mass flows are neglected. Furthermore, the measurements are always done at steady state such that the change in energy over time is zero. There is no mass accumulation is the reservoir and only one mass flow of coolant going in and out. This results in the following equation:

$$\dot{Q} - \dot{W} - \dot{m}(h_o - h_i) = 0 \tag{4.6}$$

Electrical power is transferred to the control volume, which is determined from the measured voltage and current. The sign for the electrical power in equation 4.7 is reversed, as the convention for the first law is that power is positive if it is transferred out of the control volume. The coolant is considered to be an incompressible fluid and the contribution of the pressure to the enthalpy change is assumed to be negligible compared to the total enthalpy change, such that all fluid properties can be calculated solely from the temperature (and the ethylene glycol concentration). The mass flow rate is determined from the measured volume flow rate V_o and the coolant density ρ_o . The rate of energy transfer from the control volume by the mass flow or net efflux of energy is indicated by P_{ef} . The heat transfer across the boundary as drawn in figure 4.13 is only the result of unwanted heat losses Q_{loss} . These heat losses are inevitable as the ambient temperature will be differing from the heated surface, refrigerant and coolant temperatures. Insulation is added around the tubing, reservoir and heater to reduce these losses to a minimum. For the configuration with heater, a guard heater is added to reduce heat losses even further (more details in section 5.2). The convention for the first law is to define heat transfer to the control volume as positive. We will here also change the sign,



Figure 4.13: Control volume (blue) around the reservoir, indicating mass flows (green) and electrical power (red) across the boundary.

as it is more intuitive to consider heat losses from the reservoir to the environment as a positive value. All these considerations result in the following equation:

$$\dot{Q}_{loss} = P_e - P_{ef} = V I - \rho_o(T_o) \dot{V}_o \left[h_o(T_o) - h_i(T_i) \right]$$
(4.7)

 T_o and T_i are the coolant temperatures at respectively the outlet and inlet of the condenser. They are measured by taking the average of two thermocouple measurements positioned in the flow. Ideally, the heat losses are negligible compared to the electrical and flow energy. To assess this, the relative deviation e of the energy balance is determined as:

$$e = \frac{\dot{Q}_{loss}}{P_e} = 1 - \frac{P_{ef}}{P_e} = 1 - \frac{\rho_o(T_o) \dot{V}_o \left[h_o(T_o) - h_i(T_i)\right]}{V I}$$
(4.8)

The goal is to have an energy balance that closes within \pm 3%, while a deviation of \pm 5% is still acceptable.

4.8.3 Uncertainty analysis

A detailed derivation of the measurement uncertainties and error propagation is given in appendix B. The results of the analysis are given here.

The estimated measurement uncertainties of the boiling curve are shown in figure 4.14 for four different cases. The plots on the top are made for the heater

configuration, those below are for the power module configuration. On the left a low liquid refrigerant level is used while on the right a high liquid refrigerant level is analysed, as this affects the manner of determining the saturation temperature. To estimate the boiling curve, the Cooper correlation (equation 3.53) is used with FK-649 as refrigerant at 40 °C and 45 °C. The boiling surface roughness is set to 3 μ m. The electrical power dissipation is varied from 100 W to 750 W (which relates to the estimated critical heat flux at these refrigerant temperatures). For the heater configuration, the electrical resistance of the heater is set equal to 61 Ω . A constant current of 100 A is assumed for the configuration with power module.



Figure 4.14: Uncertainty on simulated boiling curves for different saturation temperatures, liquid levels and configurations.

For all configurations, the effect of a change of saturation temperature with 5 °C is larger than the measurement uncertainty, indicating that this effect should be measurable with the designed setup. For the heater configuration, the uncertainty on the heat flux is higher than that for the power module configuration. This is mainly the result of the low current in the case of the heater, making the relative

uncertainty on the current measurement higher than that of the power module configuration. For the high liquid level measurements, the uncertainty on the surface superheat temperature is slightly lower, as for these measurements the temperature is directly measured and not derived from the measured pressure.

The estimated measurement uncertainty on the heat losses and energy balance are shown as a function of electrical power dissipation in figures 4.15 and 4.16 respectively. The electrical power dissipation is varied from 100 W to 1000 W. For this analysis, the heat losses are considered to be negligible and the temperature difference between coolant inlet and outlet is equal to 6 °C. For the heater configuration, the electrical resistance of the heater is assumed constant and equal to 61 Ω . A constant current of 100 A is assumed for the configuration with power module. The absolute uncertainty on the heat losses increases with increasing electrical power, staying below 30 W for electrical powers up to 1000 W. The uncertainty on the energy balance decreases with the electrical power, as the relative error on the voltage and current measurement decreases. For the power module configuration, this effect is negligible as the uncertainty is dominated by the uncertainty of the energy carried by the liquid, which remains constant as the coolant temperature difference is assumed to be constant. For the heater configuration, the uncertainty on the electrical power is also important. As the current is low (< 4 A), the relative uncertainty on the current measurement also adds significantly to the total uncertainty, but the uncertainty decreases with increasing current and power. For the power module configuration, the uncertainty on the energy balance should always be below 3%, while for the heater configuration it is below 5% from 120 W and below 3% from 560 W.



Figure 4.15: Uncertainty on the heat loss estimation as a function of electrical power for the heater and power module configuration.



Figure 4.16: Uncertainty on the energy balance as a function of electrical power for the heater and power module configuration.

5

Horizontal plate pool boiling

5.1 Introduction

This chapter discusses the horizontal plate pool boiling setup and measurements. The size of the horizontal plate is based on the size of the baseplate of an Infineon HybridPACK type FS400R07A1E3 inverter module. A heater is used to mimic the heat dissipation of the inverter module instead of the actual module, which will be discussed in chapter 6.

The configuration with heater has several advantages compared to the configuration with the inverter module:

- The heater configuration supplies a uniform heat flux to the boiling surface. This is not guaranteed for the power module configuration, as the heat is generated in the IGBTs and diodes, which can be seen in figure 5.1. The baseplate will act as a heat spreader in this case, but the non-uniformity of the heat generation might have an effect on the boiling heat transfer. Comparing the results of both configurations can give more insights into the effect of non-uniform heat dissipation.
- In the heater configuration, thermocouples can be placed centrally along the boiling surface, by placing them in grooves on the side which is not in contact with the refrigerant (more details can be found in section 5.2). This is not feasible for the power module configuration, as trying to place the thermocouples on the inside of the baseplate would demolish the power

module, while putting them on the outside would interfere with the boiling phenomenon.

• To achieve heat fluxes high enough to reach the critical heat flux, high currents (> 100 A) are needed. With the heater configuration, lower currents can be used to achieve similar heat fluxes.



Figure 5.1: Internal structure of Infineon HybridPACK inverter module: square components are IGBTs, rectangular components are diodes [149].

In the next sections, the heater assembly, setup controls and design of experiments are described. Further on, the results of the measurements and the influence of the varied parameters are shown. The experimental results are compared to those of a previous study of FK-649 boiling on an aluminium surface. The comparison of the measurements to the cited correlations is done together with the measurements on the power module configuration in chapter 7.

5.2 Heater assembly

Heater

In section 4.3 it is shown that a heat dissipation up to 1000 W is needed to cover the entire range from onset of nucleate boiling to critical heat flux. Several options for the electrical heater are investigated to provide the required heat flux. Film heaters are an interesting option, however the attainable heat fluxes are too low. Another option is to use cartridge heaters, which have a cylindrical shape. These heaters

need to be fitted into a housing to transfer the heat to a flat surface, which results in difficulties such as proper contact between the housing and heaters, reducing the heat losses to the environment and acquiring a uniform heat flux at the boiling surface. Ceramic heaters can have a flat surface and acquire higher heat dissipation than film heaters. For these reasons, a ceramic heater as shown in figure 5.2 is chosen.



Figure 5.2: Ceramic resistive heater.

Heat contractor

To acquire heat flows of 1000 W, the minimal size of the heater was 150 mm by 87 mm, with a thickness of 15.3 mm. This is larger than the boiling surface of 108 mm by 47 mm. To achieve this reduction in surface area, a heat spreader or, more accurately for this case, a heat 'contractor' is needed. For this purpose, a solid aluminium block with a step change is used, as is shown in figure 5.3.



Figure 5.3: Aluminium heat contractor.

Boiling plate

The smaller surface of the heat contractor is attached to the boiling plate, which is a cuboid aluminium part with a thickness of 9.7 mm and a thermal conductivity of 180 W/mK. This part is again slightly larger than the boiling surface, allowing it to be bolted to the bottom plate of the reservoir. The surface of the plate in contact with the refrigerant has an arithmetic mean surface roughness of 0.9 μ m. This value was determined by taking the average of ten measurements distributed over the surface.

In the bottom of this plate, five grooves are milled with a depth and width of 0.5 mm. In these grooves, type K mineral insulated thermocouples with a diameter of 0.5mm are placed to measure the boiling plate temperature. The tips of the thermocouples are placed in the middle of the boiling plate, spaced over the length of the boiling surface, as shown in figure 5.4. The thermocouples are spaced 20 mm apart from each other and 14 mm from the sides of the boiling surface. The average temperature of the boiling surface is determined from these thermocouple measurements using equation 4.2.



Figure 5.4: Boiling plate with five thermocouple grooves.

Guard heater

To reduce heat losses to the environment, a guard heater is placed below the main heater. The guard heater is identical to the main heater. In between both heaters, a 12.5 mm thick piece of plasterboard is placed, which has a thermal conductivity of 0.32 W/mK. Two type K thermocouples are placed at either side of the plasterboard. The power to the guard heater is controlled with a PID controller which reduces the temperature difference between both measured temperatures to zero. This ensures that the temperature difference is within the measurement uncertainty, which results in negligible heat fluxes and heat losses to the bottom.

Insulation

The entire structure (guard heater, plasterboard, main heater, heat contractor, boiling surface and reservoir) rests on a rigid foam glass insulation block, which has a thermal conductivity of 0.05 W/mK. The sides of the heaters are insulated with ceramic fibre insulation with a thermal conductivity of 0.06 W/mK and a thickness of at least 3.1 cm. After the ceramic fibre insulation, a layer of polyurethane insulation with a thickness of 5 cm and a thermal conductivity of 0.021 W/mK is placed.

Sealing

To avoid leakage of the refrigerant, a silicone gasket is placed in between the bottom plate and the sides of the aluminium boiling plate. Silicone mastic is used in the holes containing the bolts to avoid leakage along the length of the bolts.

Thermal paste

To reduce the thermal resistance from the heater to the boiling surface, thermal paste is applied on the interface between the heater and the heat contractor and between the heat contractor and the boiling plate. The thermal paste used is MTC TCTG-6.0, which has a thermal conductivity of 6 W/mK.

Assembly

A picture of the guard heater, main heater and heat contractor assembly is shown in figure 5.5. A cross-sectional view of the entire heater assembly is given in figure 5.6.



Figure 5.5: Guard heater assembly.



Figure 5.6: Heater configuration assembly. Red dots indicate thermocouple locations.

5.3 Setup controls

Main heater

As discussed in section 4.6, the power dissipation in the electrical heater is controlled with a PI-controller. For the first tests, a constant power was applied to the heater. However, this resulted in a very slow process to reach steady state, with a time constant of several hours. To accelerate this process, a varying voltage is applied to the heater with the goal to reach a certain constant average temperature measured by the thermocouple at the boiling surface.

For the controls of this setup, the derivative term is not used. This derivative term can improve the settling time of the control, however it is also sensitive to measurement noise. It is chosen to go for a slower but more stable PI-control.

To determine adequate proportional and integral constants for the controller, first the system response is acquired experimentally by setting the power with a PI-controller with manually set constants. The power and average of the measured temperatures on the boiling plate is shown in figure 5.7. The variation of the boiling plate temperature as a function of the heater power is fitted to a transfer function, the results of which are also shown in figure 5.7. This is done using the 'tfest' Matlab function of the System Identification Toolbox.

Using the system transfer function, a PI-controller can be designed. The proportional and integral term of the controller are designed using the 'pidtune' Matlab function, which tries to balance response time and robustness of the controller. For the control of the main heater, the determined proportional gain was equal to 37.2 W/K and the integration time is equal to 63 s.

The simulated step response of the system with the designed PI-controller is shown in figure 5.8. The graph indicates that the system requires about half an hour to reach the setpoint within 1% of the step size.



Figure 5.7: Variation of the boiling plate temperature with a PI-controlled power. The temperature variation is fitted using a transfer function.



Figure 5.8: Simulated step response equal to 1 °C for the boiling plate temperature with PI-control.

Guard heater

The design of the guard heater control is done in a similar way as that of the main heater. The goal of the guard heater is to have no temperature difference between the main and guard heater. Therefore instead of the actual temperature, the temperature difference is used to fit the system curve. A step change is applied for the guard heater power from 0 W to 15 W. The resulting temperature difference between the guard heater and main heater is shown in figure 5.9. After four hours, the temperature difference is still significantly increasing. This is a result of the heating of the insulation below the guard heater, which is slow due to the large size of the insulation block. The transfer function fit of this transient temperature difference is also shown in figure 5.9.



Figure 5.9: Variation of the temperature difference between the guard and main heater for a step in guard heater power. The temperature difference variation is fitted using a transfer function.

Using the fitted transfer function, a PI-controller is designed using the 'pidtune' function in Matlab. The proportional gain is equal to 3.39 W/K and the integration time is 122 s. The simulated response of the controlled system to a step change of 1 °C is shown in 5.10. This system has a faster response time than the main heater, which is also perceived during the experiments. The guard heater control is able to reduce the temperature difference below 1 °C within 5 minutes for each change of setpoint.



Figure 5.10: Simulated step response equal to 1 °C for the guard heater temperature difference with PI-control.

5.4 Selection of experimental parameter range

As discussed in the literature review, many parameters and boundary conditions can have an effect on pool boiling heat transfer. The following parameters and/or boundary conditions will be varied:

Heat flux

The boiling surface heat flux is the main variable which describes the boiling curve. To capture all effects, the measurement points should be spaced thinly in between the onset of nucleate boiling and the critical heat flux. Therefore, steps of 10 kW/m^2 to 20 kW/m^2 are aimed for in the measurements. Critical heat flux measurements are not done for this configuration. Due to the thermal resistances of the heater, heat contractor and boiling surface and of the interfaces in between, the temperature at the back of the heater reaches the limiting temperature of the insulation surrounding it. Having the heat flux increase above the critical heat flux therefore has a large risk of overheating both the insulation and the heater itself. Critical heat flux measurements are executed on the power module configuration and reported in section 6.4.

• Saturation temperature

To be useful for cooling of components of an electrical drivetrain, the refrigerant temperature should be above ambient temperature, such that it can be cooled down again by a heat exchanger. Preferably it is also not too high such that it keeps the components at the lowest possible temperature. For this reason, the range of saturation temperatures investigated in this work is from 35 °C to 50 °C. For the heater configuration, two saturation temperatures are tested: 36 °C and 41 °C. The change of saturation temperature will result in changing fluid properties and a different boiling curve.

Surface fouling

An unintentional but also interesting boundary condition is the absence or presence of fouling on the boiling surface. Fouling was perceived several times during the measurements and will be discussed in section 5.5.5. As it is hard to quantify the amount of fouling, the measurements are simply divided in a group of 'clean' and 'fouled' surface measurements, based on visual observation.

To limit the number of experiments, some variations of (possibly) influencing parameters and boundary conditions will not be tested:

Surface geometry

Only one surface will be tested in this configuration, so the surface roughness (0.9 μ m), material (aluminium), size (108 mm x 47 mm) and orientation (horizontal) remain constant.

Fluid (properties)

The fluid to be tested is FK-649. The only variation of the fluid properties to be tested is the variation of FK-649 properties with temperature.

• Subcooling

As the reservoir is sealed and only filled with refrigerant, the refrigerant will always be at saturation state. No tests are done on subcooled pool boiling.

• Gravity

All measurements are done at standard gravitational acceleration of 9.81 m/s^2 .

· Liquid refrigerant height

The height of the liquid refrigerant in the reservoir is equal to 17 cm.

5.5 Results

5.5.1 Surface temperature

Figure 5.11 shows the temperatures measured by the five thermocouples as a function of the heat flux, at a saturation temperature of 36 °C. The solid line represents the average of these five temperatures. With increasing heat flux, the variation in the measured temperature increases. Three thermocouples (TC₁, TC₃ and TC₄) remain within 0.5 °C of each other, while one thermocouple (TC₅) measures significantly higher temperatures than the average (up to 1.4 °C) temperatures and one (TC₂) significantly lower (up to 1.6 °C). When calculating the average temperature, these two temperatures have a counteracting effect. Therefore, the average temperature remains within the same range as the three other temperatures.



Figure 5.11: Measured temperature by thermocouples at five locations along the long side of the boiling plate as a function of surface heat flux with a saturation temperature of 36 °C.

The deviation of the individual temperature measurements compared to the average temperature is shown in figure 5.12, plotted as a function of the surface heat flux for a saturation temperature of 36 °C. This deviation is also plotted for all measurement points in figure 5.13 with regard to its location on the boiling surface. On the horizontal axis, 0 mm represents the start of the long side of the boiling surface, while 108 mm indicates the end. Although the variation is

consistent with the location for the different measurement points, there is no clear relation between the location and the deviation.



Figure 5.12: Deviation of thermocouple measurements compared to average temperature as a function of surface heat flux with a saturation temperature of 36 °C.

From a boiling heat transfer point of view, a quasi-uniform temperature is expected. At the boiling surface itself, temperature gradients can exist on length scales around a millimetre due to the presence of localised nucleation spots. However, the temperature is measured at the other sides of the boiling surface, where the aluminium material will have dampened the local small-scale variations in temperature. If the boiling surface is not uniform, for example with randomly distributed regions of high and low surface roughness, larger temperature variations could be explained. Inspection of the boiling surface did not reveal any major variations of the structure, so this explanation is deemed rather unlikely. Influence of the boiling surface perimeter could also affect the temperature, either by enhancing or deteriorating the heat transfer at the sides. This is however not consistent with the temperatures measured.

Several other possible explanations can be given for the two thermocouples which are deviating, which are not related to the boiling phenomenon. The temperature is measured by thermocouples placed in grooves in the boiling surface. The side of the boiling surface is in contact with the heat spreader. To reduce interface resistance and improve the contact of the thermocouples and the boiling



Figure 5.13: Deviation of thermocouple measurements compared to average temperature as a function of thermocouple location.

surface, thermal paste is first placed in the grooves and on the interface. Although both aluminium parts are machined precisely, there are regions of reduced thermal contact. These reduced thermal contact regions can result in a variation of the temperature measured by the thermocouples. Secondly, due to the high heat fluxes, a large temperature gradient will exist across the interface. Small variations in the location of the thermocouples at this interface (due to the imperfection of both surfaces) can thus result in significant temperature variations. It is deemed most likely that a combination of the thermal contact resistance and large interface temperature difference results in the measured temperature variation.

Due to the high thermal conductivity of the aluminium boiling plate, these variations will be dampened. As such, it is presumed that at the boiling surface, the temperature can be assumed more uniform than those measured by the thermocouples.

The boiling surface temperature is derived from the measured temperatures by taking into account the temperature drop over the boiling plate with equation 4.2. Both the measured temperatures and the boiling surface temperatures are shown in figure 5.14 for varying heat flux and saturation temperature equal to 36 °C. At the highest heat flux, the calculated temperature difference is equal to 4.2 °C. The correction is thus significant, but small uncertainties on the correction will result in insignificant changes to the derived boiling surface temperature as the surface



superheat temperature is several times larger (23 °C).

Figure 5.14: Measured average boiling plate temperature and derived surface temperature as a function of heat flux for a saturation temperature of 36 °C.

5.5.2 Energy balance

The energy balance is checked to analyse the heat losses of the setup. The rate of energy leaving the reservoir through the coolant, or net efflux of energy, is shown as a function of the dissipated electrical power in figure 5.15. The net efflux is always slightly lower than the electrical power, indicating that there are some heat losses to the environment.

The estimated heat losses are shown in figure 5.16. The heat losses show a slightly increasing trend as a function of the electrical power, although the trend is not significant compared to the measurement uncertainty. The losses consist of losses from the heater assembly and losses from the reservoir. The losses from the heater assembly increase with increasing heat flux due to the increasing temperatures. The losses from the reservoir however are mostly dependent on the fluid temperature and mainly independent of the heat flux. For most of the measurement points, the heat losses are significantly higher than zero.

Figure 5.17 shows the relative deviation of the energy balance. The energy balance matches within 5% for heat fluxes higher than 60 kW/m². The relative



Figure 5.15: Net efflux as a function of electrical power.



Figure 5.16: Estimated heat losses as a function of electrical power.

deviation of the energy balance decreases with increasing heat flux. This is mostly due to the almost constant heat losses from the reservoir. As these losses remain constant and the heat flux increases, the relative heat losses will decrease.



Figure 5.17: Relative deviation of the energy balance as a function of heat flux.

Due to the two heat loss paths (from the heater assembly and from the reservoir), the actual heat transferred from the surface to the fluid will lie in between the electrically dissipated power and the net efflux. For heat fluxes higher than 60 kW/m², the actual surface heat flux is within 5% of the heat flux determined from the electrical power dissipation.

5.5.3 Boiling curve

The measured boiling curve for a saturation temperature of 36 °C is shown in figure 5.18. Boiling is always occurring even at the lowest measured heat fluxes, no measurement points of pure natural convection are taken. This is a results of the PI control on the measured temperature which has a significant overshoot. The heat flux will always pass the onset of nucleate boiling, thereby activating the first nucleation sites. The boiling curve is therefore similar to one measured going from high to low heat flux.

The applied heat flux ranges from 8 kW/m² to 91 kW/m². The largest measured heat flux is lower than the expected critical heat flux (137 kW/m² according



Figure 5.18: Measured boiling curve for a saturation temperature of 36 °C. Different colours indicate different measurement dates.

to the Lienhard-Dhir correlation, equation 3.85). Heat fluxes closer to CHF were not applied, as for the highest used heat flux, the temperature at the guard heater was around 200 °C. Higher temperature would have resulted in damage to the heaters and insulation. Higher heat fluxes are measured for the configuration with power module and are reported on in section 6.4.

Repeatability of the measurements is checked by redoing the two measurements at a different date (shown in orange on figure 5.18). The two measurement points at the highest heat flux fall within the bounds of the measurements error. The other repeated point was around 13 kW/m^2 (the second and third measurement point in figure 5.18). These two points are shifted slightly as a result of the saturation temperature not being completely equal. However, also this point is within the expected boiling curve from the previous measurement points.

From the boiling curve, two regimes can be clearly perceived. The partial nucleate boiling regime occurs at heat fluxes below 15 kW/m^2 . At higher heat fluxes, the slope in the boiling curve increases and fully developed boiling takes place. As heat fluxes close to CHF were not measured, the partial dryout regime is not visible in figure 5.18.

To verify if a power-law relation exists between heat flux and surface superheat temperature (as in equation 3.66), the boiling curve is shown in figure 5.19 with logarithmic-scale axes. A power-law relation results in a straight line. The entire

boiling curve cannot be represented by a single power-law function, as there is a distinct difference between the low and high heat fluxes. On this scale, the transition to the partial dryout regime can also be perceived, as the slope of the curve starts to decrease for the highest heat fluxes measured.



Figure 5.19: Measured boiling curve for a saturation temperature of 36 °C with logarithmic-scale axes.

5.5.4 Effect of saturation temperature

Two different saturation temperatures were used during the tests: $36 \,^{\circ}C$ and $41 \,^{\circ}C$. The boiling curves for both saturation temperatures are shown in figure 5.20. An increase in saturation temperature moves the boiling curve to the left, indicating lower surface superheat temperature for the same surface heat flux.

5.5.5 Effect of fouling

The first measurements showed issues with repeatability. It was found that a fouling layer was being built up on the boiling surface. A picture of this fouling layer is shown in figure 5.21. After several days of boiling, this fouling layer no longer changed significantly.

An FTIR (Fourier transform infrared) spectroscopy analysis was done on the fouling layer, which showed signs of CH_3 , Si-C and Si-O-Si bonds. Due to the



Figure 5.20: Measured boiling curve for two different saturation temperatures.



Figure 5.21: Fouling on the boiling surface after boiling for several days in presence of silicone mastic.

presence of the silicon-oxygen bonds, it was concluded that the source of the fouling was the silicone mastic, as this mastic also showed discolouration. The silicone gaskets are also a possible source, however this was discarded as the fouling did not occur in a previous version of the setup with gaskets but without mastic.

The boiling curve at a saturation temperature of 41 °C on a clean surface is compared to that of the fouled surface at a saturation temperature of 40 °C in figure 5.22. The fouling layer causes a significant increase of the surface superheat temperature, which is more profound at higher heat fluxes. At a heat flux of 90 kW/m², the thermal resistance (between the aluminium surface and fluid) is increased by 57% due to the fouling layer. The increase in surface temperature can be approximated by adding a fouling thermal resistance of 1.4e-4 m²K/W in series to the boiling thermal resistance. It is assumed that this thermal resistance is due to conduction through the fouling layer which has a low thermal conductivity.



Figure 5.22: Boiling curve for a clean $(T_{sat} = 41 \ ^{\circ}C)$ and a fouled $(T_{sat} = 40 \ ^{\circ}C)$ surface.

5.6 Comparison with previous study

A comparison of the captured boiling curves to those reported by Forrest et al. [53] is shown in figure 5.23. Their measurements were done for boiling of FK-649 on an aluminium surface which was slightly smoother than the one tested in this study

 $(R_a = 0.41 \ \mu\text{m}$ in Forrest et al. versus $R_a = 0.9 \ \mu\text{m}$ in the current study). The boiling curves were determined for 49 °C, 71 °C and 96 °C, so no direct comparison between this datasets and the one gathered in this study is possible. The general trend of increasing heat transfer rates with increasing saturation temperature is however clear from figure 5.23.



Figure 5.23: Measured boiling curve for five different saturation temperatures. Boiling curves of 49 °C, 71 °C and 96 °C are reproduced from Forrest et al. [53].

5.7 Conclusions

A heater assembly was designed to test horizontal plate pool boiling, with the goal to determine reference boiling curves for a uniform heat flux boundary condition. A guard heater was added to reduce heat losses to the ambient. PI-controllers were designed for the main heater and guard heater control. The following conclusions can be drawn from the experimental results of this setup:

 In the experiments, the heat flux was varied from 8 kW/m² to 91 kW/m². The partial and fully developed nucleate boiling regimes were identified in the boiling curve. The boiling curve cannot be fully represented by a single power-law relation between heat flux and surface superheat temperature. The partial dryout regime and the critical heat flux were not measured, as this requires higher heat fluxes. This heat flux range will be tested using the power module configuration.

- The estimated heat losses were significantly higher than zero, but still in the same order of magnitude as the measurement accuracy. The energy balance of the setup closed within 5% for heat fluxes above 60 kW/m².
- Two fluid saturation temperatures were tested. The heat transfer rates increased with increasing saturation temperature, as is predicted by correlations. The trend of the boiling curves with saturation temperature is in agreement with the boiling curves reported by Forrest et al. [53].
- Fouling occurred on the boiling surface. This was ascribed to the silicone mastic reacting and dissolving in the fluid. The fouling layer caused an increase of up to 57% of the thermal resistance.

In the next chapter, experimental results of the configuration with power module will be discussed. These can be compared to the results of this chapter to analyse the importance of possible heat flux non-uniformity. The measurement points of both configurations will be used to analyse the predictive capabilities of the correlations described in section 3.3.

Fower module baseplate pool boiling

6.1 Introduction

This chapter discusses the measurements of pool boiling cooling of an inverter power module. The power module under study is an Infineon six-pack containing six IGBTs and six diodes. The goals are to analyse the boiling curve and the parameters influencing it, measure critical heat flux values, determine the most adequate correlations and to evaluate the feasibility of pool boiling cooling of power electronics. Section 6.2 gives more details on the power module and on how it is powered. The following section discusses the design of experiments, indicating which influences on heat transfer are tested. The measurement results are shown and analysed in section 6.4. These results are compared to those on the heater configuration in the subsequent section. After this section, an analysis of the cooling performance of pool boiling cooling of the baseplate of power modules is given. In the final section of this chapter, the main results and conclusions are highlighted.

6.2 Inverter module

Specifications

The inverter power module used for this study is an Infineon HybridPACK type FS400R07A1E3 and is shown in figure 6.1 (datasheet can be found in appendix D.4) [150]. It is a six-pack, consisting of six insulated-gate bipolar transistors (IGBTs) and six diodes. The electrical circuit diagram of the power module is

shown in figure 6.2. At terminals 1 and 2, the DC voltage is applied. Terminals 3, 4 and 5 supply the three-phase electrical waveforms. The rated DC voltage is 650 V and the rated current is 400 A. It is designed for electric vehicle applications in the range of 20 kW to 30 kW.



Figure 6.1: Infineon six-pack power module [150].



Figure 6.2: Infineon HybridPACK electrical circuit diagram [150].

Figure 6.3 shows the internal structure of the power module. The square components are IGBTs, while the smaller rectangular ones are the diodes. There are twelve IGBTs and diodes as every component is put in parallel with one other identical component. As a result, the entire circuit acts as a six-pack. The small component in the white rectangle on figure 6.3 is a negative temperature coefficient (NTC) thermistor, which is used to determine the temperature of the module substrate.

The components are placed on a ceramic substrate made of alumina (Al₂O₃), which has a relatively high thermal conductivity (72 W/mK) for an electrically non-conductive material. The substrate is positioned on a copper baseplate, which acts as a heat spreader.



Figure 6.3: Internal structure of Infineon HybridPACK inverter module: square components are IGBTs, rectangular components are diodes [149]. Green dashed rectangle indicates cooled surface.

The module has four holes in the corners to mount it to a heat sink (or the refrigerant reservoir in this study). These holes are positioned on a rectangle with sides equal to 128 mm and 57 mm. To be able to seal the interface between the module baseplate and the reservoir bottom plate, an additional 10 mm and 5 mm of the baseplate, on the long and short side respectively, are not in contact with the fluid but with the bottom plate and gasket. This results in a rectangular heat transfer surface with sides of 108 mm and 47 mm, which is used throughout this study. This surface is also shown by the green dashed-line rectangle in figure 6.3. All components are directly below the boiling surface, indicating all components are cooled uniformly.

During typical drive cycles, most heat is dissipated in the IGBTs [151]. Therefore, the focus is on the heat transfer from the IGBTs to the coolant. The thermal resistance from the junction to the case of a single IGBT is given by the datasheet and is equal to 0.120 K/W.

Electrical circuit

During the experiments, power is only dissipated in the IGBTs of the power module by applying a DC voltage and current. The power module operates under a different load than that during typical use as an inverter, during which an AC current would run through the IGBTs. To understand this choice, a closer look has to be taken to the electrical characteristics of the power module. Both the IGBTs and the diodes in the power module dissipate heat during typical traction loads. The heat dissipation in the IGBTs is higher and also the temperatures in the IGBTs are typically higher [151]. Therefore, in this study the choice was made to only impose heat dissipation to the IGBTs.

An IGBT has three terminals: the collector, emitter and gate. In an inverter, the IGBT basically operates as a switch. The collector is the positive terminal and the emitter is the negative terminal. The gate acts as the controlling terminal, to either allow current to flow through from the collector to the emitter or to prevent this.

The output characteristic of the IGBTs in the module is shown in figure 6.4. If no voltage is applied to the gate ($V_{GE} = 0$ V), the IGBT will operate in the cut-off state and negligible collector current (I_C) will flow through the module. When a high enough gate voltage is applied ($V_{GE} > 6$ V), current will be able to flow through the IGBT. For lower currents, the IGBT will operate in the saturation region. This region is shown on the left of the graph in figure 6.4, with saturation voltages (V_{CE}) ranging from 1 V to 2.5 V. If the current increases further, the IGBT can operate in the active region, shown on the right of the graph in figure 6.4. In this region, the current increases slightly for a big increase in collector-emitter voltage. The limiting value of the current is determined by the gate voltage, a higher gate voltage results in a higher current. In typical traction applications, the IGBT switches between the cut-off region and the saturation region, as the active region has higher power losses due to the higher voltage drop across the IGBT.

The losses in the IGBTs can be divided in two main categories: conduction losses and switching losses. Conduction losses are similar to Joule losses and occur when a current flows through the IGBT. Switching losses occur when the IGBT is switched from saturation state to cut-off state and vice-versa. Switching losses become the dominant loss mechanism at high switching frequencies around 10 kHz or higher, which is typically used for traction purposes. However, the power dissipated is difficult to measure accurately at these frequencies, as very high sampling frequencies (two orders of magnitude higher than the switching frequency) are required. It is therefore chosen to only apply conduction losses. To compensate for the switching losses and to reduce the required current, a low gate voltage is applied and the IGBT will always operate in the active region.

The electrical control circuit together with the power module electrical circuit is shown in figure 6.5. A programmable DC source is connected to terminals 1 and 2 of the module. Terminals 3, 4 and 5 are left unconnected. The DC source provides a DC voltage and current which branches into three parallel tracks and flows through all six IGBTs (shown in blue on figure 6.5). The maximal current of the DC source is 120 A, resulting in a maximal current of 40 A per IGBT. To operate in the active region with this current, a gate voltage of 7.2 V is maintained for all the IGBTs. In this manner the same current flows through and the same



Figure 6.4: IGBT output characteristic [150].

voltage drop occurs over all six IGBTs, resulting in an equal distribution of the heat losses. The total power dissipated in the IGBTs is controlled by adjusting the output voltage of the DC source. No current flows through and no heat is dissipated in the diodes.



Figure 6.5: Electrical circuit of the power module and power control, blue lines indicate the flow of current.

Assembly

The power module is bolted to the bottom plate using the four mounting holes in the corners. A silicone sheet is placed in between the bottom plate and module baseplate to avoid leakage of the fluid out of the reservoir. As there are only four mounting points (compared to twelve for the heater configuration), an additional metal plate was provided inside the reservoir to avoid excessive bending of the bottom plate and to ensure the silicone gasket is compressed around the entire perimeter of the boiling surface. A picture of the power module and metal plate mounted on the bottom plate is shown in figure 6.6. No boiling was perceived on the metal plate, indicating that leak heat flows through the bolts and plate are negligible.



Figure 6.6: Picture of the power module (only baseplate is visible) and supporting metal plate mounted on the bottom plate.

The temperature of the module baseplate is measured by placing five thermocouples in between the silicone sealing gasket and the baseplate, as shown in figure 6.7. These thermocouples are spaced equally along the length of the baseplate, similar to the spacing in the heater configuration. The thermocouples are in direct contact with the baseplate, but not with the fluid. This allows to measure the baseplate temperature without affecting the boiling phenomena at the baseplate and fluid interface.



Figure 6.7: Schematic of the thermocouples positioned between baseplate and sealing mat.

Voltage control

The DC voltage supplied to the power module is set using a PI-controller. The goal of the controller is to keep the measured module temperature (using the internal NTC thermistor) constant. Voltage control instead of power control was used in this configuration, as controlling the power was less accurate and resulted in larger variations.

A first set of P and I constants of the controller is determined using the 'pidtune' function of Matlab. This function requires a transfer function describing the process. To determine the transfer function, the response of the module temperature as a function of a step change of the module voltage is experimentally measured. The results of this step change are shown in figure 6.8. The module temperature is made relative to the fluid temperature, as this varied slightly during the measurement. This correction resulted in a better fit of the transfer function, which is also shown in figure 6.8.

The voltage at the module does not remain fully constant during the step. This is a result of the resistance of the lead wires connecting the DC source to the module. The voltage is kept constant at the power source. However, due to the increase in module temperature, the current through the module is also increased. This change results in a higher voltage drop in the lead wires, resulting in a slightly decreasing voltage over the module.

The 'pidtune' function returned a proportional gain of 0.3 V/K and an integration time of 42 s. These settings did not result in a stable system for all setpoints. The integration time was manually adjusted, and 300 s was found to be an adequate constant for the controller.

The simulated step response of 1 $^{\circ}$ C of the inverter module with PI-control is shown in figure 6.9. Due to the larger integration time, the system takes a relatively long time (compared to the controlled heater configuration) to reach a value within 1% of the setpoint, which is about 85 minutes.



Figure 6.8: Variation of the temperature difference between module and fluid for a step change of the module voltage. The temperature difference variation is fitted using a transfer function.



Figure 6.9: Simulated step response equal to 1 °C for the module temperature with PI-control.
6.3 Selection of experimental parameter range

Similarly to the measurements with the heater configuration, several parameters and/or boundary conditions which can influence pool boiling heat transfer will be varied:

Heat flux

To properly capture all nucleate boiling regimes, the heat flux is varied in several small steps. The goal is to have a measurement point every 10 kW/m² to 20 kW/m². This ranges from the start of the partial nucleate boiling regime (between 0 kW/m² and 10 kW/m²), up to the critical heat flux (around 140 kW/m², but dependent on the fluid saturation temperature). In the configuration with power module, the total thermal resistance from heat source (IGBT) to fluid is several times smaller than in the configuration with heater. Therefore, there are no issues with overheating in the entire nucleate boiling regime. At critical heat flux, the PI-controller reacts fast enough to reduce the voltage and thus power dissipation before the module overheats.

• Saturation temperature

As discussed in section 5.4, the saturation temperatures relevant for power electronics cooling should be above ambient temperature but at the same time low enough to have a big temperature difference to transfer heat from the components. In this configuration, saturation temperatures equal to 36 $^{\circ}$ C, 41 $^{\circ}$ C and 46 $^{\circ}$ C are experimentally investigated.

Liquid height

The setup is designed for rather high liquid levels around 17 cm. In a reallife power electronics cooling system, liquid height is preferably reduced in order to reduce the weight, volume and cost of the system. Therefore, also liquid heights of 1 cm and 6 cm are tested to analyse if the liquid height has an effect on nucleate pool boiling heat transfer.

Surface fouling

As was the case for the measurements with the heater configuration, unwanted fouling occurred on the boiling surface. Different influences on heat transfer were perceived and will be discussed in the following section.

The following parameters and boundary conditions remain constant during the experiments:

• Surface geometry

Only one power module is tested. The baseplate of the power module is copper and has an arithmetic mean surface roughness of $0.2 \,\mu\text{m}$. The boiling

surface has a size of 108 mm by 47 mm and is positioned horizontally facing upward.

• Fluid (properties)

The fluid tested in this study is FK-649. The fluid properties of FK-649 vary with temperature, but other fluids are not experimentally investigated.

Subcooling

In the sealed reservoir with no air present, the fluid is always saturated in a two-phase state. The effect of subcooling is therefore not tested.

• Gravity

The gravitational acceleration is equal to 9.81 m/s² for all measurements.

6.4 Results

6.4.1 Surface temperature

The surface temperature is measured by five thermocouples. The individual thermocouple measurements are shown in figure 6.10 as a function of the average of the five measurements. Thermocouples 2 and 5 measure respectively significantly lower and higher temperatures than the average temperature. This is also clear from figure 6.11, which shows the deviation of the thermocouples compared to the average temperature as a function of the heat flux. The absolute deviation of thermocouples 2 and 5 increases with increasing heat flux and reaches up to 6 °C. The other three thermocouples remain within 3 °C of each other.

The deviation of the temperatures measurements compared to the average surface temperature as a function of the position along the length of the baseplate is shown in figure 6.12. Halfway during the measurement campaign, the thermocouples were remounted and repositioned. The measurements before this repositioning are shown in blue, those after are shown in orange. It is clear that the remounting of the sensors resulted in better measurements since the deviation between local and average temperature was decreased. The measured temperatures remain within 2 °C of the average surface temperature.

The order of placing the thermocouples is also reversed after remounting. The variation of the temperature along the length of the baseplate does not remain consistent after altering the location of the thermocouples. Thermocouples 2 and 5 remain the most diverging thermocouples. It is concluded that these thermocouples do not measure the baseplate temperature accurately, and they are therefore excluded from further analysis. Excluding both thermocouples from the data results in a maximal change of 0.5 °C on the average surface temperature. The measured boiling curve remained unaltered after remounting the thermocouples.



Figure 6.10: Measured temperature by thermocouples at five locations along the long side of the baseplate as a function of the average baseplate temperature.



Figure 6.11: Deviation of thermocouple measurements compared to average temperature as a function of surface heat flux.



Figure 6.12: Deviation of thermocouple measurements compared to average temperature as a function of thermocouple location. Blue dots are the original measurements, orange crosses are the measurements after remounting.

6.4.2 Module temperature

The internal module temperature is monitored by an NTC thermistor. The temperature measured by the thermistor T_{int} is shown as a function of the average surface temperature in figure 6.13. As expected, the module temperature is higher than the surface temperature due to the internal thermal resistance of the module (except when there is no power applied to the module, then both temperatures are equal within the measurement uncertainty). A thermal resistance $R_{th,int}$ can be defined based on the temperature measurements:

$$R_{th,int} = \frac{T_{int} - T_w}{P_e} \tag{6.1}$$

This thermal resistance is shown as a function of the ratio of the heat flux to the surface superheat temperature in figure 6.14. If the thermistor temperature would be representative for the IGBT die temperature, an almost constant thermal resistance is expected. This is however not the case: the thermal resistance is high for low heat transfer rates and decreases significantly for increasing heat transfer rates. The variation of the thermal resistance indicates that the measured temperature is somewhere between the actual die temperature and the surface temperature.



Figure 6.13: Module temperature measured by NTC thermistor as a function of the average surface temperature.



Figure 6.14: Internal thermal resistance as a function of the boiling surface heat transfer rate.

It can be seen in figure 6.3 that the thermistor is not located in close proximity of an IGBT die. At low heat transfer rates at the boiling surface, more spreading of the heat will occur in the top layer of the module and the thermistor temperature will be close to the IGBT die temperature. At higher heat transfer rates at the boiling surface, heat will be spread less in the top substrate layer of the module but more in the baseplate. As such, the thermistor temperature will move closer to the boiling surface temperature.

An estimation of the IGBT die temperature is made using the thermal resistance reported in the power module datasheet. The estimated temperatures are shown in figure 6.15. At lower surface temperatures and lower heat transfer rates, the measured thermistor temperature and the estimated IGBT die temperature are very similar. In some cases the IGBT temperature is lower than the thermistor temperature, which is not a valid result. This might be a result of the relatively higher heat losses at low heat fluxes (see next section) or the reported datasheet thermal resistance may be an underestimation. At the highest surface temperatures and thus heat fluxes, the estimated IGBT temperature can be up to 10 °C higher than the measured thermistor temperature. It is concluded that the thermistor measurement is not a very good indicator of the IGBT temperature.



Figure 6.15: Measured module temperature and estimated IGBT die temperature as a function of the average surface temperature.

For all measurements, excluding critical heat flux measurements, the temperature measured by the thermistor and the estimated IGBT temperature remain below 81 °C and 91 °C. This is well below the allowable maximal IGBT temperature of 150 °C. During critical heat flux however, the measured thermistor temperature rises steeply and would become higher than 150 °C if the power to the power module was not decreased.

6.4.3 Energy balance

The heat losses of the setup are analysed using the experimentally measured energy balance. The net efflux of energy is shown as a function of the dissipated power in the module in figure 6.16. For most measurement points, the net efflux is slightly lower than the dissipated power, indicating some heat losses.



Figure 6.16: Net efflux as a function of electrical power.

The estimated heat losses are plotted as a function of the dissipated power in figure 6.17. At low powers (below 300 W), the estimated heat losses are significantly higher than zero. For higher heat fluxes, the absolute uncertainty on the losses increases and the losses become (mostly) smaller than the measurement uncertainty.

On average, the heat losses are smaller when compared to the heater configuration. The configuration with power module has no guard heater, thus more losses are expected from the bottom of the power module. However, due to the lower thermal resistance from the module components to the boiling surface, temperatures in the module are lower than those of the heater in the heater configuration



Figure 6.17: Estimated heat losses as a function of electrical power.

(see previous section). These lower temperatures result in lower losses.

The relative deviation of the energy balance is shown in figure 6.18. The relative deviation decreases significantly with heat flux, which is the result of the increasing heat transfer coefficient of nucleate boiling heat transfer with heat flux. From a heat flux of 65 kW/m², the energy balance closes within 5% and from 100 kW/m² it remains within 3%. This indicates that for the higher heat fluxes, the actual surface heat flux will be within 5% of the heat flux determined from the electrical power dissipation.

6.4.4 Boiling curve

Figure 6.19 shows the boiling curve for a saturation temperature of 36 °C and a liquid height of 1 cm. All measurement points are in the nucleate boiling region (except for the cross marker which is at the critical heat flux), no points of the natural convection region are included. The measurements were spread over several days, with several points repeated at different instances to check repeatability of the measurements. As shown in section 4.8.3, the measurement uncertainty on the heat flux and surface superheat temperature is lower than for the heater configuration. For the clarity of the figures, the error bars are not shown on most of the following figures.

In the boiling curve, the three different zones with different slopes can be



Figure 6.18: Relative deviation of the energy balance as a function of heat flux.



Figure 6.19: Boiling curve for a saturation temperature of 36 °C and a liquid height of 1 cm. The cross marker indicates critical heat flux.

clearly distinguished. The initial zone with lower slope lasts up to heat fluxes around 20 kW/m². The second zone, where the slope of the boiling curve is significantly higher, ranges from 20 kW/m² to about 90 kW/m². For even higher heat fluxes, the slope of the boiling curve starts to decrease again until the maximum heat flux is reached. These three zones are also clearly visible in figure 6.20, which shows the boiling curve with the superheat temperature and heat flux on logarithmic axes. The entire boiling curve cannot be fully correlated by a single power-law equation, as this would result in straight line on a log-log plot as in figure 6.20.



Figure 6.20: Boiling curve for a saturation temperature of 36 °C and a liquid height of 1 cm on a log-log plot.

6.4.4.1 Effect of liquid height

Three different liquid heights are tested: 1 cm, 6 cm and 17 cm. All measurement points for a saturation temperature of 36 °C are shown in figure 6.21. Most measurements were done at the lowest liquid level, while the full boiling curve is also measured for the highest liquid level. For the medium liquid level, only two points were taken for reference. All these points fall onto the same boiling curve. For the liquid height range between 1 cm and 17 cm, it is concluded that there is a negligible influence of the liquid height on the boiling curve. In the following section, the measurement points for different liquid heights are shown together, as also for other saturation temperatures, the liquid height does not seem to affect the



boiling curve.

Figure 6.21: Boiling curves for a saturation temperature of 36 °C and varying liquid height.

6.4.4.2 Effect of saturation temperature

The boiling curve was measured for saturation temperatures equal to 36 °C, 41 °C and 46 °C. These boiling curves are shown in figure 6.22. For saturation temperatures of 41 °C and 46 °C, the partial nucleate boiling region was not measured as this is of less importance for electronics cooling and as the uncertainty on the energy balance is relatively high. The heat transfer rates increase with increasing saturation temperature, which is in agreement with findings in literature. In the fully developed boiling region, the slope of the boiling curve decreases with increasing saturation temperature. Most correlations use a constant exponent to describe the boiling curve. Only the Gorenflo and the Leiner correlation (which is based on the Gorenflo correlation) take into account a variation of the exponent with the saturation pressure (and thus saturation temperature). These correlations also predict a decrease in the slope for an increase in saturation temperature.



Figure 6.22: Boiling curves for varying saturation temperatures.

6.4.4.3 Effect of fouling

Fouling occurred several times during the measurements on the power module configuration. No more silicone mastic is used in this configuration, as this is assumed to be the source of the fouling perceived in the heater configuration. Visually, there was a clear distinction between fouling in the heater configuration (figure 5.21) and in the power module configuration, which is shown in figure 6.23. The fouling layer which contained silicone was quite uniform, while the fouling without the silicone was more localized. It is assumed that these local fouling spots are around nucleation sites.

The composition of these fouling spots was not tested, so only speculations can be made about the source of the fouling. Oxidation was seen on both stainless steel and copper surfaces present in the reservoir. This indicates that air was present in the reservoir, which enters through small leaks. Water vapour will also enter the reservoir with the air and can additionally enter through the polycarbonate windows, which are slightly permeable to water vapour. FK-649 reacts with water to form an acid [137]. This acid can react with the metal oxides to form salts which can dissolve in the refrigerant and deposit on the surface. Additionally, leftover organic materials on the silicone gaskets can also dissolve in the refrigerant and deposit on the boiling surface.

The change in the boiling curve for a clean and fouled surface is shown in fig-



Figure 6.23: Fouling on the power module baseplate.

ure 6.24 for a saturation temperature of 36 °C. Remarkably and in contrast to the measurements on the heater configuration, heat transfer rates are increased for the fouled surface. This is attributed to the different type of fouling. In the heater configuration, a uniform layer is deposited on the surface, which presumably results in a conductive thermal resistance. In the power module configuration, due to the absence of silicone mastic, another type of fouling occurs which causes very local spots and does not cover the entire surface. These spots result in an increase of the surface roughness (and might also affect the wettability of the surface). The increased surface roughness aids nucleation, while the local spots only cause a negligible conductive thermal resistance. This phenomenon is also reported on for pool boiling heat transfer with initial calcium sulfate deposits, where heat transfer rates are also increased [111]. This could also indicate that the depositions in the power module configuration are related to dissolved salts in the refrigerant.

6.4.4.4 Nucleate boiling regimes

The boiling surface is visually accessible through the polycarbonate windows. Videos have been made of the boiling phenomenon for varying heat fluxes and a saturation temperature of 46 °C. These videos can be accessed on-line (https://users.ugent.be/~itjollyn/video). Still frames of these videos are shown in figure 6.25.

The partial nucleate boiling regime can be seen in figure 6.25 (a) and (b). In this region, not all nucleation sites on the boiling surface are yet active. The nucleation site density increases with heat flux in this region.

From about 20 kW/m², the fully developed nucleate boiling regime occurs.



Figure 6.24: Boiling curves for a saturation temperature of 36 °C on clean and fouled surfaces.

This is visible in figure 6.25 (c), (d) and (e). The entire boiling surface is covered with bubble generating locations. Individual vapour bubbles emerge from the surface at the lower heat fluxes, while at higher heat fluxes bubbles begin to merge and more vigorous bubble dynamics can be perceived.

When the heat flux increases even further, above 90 kW/m^2 , the partial dryout regime starts. Vapour bubbles start to merge and gather close to the boiling surface, which can be seen in figure 6.25 (f) and (g). Vapour moves upward from the boiling surface as large vapour slugs.

When the critical heat flux is reached, the boiling behaviour changes drastically. This can be seen on figure 6.25 (h). A vapour film is formed covering the entire boiling surface, which emits vapour bubbles at a lower rate than during nucleate boiling.

The different behaviour of boiling during the three regimes can also be seen in the boiling curve. The measured boiling curve for a saturation temperature of 36 °C is shown on a log-log plot in figure 6.26. A distinction is made between the three nucleate boiling regimes. The three regimes can be represented by three power-law relations, which are fitted to the measurement data and shown in the figure.

In the partial nucleate boiling region, heat flux varies with the surface superheat temperature raised to the power 1.36. This power corresponds well to the power



(a) 11 kW/ m^2





(c) 40 kW/m^2





(e) 78 kW/m^2





 $(g) \, 119 \, kW/m^2$

(h) CHF (146 kW/m^2)

Figure 6.25: Still frames of nucleate boiling videos for different heat fluxes.



Figure 6.26: Boiling curve for a saturation temperature of 36 °C with an indication of the three nucleate boiling regimes.

found in natural convection heat transfer, indicating enhanced natural convection may be the dominant heat transfer mechanism in this region.

A far steeper rise in heat flux with surface superheat temperature is measured in the fully developed boiling region. For the saturation temperature regarded here (36 °C), heat flux varies with surface superheat raised to a power of 5.29. This value seems to be dependent on the saturation temperature, as less steep increases are measured for higher saturation temperatures (see figure 6.22).

For higher heat fluxes closer to CHF, in the partial dryout region, surface superheat temperatures increase more with heat flux than for the fully developed boiling region. The relation between heat flux and surface superheat is best correlated by a power-law with a power equal to 1.07, which indicates an almost linear relation between heat flux and surface superheat temperature.

To verify if this behaviour for the three nucleate boiling regimes is a general trend, more measurements with other boundary conditions (fluid, surface, saturation temperature) are needed.

6.4.5 Critical heat flux

6.4.5.1 Effect of saturation temperature

The critical heat flux as a function of the refrigerant saturation temperature is shown in figure 6.27. In the range investigated in this study (34 °C to 46 °C), CHF increases with saturation temperatures. The measured CHF ranges from 121 kW/m² to 146 kW/m².



Figure 6.27: Critical heat flux as a function of saturation temperature for varying liquid height.

6.4.5.2 Effect of liquid height

Figure 6.27 also shows the influence of the liquid height on CHF. For similar saturation temperatures, the measured CHF lies within 3% for the different liquid heights (1 cm, 6 cm and 17 cm). The small differences show no clear correlation with liquid height and are attributed to the uncertainties in determining the CHF. In the range of liquid heights tested in this study (1 cm to 17 cm), the influence of the liquid height on the CHF is thus negligible.

6.4.5.3 Effect of fouling

The effect of fouling on the critical heat flux is shown in figure 6.28. CHF on the fouled surface is consistently higher than on the clean surface by 15% over the



range of saturation temperatures tested. This is in agreement with other studies of CHF on fouled surfaces (see section 3.5.2).

Figure 6.28: Critical heat flux as a function of saturation temperature on a clean and a fouled surface.

6.5 Comparison with heater configuration

Most parameters and boundary conditions are similar when comparing the heater and power module configurations. The size of the boiling surface, orientation and refrigerant are equal during all the experiments, while saturation temperatures and liquid height are equal for certain measurement points. There are three main differences between both configurations. The heat flow applied in the heater configuration is uniformly distributed over the boiling surface. In the power module configuration, heat is dissipated in the component and subsequently spreads through the baseplate. Secondly, the roughness of the surface in the heater and power module configuration are respectively 0.9 μ m and 0.2 μ m. Finally, the boiling surface in the heater configuration is made of aluminium, while the baseplate of the power module is copper.

Figure 6.29 shows the boiling curve for both configurations with a saturation temperature equal to 36 °C. In the partial nucleate boiling region, heat transfer rates are higher for the power module configuration. However, at these low heat



fluxes, the relative deviation of the energy balance is rather high, indicating that this difference between configurations can be a result of a difference in heat losses.

Figure 6.29: Boiling curves for the heater and power module configuration with a saturation temperature of 36 °C.

For the fully developed nucleate boiling region, the boiling curve of the heater configuration is shifted slightly to the left compared to that of the power module configuration. This is most probably the result of the higher surface roughness of the aluminium boiling surface. The difference in surface superheat between the two configurations in the fully developed nucleate boiling region is smaller than 10%, indicating that the boiling heat transfer behaviour for both configurations is very similar. From these results, it is assumed that the baseplate of the inverter module provides adequate spreading to achieve a quasi-uniform heat flux at the baseplate-fluid interface.

This assumption of uniform heat flux is also checked by visually inspecting the boiling phenomenon. The onset of nucleation occurs at the boiling surface edges and random locations on the boiling surface. With increasing heat flux, the nucleation sites spread evenly of the boiling surface. Both for the onset of nucleation and throughout of the nucleate boiling regimes up to critical heat flux, bubble and vapour generation occurs uniformly on the boiling surface. No regions of more vigorous vapour generation coinciding with the location of the module components were perceived. From these visual observations and the similarity of the boiling curves, it is concluded that a quasi-uniform heat flux is present at the boiling surface. This is of importance for the design of pool boiling cooling systems for power modules, as there are no local hot spots and the existing correlations for uniform heat flux are applicable.

6.6 Cooling performance analysis

The measured cooling performance of pool boiling with FK-649 can be compared to the other cooling methods which are summarized in table 2.1 (and table 6.1). The nucleate boiling heat transfer coefficient ranges from 973 W/m²K (partial nucleate boiling region and low saturation temperature) to 4965 W/m²K (partial dryout region and high saturation temperature). This is higher than water-glycol cold plate cooling and is comparable to direct baseplate water-glycol cooling. However, the heat transfer coefficient is lower than for pool boiling with HFE-7100 and R134a. Water-glycol cooling with fins integrated in the baseplate also results in higher heat transfer rates.

Pool boiling cooling with R134a and also to a lesser extent with HFE-7100 outperforms cooling with FK-649. This is related to the superior thermal properties of the fluids. However, as shown in section 2.5, these fluids have other drawbacks which make them inapplicable for cooling of (mobile) electric drivetrains.

Water-glycol cooling of the power module baseplate with integrated fins also thermally outperforms FK-649 pool boiling cooling. The large difference in equivalent heat transfer coefficient is mainly related to the fins which increase the heat transfer area up to five times and also increase the convective heat transfer coefficient.

Next to the heat transfer rates, the maximal heat flux is very important. This heat flux can be limited by the critical heat flux, but also by the temperature limit of the IGBTs. The maximal continuous operating temperature of the IGBTs is 150 °C. When considering a typical water-glycol coolant temperature of 65 °C, a temperature difference of 85 °C is available for cooling. Taking into account the total thermal resistance (internal module thermal resistance and heat sink thermal resistance), the maximal heat flux for the different types of heat sinks can be calculated.

Table 6.1 gives an overview of the heat transfer coefficients (U), maximal heat fluxes for an IGBT temperature equal to 150 °C (\dot{q}_{max}) and critical heat fluxes. For all two-phase cooling methods, the maximal heat flux is limited by the critical heat flux, not by the temperature limit of the IGBTs.

The maximal heat flux measured for a clean surface is equal to 146 kW/m². This is lower than the heat fluxes attainable with cold plate cooling (186 kW/m²). For the method to become feasible, strategies to increase CHF need to be investigated. Increasing the heat transfer area will increase the maximal heat flux attainable from the module. Furthermore, higher saturation temperatures can increase

	U	\dot{q}_{max}	\dot{q}_{CHF}
Method	$[W/m^2K]$	$[kW/m^2]$	$[kW/m^2]$
WG cold plate [19]	2814	186	-
WG baseplate [20]	7880	372	-
WG pin fin [19]	46000	690	-
WG straight fin [22]	24000	594	-
HFE-7100 boiling [26]	13000	476	210
R134a boiling [26]	50000	699	600
FK-649 boiling	4965	281	146

Table 6.1: Maximal heat transfer coefficients and heat fluxes, based on baseplate area, for different cooling methods.

the CHF up to around 200 kW/ m^2 , but this also decreases the temperature difference available for cooling. Using subcooled liquid as coolant is also a viable strategy to increase CHF.

It is clear that FK-649 pool boiling on a flat baseplate does not achieve the same cooling performance as most state-of-the-art water-glycol cooling methods. Increasing the heat transfer area is the most straightforward solution to increase both the heat transfer rates and maximal heat flux. A possibility in this regard is to use a power module with a baseplate with integrated pin fins. It is advised to conduct new experiments on this geometry, as the heat transfer rates and critical heat flux on finned surfaces are presumably lower than those of a flat plate, when recalculating to the total heat transfer area.

Fouling on the boiling surface can cause both a decrease and an increase in the heat transfer rates in the nucleate boiling region. The critical heat flux is increased for a fouled surface. Although the most limiting factor for cooling is the CHF, excessive fouling might cause a thermal resistance high enough to induce overheating of the module. It is therefore advised to take measures to reduce fouling, such as using silica packs to avoid contact with water and adding (carbon) filters to remove contaminants. The cooling design should be based on the CHF of a clean surface. As such, if some fouling occurs, an additional margin of safety is present as the CHF increases for a fouled surface.

6.7 Conclusions

Pool boiling cooling of the baseplate of a power module with FK-649 was experimentally tested. Heat dissipation was induced in the IGBTs of the power module, while the internal temperature and baseplate temperature are monitored. Heat dissipation, coolant temperature and liquid height were varied to analyse their effect on the baseplate and module internal temperature. The following conclusions can be drawn from the measurements:

- The internal thermistor measurement was found to underestimate the actual IGBT temperature. For heat fluxes below the critical heat flux, the measured internal temperature was always below 81 °C, while the estimated IGBT temperature reached up to 91 °C. Both values are well below the allowable maximal temperature of 150 °C for the power module components.
- Three nucleate boiling regimes were perceived visually and in the measurements: partial nucleate boiling, fully developed nucleate boiling and partial dryout boiling. The boiling curve behaves differently in these three regimes: it has a slope similar to that of natural convection in the partial nucleate boiling region, a steeper slope in the fully developed nucleate boiling region and a lower slope in the partial dryout region. The entire nucleate boiling curve cannot be represented by a single power-law relation.
- The critical heat flux was determined through the measurements, the highest value measured was 146 kW/m². This is a limit for the maximal heat dissipation of the power module.
- The liquid height was varied, but had no effect on the boiling curve and the critical heat flux.
- An increase in saturation temperature resulted in an increase in heat transfer rate and of critical heat flux in the range tested in this study.
- Fouling occurred on the boiling surface. This resulted in an increase in heat transfer rates and critical heat flux (by about 15%).
- No influence of the non-uniformity of the heat flux was noticed visually. The measured boiling curves were very similar to those of the heater configuration with uniform heat flux.

Finally, the measured cooling performance is compared to that of other cooling methods. The heat transfer coefficients are higher than those of water-glycol cold plate cooling and similar to direct baseplate water-glycol cooling. However, attainable heat fluxes are lower due to the limiting critical heat flux value. Higher heat transfer coefficients and heat fluxes are featured for water-glycol cooling of module with pin fins and straight fins integrated in the baseplate. Immersion boiling cooling of a flat baseplate is therefore significantly inferior to baseplate waterglycol cooling with integrated fins. The most straight-forward next step in the research is to test pool boiling on baseplates with fins, as this is also the strategy used to reduce the thermal resistance in single-phase cooling.

Assessment of nucleate pool boiling correlations

7.1 Introduction

This chapter intends to make an assessment of the predictive quality of nucleate pool boiling heat transfer and critical heat flux correlations for the experimental data of the previous two chapters. In the following section, an overview is given of the assessed correlations, the inputs of these correlations and the performance parameters that are used. Next, the heat transfer correlations are compared to the experimental data. A global assessment is made, but the effects of heat flux, saturation temperature and boiling surface properties are also evaluated independently. A similar assessment is done for the critical heat flux correlations. Using the correlations which performed best, an assessment is done of heat transfer rates and critical heat fluxes for other working fluids. Finally, some general conclusions are made with regards to the predictive quality of the correlations.

7.2 Methodology

To assess the predictive quality of the correlations described in section 3.3 and 3.5, the boiling curve and critical heat flux predicted by the correlations are compared to those gathered from the measurements.

The heat transfer correlations are assessed by determining the surface super-

heat temperature based on the different correlations and comparing this to the measured surface superheat temperature. For the critical heat flux correlations, the measured and predicted fluxes are compared.

7.2.1 Correlations

In the literature review on nucleate boiling heat transfer correlations (section 3.3), thirteen correlations have been described. For the Rohsenow and Pioro correlations, two predictions are made using either a priori determined constants from other studies or by fitting the constants to the measurements of this study. This results in fifteen different heat transfer correlations to be analysed:

- Kruzhilin
- Rohsenow a priori
- Rohsenow fit
- Forster-Zuber
- Kutateladze-Borishanskii
- Borishanskii-Mostinski
- Shekriladze-Ratiani
- Labuntsov
- Gorenflo
- Stephan-Abdelsalam
- Cooper
- Kutateladze
- Leiner
- Pioro a priori
- Pioro fit

Six correlations for CHF are described in section 3.5. The Chang-You correlation is used to evaluate the effect of heat transfer surface inclination. Since this parameter is not varied in this study, this correlation will not be evaluated. The Kandlikar correlation takes into account heat transfer surface inclination and the effect of the static contact angle. As the static contact angle is not measured in this study, also this correlation cannot be evaluated. Four correlations can thus be considered for CHF prediction:

- Kutateladze
- Zuber
- Lienhard-Dhir
- Mudawar et al.

7.2.2 Inputs

All heat transfer correlations require the measured heat flux as an input parameter. Besides the heat flux, all correlations require information on the state of the fluid and/or some of the fluid properties. The state of the fluid is always at saturation and either the measured temperature is used (if the liquid level is 17 cm) or the measured pressure with the correction for the hydrostatic head of equation 4.3 is used (if the liquid level is lower than 17 cm). The potentially required fluid characteristics and properties are determined using the REFPROP program [152], which determines the equation of state from Mclinden et al. [148], liquid viscosity from Wen et al. [153], liquid thermal conductivity from Perkins et al. [154] and surface tension from Cui et al. [155].

Several correlations use the gravitational acceleration as a parameter, which is in this case equal to 9.81 m/s². The arithmetic mean roughness is required for some correlations, which is equal to 0.9 μ m for the heat configuration and 0.2 μ m for the power module configuration. If $R_{p,DIN}$ is used as a parameter, it is estimated from the arithmetic mean roughness with equation 3.69.

The Stephan-Abdelsalam correlation (equation 3.52) uses the static contact angle, which is assumed here equal to 35°, as is also done in the analysis of Stephan and Abdelsalam for refrigerants [79]. This parameter was not measured, so it is merely a best guess.

The Gorenflo correlation (equation 3.46) requires the property $k_w \rho_w c_{p,w}$ of the wall material, which is equal to 561 kJ²/(m⁴K²s) for the heater configuration (aluminium) and equal to 1250 kJ²/(m⁴K²s) for the power module configuration (copper).

The Shekriladze-Ratiani correlation (equation 3.44) uses the radius of the critical nucleus as a parameter for the microgeometry of the boiling surface. As it is unclear how to determine this parameter, a value of 5 μ m is used as is suggested by Shekriladze [74]. This value is used for both tested boiling surfaces.

Both the Rohsenow and the Pioro correlations (respectively equation 3.31 and 3.64) require fitting constants based on the surface-fluid combination. Two strategies are followed for both correlations: first the correlation is assessed using parameters found in previous studies and secondly the parameters are fitted using the measurements of this study. In the Rohsenow correlation, three parameters can be varied: C_{sf} , r and s. r and s are kept to their standard values of 0.33 and 1.7 respectively. For the heater configuration, a value of 0.0037 is used based on the measurements of Forrest et al. [53] on an aluminium surface. A value of 0.0050 is used for the power module configuration, based on the work of Cao et al. [143] for boiling on a copper surface. A least-squares fitting was used to determine the parameter based on the measurement performed on both configurations. Remarkably, the fitted parameter was equal for both configurations at 0.0051 after rounding to the fourth decimal.

Two parameters should be fitted for each surface-fluid combination in the Pioro correlation: C_{sf}^* and m. No values for these parameters are available for FK-649. The values for another refrigerant, R-113, are used instead and are gathered from [83]. For aluminium, values of 45620 and -2.35 are used for C_{sf}^* and m respectively, while for copper these parameters are equal to 168885 and -3.14 respectively. Using a least-squares fitting procedure, the following values were found for C_{sf}^* and m: 50111 and -2.29 for the aluminium boiling surface and 34405 and -2.13 for the copper boiling.

7.2.3 Performance parameters

The predictive performance of the correlations is determined by calculating the normalized mean absolute error (NMAE). This is the average value of the absolute deviation of the predicted value to the measured value, scaled to the measured value. The lower the NMAE, the closer the correlation matches the experiments. Another interesting parameter is the maximal normalized absolute error (NAE_{max}) , which gives the maximal deviation of the correlation.

For the boiling curve, NMAE and NAE_{max} are determined as:

$$NMAE = \frac{1}{N_m} \sum_{i=1}^{N_m} \frac{|\Delta T_{s,m,i} - \Delta T_{s,p,i}|}{\Delta T_{s,m,i}}$$
(7.1)

$$NAE_{max} = max \left(\frac{|\Delta T_{s,m} - \Delta T_{s,p}|}{\Delta T_{s,m}}\right)$$
(7.2)

In these equations, N_m is the number of measurement points and the indices m and p indicate measured and predicted values. The measurements points with a fouled boiling surface and of critical heat flux are not included in the analysis of the boiling curve prediction, as predicting these trends is not included in the correlations.

For the critical heat flux correlations, the definition of NMAE and NAE_{max} are as follows:

$$NMAE = \frac{1}{N_m} \sum_{i=1}^{N_m} \frac{|\dot{q}_{CHF,m,i} - \dot{q}_{CHF,p,i}|}{\dot{q}_{CHF,m,i}}$$
(7.3)

$$NAE_{max} = max \left(\frac{|\dot{q}_{CHF,m} - \dot{q}_{CHF,p}|}{\dot{q}_{CHF,m}} \right)$$
(7.4)

Only critical heat flux measurements on clean surfaces are evaluated.

7.3 Nucleate boiling heat transfer

7.3.1 Predictive performance

A total of 91 measurements points is used in the assessment of the correlations. Figure 7.1 shows the predicted surface superheat temperature as a function of the measured surface superheat temperature for the fifteen correlations. The corresponding NMAE for the different correlations is shown in table 7.1.

Correlation	NMAE	NAE_{max}
Pioro - fit	7.4%	26.4%
Rohsenow - fit	7.5%	25.6%
Labuntsov	7.5%	29.1%
Rohsenow - a priori	10.6%	46.4%
Forster-Zuber	13.7%	45.9%
Stephan-Abdelsalam	17.2%	46.8%
Cooper	25.2%	44.6%
Shekriladze-Ratiani	26.3%	42.2%
Kruzhilin	30.6%	54.6%
Leiner	31.7%	47.4%
Gorenflo	36.5%	51.8%
Kutateladze	45.6%	59.2%
Borishanskii-Mostinski	47.9%	71.6%
Kutateladze-Borishanskii	50.4%	73.0%
Pioro - a priori	94.8%	142.5%

Table 7.1: NMAE and NAE_{max} of the heat transfer correlations with regard to the measurements.

Unsurprisingly, the fitted correlations (Rohsenow and Pioro) perform best. Although Pioro has two fitted parameters, it performs negligibly better than the Rohsenow correlation which only has one fitting parameter. The Rohsenow correlation also performs very well using parameters from literature for the same surface material and fluid combination. For the Pioro correlation, parameters for another refrigerant were used, which results in highly diverging results. This indicates that the correlation cannot be used if no measurements for fitting or parameters for the required fluid-surface combination are available.



Figure 7.1: Predicted surface superheat temperature as a function of the measured surface superheat temperature for fifteen correlations.

Of the correlations without fitting constants, the Labuntsov correlation has the same NMAE as the fitted Rohsenow correlation, which is a very remarkable feat. It predicts all measurement points within 30%.

Next in line (after the Labuntsov and Rohsenow correlations) is the Forster-Zuber correlation. This result is quite unexpected, as this correlation is fitted to critical heat flux data and has a very diverging relation between heat flux and surface superheat temperature when compared to other correlations. On figure 7.1, it can be seen that the correlation is indeed very accurate for the high surface superheats (near critical heat flux). The maximal deviation of all measurement points (45.9%) is however in line with most other correlations.

The Stephan-Abdelsalam, Cooper, Shekriladze-Ratiani, Kruzhilin, Leiner and Gorenflo correlations all have an NMAE ranging from 17% to 37% and a maximal deviation within 42% and 55%. These correlations are not quite able to properly predict the boiling curves measured in this study.

The Kutateladze-Borishanskii, Borishanskii-Mostinski and Kutateladze correlations all have higher NMAE and maximal deviations. These correlations are not suitable to predict heat transfer rates for the conditions in this study.

Although the NMAE and the maximal deviation are the most crucial parameters to assess the predictive quality of the correlations, they do not render a full picture of the adequacy of the correlation. As only one fluid is tested for a limited range of saturation temperatures, the NMAE and maximal deviation mostly analyse if the correlation is able to predict the magnitude of the overall heat transfer rate for the regarded fluid. The correlation's prediction of the variation of surface superheat with heat flux, saturation temperature and boiling surface is therefore not included in these results. The following section will go more into detail on these effects and how well the correlations are able to take them into account.

7.3.2 Effect of heat flux

To analyse how the correlations take into account the effect of heat flux on the surface superheat temperature, the measured boiling curve is plotted along with those predicted by the correlations. The boiling curve with the most measurement points is chosen, which is the curve for the power module configuration with a saturation temperature of 36 °C. To avoid overcrowding of the figure, it is divided in three plots with each five correlations. A logarithmic scale is used for both axes, which transforms power-law relations to straight lines. As all correlations use a power-law relation between surface superheat and heat flux, the correlations will all be plotted as a straight line.

The boiling curve comparison between measurements and predictions by correlations is shown in figures 7.2, 7.3 and 7.4. The first conclusion that can be drawn is similar to that of the previous section: the Rohsenow (a priori and fitted), Labuntsov and Pioro (fitted) correlation provide the best match between measurement and correlation for the entire nucleate boiling range.



Figure 7.2: Measured boiling curve and boiling curves according to correlations (1/3).

As all correlations use a power-law relation between surface superheat temperature and heat flux, none of the correlations are able to grasp the differences between the three nucleate boiling regimes.

The Forster-Zuber correlation has a significantly too low slope to be able to properly represent the entire boiling curve. However, on figure 7.2 it is also clear that it very closely predicts the heat transfer rates close to CHF.

Gorenflo and derived Leiner correlation have a significantly higher slope than all the other correlations. This slope matches well with the slope of the measured boiling curve in the fully developed nucleate boiling region. Both correlations however predict too high heat transfer rates.

All other correlations have a similar slope, corresponding to an exponent n in the range of 3 to 3.33. This slope matches well for the general trend over the three boiling regions. To properly predict heat transfer rates in all three regions, different values for the exponent are required. For further improvement of the prediction of heat transfer rates in nucleate boiling, the author therefore recommends to make a distinction between the three regimes in analysing and correlating heat transfer data.



Figure 7.3: Measured boiling curve and boiling curves according to correlations (2/3).



Figure 7.4: Measured boiling curve and boiling curves according to correlations (3/3).

7.3.3 Effect of saturation temperature

The change of saturation temperature results in a variation in the fluid properties which affects the boiling curve. To analyse which correlations predict these changes best for FK-649 and the saturation temperature range tested in this study, the measured heat flux is plotted as a function of the product of the measured surface superheat and the function $F(p_r)$. The function $F(p_r)$ is defined by equation 3.68 and is specific to each correlation. The function is dependent on the reduced pressure (or equivalently the saturation temperature) and is equal to 1 at a reduced pressure of 0.1. If the correlation accurately predicts the effect of changing saturation pressure, the boiling curves of the different saturation temperatures should combine into one curve.

For this analysis, the measurements on the power module configuration are used, as more variation in saturation temperature was tested for this configuration than for the heater configuration. On the top left of figure 7.5, the original measurements are shown without any corrections for the saturation temperature. The three levels of saturation temperature are shown in different colours.

Only one plot is shown for the Rohsenow correlation, as the value of the fitting constant has no influence on the effect of saturation temperature in the correlation. For the Pioro correlation, two plots are shown, as the second fitting parameter m is the exponent of the Prandtl number in the correlation and as such does influence the variation with saturation temperature.

For all correlations, the boiling curves of the three saturation temperatures move closer to each other. The best results are found for the Rohsenow, Labuntsov, Gorenflo, Leiner and fitted Pioro correlations. The Rohsenow, Labuntsov and fitted Pioro correlations also performed well for the boiling curve prediction, while the Gorenflo and Leiner correlations were not as successful. This indicates that for the Gorenflo and Leiner correlations, the function $F(p_r)$ performs adequately for FK-649, but that the part of the correlation which takes into account differences between fluids does not work properly for FK-649.

7.3.4 Effect of boiling surface

A similar approach as for the effect of the saturation temperature is followed to analyse the effect of the boiling surface and how it is handled by the correlations. The measured heat flux is plotted as a function of the product of the measured surface superheat temperature and any factor in the correlations that is influenced by the boiling surface. As the numerical value of this product has no physical significance, the numerical values on the axes of the plots have been omitted.

Measurement points with a saturation temperature of 36 °C are used. The boiling curve of the heater configuration is compared to that of the power module configuration and is shown in the top left corner of figure 7.6. Measurement points



Figure 7.5: Measured heat flux as a function of the product of the measured surface superheat temperature and $F(p_r)$ for the measurements on the power module configuration. The functions $F(p_r)$ are based on the heat transfer correlations.

of the heater configuration are shown in blue, while those of the power module configuration are shown in red. If the correlations are able to predict the effect of the boiling surface accurately, the boiling curve of both configurations should align.

Not all correlations take into account effects of the boiling surface. The Kruzhilin, Forster-Zuber, Kutateladze-Borishanskii, Borishanskii-Mostinski, Labuntsov and Kutateladze correlation have no parameters related to the boiling surface. The Shekriladze-Ratiani correlation uses the radius of the critical nucleus as a parameter of the boiling surface. However, as this parameter is unknown for the surfaces in this study, its effect cannot be analysed.

The Rohsenow and Pioro correlations take into account effects of the boiling surface using fitting parameters. The correction proposed by Stephan [92] using the arithmetic mean roughness is used in the Stephan-Abdelsalam, Gorenflo and Leiner correlations. The Cooper correlation also takes into account surface roughness, but uses a different parameter ($R_{p,DIN}$) and correlates the effect of roughness together with reduced pressure. In the Gorenflo correlation, also the effusivity of the boiling surface material is taken into account.

The results are shown in figure 7.6. Utilizing fitting coefficients from previous studies in the Rohsenow and Pioro correlations does not result in a good agreement between the boiling curves. This was to be expected, as surface roughness also affects the fitting parameters. It is unlikely that adequate constants can be found in literature for the correct combination of fluid, surface material and surface roughness. When fitting the constants to the measurements of this study, naturally a good agreement between the boiling curves can be found.

The correction proposed by Stephan moves the boiling curves further away from each other. Two possible explanations can be given. The first possibility is that the arithmetic mean roughness is not an adequate parameter to describe the effect of the surface microgeometry on the boiling curve. The second option is that the boiling surface material properties also affect the boiling curve. As both influences were not tested independently in this study, no conclusion can be drawn as to which explanation is correct.

The correction proposed by Cooper moves the boiling curves even further away from each other. As the reduced pressure for a saturation temperature of 36 °C is equal to 0.0337, the exponent to which the roughness parameter is raised is equal to 0.295. This is significantly higher than that of the other corrections. The results of this study do not seem to agree with the assumption of a large influence of surface roughness at relatively low reduced pressures.

The Gorenflo correlation utilizes two corrections based on the surface roughness and surface thermal effusivity. These effects cancel out for the combinations tested in this study, which results in a good fit with the measured boiling curves.

Due to the combination of two effects (surface microgeometry and surface



Figure 7.6: Measured heat flux as a function of the product of the measured surface superheat temperature and factors based on the boiling surface according to the heat transfer correlations.

material) which were not tested independently, no firm conclusions can be made on the effects of both parameters separately. For more conclusive results, additional measurements are required.

7.4 Critical heat flux

7.4.1 Predictive performance

The critical heat flux was determined on 8 different occasions (for a clean boiling surface). These measurement points are compared to the predictions of the four selected correlations in figure 7.7. The corresponding values of the NMAE and the maximal deviation are shown in table 7.2.



Figure 7.7: Predicted CHF by the correlations as a function of the measured CHF.

The Kutateladze correlation overestimates the critical heat flux by about 15%, while the other correlations predict CHF within 11%. Both the Lienhard-Dhir and Mudawar et al. correlations overestimate CHF by about 7%, while the Zuber correlation predicts CHF values of around 6% lower. The differences in predictive performance for these three correlations are small. For the measurements in this study, the Zuber correlation has the best prediction, followed by the Lienhard-Dhir and Mudawar et al. correlations.
Correlation	NMAE	$\mathbf{NAE}_{\mathbf{max}}$
Zuber	6.1%	9.0%
Lienhard-Dhir	6.6%	9.8%
Mudawar et al.	7.4%	10.7%
Kutateladze	14.5%	17.9%

Table 7.2: NMAE and NAE_{max} of the CHF correlations with regard to the measurements.

7.4.2 Effect of saturation temperature

The effect of the saturation temperature on CHF is shown in figure 7.8 for both the measurement and the correlations. All correlations follow the trend with saturation temperature adequately.



Figure 7.8: CHF as a function of saturation temperature for the measurements and correlations.

From the literature study (section 3.5.2) it is known that the critical heat flux increases and reaches a maximal value for increasing saturation temperature. This maximal critical heat flux can be analysed by plotting the correlations over a larger saturation temperature range, as is done in figure 7.9. The maximal CHF for FK-649 will be reached for saturation temperatures in the range of 100 °C to 110 °C and ranges from 190 kW/m² to 210 kW/m² (excluding the prediction with the

Kutateladze correlation as this had the highest deviation compared to the measurements). This is an increase of 30% to 40% compared to the highest CHF measured in this study. If the CHF is the limiting factor in the design of the cooling system, it can thus be beneficial to increase the saturation temperature of the coolant.



Figure 7.9: CHF as a function of saturation temperature for the measurements and correlations with extrapolation to higher temperatures.

7.5 Assessment of other working fluids

FK-649 was chosen as working fluid mostly for legislative and practical reasons. With the best performing correlations for heat transfer and critical heat flux, this choice of fluid can be evaluated on a thermal basis. A comparison is made between FK-649 and several other possible fluids covered in section 2.5. The results are summarized in table 7.3. The fluids are ordered based on their molar mass as there is a correlation with several other parameters. The saturation pressure and reduced pressure are also shown for a saturation temperature of 46 $^{\circ}$ C.

The expected surface superheat temperature is determined with the Labuntsov correlation and for a saturation temperature of 46 °C and a heat flux of 100 kW/m^2 . The surface superheat temperature can be roughly correlated to the reduced pressure. For the fluids with low molar mass, the reduced pressure is mostly higher as they typically have higher working pressures for equal saturation temperatures. HFO-1336mzz(Z), the second suitable candidate fluid besides FK-649, has a slight-

ly lower surface superheat temperature but it is comparable to FK-649. The fluids HFO-1234yf and HFO-1234ze(E) have significantly lower surface superheat temperatures, but require high working pressures (about 12 and 9 bar respectively) to operate in the two-phase region at the desired saturation temperature.

A prediction for the critical heat flux is made using the Zuber correlation, based on a saturation temperature of 46 °C. For these fluids, the critical heat flux is quite well correlated to the molar mass. Lower molar masses result in higher critical heat fluxes. HFO-1336mzz(Z) has a significantly lower molar mass than FK-649 (164 g/mol and 316 g/mol respectively). The predicted critical heat flux for HFO-1336mzz(Z) is 80% higher than that for FK-649, indicating that this fluid is potentially interesting to investigate as alternative to FK-649. If even higher critical heat fluxes are needed, HFO-1234yf and HFO-1234ze(E) are potential fluids, with the drawback of requiring high working pressures.

	M	p_c	p_{sat} ^a	$p_r^{\ a}$	$\Delta T_s^{a,b}$	\dot{q}_{CHF} a,c
Fluid	[g/mol]	[bar]	[bar]	[-]	[K]	$[kW/m^2]$
HFO-1234yf	114	33.82	11.82	0.350	2.7	336
HFO-1234ze(E)	114	36.35	8.99	0.247	4.2	386
HCFO-1233zd(E)	130	36.24	2.60	0.072	13.8	303
HCFO-1224yd(Z)	148	33.37	2.95	0.088	12.0	294
HFO-1336mzz(Z)	164	29.03	1.57	0.054	19.1	240
HFE-7000	200	24.78	1.53	0.062	18.1	202
HFE-7100	250	22.30	0.62	0.028	28.4	139
FK-649	316	18.69	0.91	0.049	21.8	134

Table 7.3: Comparison of different working fluids.

^a evaluated at 46 °C

^b determined at 100 kW/m² with the Labuntsov correlation

^c determined with the Zuber correlation

7.6 Conclusions

The measurements were compared with predictions by correlations for heat transfer and the critical heat flux. For heat transfer, the correlations with fitting constants (Rohsenow and Pioro) naturally performed best. Of the correlations without any fitting constants, the Labuntsov correlation performed best with similar accuracy to the correlations with fitting constants (all data predicted within 29.1%). The change in the boiling curve for the three nucleate boiling regimes is not predicted by any of the correlations. The effect of saturation temperature on the boiling curve was predicted well by several correlations (Rohsenow, Labuntsov, Gorenflo, Leiner and fitted Pioro correlation). This indicates that both the approach using fluid properties and using the theory of thermodynamic similarity are able to predict changes in heat transfer due to varying saturation temperature. For the effect of the boiling surface (surface microgeometry and material) on heat transfer and the predictive performance of the correlations for these effects, additional measurements are required to make firm conclusions.

Critical heat fluxes are predicted best by the correlations of Zuber and Lienhard-Dhir, with all measurement points predicted within 10%. The effect of saturation temperature is properly captured by all the correlations. Overall, the accuracy of the critical heat flux correlations is far better than those of the heat transfer correlations.

Using the best performing correlations, an assessment is done of different working fluids. For higher heat transfer rates and critical heat fluxes, fluids with lower molar mass and higher working pressures should be used. HFO-1336mzz(Z) has an 80% higher critical heat flux values at a working pressure of 1.57 bar. Even higher critical heat fluxes can be achieved with fluids at higher pressures.

8 Conclusions

To conclude this study, it is a good idea to recap the objective stated in chapter 1, which is to analyse two-phase cooling of power electronics. This was to be done with an experimental setup which can determine the heat transfer performance. A suitable low-GWP fluid needed to be selected as coolant.

A setup was constructed to measure the heat transfer from a solid component to a boiling fluid. This setup is described in detail in chapter 4. The heat generating component in the setup is interchangeable to be able to test the influence of the surface material and boundary conditions. The setup is supplied with sensors to accurately measure the electrical heat dissipation and temperatures of the boiling surface and fluid. The energy balance is also determined by measuring the net efflux of energy. Controls are added to the setup to be able to regulate the fluid saturation temperature and boiling surface temperature.

The fluid FK-649 was chosen for this analysis. Besides the low global warming potential equal to 1, this fluid met with the safety concerns, is dielectric and the atmospheric boiling temperature is in the suitable range for power electronics cooling.

The literature review of chapters 2 and 3 resulted in several research questions to be answered to reach the goal of this study. These research questions were divided into fundamental research questions and technical research questions. A summarized answer to these questions is given here, based on the results and conclusions of chapters 5, 6 and 7.

Fundamental research questions

• Which correlation is most suited to predict heat transfer rates during pool boiling with FK-649? Can a correlation without fitting constants provide adequate enough predictions? Do the correlations adequately predict the variation of the boiling curve with varying saturation pressure?

From the literature study, a list of thirteen frequently used nucleate pool boiling heat transfer correlations was generated. Two of these correlations (the Rohsenow and Pioro correlation) contain constants which need to be fitted to the specific surface-fluid combination. All correlations use fluid properties, some also take into account properties of the boiling surface.

The predictions of pool boiling heat transfer are compared to the measurements done in this study for pool boiling of FK-649. As expected, the correlations which contain fitting constants agree best with the data. The Rohsenow correlation is the preferred correlation, as this correlation only requires one fitting constant (the Pioro correlation requires two) but has the same predictive performance as the Pioro correlation. The normalized mean absolute error when comparing the measurement data to the predictions with the Rohsenow correlation is equal to 7.5%, while the maximal deviation is 25.6%. These values indicate that the fitted Rohsenow correlation matches fairly well with the measurement data.

If no data is available to fit the constants, the Labuntsov correlation gave the best results. Remarkably, the normalized mean absolute error of this correlations was also equal to 7.5%, while no fitting parameters were used. The maximal deviation was equal to 29.1%, which is slightly larger than for the fitted Rohsenow correlation. The Labuntsov correlation is thus the best correlation for the prediction of heat transfer rates for pool boiling with FK-649 and works almost as well as the fitted correlations.

The effect of varying saturation temperature was also analysed separately. It was shown that both the Rohsenow and the Labuntsov correlation were able to predict the variation of the heat transfer rates accurately. Both these correlations can be used to predict heat transfer rates and design pool boiling cooling systems. Care has to be taken when using the Labuntsov correlation for a boiling surface with highly deviating boiling surface microgeometry and material from those tested in this study, as this correlation does not take into account any effects of the boiling surface on the heat transfer rate.

• Do the correlations properly describe the variation of the boiling curve with heat flux? Is it necessary to take into account the different regimes of nucleate boiling when using a correlation?

The boiling curve is the relation between surface superheat temperature and surface heat flux. In literature, three different nucleate boiling regimes are described: partial nucleate boiling at low heat fluxes, fully developed nucleate boiling at intermediate heat fluxes and partial dryout boiling near the critical heat flux. From different sources, it was also shown that different slopes of the boiling curve are apparent in the three regimes (lower slopes in the partial nucleate and partial dryout boiling regimes compared to the higher slope in the fully developed nucleate boiling regime).

None of the correlations take into account the differences between the three regimes. The relation between surface superheat temperature and surface heat flux is correlated by a power-law relation, where the exponent varies between the different correlations. This correlation is the first of its sort to be proposed for nucleate boiling heat transfer.

The experimental results clearly show the different behaviour of the boiling curve in the three nucleate boiling regimes. Although some correlations can give an overall good estimation of the boiling curve, the differences between the regimes are not captured. A good fit of the measurement data can be made by using three different power-law relations with different values for the exponent for the three regimes.

For better understanding of boiling heat transfer and the parameters influencing the boiling curve, the three regimes should be studied separately. More measurements on fluid and surfaces with different parameters should be conducted to determine if the subdivision in the three regimes is universally valid and if the exponents in the different regimes determined in this study can also be used for these other configurations. For design calculations, if some degree of inaccuracy is allowable, the correlations of Rohsenow and Labuntsov can still be used.

• Which correlation is most suited to predict the critical heat flux for FK-649? Do the correlations adequately predict the variation of the critical heat flux with varying saturation pressure?

The critical heat flux values measured in this study were predicted best by the Zuber and Lienhard-Dhir correlations. The normalized mean absolute error was equal to 6.1% and 6.6% respectively, while the maximal error was equal to 9.0% and 9.8% respectively. The critical heat flux correlations are thus a lot more accurate than the heat transfer correlations. Also the effects of saturation temperature were captured accurately by the correlations, which had not yet been described for FK-649 in scientific literature. The Zuber correlation slightly underpredicts the critical heat flux values, so this correlation will result in a small safety margin when used in design calculations.

Technical research questions

• Is the effect of liquid height on the boiling curve and critical heat flux negligible in the range of interest for power electronics cooling? Liquid heights ranging from 1 cm to 17 cm were tested. No differences were found in the boiling curves measured with different liquid heights. If the liquid level is decreased even further, at some point the heat transfer rates will be deteriorated as liquid rewetting of the surface might be hampered.

• Is there an effect of the non-uniform heat flux from the semiconductor dies on the boiling phenomenon? Will this affect the boiling curve and/or the critical heat flux?

Measurements were performed on two configurations. In the first batch of measurements, an electrical heater was used to provide a uniform heat flux to the boiling surface. In the second batch, a power module was used to generate heat and boiling occurred directly on the baseplate, which resulted in a unique dataset. As the heat generating components are smaller than the baseplate, a non-uniformity of the heat flux at the boiling surface could be possible.

The boiling phenomenon was analysed visually for both configurations. At the onset of nucleate boiling, nucleation sites appeared at random locations for both configurations. No correlation with the location of the IGBTs which generate the heat was perceived. Also for higher heat fluxes, bubble generation remained uniformly spread over the surface. At critical heat flux, vapour covered the entire surface and was not seen to originate from any specific location. Overall, visually no effect of heat flux non-uniformity was perceived.

The boiling curves of both configurations were compared. The difference in surface superheat between both configurations was smaller than 10%. These small differences could be attributed to the different surface roughness and surface material in both configurations.

The critical heat fluxes measured for the power module agreed well with the correlations which are based on a uniform heat flux. This indicates that critical heat flux does not occur at lower heat fluxes due to potentially higher heat fluxes directly below the IGBT dies.

Overall, it is concluded that due to spreading of the heat in the baseplate of the power module, a quasi-uniform heat flux is achieved at the boiling surface. Correlations for pool boiling heat transfer and critical heat flux for uniform heat fluxes can be directly applied.

• Is two-phase immersion a feasible method for cooling of power electronics for high-power density drivetrains?

Well, yes and no.

No, because the critical heat flux as measured in this study limits the maximal heat dissipation to lower values than for water-glycol cooling methods, while the complexity and cost of a two-phase system is higher. Heat transfer rates with FK-649 are in the same range as water-glycol baseplate cooling, which is lower

than water-glycol cooling of power modules with fins integrated in the baseplate. There is a risk of fouling of the boiling surface due to the deposition of dissolved contaminants. At this time, two-phase cooling of power modules with FK-649 is not competitive with water-glycol cooling.

Yes, because the potential of two-phase cooling is clear. There are many possibilities to significantly increase the critical heat flux, which include subcooling of the fluid, utilizing higher saturation temperatures and using flow boiling instead of pool boiling. Higher saturation temperatures and flow boiling will also result in higher heat transfer rates. Higher heat transfer rates will be achieved when using power modules with integrated fins. Fouling can be avoided by adding carbon filters and by identifying the sources of the contaminants. Using dielectric fluids as coolant can also enable direct cooling of the active parts of the power module, removing the thermal resistance of the electrical insulation layer. Finally, FK-649 does not feature as high heat transfer rates and critical heat flux values as some other fluids due to inferior thermal properties. An alternative fluid to test can be HFO-1336mzz(Z).

All these options require further research to evaluate their potential. If these researches prove successful, two-phase can be the future for highly effective cooling of power electronics (and by extension electric drivetrains). Until then, waterglycol cooled methods still have the advantage.

Lessons learned

Six years have passed between the initial idea for this experimental study and the writing of this thesis. During this time, I made numerous mistakes and learned just as many lessons from them. With this section, I would like to sum up some of the important, mostly practical lessons learned over the years, for anyone who may be interested.

- Leakages are everywhere in two-phase systems. Air leaking in the system can result in altered heat transfer rates, affect the desired setpoints and cause reactions with the fluid and materials inside. Leakage of the fluid from the setup can be costly at best and dangerous at worst. During design of the system, always choose the option which is thought to provide the best sealing. O-rings typically work better than gaskets. Many polymers such as polycarbonate, PMMA and POM are permeable by water vapour and are therefore best avoided.
- Transients in thermal systems can be very slow. Especially if the measurement setup is well insulated to avoid heat losses, the time constants can be very large. It can take several hours for a surface temperature to increase by 1 °C. This makes it very hard to judge if the system is operating in steady

state. Adding guard heaters to the system can help by both reducing the heat losses and by making the system respond faster. Using a PID-controller on the applied heat dissipation to achieve the desired setpoint can also greatly speed up measurements.

Accurate temperature measurements (with an uncertainty in the order of magnitude of 0.1 °C) require proper design and thorough attention. When using thermocouples, an accurate calibration is a necessity. When a solid temperature is to be measured, the contact between thermocouple and the solid is crucial. Ideally the thermocouple is soldered. If this is not possible, contact should be made with enough contact pressure, preferably with thermal paste applied to provide a good thermal contact. The thermocouple leads not in contact with the solid should be thermally insulated.

Recommendations for future work

This study is not the end point for research on two-phase cooling. In several research tracks, there are still open questions left to be answered.

There is still no consensus on the fundamental heat transfer mechanisms which govern the heat transfer during nucleate pool boiling. More measurements with temporally and spatially discretized temperature and heat flux measurements for widely varying reduced pressures and fluids can give more insights. Special attention should be given to the differences between the three nucleate boiling regimes.

With regard to FK-649, a further study with more variation of the boundary conditions is of interest. A larger range of saturation temperature and independently varying the boiling surface material and surface roughness will result in a better understanding of pool boiling heat transfer with FK-649. Appendix C gives an overview of adaptations required to the experimental setup to be able to test this wider range of boundary conditions.

The different behaviour of the boiling curve in the three nucleate boiling regimes needs further research. It should be studied if the behaviour is similar at widely different reduced pressures and for other fluids.

Ideally, a large database of pool boiling heat transfer measurements should be formed, containing measurement data from different setups and institutions worldwide. Measurement uncertainties should be included in this database, to be able to analyse the quality of the data. This database can result in an evaluation of the heat transfer correlations with regards to a significantly larger dataset with more variation in boundary conditions. Existing correlations can subsequently be improved or a new correlation can be developed.

For two-phase power electronics cooling, the next step would be to test pool boiling from a power module with integrated (pin) fins. Heat transfer rates and critical heat fluxes will be higher, but presumably these effects cannot be simply captured by scaling with the heat transfer area.

Another interesting track to investigate is direct cooling of the active components of the power module. This will require a different design, where the ceramic substrate can be traded for a cheaper material with lower thermal conductivity, as heat does not need to be transferred through this electrical insulation layer. Heat spreading will still be a necessity. Copper conductors carrying the electrical current can carry out a second task by also acting as heat spreaders.

FK-649 was found to have lower heat transfer rates and critical heat fluxes than R134a and HFE-7100. However, the latter two fluids have a GWP which is too high for application in mobile applications. HFO-1336mzz(Z) might be a good alternative and as such is interesting to test as coolant for two-phase power electronics cooling.

References

- Communication from the Commission to the European Parliament, the European Council, the Council, the European Economic and Social Committee and the Committee of The Regions - The European Green Deal, November 2019.
- [2] Communication from the Commission to the European Parliament, the Council, the European Economic and Social Committee and the Committee of the Regions - 'Fit for 55': Delivering the EU's 2030 Climate Target on the Way to Climate Neutrality, July 2021.
- [3] Proposal for a Regulation of the European Parliament and of the Council Amending Regulation (EU) 2019/631 as Regards Strengthening the CO2 Emission Performance Standards for New Passenger Cars and New Light Commercial Vehicles in Line with the Unions Increased Climate Ambition, July 2021.
- [4] F. Un-Noor, S. Padmanaban, L. Mihet-Popa, M. N. Mollah, and E. Hossain. A Comprehensive Study of Key Electric Vehicle (EV) Components, Technologies, Challenges, Impacts, and Future Direction of Development. *Energies*, 10(8):1217, August 2017. Number: 8 Publisher: Multidisciplinary Digital Publishing Institute.
- [5] Proposal for a Directive of the European Parliament and of the Council on Energy Efficiency (Recast), July 2021.
- [6] S. M. Lambert, B. C. Mecrow, R. Abebe, G. Vakil, and C. M. Johnson. Integrated Drives for Transport - A Review of the Enabling Thermal Management Technology. In 2015 IEEE Vehicle Power and Propulsion Conference (VPPC), pages 1–6, October 2015.
- [7] A. Grassmann, O. Geitner, W. Hable, C. Neugirg, A. Schwarz, F. Winter, and I. Yoo. Double Sided Cooled Module concept for High Power Density in HEV Applications. In *Proceedings of PCIM Europe 2015; International Exhibition and Conference for Power Electronics, Intelligent Motion, Renewable Energy and Energy Management*, pages 1–7, May 2015.

- [8] T. Sebastian. Temperature effects on torque production and efficiency of PM motors using NdFeB magnets. *IEEE Transactions on Industry Applications*, 31(2):353–357, March 1995. Conference Name: IEEE Transactions on Industry Applications.
- [9] Y. A. Cengel. *Heat Transfer A Practical Approach*. McGraw-Hill, 2nd edition, 2003.
- [10] K. Rajashekara. Present Status and Future Trends in Electric Vehicle Propulsion Technologies. *IEEE Journal of Emerging and Selected Topics in Power Electronics*, 1(1):3–10, March 2013. Conference Name: IEEE Journal of Emerging and Selected Topics in Power Electronics.
- [11] A. D. Kraus and A. Bar-Cohen. *Thermal analysis and control of electronic equipment*. January 1983.
- [12] ASHRAE. Standard 34-2019 Designation and Safety Classification of Refrigerants. Technical report, 2019.
- [13] Audi MediaCenter, 2021.
- [14] A. Emadi, Y. J. Lee, and K. Rajashekara. Power Electronics and Motor Drives in Electric, Hybrid Electric, and Plug-In Hybrid Electric Vehicles. *IEEE Transactions on Industrial Electronics*, 55(6):2237–2245, June 2008. Conference Name: IEEE Transactions on Industrial Electronics.
- [15] Z. Huang, S. Nategh, V. Lassila, M. Alakla, and J. Yuan. Direct oil cooling of traction motors in hybrid drives. In 2012 IEEE International Electric Vehicle Conference, pages 1–8, March 2012.
- [16] SingleOilCnL ICON Single oil-based active cooling and lubrication of electrical drivetrains, 2020.
- [17] S. S. Kang. Advanced Cooling for Power Electronics. In 2012 7th International Conference on Integrated Power Electronics Systems (CIPS), pages 1–8, March 2012.
- [18] R. Hu, T. Guo, X. Zhu, S. Liu, and X. Luo. A small flat-plate vapor chamber fabricated by copper powder sintering and diffusion bonding for cooling electronic packages. In 2013 IEEE 63rd Electronic Components and Technology Conference, pages 2280–2285, May 2013. ISSN: 2377-5726.
- [19] J. Nonneman and M. De Paepe. HIPERCOOL ICON Power electronics cooling models and validation. Technical report, Ghent University, 2020.

- [20] J. Nonneman, S. Schlimpert, I. T'Jollyn, P. Sergeant, and M. De Paepe. Experimental investigation of direct contact baseplate cooling for electric vehicle power electronics. In *Proceedings of the 2019 Eighteenth IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm 2019)*, pages 1220–1227, 2019. ISSN: 1936-3958, 2577-0799.
- [21] K. Olesen, R. Bredtmann, and R. Eisele. "ShowerPower" New Cooling Concept for Automotive Applications. Paris, 2006.
- [22] M. Reeves, J. Moreno, P. Beucher, S.-J. Loong, and D. B. M. Division. Investigation on the Impact on Thermal Performances of New Pin and Fin Geometries Applied to Liquid Cooling of Power Electronics. *International Journal of Microscale and Nanoscale Thermal and Fluid Transport Phenomena*, 3(3):7, 2012.
- [23] Evaluation Kit for Applications with Infineon HybridPACK1 Module, 2012.
- [24] P. Wang, P. McCluskey, and A. Bar-Cohen. Two-Phase Liquid Cooling for Thermal Management of IGBT Power Electronic Module. *Journal of Electronic Packaging*, 135(2), March 2013.
- [25] I. Aranzabal, I. M. de Alegra, N. Delmonte, P. Cova, and I. Kortabarria. Comparison of the Heat Transfer Capabilities of Conventional Single- and Two-Phase Cooling Systems for an Electric Vehicle IGBT Power Module. *IEEE Transactions on Power Electronics*, 34(5):4185–4194, May 2019. Conference Name: IEEE Transactions on Power Electronics.
- [26] G. Moreno. Two-Phase Cooling Technology for Power Electronics with Novel Coolants. Technical report, National Renewable Energy Laboratory (NREL), 2011.
- [27] C. M. Barnes and P. E. Tuma. Immersion Cooling of Power Electronics in Segregated Hydrofluoroether Liquids. pages 719–725. American Society of Mechanical Engineers Digital Collection, July 2009.
- [28] J. Nonneman, I. T'Jollyn, N. Clarie, S. Weckx, P. Sergeant, and M. De Paepe. Model-based comparison of thermo-hydraulic performance of various cooling methods for power electronics of electric vehicles. In *Proceedings of the 17th IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm 2018)*, pages 398– 409. IEEE, 2018. ISSN: 1936-3958.
- [29] Regulation (EC) No 1005/2009 of the European Parliament and of the Council of 16 September 2009 on substances that deplete the ozone layer (Text with EEA relevance), October 2009. Legislative Body: EP, CONSIL.

- [30] Amendment to the Montreal Protocol on Substances that Deplete the Ozone Layer, 2016.
- [31] Directive 2006/40/EC of the European Parliament and of the Council of 17 May 2006 relating to emissions from air conditioning systems in motor vehicles and amending Council Directive 70/156/EEC (Text with EEA relevance), June 2006. Code Number: 161.
- [32] Montreal Protocol on Substances that Deplete the Ozone Layer, 1987.
- [33] Kyoto Protocol to the United Nations Framework Convention on Climate Change. | UNFCCC, 1997.
- [34] Heat transfer applications using 3M Novec Engineered Fluids, 2018.
- [35] S. Nukiyama. The maximum and minimum values of the heat q transmitted from metal to boiling water under atmospheric pressure. *International Journal of Heat and Mass Transfer*, 27(7):959–970, July 1984.
- [36] S. Kandlikar and J. Chung. Boiling and Condensation. In C. T. Crowe, editor, *Multiphase flow handbook*, Mechanical engineering series, pages 131– 190. CRC : Taylor & Francis, Boca Raton, FL, 2006.
- [37] J. Kim. Review of nucleate pool boiling bubble heat transfer mechanisms. *International Journal of Multiphase Flow*, 35(12):1067–1076, December 2009.
- [38] G. F. Hewitt. Boiling. In W. M. Rohsenow, J. P. Hartnett, and Y. I. Cho, editors, *Handbook of heat transfer*, McGraw-Hill handbooks, pages 989– 1156. McGraw-Hill, New York, 3rd ed edition, 1998.
- [39] B. B. Mikic and W. M. Rohsenow. A New Correlation of Pool-Boiling Data Including the Effect of Heating Surface Characteristics. *Journal of Heat Transfer*, 91(2):245–250, May 1969.
- [40] R. L. Judd and K. S. Hwang. A Comprehensive Model for Nucleate Pool Boiling Heat Transfer Including Microlayer Evaporation. *Journal of Heat Transfer*, 98(4):623–629, November 1976.
- [41] D. D. Paul and S. I. Abdel-Khalik. A statistical analysis of saturated nucleate boiling along a heated wire. *International Journal of Heat and Mass Transfer*, 26(4):509–519, April 1983.
- [42] P. Stephan and J. Hammer. A new model for nucleate boiling heat transfer. *Heat and Mass Transfer*, 30(2):119–125, November 1994.

- [43] E. Wagner and P. Stephan. High-Resolution Measurements at Nucleate Boiling of Pure FC-84 and FC-3284 and Its Binary Mixtures. *Journal of Heat Transfer*, 131(12):121008, December 2009.
- [44] C. Gerardi, J. Buongiorno, L.-w. Hu, and T. McKrell. Study of bubble growth in water pool boiling through synchronized, infrared thermometry and high-speed video. *International Journal of Heat and Mass Transfer*, 53(19):4185–4192, September 2010.
- [45] V. K. Dhir. Boiling Heat Transfer. Annual review of fluid mechanics, 30(1):365–401, 1998.
- [46] A. Richenderfer, A. Kossolapov, J. H. Seong, G. Saccone, M. Bucci, and J. Buongiorno. Direct Measurement of Heat Flux Partitioning in Boiling Heat Transfer. American Society of Mechanical Engineers Digital Collection, October 2017.
- [47] G. Y. Su, C. Wang, L. Zhang, J. H. Seong, R. Kommajosyula, B. Phillips, and M. Bucci. Investigation of flow boiling heat transfer and boiling crisis on a rough surface using infrared thermometry. *International Journal of Heat and Mass Transfer*, 160:120134, October 2020.
- [48] K. Stephan. *Heat Transfer in Condensation and Boiling*. International Series in Heat and Mass Transfer. Springer Berlin Heidelberg, Berlin, Heidelberg, 1992.
- [49] C. E. Brennen. Fundamentals of multiphase flow. Cambridge University Press, Cambridge [England]; New York, 2005.
- [50] W. Fritz. Berechnung des Maximalvolumens von Dampfblasen. Psysikalische Zeitschrift, 36(11):379–384, June 1935.
- [51] S. S. Kutateladze and I. I. Gogonin. Growth Velocity and Departure Diameter of Vapor Bubbles in Saturated Liquids in free Convective Flow. *Heat transfer, mechanics of liquid, gas and plasma*, 17(4):792–797, 1979.
- [52] I. L. Pioro, W. Rohsenow, and S. S. Doerffer. Nucleate pool-boiling heat transfer. II: assessment of prediction methods. *International Journal of Heat and Mass Transfer*, 47(23):5045–5057, November 2004.
- [53] E. C. Forrest, L.-W. Hu, J. Buongiorno, and T. J. McKrell. Pool Boiling Heat Transfer Performance of a Dielectric Fluid With Low Global Warming Potential. *Heat Transfer Engineering*, 34(15):1262–1277, December 2013.
- [54] R. S. Bartle, K. Menon, and E. Walsh. Pool boiling of resin-impregnated motor windings geometry. *Applied Thermal Engineering*, 130:854–864, February 2018.

- [55] R. S. Bartle. Pool Boiling in Electrical Systems. Master's thesis, University of Oxford, Oxford, 2019.
- [56] J. G. Collier. Boiling and Evaporation. In E. U. Schlnder, editor, *Heat exchanger design handbook*, pages 602–714. VDI-Verl. [u.a.], Dsseldorf, 1983.
- [57] G. Kruzhilin. Free-convection transfer of heat from a horizontal plate and boiling liquid. *Doklady AN SSSR (Reports of the USSR Academy of Sciences)*, 58(8):1657–1660, 1947.
- [58] W. Rohsenow. A Method of Correlating Heat-Transfer Data for Surface Boiling Liquids. *Transactions of the ASME*, 74(6):969–976, 1952.
- [59] M. Jakob. Kondensation und Verdampfung. Zeitschrift des Vereins Deutscher Ingenieure, 76(48):1161–1170, 1932.
- [60] M. Jakob. Local Temperature Differences as Occurring in Evaporation, Condensation, and Catalytic Reaction. *Temperature, Its Measurement and Control in Science and Industry, p. &34, Reinhold, New York*, 1, 1941.
- [61] R. I. Vachon, G. H. Nix, and G. E. Tanger. Evaluation of Constants for the Rohsenow Pool-Boiling Correlation. *Journal of Heat Transfer*, 90(2):239– 246, May 1968.
- [62] I. Pioro. Experimental evaluation of constants for the Rohsenow pool boiling correlation. *International Journal of Heat and Mass Transfer*, 42(11):2003–2013, June 1999.
- [63] V. K. Dhir. Nucleate and transition boiling heat transfer under pool and external flow conditions. *International Journal of Heat and Fluid Flow*, 12(4):290–314, December 1991.
- [64] I. G. Malenkov. Detachment frequency as a function of size for vapor bubbles. *Journal of engineering physics*, 20(6):704–708, June 1971.
- [65] H. K. Forster and N. Zuber. Dynamics of vapor bubbles and boiling heat transfer. *AIChE Journal*, 1(4):531–535, December 1955.
- [66] M. T. Cichelli and C. F. Bonilla. Heat transfer to liquids boiling under pressure. *Transactions of the American Institute of Chemical Engineers*, 41, 1945. Publisher: Johns Hopkins University.
- [67] W.-T. Ji, C.-Y. Zhao, Y.-L. He, and W.-Q. Tao. Experimental validation of Cooper correlation at higher heat flux. *International Journal of Heat and Mass Transfer*, 90:1241–1243, November 2015.

- [68] S. S. Kutateladze and V. M. Borishanskii. *Handbook on Heat Transfer*. 1958.
- [69] V. M. Borishanskii. Accounting for the Effect of Pressure on Heat Transfer and Critical Heat Flux during Boiling Using the Theory of Thermodynamic Similarity. In S. S. Kutateladze, editor, *Problems of Heat Transfer and Hydraulics in Two-Phase Media*, pages 18–36. 1961.
- [70] V. M. Borishanskii. Correlation of the Effect of Pressure on the Critical Heat Flux and Heat Transfer Rates Using the Theory of Thermodynamic Similarity. In S. S. Kutateladze, editor, *Problems of Heat Transfer and Hydraulics of Two-Phase Media*, pages 16–37. Pergamon, January 1969.
- [71] I. L. Mostinski. Calculation of heat transfer and critical heat flux in boiling liquids based on the law of corresponding state. *Teploenergetika*, 10:66–71, 1963.
- [72] M. G. Cooper. Heat Flow Rates in Saturated Nucleate Pool Boiling-A Wide-Ranging Examination Using Reduced Properties. In J. P. Hartnett and T. F. Irvine, editors, *Advances in Heat Transfer*, volume 16, pages 157–239. Elsevier, January 1984.
- [73] I. Shekriladze and G. Ratiani. On the basic regularities of developed nucleate boiling heat transfer. *Bull. Acad. Sci. Georg. SSR*, 42(1):145–150, 1966.
- [74] I. G. Shekriladze. Developed boiling heat transfer. *International Journal of Heat and Mass Transfer*, 24(5):795–801, May 1981.
- [75] I. G. Shekriladze. MTD-MFC: Unified Framework for Investigation of Diversity of Boiling Heat Transfer Curves. page 16, 2009.
- [76] D. A. Labuntsov. Heat Transfer Problems with Nucleate Boiling of Liquids. *Teploenergetika*, 19(9):21–28, 1972.
- [77] D. Gorenflo, P. Sokol, and S. Caplanis. Pool boiling heat transfer from single plain tubes to various hydrocarbons. *International Journal of Refrigeration*, 13(5):286–292, September 1990.
- [78] D. Gorenflo and D. Kenning. Pool Boiling. In P. Stephan, editor, VDI Heat Atlas, pages 757–792. Springer Berlin Heidelberg, Berlin, Heidelberg, 2nd edition, 2010.
- [79] K. Stephan and M. Abdelsalam. Heat-transfer correlations for natural convection boiling. *International Journal of Heat and Mass Transfer*, 23(1):73–87, January 1980.

- [80] K. Nishikawa, Y. Fujita, H. Ohta, and S. Hitaka. Effects of System Pressure and Surface Roughness on Nucleate Boiling Heat Transfer. *Memoirs of the Faculty of Engineering, Kyushu University*, 42(2):95–123, June 1982. Publisher: Elsevier Limited.
- [81] S. S. Kutateladze. Natural Convection Boiling of Pure Saturated Fluids. In *Heat Transfer and Hydrodynamic Resistance: Handbook*, pages 197–200. 1990.
- [82] W. Leiner. Heat transfer by nucleate pool boilinggeneral correlation based on thermodynamic similarity. *International Journal of Heat and Mass Transfer*, 37(5):763–769, March 1994.
- [83] I. L. Pioro, W. Rohsenow, and S. S. Doerffer. Nucleate pool-boiling heat transfer. I: review of parametric effects of boiling surface. *International Journal of Heat and Mass Transfer*, 47(23):5033–5044, November 2004.
- [84] M. S. El-Genk and H. Bostanci. Saturation boiling of HFE-7100 from a copper surface, simulating a microelectronic chip. *International Journal of Heat and Mass Transfer*, 46(10):1841–1854, May 2003.
- [85] R. Raj, J. Kim, and J. McQuillen. Subcooled Pool Boiling in Variable Gravity Environments. *Journal of Heat Transfer*, 131(9), June 2009.
- [86] M.-G. Kang. Experimental investigation of tube length effect on nucleate pool boiling heat transfer. *Annals of Nuclear Energy*, 25(4):295–304, March 1998.
- [87] A. Gupta, R. Kumar, and V. Kumar. Nucleate pool boiling heat transfer over a bundle of vertical tubes. *International Communications in Heat and Mass Transfer*, 37(2):178–181, February 2010.
- [88] M.-G. Kang. Effect of Tube Inclination on Pool Boiling Heat Transfer. Journal of Heat Transfer, 122(1):188–192, October 1999.
- [89] K. Niskikawa, Y. Fujita, S. Uchida, and H. Ohta. Effect of surface configuration on nucleate boiling heat transfer. *International Journal of Heat and Mass Transfer*, 27(9):1559–1571, September 1984.
- [90] M. S. El-Genk and Z. Guo. Transient boiling from inclined and downwardfacing surfaces in a saturated pool. *International Journal of Refrigeration*, 16(6):414–422, January 1993.
- [91] H. M. Kurihara and J. E. Myers. The effects of superheat and surface roughness on boiling coefficients. *AIChE Journal*, 6(1):83–91, 1960. _eprint: https://aiche.onlinelibrary.wiley.com/doi/pdf/10.1002/aic.690060117.

- [92] K. Stephan. Mechanismus und Modellgesetz des Wrmebergangs bei der Blasenverdampfung. *Chemie Ingenieur Technik*, 35(11):775–784, 1963. eprint: https://onlinelibrary.wiley.com/doi/pdf/10.1002/cite.330351105.
- [93] G. Ribatski and J. M. S. Jabardo. Experimental study of nucleate boiling of halocarbon refrigerants on cylindrical surfaces. *International Journal of Heat and Mass Transfer*, 46(23):4439–4451, November 2003.
- [94] B. J. Jones, J. P. McHale, and S. V. Garimella. The Influence of Surface Roughness on Nucleate Pool Boiling Heat Transfer. *Journal of Heat Transfer*, 131(12), October 2009.
- [95] A. Luke. Active and Potential Bubble Nucleation Sites on Different Structured Heated Surfaces. *Chemical Engineering Research and Design*, 82(4):462–470, April 2004.
- [96] J. M. S. Jabardo, G. Ribatski, and E. Stelute. Roughness and surface material effects on nucleate boiling heat transfer from cylindrical surfaces to refrigerants R-134a and R-123. *Experimental Thermal and Fluid Science*, 33(4):579–590, April 2009.
- [97] U. Magrini and E. Nannei. On the Influence of the Thickness and Thermal Properties of Heating Walls on the Heat Transfer Coefficients in Nucleate Pool Boiling. *Journal of Heat Transfer*, 97(2):173–178, May 1975.
- [98] T. L. Chuck and J. E. Myers. The effect of heater plate thickness on boiling heat-transfer coefficients. *International Journal of Heat and Mass Transfer*, 21(2):187–191, February 1978.
- [99] X. Zhou and K. Bier. Influence of the heat conduction properties of the wall material and of the wall thickness on pool boiling heat transfer. In *Proceedings of EUROTHERM Seminar No. 48: Pool Boiling*, volume 2, pages 43–54, 1996.
- [100] R. N. Wenzel. Resistance of solid surfaces to wetting by water. *Industrial & Engineering Chemistry*, 28(8):988–994, 1936. Archive Location: world Publisher: American Chemical Society.
- [101] C. H. Wang and V. K. Dhir. Effect of Surface Wettability on Active Nucleation Site Density During Pool Boiling of Water on a Vertical Surface. *Journal of Heat Transfer*, 115(3):659–669, August 1993.
- [102] H. T. Phan, N. Caney, P. Marty, S. Colasson, and J. Gavillet. Surface wettability control by nanocoating: the effects on pool boiling heat transfer and nucleation mechanism. *International Journal of Heat and Mass Transfer*, 52(23-24):5459–5471, 2009. Publisher: Elsevier.

- [103] D. Y. Kwok and A. W. Neumann. Contact angle measurement and contact angle interpretation. *Advances in Colloid and Interface Science*, 81(3):167– 249, September 1999.
- [104] P. Di Marco, H. Ohta, and J. Kim. Gravity Effects on Pool Boiling Heat Transfer. In *Encyclopedia of Two-Phase Heat Transfer and Flow II*, pages 217–252. WORLD SCIENTIFIC, March 2015.
- [105] J. Straub. Boiling Heat Transfer and Bubble Dynamics in Microgravity. In J. P. Hartnett, T. F. Irvine, Y. I. Cho, and G. A. Greene, editors, *Advances in Heat Transfer*, volume 35, pages 57–172. Elsevier, January 2001.
- [106] C. D. Henry and J. Kim. A study of the effects of heater size, subcooling, and gravity level on pool boiling heat transfer. *International Journal of Heat* and Fluid Flow, 25(2):262–273, April 2004.
- [107] O. Kannengieser, C. Colin, and W. Bergez. Influence of gravity on pool boiling on a flat plate: Results of parabolic flights and ground experiments. *Experimental Thermal and Fluid Science*, 35(5):788–796, July 2011.
- [108] M. Y. Shukla and S. G. Kandlikar. Influence of Liquid Height on Bubble Coalescence, Vapor Venting, Liquid Return, and Heat Transfer in Pool Boiling. *International Journal of Heat and Mass Transfer*, 173:121261, July 2021.
- [109] P. E. Tuma. Design considerations relating to non-thermal aspects of passive 2-phase immersion cooling. In 2011 27th Annual IEEE Semiconductor Thermal Measurement and Management Symposium, pages 1–9, San Jose, CA, USA, March 2011. IEEE.
- [110] Y. Fukada, I. Haze, and M. Osakabe. The effect of fouling on nucleate pool boiling of small wires. *Heat TransferAsian Research*, 33(5):316–329, 2004. _eprint: https://onlinelibrary.wiley.com/doi/pdf/10.1002/htj.20016.
- [111] S. M. Peyghambarzadeh, A. Vatani, and M. Jamialahmadi. Application of asymptotic model for the prediction of fouling rate of calcium sulfate under subcooled flow boiling. *Applied Thermal Engineering*, 39:105–113, June 2012.
- [112] M. M. Sarafraz and F. Hormozi. Nucleate pool boiling heat transfer characteristics of dilute Al2O3ethyleneglycol nanofluids. *International Commu*nications in Heat and Mass Transfer, 58:96–104, November 2014.
- [113] D. Wang, L. Zhao, X. Nie, Y. Lu, and S. Deng. Experimental study on flow boiling characteristics of R-245fa in circular tube under non-uniform

heat flux. *International Journal of Heat and Mass Transfer*, 143:118570, November 2019.

- [114] G. Liang and I. Mudawar. Pool boiling critical heat flux (CHF) Part 1: Review of mechanisms, models, and correlations. *International Journal of Heat and Mass Transfer*, 117:1352–1367, February 2018.
- [115] G. Liang and I. Mudawar. Pool boiling critical heat flux (CHF) Part 2: Assessment of models and correlations. *International Journal of Heat and Mass Transfer*, 117:1368–1383, February 2018.
- [116] S. G. Kandlikar. A Theoretical Model to Predict Pool Boiling CHF Incorporating Effects of Contact Angle and Orientation. *Journal of Heat Transfer*, 123(6):1071–1079, April 2001.
- [117] W. M. Rohsenow and P. Griffith. Correlation of Maximum Heat Flux Data for Boiling of Saturated Liquids. *Chemical engineering progress symposium series*, 52:47–49, 1955.
- [118] N. Zuber. On the stability of boiling heat transfer. Trans. Am. Soc. Mech. Engrs., 80, 1958. Publisher: Univ. of California, Los Angeles.
- [119] Y. Haramura and Y. Katto. A new hydrodynamic model of critical heat flux, applicable widely to both pool and forced convection boiling on submerged bodies in saturated liquids. *International Journal of Heat and Mass Transfer*, 26(3):389–399, March 1983.
- [120] V. V. Yagov. Is a crisis in pool boiling actually a hydrodynamic phenomenon? *International Journal of Heat and Mass Transfer*, 73:265–273, June 2014.
- [121] I. Mudawar, A. H. Howard, and C. O. Gersey. An analytical model for nearsaturated pool boiling critical heat flux on vertical surfaces. *International Journal of Heat and Mass Transfer*, 40(10):2327–2339, July 1997.
- [122] S. Kutateladze. On the transition to film boiling under natural convection. *Kotloturbostroenie*, 3:10–12, 1948.
- [123] V. Borishanskii. An equation generalizing experimental data on the cessation of bubble boiling in a large volume of liquid. *Zh. Tekh. Fiz.*, 26:452– 456, 1956.
- [124] V. K. Dhir and J. H. Lienhard. Peak Pool Boiling Heat Flux in Viscous Liquids. *Journal of Heat Transfer*, 96(1):71–78, February 1974.

- [125] L. Wang, Y. Li, F. Zhang, F. Xie, and Y. Ma. Correlations for calculating heat transfer of hydrogen pool boiling. *International Journal of Hydrogen Energy*, 41(38):17118–17131, October 2016.
- [126] J. H. Lienhard and V. K. Dhir. Hydrodynamic Prediction of Peak Poolboiling Heat Fluxes from Finite Bodies. *Journal of Heat Transfer*, 95(2):152–158, May 1973.
- [127] J. Y. Chang and S. M. You. Heater Orientation Effects on Pool Boiling of Micro-Porous-Enhanced Surfaces in Saturated FC-72. *Journal of Heat Transfer*, 118(4):937–943, November 1996.
- [128] M. Arik and A. Bar-Cohen. Effusivity-based correlation of surface property effects in pool boiling CHF of dielectric liquids. *International Journal of Heat and Mass Transfer*, 46(20):3755–3764, September 2003.
- [129] S. M. Kwark, G. Moreno, R. Kumar, H. Moon, and S. M. You. Nanocoating characterization in pool boiling heat transfer of pure water. *International Journal of Heat and Mass Transfer*, 53(21):4579–4587, October 2010.
- [130] H. J. Ivey and D. J. Morris. On the relevance of the vapour-liquid exchange mechanism for sub-cooled boiling heat transfer at high pressure. Technical report, Reactor Development Division, Atomic Energy Establishment, Dorset, England, 1962. OCLC: 636077040.
- [131] Y. Elkassabgi and J. H. Lienhard. Influences of Subcooling on Burnout of Horizontal Cylindrical Heaters. *Journal of Heat Transfer*, 110(2):479–486, May 1988.
- [132] H. J. Ivey. Acceleration and the Critical Heat Flux in Pool Boiling Heat Transfer. *Proceedings of the Institution of Mechanical Engineers*, 177(1):15–42, June 1963. Publisher: IMECHE.
- [133] T. D. Bui and V. K. Dhir. Transition Boiling Heat Transfer on a Vertical Surface. *Journal of Heat Transfer*, 107(4):756–763, November 1985.
- [134] S. J. Kim, I. C. Bang, J. Buongiorno, and L. W. Hu. Surface wettability change during pool boiling of nanofluids and its effect on critical heat flux. *International Journal of Heat and Mass Transfer*, 50(19):4105–4116, September 2007.
- [135] R. Revellin, J. M. Quibn, J. Bonjour, and J. R. Thome. Effect of Local Hot Spots on the Maximum Dissipation Rates During Flow Boiling in a Microchannel. *IEEE Transactions on Components and Packaging Technologies*, 31(2):407–416, June 2008. Conference Name: IEEE Transactions on Components and Packaging Technologies.

- [136] 3M Novec 649 Engineered Fluid Technical Datasheet, 2009.
- [137] P. E. Tuma. Fluoroketone C2F5C(O)CF(CF3)2 as a Heat Transfer Fluid for Passive and Pumped 2-Phase Applications. In 2008 Twenty-fourth Annual IEEE Semiconductor Thermal Measurement and Management Symposium, pages 173–179, March 2008. ISSN: 1065-2221.
- [138] E. Forrest, J. Buongiorno, T. McKrell, and L.-W. Hu. Pool boiling performance of Novectm 649 engineered fluid. 2009.
- [139] J. L. Gess, S. H. Bhavnani, and R. W. Johnson. Experimental Investigation of a Direct Liquid Immersion Cooled Prototype for High Performance Electronic Systems. *IEEE Transactions on Components, Packaging and Manufacturing Technology*, 5(10):1451–1464, October 2015. Conference Name: IEEE Transactions on Components, Packaging and Manufacturing Technology.
- [140] R. Kaniowski, R. Pastuszko, and u. Nowakowski. Effect of geometrical parameters of open microchannel surfaces on pool boiling heat transfer. *EPJ Web of Conferences*, 143:02049, 2017. Publisher: EDP Sciences.
- [141] R. Bartle, K. Menon, and E. Walsh. Effect of Spatial Confinement and Liquid Level on Nucleate Boiling. In *Proceedings of the 15th UK Heat Transfer Conference (UKHTC2017)*, pages 166–167, 2017.
- [142] O. Ghaffari, F. Grenier, J.-F. Morissette, M. Bolduc, S. Jasmin, and J. Sylvestre. Pool Boiling Experiment of Dielectric Liquids and Numerical Study for Cooling a Microprocessor. In 2019 18th IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm), pages 540–545, May 2019. ISSN: 2577-0799.
- [143] Z. Cao, Z. Wu, and B. Sundn. Heat transfer prediction and critical heat flux mechanism for pool boiling of NOVEC-649 on microporous copper surfaces. *International Journal of Heat and Mass Transfer*, 141:818–834, October 2019.
- [144] J. Nonneman. Immersion boiling cooling of high-power density electronic components. Master's thesis, Ghent University, 2016.
- [145] J. P. Meyer and M. Everts. Single-phase mixed convection of developing and fully developed flow in smooth horizontal circular tubes in the laminar and transitional flow regimes. *International Journal of Heat and Mass Transfer*, 117:1251–1273, 2018. Publisher: Elsevier.
- [146] A. Bakker, R. D. LaRoche, and E. M. Marshall. Laminar flow in static mixers with helical elements. *The online CFM book*, 546, 2000.

- [147] I. H. Bell, J. Wronski, S. Quoilin, and V. Lemort. Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp. *Industrial & Engineering Chemistry Research*, 53(6):2498–2508, February 2014.
- [148] M. O. McLinden, R. A. Perkins, E. W. Lemmon, and T. J. Fortin. Thermodynamic Properties of 1, 1, 1, 2, 2, 4, 5, 5, 5-Nonafluoro-4-(trifluoromethyl)-3-pentanone: Vapor pressure,(p,, T) Behavior, and Speed of Sound Measurements, and an Equation of State. *Journal of Chemical & Engineering Data*, 60(12):3646–3659, 2015. Publisher: ACS Publications.
- [149] L. Hallemans, J. Zwysen, S. Ravyts, G. Van den Broeck, S. Schlimpert, G. Beckers, and J. Driesen. An Experimental Comparison of Thermal Modelling Techniques for IGBT Modules in Electrical Drivetrains. Nottingham, UK, 2020.
- [150] Infineon HybridPACK 1 Module FS400R07A1E3 Datasheet, 2018.
- [151] T. Kojima, Y. Yamada, Y. Nishibe, and K. Torii. Novel RC Compact Thermal Model of HV Inverter Module for Electro-Thermal Coupling Simulation. In 2007 Power Conversion Conference - Nagoya, pages 1025–1029, April 2007.
- [152] E. Lemmon, I. H. Bell, M. Huber, and M. McLinden. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 10.0, National Institute of Standards and Technology. *Standard Reference Data Program, Gaithersburg*, 2018.
- [153] C. Wen, X. Meng, M. L. Huber, and J. Wu. Measurement and Correlation of the Viscosity of 1,1,1,2,2,4,5,5,5-Nonafluoro-4-(trifluoromethyl)-3pentanone. *Journal of Chemical & Engineering Data*, 62(10):3603–3609, October 2017. Publisher: American Chemical Society.
- [154] R. A. Perkins, M. L. Huber, and M. J. Assael. Measurement and Correlation of the Thermal Conductivity of 1,1,1,2,2,4,5,5,5-Nonafluoro-4-(trifluoromethyl)-3-pentanone. *Journal of Chemical & Engineering Data*, 63(8):2783–2789, August 2018. Publisher: American Chemical Society.
- [155] J. Cui, S. Yan, S. Bi, and J. Wu. Saturated Liquid Dynamic Viscosity and Surface Tension of trans-1-Chloro-3,3,3-trifluoropropene and Dodecafluoro-2-methylpentan-3-one. *Journal of Chemical & Engineering Data*, 63(3):751–756, March 2018. Publisher: American Chemical Society.
- [156] J. Taylor. Introduction to error analysis, the study of uncertainties in physical measurements. 1997.



Publications as first author in peer-reviewed international journals

I. T'Jollyn, S. Lecompte, B. Vanslambrouck, and M. De Paepe, *Energetic and financial assessment of the implementation of an absorption heat pump in an industrial drying system*, Drying Technology, vol. 37, no. 15, pp. 1939 - 1953, 2019.

Publications as co-author in peer-reviewed international journals

M. C. Tran, I. T'Jollyn, J. Logie, M. van den Broek, M. De Paepe, and B. Vanslambrouck, *Performance of heat pump drying system integrated into a blood dryer*, Drying Technology, vol. 34, no. 14, pp. 1677 - 1689, 2016.

A. Sharif, B. Ameel, I. T'Jollyn, S. Lecompte, and M. De Paepe, *Comparative performance assessment of plate heat exchangers with triangular corrugation*, Applied Thermal Engineering, vol. 141, pp. 186 - 199, 2018.

W. Beyne, K. Couvreur, I. T'Jollyn, R. Tassenoy, S. Lecompte, and M. De Paepe, *A charging time energy fraction method for evaluating the performance of a latent thermal energy storage heat exchanger*, Applied Thermal Engineering, vol. 195, 2021.

Publications in proceedings of international conferences

I. T'Jollyn, K. De Kerpel, and M. De Paepe, *Determination of the ice concentration of ice slurries using impedance measurement*, in IIR International Congress of Refrigeration, Proceedings, Yokohama, Japan, 2015, pp. 1 - 8.

I. T'Jollyn, K. De Kerpel, and M. De Paepe, *Ice fraction estimation for ice slurries through impedance measurements,* in 11th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, Proceedings, Kruger National Park, South Africa, 2015, pp. 625 - 630.

M. C. Tran, I. T'Jollyn, J. Logie, M. van den Broek, M. De Paepe, and B. Vanslambrouck, *Case study of heat pump integration in a blood dryer*, in 8th Asia-Pacific Drying Conference, Proceedings, Kuala Lumpur, Malaysia, 2015, pp. 130 - 137.

I. T'Jollyn and M. De Paepe, *Simulations of absorption heat pump implementation in a drying system: an industrial case study,* presented at the 7th European Thermal-Sciences Conference, Krakow, Poland, 2016, pp. 1 - 8.

I. T'Jollyn and M. De Paepe, *Thermal analysis of absorption heat pump implementation in an industrial dryer*, in HEFAT 2016, Proceedings, Malaga, Spain, 2016, pp. 1427 - 1432.

I. T'Jollyn, T. Pujol, M. De Paepe, A. Massaguer, and L. Montoro, *Plate fin heat sink modelling and design considerations for thermoelectric generators*, Renewable Energy & Power Quality Journal, vol. 1, no. 15, pp. 551 - 556, 2017.

I. T'Jollyn, T. Pujol, A. Massaguer, L. Montoro, and M. De Paepe, *Experimental assessment of plate fin heat sinks for thermoelectric generators,* presented at the 9th World Conference on Experimental Heat Transfer, Fluid Mechanics and Thermodynamics, Iguazu Falls, Brazil, 2017.

I. T'Jollyn, B. Ameel, S. Devos, J. Bienstman, S. Schlimpert, and M. De Paepe, *Experimental study of design parameter influence on thermal and hydraulic performance of cold plates*, in Proceedings of the Sixteenth Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems ITherm 2017, Orlando, US, 2017, pp. 551 - 557.

I. T'Jollyn, J. Nonneman, and M. De Paepe, *On the design and advantages of a vapor chamber for battery packs and the discrepancy in vertical channel boiling correlations at small gap spacing*, in 13th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, Proceedings, Portoroz, Slovenia, 2017, p. 243-248.

I. T'Jollyn, J. Nonneman, W. Beyne, and M. De Paepe, *1D simulations of thermally buffered prismatic batteries through the application of PCMs*, in International Heat Transfer Conference 16, Beijing, 2018, pp. 4071 - 4078.

J. Nonneman, N. Clarie, I. T'Jollyn, S. Schlimpert, P. Sergeant, and M. De Paepe, *Advanced lumped parameter model for switched reluctance motors with high performance cooling*, in Proceedings of the 16th International Heat Transfer Conference, IHTC-16, Beijing, 2018, pp. 1833 - 1841.

J. Nonneman, I. T'Jollyn, N. Clarie, S. Weckx, P. Sergeant, and M. De Paepe, *Model-based comparison of thermo-hydraulic performance of various cooling methods for power electronics of electric vehicles*, in Proceedings of the 17th Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm 2018), San Diego, CA, 2018, pp. 398 - 409.

I. T'Jollyn, J. Nonneman, and M. De Paepe, *Pool boiling heat transfer measurements on a horizontal plate with a low global warming potential refrigerant,* in Proceedings of the 25th IIR International Congress of Refrigeration, Montral, Canada, 2019, pp. 1558 - 1564.

I. T'Jollyn, M. Callewaert, J. Nonneman, J. Van de Wauw, B. Ameel, and M. De Paepe, *Combined conduction and natural convection cooling of offshore power cables in porous sea soil*, in Proceedings of the 2019 Eighteenth IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm 2019), Las Vegas, NV, USA, 2019, pp. 1307 - 1312.

J. Nonneman, S. Schlimpert, I. T'Jollyn, P. Sergeant, and M. De Paepe, *Experimental investigation of direct contact baseplate cooling for electric vehicle power electronics*, in Proceedings of the 2019 Eighteenth IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm 2019), Las Vegas, 2019, pp. 1220 - 1227.

I. T'Jollyn, J. Nonneman, and M. De Paepe, *Thermohydraulic modelling of microchannel winding cooling for electric machines*, in 2020 International Con-

ference on Electrical Machines (ICEM), Gothenburg, Sweden (Electr Network), 2020, pp. 1004 - 1010.

I. T'Jollyn, J. Nonneman, L. Hallemans, S. Ravyts, J. Driesen, and M. De Paepe, *Experimental study of inverter base plate cooling with two-phase pool boiling*, in 33rd International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems (ECOS 2020), Osaka, Japan, 2020, pp. 83 - 91.

J. Nonneman, I. T'Jollyn, P. Sergeant, and M. De Paepe, *Quality assessment of a 2D FE based lumped parameter electric motor thermal model using 3D FE models*, in 2020 International Conference on Electrical Machines (ICEM), Gothenburg, Sweden (Electr Network), 2020, pp. 973 - 980.

J. Nonneman, S. Schlimpert, I. T'Jollyn, and M. De Paepe, *Experimental study* of a switched reluctance motor stator tooth with slot and end winding cooling, in 33rd International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems (ECOS 2020), Osaka, Japan, 2020, pp. 2425 - 2435.

J. Nonneman, S. Schlimpert, I. T'Jollyn, and M. De Paepe, *Modelling and validation of a switched reluctance motor stator tooth with direct coil cooling*, in 2020 19th IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm), Proceedings, Orlando, FL, 2020, pp. 306 -314.

B

Measurement uncertainty analysis

The analyses made in this section are based on the work of Taylor [156]. δ indicates the absolute error of the considered value, while δ_{rel} refers to the relative error on that value.

B.1 Boiling curve

The error on the surface superheat temperature is determined from the propagation of measurement uncertainties. From equation 4.1, the following uncertainty can be derived:

$$\delta(\Delta T_s) = \sqrt{\delta T_w^2 + \delta T_{sat}^2} \tag{B.1}$$

For the heater configuration, the temperature difference due to conduction in the aluminium plate is also taken into account (equation 4.2). The uncertainty of the calculated wall temperature can be determined as:

$$\delta T_w = \sqrt{\delta T_{TC}^2 + \left(\frac{d_{Al}}{k_{Al}}\right)^2 \delta \dot{q}^2 + \left(\frac{\dot{q}}{k_{Al}}\right)^2 \delta d_{Al}^2 + \left(\dot{q}\frac{d_{Al}}{k_{Al}^2}\right)^2 \delta k_{Al}^2} \qquad (B.2)$$

The uncertainty on the determination of the heat flux $\delta \dot{q}$ is given by equation B.4. The uncertainties of the plate thickness and thermal conductivity are respectively 0.1 mm and 10 W/mK. The temperature T_{TC} is the average of five thermocouple measurements. When considering the uncertainties on these measurements as independent from each other, the uncertainty on the average is $\sqrt{5}$ times lower than that of a single measurement. However, since the thermocouples are calibrated by the same procedure, it is not appropriate to consider the uncertainties to be independent. For example, a constant offset in the measurement of all values propagates one on one to the offset on the average of the values. Therefore, the uncertainty of the average is taken equal to the uncertainty of a single thermocouple measurement. This reasoning is not made for the analysis of the uncertainty of the temperature difference in equation B.1, as here a constant offset of both measurements would be negated by subtracting both values.

If the saturation temperature is directly measured, the uncertainty on this measurement is equal to the uncertainty of the thermocouple measurement. If it is derived from the pressure measurement, the uncertainty is derived as:

$$\delta T_{sat} = \frac{dT_{sat}}{dp_{sat}} \,\delta p_{sat} \approx \frac{dT_{sat}}{dp_{sat}} \,\delta p_m \tag{B.3}$$

As this determination of the saturation temperature is only used for low liquid levels, the correction for the static pressure is small compared to the measurement uncertainty and it is therefore assumed that this introduces no additional uncertainty in the saturation temperature. The derivative of the saturation temperature to the saturation pressure is determined from the vaporization curve of the refrigerant.

The uncertainty on the heat flux for the configuration with the heater is determined from equation 4.4:

$$\delta \dot{q} = \dot{q} \sqrt{\left(\frac{\delta V}{V}\right)^2 + \left(\frac{\delta I}{I}\right)^2} \tag{B.4}$$

If the power module configuration is used, the uncertainty on the voltage measurement is the combination of an absolute and a relative uncertainty, respectively δV and $\delta_{rel}V$, and the uncertainty on the heat flux becomes:

$$\delta \dot{q} = \dot{q} \sqrt{\left(\frac{\delta V}{V} + \delta_{rel} V\right)^2 + \left(\frac{\delta I}{I}\right)^2} \tag{B.5}$$

B.2 Energy balance

The uncertainty on the electrical power P_e is equal to:

$$\delta P_e = A \,\delta \dot{q} \tag{B.6}$$

The energy flow out of the reservoir by the coolant P_{ef} is equal to:

$$P_{ef} = \rho_o(T_o) \, \dot{V}_o \left[h_o(T_o) - h_i(T_i) \right] \tag{B.7}$$

The uncertainty on the determination of this energy flow is derived through the propagation of uncertainties. By neglecting the error on the fluid property determination, the following equation is obtained:

$$\delta P_{ef} = P_{ef} \sqrt{\left(\frac{d\rho}{dT} \frac{\delta T_o}{\rho_o(T_o)}\right)^2 + \delta_{rel} \dot{V}_o^2 + \frac{\left(\frac{dh}{dT} \delta T_o\right)^2 + \left(\frac{dh}{dT} \delta T_i\right)^2}{[h_o(T_o) - h_i(T_i)]^2}}$$
(B.8)

The relative uncertainty on the density due to the uncertainty on the temperature measurement (between brackets in the first term in the square root of equation B.8) is equal to 0.003% for a temperature of 20 °C. This is negligible compared to the relative error of the flow rate measurement (1.5%), thus it is not taken into account for the calculation. With the effect of pressure on the enthalpy neglected, the specific heat c_p can be introduced in equation B.8. With the additional assumption that the specific heat does not vary significantly between inlet and outlet condition, equation B.8 can be simplified to:

$$\delta P_{ef} = P_{ef} \sqrt{\delta_{rel} \dot{V}_o^2 + \frac{\left(c_p \,\delta T_o\right)^2 + \left(c_p \,\delta T_i\right)^2}{[c_p \,(T_o - T_i)]^2}} = P_{ef} \sqrt{\delta_{rel} \dot{V}_o^2 + \frac{2 \,\delta T_{TC}^2}{(T_o - T_i)^2}}$$
(B.9)

The inlet and outlet temperature are determined as the average of two thermocouple measurements. Also here, the same reasoning is used for the propagation of the uncertainty for the average value, taking the uncertainty of the average equal to the uncertainty of a single thermocouple measurement (δT_{TC}). For a maximal relative uncertainty of 3% on the energy flow out of the reservoir, the minimal temperature difference is 3.8 °C. The volumetric flow rate is therefore always set low enough such that the temperature difference is above this value. The minimal volumetric flow rate that can be measured is 0.2 L/min. The volumetric flow rate range as a function of the electrical power dissipation is shown in figure B.1. The maximal flow rate that should be able to be measured is about 4.2 L/min, which falls within the range of the flow rate meter.

The uncertainties on the estimation of the heat losses and on the deviation of the energy balance are determined from equations B.6 and B.9:

$$\delta \dot{Q}_{loss} = \sqrt{\delta P_e^2 + \delta P_{ef}^2} \tag{B.10}$$

$$\delta e = \sqrt{\left(\frac{\delta P_e}{P_e}\right)^2 + \left(\frac{P_e \,\delta P_{ef}}{P_{ef}^2}\right)^2} \tag{B.11}$$



Figure B.1: Flow rate range for which the uncertainty on the energy carried by the coolant is below 3% as a function of the electrical power.

Future setup design recommendations

One of the recommendations for future research is to gather more experimental data with FK-649 and other fluids. This dataset should cover a wide range of boundary conditions and parameters which are independently varied. The setup developed during this study, in its current state, is not able to handle this wide range. Some suggestions are made here for adaptations to the setup (or a new setup design) to be able to test this wide range.

Sealing

The setup features several small leaks. For FK-649, as the saturation pressure at ambient temperature is below atmospheric pressure, this results in air leaking into the setup. This make de-airing before each measurement set necessary, which is quite time-consuming. Next to this, leakage of water vapour can lead to reactions with FK-649 and fouling on the boiling surface, which will be covered in the next section. For fluids with pressures above atmospheric pressure, leakages will results in loss of fluid.

In future designs, leakages should be avoided even more than in the current design. As much of the reservoir should be welded as possible. The boiling surface assembly should be connected to the reservoir utilizing an O-ring. The windows in the reservoir should be removed if not needed, or made smaller than they are now and sealed with O-rings. A glass material should be used, as most polymeric materials are permeable to water vapour. If materials other than metals are used elsewhere in the reservoir, the permeability of these materials should be assessed.

• Fouling

Non-metallic materials in contact with FK-649 should be avoided. To avoid water coming in contact with the fluid, a desiccant (silica gel packets) can be used to absorb any water leaking into the reservoir. An active carbon filter can be added to remove any contaminants in the fluid. If other fluids are used in the setup, compatibility of these fluids with the dessicant and filter should be checked.

• Operating range

The setup is currently designed to operate at saturation temperature between 20 °C and 50 °C and pressure ranging from 0 bar to 1 bar. In future designs, higher temperatures and pressures should be possible. With the redesigned sealing, the reservoir should be leak-tested at higher pressures before filling with the fluid. All materials of the setup should be able to withstand higher temperatures. Mostly the sealings (O-rings), any polymeric materials and insulating material should be checked for their maximal operating temperature. Another chiller might be necessary, which can provide a water-glycol flow at higher temperatures.

Boiling surfaces

For tests on the effect of the boiling surface material and microgeometry, the heater configuration is preferred. A design with cartridge heater might be advantageous compared to the current design, as it will be more compact, have lower heat losses and lower temperature at the heater.

The boiling surface should be made from a certain material (of which the thermal properties are known or measured) and with a desired surface roughness (which should be measured). The boiling curve should subsequently be measured, potentially at different saturation temperatures and with different fluids (see next bullet point). Afterwards, the boiling surface should be processed with a surface finishing technique to obtain a different surface roughness. New boiling curves are subsequently measured with the same material but different surface roughness. The entire process should then be repeated for a different boiling surface material. Preferably, the same surface roughness is aimed for when processing the boiling surface of the new material.

Next to practical considerations, the boiling surface materials should be chosen to represent a range of thermal properties. At least three different materials should be tested, with independently varying thermal conductivity and thermal effusivity, such that effects of both properties can be separated.

• Other fluids
Other fluids than FK-649 should be tested. This entails different pressure levels in the reservoir for the same temperature range, for which the setup has to be designed. Compatibility of the reservoir materials with the other fluids has to be checked.



D.1 3M Novec 649 (FK-649)



3M[™] Novec[™] 649 Engineered Fluid

Introduction

 $3M^{\circ\circ}$ Novec^{\$\vee\$} 649 Engineered Fluid is a clear, colorless and low odor fluid, one in a line of 3M products designed as replacements for ozone depleting substances (ODSs) and compounds with high global warming potentials (GWPs) such as sulfur hexafluoride (SF₆) and hydrofluorocarbons (HFCs), such as HFC-134a and HFC 245fa.

3M Novec 649 Engineeree Fluid is an advanced heat transfer fluid, balancing customer needs for physical, thermal and electrical properties, with desirable environmental properties.

Typical Applications

Novec 649 fluid is an effective heat transfer fluid with a boiling point of 49°C. Novec 649 fluid is useful in heat transfer particularly where non-flammability or environmental factors are a consideration. Examples of systems which benefit from use Novec 649 fluid include:

- Organic Rankine Cycle
 Diesel Engines
 Generators
- Electronics Cooling (Single or Dual Phase)
 Power Electronics such as IGBTs or inverters
- Transformers and other equipment (SF6 replacement)
- Computer/Data Center Cooling
- Geothermal Applications - Solar Applications

Properties Description

Composition of 3M [™] Novec [™] 649 Fluid	
Dodecafluoro-2-methylpentan-3-one	99.0 mole %, minimum
Chemical Formula	CF ₂ CF ₂ C(0)CF(CF ₂) ₂

Typical Physical Properties

	3M™ Novec™ 649 Fluid
Boiling Point(°C)	49
Pour Point (°C)	-108
Molecular Weight (g/mol)	316
Critical Temperature (°C)	169
Critical Pressure (MPa)	1.88
Vapor Pressure (kPa)	40
Heat of Vaporization (kJ/kg)	88
Liquid Density (kg/m ³)	1600
Coefficient of Expansion (K ⁻¹)	0.0018
Kinematic Viscosity (cSt)	0.40
Absolute Viscosity (cP)	0.64
Specific Heat (J/kg-K)	1103
Thermal Conductivity (W/m-K)	0.059
Surface Tension (mN/m)	10.8
Solubility of Water in Fluid (ppm by wt)	20
Dielectric Strength, 0.1" gap (kV)	>40
Dielectric Constant @ 1kHz	1.8
Volume Resistivity (Ohm-cm)	1012
Global Warming Potential (GWP)	1



2

3M[™] Novec[™] 649 Engineered Fluid

Thermophysical Properties



Liquid Thermal Conductivity (Wm-K) = $0.063403 - 0.000188 T(^{\circ}C)$



Liquid Specific Heat $(J/kg-K) = 1091.9 + 0.3419 T(^{\circ}C) + 0.0039T^{2}$

Vapor Pressure ln(P[Pa]) = -3545.3/T(K) + 22.492



Novec 649 Fluid Viscosity vs. Temperature

Features

The environmental profile, margin of safety, low viscosity, high molecular weight, low pour point and heat transfer performance of $3M^{-1}$ Novec⁻⁻ 649 Engineered Fluid make it an ideal candidate for a variety of heat transfer applications.

Novec 649 fluid is compatible with a wide range of materials of construction and requires no special piping or handling systems, and is very stable in storage. Its high dielectric constant makes it safe for direct contact in most electronics/computing applications.

Physical Properties

Properties	3M™ Novec™ 649 Fluid		HFC-245fa	HFC-134a
Ozone Depletion Potential (ODP)1	0.0	0.0	0.0	0.0
Global Warming Potential ²	1	23900	1030	1,300
Atmospheric Lifetime (years)	0.014	3200	7.6	140

¹ World Meteorological Organization (WMO) 1998, Model-Derived Method.
² Intergovernmental Panel on Climate Change (IPCC) 2007 Method, 100 Year ITH.

Stability

Novec 649 fluid should be used in a sealed system to prevent interaction with water. Fluoroketones like Novec 649 fluid, hough reactive with liquid water (ie. a separate water phase), are remarkably stable in its absence to over 300°C. 3M's applications engineers are available to discuss system design and trade-offs for Novec 649 fluid vs. alternative heat transfer fluids or solutions.

Environmental, Health and Safety

Studies by a third party laboratory (Massachusetts Institute of Technology) have shown that Novec 649 fluid has an estimated atmospheric lifetime of five days due to photolysis in sunlight.¹

The potential for Novec 649 fluid to impact the radiative balance in the atmosphere (i.e., climate change) is limited by this very short atmospheric lifetime. [Using a measured IR cross-section and the method of Pinnock et.al., the instantaneous radiative forcing for Novec 649 fluid is calculated to be 0.50 Wm-2ppbV-11, This radiative forcing and a 5-day atmospheric lifetime result in a GWP value of about 1 [using the WMO 1999 method over a 100-year integration time horizon].

The photolysis of Novec 649 fluid is expected to rapidly produce fluorinated alkyl radicals similar to those produced by other fluorochemicals. Studies of the atmospheric chemistry of these radical species and their degradation products have concluded that they have no impact on stratospheric ozone. This combined with its very short atmospheric lifetime, leads to the conclusion that Novec 649 fluid has an ozone depletion of zero.¹

Before using this product, please read the current product Material Safety Data Sheet (available online or through your 3M sales or technical service representative) and the precautions and directions for use on the product package. Follow all applicable precautions and directions for use.

N. Taniguchi, J. Wallington, M.D. Hurley, A.G. Guschin, L.T. Molina and M.J. Molina, Atmospheric Chemistry of C₂F₂C(0)CF(CF₂), Photolysis and Reaction with Cl Atoms, OH Radicals and Ozone. J Phys Chem A, 2003, 107, 2674 – 2679.

Toxicity Profile

3M carefully characterizes the toxicity of new materials early in the product development process. These early studies and the subsequent studies conducted by independent laboratories indicate that Novec 649 fluid is very low in both acute and repeat dose toxicity. The No Observed Adverse Effect Level (NOAEL) for all endpoints of acute toxicity is 10% (100,000 pmV) based on a cardiac sensitization study and a 4-hour acute inhalation study. The 8-hour time weighed average (TWA) exposure guideline for Novec 649 fluid is 150 ppmV. On this basis, foreseeable use under normal operating conditions results in a large margin of safety between anticipated exposure and the exposure guideline.

3M[™] Novec[™] 649 Engineered Fluid

Recycle and Disposal Options

Used Fluid Return Program

3M offers a program for free pickup and return of used 3M specialty fluids in the U.S. A pre-negotiated handling agreement between users and our authorized service provider offers users broad protection against future liability for used 3M product. The fluid return program is covered by independent third-party financial and environmental audits of treatment, storage and disposal facilities. Necessary documentation is provided. A minimum of 30 gallons of used 3M specialty fluid is required for participation in this free program.

For additional information on the 3M Used Fluid Return Program, contact your local 3M representative or call 3M Customer Service at 800.810.8513.

Resources

3M™ Novec™ Engineered Fluids are supported by global sales, technical and customer service resources, shi brocc Engineered nataries in the U.S. Europe, Japan, Latin America and Southeast Sais. Users benefit from 3M's broad technology base and continuing attention to product development, performance, safety and environmental issues. For additional technical information on 3M^m Novec^m 649 Engineered Fluid in the United States or for the name of a local authorized distributor, call 3M Electronics Markets Materials Division: 800 810 8513.

The 3M [™] Novec [™]	The Novec brand is the hallmark for a variety of patented 3M products. Although each has its own unique formula and performance properties, all
Brand Family	Novec products are designed in common to address the need for safe, effective, sustainable solutions in industry-specific applications. These include
1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1	precision and electronics cleaning, heat transfer, fire protection, lubricant deposition and several specialty chemical applications.

31 Wovec" Engineered Fluids • 31 Novec" Aerosol Cleaners • 31 Wovec" 1230 Fire Protection Fluid • 31 Novec" Electronic Coatings • 31 Novec" Electronic Surfactants

United States	China	Europe	Japan	Korea
3M Electronics Markets	3M China Ltd.	3M Belgium N.V.	Sumitomo 3M Limited	3M Korea Limited
Materials Division	86 21 6275 3535	32 3 250 7521	813 3709 8250	82 2 3771 4114
800 810 8513				

Product lise: All statements, technical information and recommendations contained in this document are based on tests or experience that 3M believes are reliable. However, **Product Use:** An statistication, technical information and recommensational contained and particular application.
Introduct Use: An statistication, technical information and recommensation and product in a particular application, including conditions under which the product is used and the time and environmental conditions in which the product is supercluded particular application.
Include a particular applications and the application and recommensation and recommensation and recommensation and recommensation and recommensation and recommensation application.
Include a particular purposes a state of the product is application.
Include a particular purposes and state of the true user's method or application.

Warranty and Limited Remedy: Unless stated otherwise in SM's product literature, packaging inserts or product packaging for individual products, 3M warrants that each 3M product meets the applicable specifications at the time 3M sings the product. Individual products may have additional or different warrantes as stated on product literature, package inserts or product packages. 3M MARES NO THEW WARRANTIES, EXPRESS OR IMPLEED, INCLDING BUT NOT LIMITED TO, ANY IMPLEED WARRANTIES OF MERCINATIABILITY OR FITNESS FOR A PARTICULAR PURPOSE OR ANY IMPLEED WARRANTY SAINS OUT OF A COURSE OF DEALING, CUSTOM OR USAGE OF TRADE. User is responsible for determining whether the 3M product is fit for a particular purpose and suitable for user's application. If the 3M product is defective within the warranty period, your exclusive remedy and 3M's and seller's sole obligation will be, at 3M's option, to replace the product or refund the purchase price.

Limitation Of Liability: Except where prohibited b.incidental, or consequential regardless of the legal theory asserted, including warranty, contract, negligence or strict liability.

3M

Electronics Markets Materials Division 3M Center, Building 224-3N-11 St. Paul, MN 55144-1000 www.3M.com/novec rec 1-800-810-8513

Please recycle. Printed in USA. Issued: 9/09 © 3M 2009. All rights reserved. 7009HB 60-5002-0392-6

3M and Novec are trademarks of 3M. ies and affiliatee ed under license by 3M subsi

D.2 NESLAB Merlin Series Recirculating Chillers

The NESLAB Merlin Series recirculating chillers provide a wide range of cooling capacity all in a highly reliable, spacesaving package for your critical application needs.

NESLAB Merlin Series Recirculating Chillers

Delivers a continuous supply of cooling capacity up to 5045 watts

0



Typical applications for the Merlin Series: • Lasers • Analytical instrumentation • Printing equipment • Electron microscopes • Semiconductor manufacturing • Medical imaging equipment • Reaction vessels • Condensers



Feature-Packed Compact Chillers

The NESLAB Merlin Series of recirculating chilles provide precise temperature control in the most demanding cooling applications. Engineered for process-fluid cooling, these feature-packed compact chillers include easy-to-use controls, effortless maintenance, and quiet operation. Plus adjustable high and que temperature safeties, an audible alarm, and temperature stability of up to 40.1°C.

The environmentally-friendly CFC-free air-cooled refrigeration system provides reliable and powerful cooling. With a variety of pumps to choose from, you can configure a chiller that is ideal for your specific application requirements. While each Merlin recirculating chiller has many standard features, a variety of options and accessories are available to meet specific application needs.

Backed By a Two-Year Warranty

When purchased new from Thermo Electron Corporation, each Merlin unit is warrantied for 24 months. For you, it means peace of mind. For us, it means we believe in our products.



Analyze + Detect + Measure + Control™



System Specifications

Thermo Electron Corporation has a well-established reputation in temperature control through its NESLAB and HAAKE product lines. Theme because of a provide the solution of the solutions that help you analyze, detect, measure, and control your application with increasingly advanced precision.

NESLAB Merlin Series Specifications

	M 25	M 33	M 75	M 100	M 150
Cooling capacity @ 20°C					
60 Hz	810 watte	1250 watte	2/10 watte	3500 watte	50/15 watte
00112	2765 BTU	4266 RTH	8225 BTH	11946 BTH	17219 BTH
	697 Kcal	1075 Kcal	2073 Kcal	3010 Kcal	4339 Kcal
50 Hz	725 watts	1125 watts	2180 watts	3500 watts	4540 watts
00112	2474 BTU	3840 BTU	7440 BTU	11946 BTU	15495 BTU
	624 Kcal	968 Kcal	1875 Kcal	3010 Kcal	3905 Kcal
Refrigerant	B134A	B134A	B134A	R404A	B404A
Temperature range					
standard	+5°C to 35°C	+5°C to 35°C	+5°C to 35°C	+5°C to 35°C	+5°C to 35°C
low temperature option	-15°C to 35°C	-15°C to 35°C	-15°C to 35°C	-15°C to 35°C	-15°C to 35°C
Ambient temperature range					
c · ·	13° to 35°	13° to 35°	13° to 35°	13° to 35°	13° to 35°
F	55° to 95°	55° to 95°	55° to 95°	55° to 95°	55° to 95°
Temperature stability	+/- 0.1°C	+/- 0.1°C	+/- 0.15°C	+/- 0.15°C	+/- 0.15°C
Pumps					
MD 30 – Mag Drive*	yes	yes	n/a	n/a	n/a
PD 1 – Positive Displacement	yes	yes	yes	n/a	n/a
PD 2 – Positive Displacement	n/a	yes	yes	yes	yes
CP 55 – Centrifugal Pump	n/a	n/a	n/a	yes**	yes**
Reservoir volume					
gallon	0.5	0.5	0.5	0.5	0.5
liter	1.8	1.8	1.8	1.8	1.8
Unit dimensions					
H x W x D in	23.50 x 12.62 x 20.85	23.50 x 12.62 x 20.875	26.125 x 16.25 x 24	30.375 x 21.25 x 29.25	30.375 x 21.25 x 29.25
H x W x D cm	59.7 x 32.1 x 53.0	59.7 x 32.1 x 53.0	66.4 x 41.3 x 61.0	77.2 x 54.0 x 74.3	77.2 x 54.0 x 74.3
Power requirements					
option 1	115V, 60 Hz, 10 amps	115V, 60 Hz, 13 amps	200-230V, 60 Hz, 9 amps	208-230V, 60 Hz, 13 amps	208-230V, 60 Hz, 15 amps
	100V, 50 Hz, 10 amps	100V, 50 Hz, 13 amps	200V, 50 Hz, 9 amps	200V, 50 Hz, 13 amps	200V, 50 Hz, 15 amps
option 2	230V, 50 Hz, 5 amps	200-230V, 60 Hz, 7 amps	230, 50 Hz, 9 amps	230V, 50 Hz, 13 amps	230V, 50 Hz, 16 amps
		200V, 50 Hz, 7 amps			
option 3	n/a	230, 50 Hz, 7 amps	n/a	400V, 50 Hz, 3 ph 8 amps	400V, 50 Hz, 3 ph 8 amps
Unit weight					
lb	118	120	176	254	254
kg	54	54	80	115	115
Compliance					
60 Hz	UL/cUL, CE	UL/cUL, CE	UL/cUL, CE	UL/cUL, CE	UL/cUL, CE
50 Hz	CE	CE	UL/cUL, CE	CE	CE

Specification listed for standard units circulating water at 20°C fluid temperature and 20°C ambient. Other fluids, fluid temperatures, or ambient tempera-tures will affect performance. Cooling capacity and amperage ratings based on units with PD 1 pumps (PD 2 with M 100 and M 150). Other pumps will affect performance. Specifications are subject to change. *Available in 60 k2 configuration only. **M 100 and M 150 units with CP 55 pump have a height of 36.4", and unit weight of 275 lbs.

Thermo

Standard Features

Feature	Benefit
Variety of pumps	Meets your specific application needs
Refrigeration system	Provides CFC-free refrigeration system for precise temperature control and optimum stability
Hot gas bypass	Extends chiller life
Reservoir isolation valves	Eliminates reservoir overflow when pumping vertically
Fluid reservoir	Enables you to view the liquid level and fluid condition at a glance
Removable front grille	Allows quick release of front grille for routine cleaning
Integrated fluid pressure and flow adjustment	Meets specific application needs
Easy access reservoir drain	Allows fast and easy access when changing fluids
Pressure gauge	Provides fluid system diagnostics at a glance
Digital temperature controller	Provides intuitive controller with an easy-to-view digital display for simple operation
High/low temperature safeties	Ensures your application maintains critical operating parameters
Compact design	Maximizes valuable floor space
Two-year warranty	Provides you with peace of mind

Options

Feature	Benefit
Communications package:	
 RS-232 or RS-485 	Allows for remote operation, monitoring, and data logging
Remote start/stop	Allows the unit to be remotely turned on and off for convenient operation
Status relay	Allows for remote monitoring
 Remote sensor port 	Allows remote temperature control of the application
Low level/low flow safety package	Ensures fluid level and fluid flow safety to ensure continuous operation

Accessories

Feature	Benefit
NESLAB NEScom software	Allows you to easily program and automate your temperature control process from a PC system
Plumbing kits	Provides tubing, insulation, and connectors for easy installation
External temperature gauge	Allows you to monitor system operating temperature any point in-line between the
	chiller and the application
External pressure gauge	Allows you to monitor system pressure any point in-line between the chiller and the application
Air filter kits	Protects your chiller from dusty environments
Deionized water package	Maintains a water resistivity level between 1 and 3 megohm/cm2 for cooling application
	requiring ultra pure water
Fluid particulate filters	Maintains particulate-free operating fluid. Available with 5, 10, 25 or 40 micron filters
Ethylene glycol	Allows circulation to temperatures down to -20°C in a 50/50 mixture with water

A = M 150, 60 Hz B = M 150, 50 Hz C = M 100, 50 Hz, 60 Hz

Performance Curves

Cooling Capacity









Thermo

France

16 Avenue du Québec - Silic 76 91963 Courtaboeuf Cedex Tel. +33 (0) 1 60 92 48 00 info.tc.fr@thermo.com

	United Kingdom
65	Unit 5, The Ringway Centr
	Basingstoke, Hampshire
	RG21 6YH
	Tel. +44 (0) 870 609 9254
	info.tc.uk@thermo.com

Benelu

USA 25 Nimble Hill Rd.

Newington, NH 03801 Tel. 603-436-9444 info.tc.us@thermo.com



Benelux	International/Germany
Takkebijsters	1 Dieselstr. 4
4817 BL Breda	76227 Karlsruhe
Tel. +31 (0) 76 5 87 98 88	Tel. +49 (0) 721 4 09 44 44
info.tc.nl@thermo.com	info.tc.de@thermo.com
	www.thermo.com/to



25 30 35

M 100, M 150

6000

5000

1000

5

Removal (Watts) 4000 3000 Heat 2000

D.3 GEMS 2200 Series Pressure Transducers



Shock

Output

PRESSURE TRANSDUCERS





D.4 Infineon HybridPACK 1 Module FS400R07A1E3



Final Data Sheet

V3.4, 2018-08-09

Automotive High Power



1 Features / Description

HybridPACK™ 1 module with Trench/Fieldstop IGBT3 and Emitter Controlled 3 diode and NTC



Typical Applications

- Automotive Applications
- · Hybrid Electrical Vehicles (H)EV
- Commercial Agriculture Vehicles
- Motor Drives
- Optimized for automotive applications with DC link voltages up to 420 V

Electrical Features

- · Low Switching Losses
- Low V_{CEsat}
- T_{vj op} = 150°C
- · V_{CEsat} with positive Temperature Coefficient

Mechanical Features

- · 2.5kV AC 1min Insulation
- · Al₂O₃ Substrate with Low Thermal Resistance
- High mechanical robustness
- Integrated NTC temperature sensor
- Copper Base Plate
- · RoHS compliant



 $V_{CES} = 650V$ $I_{C nom} = 400A$

Description

Infineon[®]s HybridPACK[™] 1 is an automotive qualified power module designed for electric vehicle applications for a power range up to 20–30kW. Designed for a 150°C junction operation temperature, the module accommodates a 3-phase Six-Pack configuration of Trench-Field-Stop IGBT3 and matching emitter controlled diodes.

The HybridPACK[™] 1 power module is built on Infineon's long time experience in the development of IGBT power modules, intense research efforts of new material combinations and assembly technologies. HybridPACK[™] 1 is suitable for air or liquid cooling. The copper base plate combined with high-performance ceramic substrate and Infineon's enhanced wire-bonding process provides unparalleled thermal and power cycling capability and highest reliability for mild hybrid inverter or generator applications. For a compact design the driver stage PCB can easily be soldered on top of the module. All power connections are realized with screw terminals.

Product Name	Ordering Code
FS400R07A1E3	SP000663446



2 IGBT,Inverter

Parameter	Conditions		Symbol		Value		Unit
Collector-emitter voltage	T _{id} = 25°C		Vces		650		V
Continuous DC collector current	$T_{C} = 65^{\circ}C, T_{vj max} = 175^{\circ}C$ $T_{C} = 25^{\circ}C, T_{vj max} = 175^{\circ}C$		Ic nom		400 500		A
Repetitive peak collector current	t _P = 1 ms		ICRM		800		А
Total power dissipation	T _C = 25°C, T _{vj max} = 175°C		Ptot		1250		w
Gate-emitter peak voltage			V _{GES}		+/-20		V
2.2 Characteristic Values			^ 	min	tvn	may	
Collector-emitter saturation voltage		$T_{vj} = 25^{\circ}C$ $T_{vj} = 125^{\circ}C$ $T_{vj} = 150^{\circ}C$	VCE sat		1.45 1.60 1.70	1.90	v
Gate threshold voltage	I _C = 6.40 mA, V _{CE} = V _{GE}	T _{vj} = 25°C	V _{GEth}	4.90	5.80	6.50	V
Gate charge	V _{GE} = -15 V 15 V		Q _G		4.30		μC
Internal gate resistor		T _{vj} = 25°C	R _{Gint}		1.0		Ω
Input capacitance	f = 1 MHz, V _{CE} = 25 V, V _{GE} = 0 V	T _{vj} = 25°C	Cies		26.0		nF
Reverse transfer capacitance	f = 1 MHz, V _{CE} = 25 V, V _{GE} = 0 V	T _{vj} = 25°C	Cres		0.76		nF
Collector-emitter cut-off current	V _{CE} = 650 V, V _{GE} = 0 V	T _{vj} = 25°C	ICES			1.0	mA
Gate-emitter leakage current	Vce = 0 V, Vge = 20 V	T _{vj} = 25°C	IGES			400	nA
Turn-on delay time, inductive load		$\begin{array}{l} T_{vj} = 25^{\circ}C \\ T_{vj} = 125^{\circ}C \\ T_{vj} = 150^{\circ}C \end{array}$	t _{d on}		0.10 0.11 0.12		μs
Rise time, inductive load	$ I_{C} = 400 \text{ A}, V_{CE} = 300 \text{ V} \\ V_{GE} = \pm 15 \text{ V} \\ R_{Gon} = 1.8 \Omega $	$T_{vj} = 25^{\circ}C$ $T_{vj} = 125^{\circ}C$ $T_{vj} = 150^{\circ}C$	tr		0.08 0.08 0.08		μs
Turn-off delay time, inductive load	$\begin{array}{c} I_{C} = 400 \text{ A}, V_{CE} = 300 \text{ V} \\ V_{GE} = \pm 15 \text{ V} \\ R_{Goff} = 1.8 \ \Omega \end{array}$	$\begin{array}{l} T_{vj} = 25^{\circ}C \\ T_{vj} = 125^{\circ}C \\ T_{vj} = 150^{\circ}C \end{array}$	t _{d off}		0.46 0.50 0.50		μs
Fall time, inductive load	$ I_{C} = 400 \text{ A}, V_{CE} = 300 \text{ V} \\ V_{GE} = \pm 15 \text{ V} \\ R_{Goff} = 1.8 \Omega $	$\begin{array}{l} T_{vj} = 25^{\circ}C \\ T_{vj} = 125^{\circ}C \\ T_{vj} = 150^{\circ}C \end{array}$	ŧ		0.05 0.07 0.08		μs
Turn-on energy loss per pulse	$ \begin{array}{l} I_{C} = 400 \; A, V_{CE} = 300 \; V, L_{S} = 25 \; nH \\ V_{GE} = \pm 15 \; V, di/dt = 5500 \; A/\mu s \; (T_{vj} = 150^{\circ}C) \\ R_{Gon} = 1.8 \; \Omega \end{array} $	$\begin{array}{l} T_{vj} = 25^{\circ}C \\ T_{vj} = 125^{\circ}C \\ T_{vj} = 150^{\circ}C \end{array}$	Eon		2.90 4.20 4.50		mJ
Turn-off energy loss per pulse	$ \begin{array}{l} I_{C} = 400 \ \text{A}, \ \text{V}_{CE} = 300 \ \text{V}, \ \text{L}_{S} = 25 \ \text{nH} \\ \text{V}_{GE} = \pm 15 \ \text{V}, \ \text{du/dt} = 3000 \ \text{V/\mus} \ (T_{vj} = 150^{\circ}\text{C}) \\ \text{R}_{\text{Goff}} = 1.8 \ \Omega \end{array} $	$\begin{array}{l} T_{vj} = 25^{\circ}C \\ T_{vj} = 125^{\circ}C \\ T_{vj} = 150^{\circ}C \end{array}$	Eoff		13.0 16.0 17.0		mJ
SC data	$ \begin{array}{ll} V_{GE} \leq 15 \ V, \ V_{CC} = 360 \ V & t_P \leq 8 \ \mu s \\ V_{CEmax} = V_{CES} \ -L_{sCE} \ \cdot di/dt & t_P \leq 6 \ \mu s \end{array} $, T _{vj} = 25°C , T _{vj} = 150°C	Isc		2800 2000		А
Thermal resistance, junction to case	per IGBT		RthJC			0.120	K/W
Thermal resistance, case to heatsink	per IGBT $\lambda_{Paste} = 1 W/(m \cdot K) / \lambda_{grease} = 1 W/(m \cdot K)$		R _{thCH}		0.080		к/w
Temperature under switching conditions	top continuous		Tviop	-40		150	°C

3

Final Data Sheet

V3.4, 2018-08-09



3 Diode, Inverter

3.1 Maximum Rated Values							
Parameter	Conditions		Symbol		Value		Unit
Repetitive peak reverse voltage	$T_{vj} = 25^{\circ}C$		VRRM		650		V
Continuous DC forward current			IF		400		Α
Repetitive peak forward current	t _P = 1 ms		IFRM		800		А
l²t - value	$ \begin{array}{l} V_{\rm R} = 0 \; V, t_{\rm P} = 10 \; ms, T_{vj} = 125^{\circ} C \\ V_{\rm R} = 0 \; V, t_{\rm P} = 10 \; ms, T_{vj} = 150^{\circ} C \end{array} $		l²t		8800 8500		A²s A²s
3.2 Characteristic Values				min.	typ.	max.	
Forward voltage	$ \begin{array}{l} I_F = 400 \; A, \; V_{GE} = 0 \; V \\ I_F = 400 \; A, \; V_{GE} = 0 \; V \\ I_F = 400 \; A, \; V_{GE} = 0 \; V \end{array} $	$\begin{array}{l} T_{vj} = 25^{\circ}C \\ T_{vj} = 125^{\circ}C \\ T_{vj} = 150^{\circ}C \end{array}$	VF		1.55 1.50 1.45	1.95	v
Peak reverse recovery current	$ \begin{array}{ c c c c c } I_F = 400 \ \text{A}, - di_F/dt = 5500 \ \text{A}/\mu s \ (T_{vj} = 150^\circ \text{C}) \\ V_R = 300 \ \text{V} \\ V_{GE} = -15 \ \text{V} \end{array} $	$\begin{array}{l} T_{vj} = 25^{\circ}C \\ T_{vj} = 125^{\circ}C \\ T_{vj} = 150^{\circ}C \end{array}$	IRM		210 280 300		А
Recovered charge	$ \begin{array}{l} I_{F} = 400 \; A, - di_{F}/dt = 5500 \; A/\mu s \; (T_{vj} = 150^{\circ}C) \\ V_{R} = 300 \; V \\ V_{GE} = -15 \; V \end{array} $	$\begin{array}{l} T_{vj} = 25^{\circ}C \\ T_{vj} = 125^{\circ}C \\ T_{vj} = 150^{\circ}C \end{array}$	Qr		18.0 30.0 34.0		μC
Reverse recovery energy	$ \begin{array}{l} I_{\rm F} = 400 \; A, - d_{\rm F}/dt = 5500 \; A/\mu s \; (T_{\rm vj} = 150^{\circ} C) \\ V_{\rm R} = 300 \; V \\ V_{\rm GE} = -15 \; V \end{array} $	$\begin{array}{l} T_{vj} = 25^{\circ}C \\ T_{vj} = 125^{\circ}C \\ T_{vj} = 150^{\circ}C \end{array}$	Erec		3.60 7.25 8.30		mJ
Thermal resistance, junction to case	per diode		RthJC			0.200	K/W
Thermal resistance, case to heatsink	per diode $\lambda_{Paste} = 1 W/(m \cdot K) / \lambda_{grease} = 1 W/(m \cdot K)$		RthCH		0.085		к/w
Temperature under switching conditions	t _{op} continuous		T _{vj op}	-40		150	°C

4 NTC-Thermistor

4 NTC-Thermistor			min.	typ.	max.	
Parameter	Conditions	Symbol		Value		Unit
Rated resistance	Tc = 25°C	R25		5.00		kΩ
Deviation of R100	T _C = 100°C, R ₁₀₀ = 493 Ω	ΔR/R	5		5	%
Power dissipation	T _C = 25°C	P ₂₅			20.0	mW
B-value	R2 = R25 exp [B25/50(1/T2 - 1/(298,15 K))]	B _{25/50}		3375		к
B-value	R ₂ = R ₂₅ exp [B _{25/80} (1/T ₂ - 1/(298,15 K))]	B _{25/80}		3411		к
B-value	R ₂ = R ₂₅ exp [B _{25/100} (1/T ₂ - 1/(298,15 K))]	B _{25/100}		3433		к

4

Specification according to the valid application note.



5 Module

Parameter	Conditions	Symbol		Value		Unit
Isolation test voltage	RMS, f = 50 Hz, t = 1 min.	VISOL		2.5		kV
Material of module baseplate				Cu		
Internal isolation	basic insulation (class 1, IEC 61140)			AI_2O_3		
Creepage distance	terminal to heatsink terminal to terminal	d _{Creep}		12.0 6.1		mm
Clearance	terminal to heatsink terminal to terminal	d _{Clear}		12.0 6.1		mm
Comperative tracking index		CTI		> 200		
			min.	typ.	max.	
Stray inductance module		Lsce		30		nH
Module lead resistance, terminals - chip	T _c = 25 °C, per switch	R _{CC'+EE'}		1.00		mΩ
Storage temperature		T _{stg}	-40		125	°C
Mounting torque for modul mounting	Screw M5 baseplate to heatsink	М	3.00		6.00	Nm
Terminal connection torque	Screw M6	М	3.0	-	6.0	Nm
Weight		G		485		g

5

Der Kollektor-Dauergleichstrom / Dioden-Dauergleichstrom ist durch die Lastanschlüsse begrenzt. DC-collector current / diode forward current limited by power terminals.



FS400R07A1E3 HybridPACK[™] 1 Module

6 **Characteristics Diagrams**







output characteristic IGBT,Inverter (typical) $I_{\rm C}$ = f (V_{CE}) $T_{\rm vj}$ = 150°C



 $\begin{array}{l} \mbox{switching losses IGBT,Inverter (typical)} \\ E_{cn} = f \left(l_{c} \right), E_{off} = f \left(l_{c} \right) \\ V_{GE} = \pm 15 \ V, R_{Gon} = 1.8 \ \Omega, R_{Goff} = 1.8 \ \Omega, V_{CE} = 300 \ V \end{array}$

C

6



Final Data Sheet

V3.4, 2018-08-09







transient thermal impedance IGBT,Inverter $Z_{\text{thuc}} = f(t)$

- Zturc : IGBT

0,1

7





forward characteristic of Diode, Inverter (typical) $I_{\rm F}$ = f (V_{\rm F})



Final Data Sheet

V3.4, 2018-08-09

10



FS400R07A1E3 HybridPACK[™] 1 Module







6 NTC-Thermistor-temperature characteristic (typical) R = f (T)

8 10 R_G [Ω]

12 14 16 18



8

Final Data Sheet

V3.4, 2018-08-09



FS400R07A1E3 HybridPACK™ 1 Module

7 Circuit diagram



9

Final Data Sheet

V3.4, 2018-08-09





10

Final Data Sheet

FS400R07A1E3 HybridPACK[™] 1 Module



9 Label Codes

9.1 Module Code				
Code Format	Data Matrix			
Encoding	ASCII Text			
Symbol Size	16x16			
Standard	IEC24720 and IEC16022			
Code Content	Content Module Serial Number Module Material Number Production Order Number Datecode (Production Year) Datecode (Production Week)	Digit 1 - 5 6 - 11 12 - 19 20 - 21 22 - 23	Example (below) 71549 142846 55054991 15 30	
Example		715491	42846550549911530	

9.2 Packing Code

the rubbing obud		-		
Code Format	Code128			
Encoding	Code Set A			
Symbol Size	34 digits			
Standard	IEC8859-1			
Code Content	Content Backend Construction Number Production Lot Number Serial Number Date Code Box Quantity	Identifier X 1T S 9D Q	Digit 2 - 9 12 - 19 21 - 25 28 - 31 33 - 34	Example (below) 95056609 2X0003E0 754389 1139 15
Example		50566091T2X	0003E0S754	

11



Revision History

Major changes since previous revision

Revision History					
Reference	Date	Description			
V1.0	2009-08-26	Initial Version			
V1.1	2009-10-14	target data			
V2.0	2009-11-19	preliminary data			
V3.0	2010-04-21	final data			
V3.1	2012-02-29	final data			
V3.2	2016-11-08	new datasheet format			
V3.3	2017-03-07	-			
V3.4	2018-08-09	update of "Terms & Conditions of usage"			

12



FS400R07A1E3 HybridPACK[™] 1 Module

Terms & Conditions of usage

Edition 2018-08-01

Published by Infineon Technologies AG 81726 Munich, Germany © 2018 Infineon Technologies AG All Rights Reserved

Legal Disclaimer The information given in this document shall in no event be regarded as a guarantee of conditions or characteristics. With respect to any examples on hints given herein, any typical values stated herein and/or any information regarding the application of the device, infineon Technologies hereby disclaims any and all warranties and liabilities of any kind, including without limitation, warranties of non-infringement of intellectual property rights of any third party.

Information For further information on technology, delivery terms and conditions and prices, please contact the nearest Infineon Technologies Office (http://www.infineon.com)

Warnings

Warnings Due to technical requirements, components may contain dangerous substances. For information on the types in question, please contact the nearest Infineon Technologies Office. These components are not designed for "special applications" that demand extremely high reliability or safety such as aerospace, defense or life support devices or systems (Class III medical devices). If you intend to use the components in any of these special applications, please contact your local representative at International Rectifier HiRel Products, lic. or the Infineon support (https://www.infineon.com/support) to review product requirements and reliability testing. Ill medical devices are intended to be implaned in the human body or to support and/or maintain and sustain and/or protect human life. If they fail, it is reasonable to assume that the health of the user or other persons may be endangered.

Trademarks

Trademarks of Infineon Technologies AG

AURIX[™], C166[™], CanPAK[™], CIPOS[™], CIPURSE[™], EconoPACK[™], CoolMOS[™], CoolSET[™], CORECONTROL[™], CROSSAVE[™], DAVE[™], DI-POL[™], EasyPiM[™], EconoBRIDGE[™], EconoDUAL[™], EconoPM[™], EconoPACK[™], ElceDRIVER[™], euger[™], FCOS[™], HITTET[™], HyoridPACK[™], PR[™], ISIO-E[™], IsoPACK[™], MIPAQ[™], ModSTACK[™], my[™], NovaithiC[™], OptiMOS[™], ORG[™], POWERCODE[™], PRIMARION[™], PrimePACK[™], PrimeSTACK[™], PRO-SIL[™], PAOFE[™], RASIC[™], ReverSave[™], SatRIC[™], SIGET[™], SINDRION[™], SIPMOS[™], SmartLEWS[™], SOLD FLASH[™], TEMPFET[™], MINAQ[™], TRANCHSTOP[™], TICOR[™], SIEGET[™], SINDRION[™], SIPMOS[™], SmartLEWS[™], SOLD FLASH[™], TEMPFET[™], MINAQ[™], TRANCHSTOP[™], TICOR[™], SIEGET[™], SINDRION[™], SINDRION[™], SINDRIN[™], SINDRION[™], SINDRIN[™], SINDRIN[™]

Other Trademarks

Advance Design System[™] (ADS) of Agilent Technologies, AMBA[™], ARM[™], MULTI-ICE[™], KEIL[™], PRIMECELL[™], REALVIEW[™], THUMB[™], Vision[™] of ARM Limited, UK. AUTOSAR[™] is licensed by AUTOSAR development partnership. Bluetooth[™] of Bluetooth SIG Inc. CAT-iq[™] of DECT Forum. COLOSSUS[™], FirstCPS[™] of Timble Navigation Ltd. EMV[™] of EMVCo, LLC (Visa Holdings Inc.). EPCOS[™] of Epcos AG. FIEXGO[™] of Microsoft Corporation. FieXR[™] is licensed by PlexRay Consortium. HYPERTERNINAL[™] of Higrese Incorporated. IEC[™] of Commission Electrotechnique Internationale. IrDA[™] of Infrared Data Association Corporation. ISO[™] of INTERNATIONAL ORGANIZATION FOR STANDARDIZATION. MATLAB[™] of Mairia Wicks, Inc. MANIM[™] of Maxim Integrated Products, Inc. MICROTECC[™], NUCLEUS[™] of Mentor Graphics Corporation. MIPI[™] of MIPI Alliance, Inc. MIPS[™] of MIPS Technologies, Inc., USA. multata[™] of MURATA MANUPACTURING CO., MICROWAVE OFFICE[™] (MWO) of Applied Wave Research Inc., ComiNision[™] of OmNIJsion Technologies, Inc. Openwave Systems Inc. RED HAT[™] Red Hat, Inc. RFMD[™] RF Micro Devices, Inc. SIRUS[™] of Timus Satellite Radio Inc. SOLARIS[™] of Sun Microsystems, Inc. SPANSION[™] of Spansion LLC LLd. Symbian[™] of Siruis Satellite Radio Inc. SOLARIS[™] of Sun CERLICE[™] of CEVA, Inc. TEKTRONIX[™] of Tektronix Inc. TOKO[™] of TOKO KABUSHIKI KAISHA TA. UNIX[™] d/ X/Open Company Limited. VERILOG[™], PLLALDIUM[™] of Cadence Design Systems, Inc. UXYNQ[™] of Texas Instruments Incorporated. VXWORKS[™]</sup>, WIND RIVER[™] of WIND RIVER SYSTEMS, INC. ZETEX[™] of Diodes Zetex Limited.

Last update

2011-11-11

Final Data Sheet

V3.4, 2018-08-09

www.infineon.com

Published by Infineon Technologies AG



