

**Flow boiling of HFE-7100 in a single
microchannel**

**A Thesis Submitted for the
Degree of Doctor of
Philosophy**

By

Joseph Jake Widginton

**Department of Mechanical and Aerospace
Engineering,
Brunel University London**

2026

ABSTRACT

The global drive towards electrification and carbon reduction has created an urgent requirement for compact, high-performance thermal management systems capable of dissipating high heat fluxes while maintaining component reliability. Flow boiling in microchannels has emerged as a promising solution due to its high heat transfer capability arising from latent heat effects and large surface area to volume ratios. However, industrial implementation remains limited by incomplete understanding of microscale flow behaviour, heat transfer, pressure drop, and critical heat flux (CHF). Existing predictive methods are often empirical, fluid-specific, and lack general applicability. This thesis addresses these limitations through an experimental investigation of flow boiling of HFE-7100 in a single rectangular microchannel.

An experimental facility was designed and constructed to obtain accurate measurements of wall temperature, pressure drop, and high-speed flow visualisation. Two oxygen-free copper test sections with rectangular channels and hydraulic diameters of 0.75 mm were fabricated, with heated lengths of 30 mm and 75 mm to investigate length effects. Experiments were conducted over mass fluxes from 100–1000 kg/m²s, wall heat fluxes up to CHF, and inlet pressures of 1–2 bar. HFE-7100 was selected due to its dielectric properties and lack of prior single-channel flow boiling data. Single-phase validation and uncertainty analysis confirmed the reliability of the methodology.

Flow visualisation identified bubbly, slug, churn, and annular flow regimes, with transitions strongly dependent on mass flux, heat flux, inlet pressure, and heated length. Stratified flow was not observed, confirming the confined nature of microscale boiling. Nucleation consistently initiated at channel corners due to geometric confinement. Increasing mass flux and pressure altered bubble coalescence and stability, shifting transition boundaries accordingly. Shorter heated lengths promoted earlier slug formation due to accelerated bubble growth at higher local heat fluxes. Transition maps demonstrated that existing predictive models only partially capture the combined effects of confinement, evaporation momentum, interfacial shear, and geometry.

Heat transfer measurements showed that boiling incipience occurred at lower wall superheats with increasing inlet pressure. Heat flux significantly enhanced heat transfer coefficients through increased nucleation activity, while mass flux primarily extended the subcooled boiling region. Higher inlet pressures further improved heat transfer by stabilising nucleation. The 30 mm channel produced higher average heat transfer coefficients than the 75 mm channel at equivalent outlet vapour qualities due to intensified nucleation from concentrated heat input. Nucleate boiling persisted throughout the vapour quality range, limiting the applicability of convective boiling models. Evaluation of twelve existing

correlations showed that the best-performing models were those capable of accounting for nucleate boiling dominance.

Two-phase pressure drop increased with heat flux due to increased vapour generation and interfacial shear. Increasing pressure generally reduced pressure drop by delaying flow regime transitions through changes in fluid properties. Shorter heated lengths produced steeper pressure gradients due to intensified bubble activity and frictional losses. Existing pressure drop correlations showed considerable discrepancies, although the homogeneous flow model provided reasonable agreement.

CHF investigations in the 30 mm channel showed that dryout occurred primarily through liquid film depletion in annular flow. Higher mass flux and pressure increased CHF, with mass flux having the strongest influence due to improved liquid replenishment and bubble removal. Existing CHF correlations performed reasonably well overall but did not accurately capture the weak pressure dependence observed experimentally.

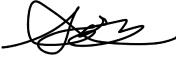
Finally, a novel neural network framework for predicting microscale flow pattern transitions was developed. A multi-stage feed-forward neural network architecture was trained using a large experimental database with robust preprocessing, cross-validation, and hyperparameter optimisation. The model achieved an overall classification accuracy of 95.5%, outperforming conventional empirical approaches and demonstrating the potential of data-driven methods for predicting complex microscale two-phase flow behaviour.

DECLARATION

I, Joseph Jake Widgington, hereby declare that the present study is my own original and independent work except where otherwise specified, or where acknowledgements are made via references. This research was conducted at the Department of Mechanical and Aerospace Engineering, Brunel University London, under the supervision of Professor Tassos G. Karayiannis and Doctor Atanas Ivanov.

I further declare that any use of Artificial Intelligence (AI) tools in the preparation of this work was limited strictly to non-creative, non-text-generative tasks. AI assistance was used for repetitive coding activities, such as formatting figures and automating routine operations in MATLAB. AI tools were not used for generating text, developing arguments, interpreting results, or contributing original thought to the research or written thesis.

This work has not been submitted for another degree or award to any other institution.

Signed:  Date: 31/08/2026

ACKNOWLEDGMENTS

I would firstly like to show my deepest gratitude and sincerest thanks to my supervisor, Professor Tassos G. Karayiannis. The knowledge imparted by Professor Karayiannis, his emphasis on good research practices, his patience and his kindness have all significantly improved my skills as a researcher. His keen insight and enthusiastic engagement throughout the project were instrumental in enabling this work. Sincere gratitude is also extended to Dr. Mohamed M. Mahmoud for his beneficial insights at the initial stages of this project and to Dr. Ali Al-Zaidi for providing his invaluable knowledge on flow boiling and friendship whilst conducting this research. I would like to express my gratitude to my second supervisor Dr. Atanas Ivanov for his support, particularly in the development of the experimental facility. I extend my thanks to Dr Fang Wang for generously sharing her time and expertise in machine learning, and for her assistance in reviewing this work.

I would like to thank Mr. Costas Xanthos, not only for his help in the preparation of the experimental facility but also for the many interesting conversations in the laboratory that have made conducting research in his presence so enjoyable.

To my mother and father, your unconditional love has provided me with strength to continue in the face of the many obstacles I have faced.

To my sister Hannah, thank you for being the voice of reason during difficult times and providing unwavering support. To my Aunt Janet, thank you for instilling the value of education within me and supporting me in more ways than I could ever list. This work would most certainly have not happened without either of you.

Most importantly, I would like to dedicate this work to my beautiful wife, Emily Widginton, who has supported me both emotionally and financially more than could ever be expected of her. This achievement is as much yours, as it is mine. I now hope to make it up to you with a lifetime of happiness.

TABLE OF CONTENTS

Abstract.....	i
Declaration.....	iii
Acknowledgments.....	iv
Table of Contents.....	v
List of Figures.....	xii
List of Tables	xxiii
Nomenclature.....	xxiv
CHAPTER 1	1
Introduction.....	1
1.1 Background and Motivation.....	1
1.2 Thesis Structure.....	13
CHAPTER 2	15
Literature Review.....	15
2.1 Introduction.....	15
2.2 Defining the Microscale.....	15
2.3 Flow Boiling Patterns.....	21
2.3.1 Flow Patterns in the Conventional/Macro Scale.....	22
2.3.2 Flow Patterns in the Microscale.....	24
2.3.3 Microscale Flow Pattern Prediction Tools.....	52
2.4 Microscale Flow Boiling Heat Transfer.....	61
2.4.1 Flow Boiling Heat Transfer Mechanisms.....	61
2.4.2 Parametric Effects on Microscale Flow Boiling Heat Transfer.....	63
2.4.3 Existing Heat Transfer Prediction Tools.....	72
2.5 Microscale Flow Boiling Pressure Drop.....	76

2.5.2	Existing Pressure Drop Prediction Tools.....	79
2.6	Critical Heat Flux.....	83
2.6.1	Critical Heat Flux Mechanisms	84
2.6.2	Microscale Critical Heat Flux Prediction Tools	87
2.7	Flow Boiling and Artificial intelligence.....	91
2.7.1	Flow Patterns	92
2.7.2	Other Applications in Flow Boiling.....	96
2.8	Conclusions	98
2.8.1	Defining the Microscale.....	99
2.8.2	Flow Boiling Patterns	100
2.8.3	Heat Transfer	100
2.8.4	Pressure Drop.....	101
2.8.5	Critical Heat Flux.....	101
2.8.6	Flow Boiling and Artificial Intelligence	102
2.9	Research Objectives	12
CHAPTER 3		103
Experimental Methodology		103
3.1	Introduction	103
3.2	Initial Experimental Facility.....	103
3.2.1	Main Closed Flow Loop	105
3.2.2	Auxiliary Cooling Loop.....	109
3.2.3	Data Acquisition System (DAQ)	111
3.2.4	High-Speed Camera and Microscope	111
3.2.5	Software	112
3.2.6	Initial Test Section Design.....	112
3.2.7	Fluid Selection	115
3.3	Modifications to Existing Facility.....	116
3.3.1	Replacing the Pre-Heater	116

3.3.2	Change of Test Section Design.....	117
3.3.3	Replacing Variac for Test Section	126
3.3.4	Inclusion of Gas-Loaded Accumulator	127
3.4	Surface Analysis.....	129
3.5	Data Reduction.....	133
3.5.1	Single-phase Data Reduction	133
3.5.2	Flow Boiling Data Reduction	136
3.6	Experimental Parameters.....	141
3.7	Instrumentation Calibration Procedures.....	142
3.7.1	Thermocouple Calibration Procedure	143
3.7.2	Absolute Pressure Transducer Calibration Procedure	145
3.8	Experimental Procedure	146
3.8.1	System Degassing Procedure.....	146
3.8.2	Single-Phase Validation Experimental Procedure	147
3.8.3	Flow Boiling Experimental Procedure.....	148
3.9	Estimating Heat Loss in the Test Section.....	149
3.9.1	Heat Loss Estimation Experimental Procedure	149
3.9.2	Heat Loss Estimation Data Reduction	150
3.10	Single-Phase Validation	151
3.11	Repeatability	155
3.12	Summary	157
CHAPTER 4		159
Flow Boiling Pattern Results		159
4.1	Introduction	159
4.2	Flow Visualisation	159
4.3	Experimental Flow Pattern Maps	163
4.3.1	Bubbly to Slug Transition.....	166
4.3.2	Slug to Churn Transition.....	168

4.3.3	Churn to Annular Transition.....	170
4.3.4	Effect of System Pressure.....	173
4.3.5	Effect of Heated Length.....	175
4.4	Comparison to Existing Flow Boiling Pattern Maps.....	178
4.4.1	Comparison to Chen (2006).....	178
4.4.2	Comparison to Mahmoud and Karayiannis (2016).....	179
4.4.3	Comparison to Revellin and Thome (2007) and Ong and Thome (2011)	185
4.4.4	Comparison to Harirchian and Garimella (2010) and Harirchian and Garimella (2012) 188	
4.4.5	Comparison to Tibiriçá et al. (2017).....	191
4.4.6	Comparison to Choi et al. (2017).....	191
4.4.7	Comparison to Jige et al. (2018).....	193
4.5	Conclusions.....	194
CHAPTER 5		197
Heat Transfer Results.....		197
5.1	Introduction.....	197
5.2	Boiling Incipience	197
5.3	Effect of Heat Flux on Heat Transfer.....	201
5.4	Effect of Mass Flux on Heat Transfer.....	203
5.5	Effect of Channel Inlet Pressure on Heat Transfer.....	206
5.6	Effect of Heated Length on Heat Transfer	211
5.7	Comparison to Existing Heat Transfer Prediction Tools	213
5.7.1	Comparison to Shah (1982)	214
5.7.2	Comparison to Warriar et al. (2002).....	215
5.7.3	Comparison to Lee and Mudawar (2005)	216
5.7.4	Comparison to Mahmoud and Karayiannis (2013).....	217
5.7.5	Comparison to Li and Jia (2015)	218
5.7.6	Comparison to Thiangtham et al. (2016)	219

5.7.7	Comparison to Shah (2017)	221
5.7.8	Comparison to Wen et al. (2019).....	221
5.7.9	Comparison to Zhang et al. (2025)	223
5.7.10	Comparison to Patra et al. (2025).....	224
5.8	Conclusions	226
CHAPTER 6		228
Pressure Drop Results		228
6.1	Introduction	228
6.2	Effect of Heat Flux and mass flux on Pressure Drop	228
6.3	Effect of System Pressure on Pressure Drop.....	230
6.4	Effect of Heated Length on Pressure Drop	236
6.5	Comparison to Existing Flow Boiling Pressure Drop Correlations	236
6.5.1	Comparison to Lockhart and Martinelli (1949).....	237
6.5.2	Comparison to Homogeneous Flow Model	238
6.5.3	Comparison to Qu and Mudawar (2003)	240
6.5.4	Comparison to Lee and Garimella (2008).....	241
6.5.5	Comparison to Keepaiboon et al. (2016)	242
6.5.6	Comparison to Huang and Thome (2017)	243
6.5.7	Comparison to Markal et al. (2019).....	244
6.5.8	Comparison to Gao et al. (2019).....	245
6.5.9	Comparison to Moradkhani et al. (2021).....	246
6.5.10	Comparison to Al-Zaidi et al. (2024)	248
6.6	Conclusions	250
CHAPTER 7		252
Critical Heat Flux Results		252
7.1	Introduction	252
7.2	Critical Heat Flux Mechanisms.....	252
7.3	Comparison to Existing Microscale Critical Heat Flux Prediction Tools	260

7.4	Conclusions	262
CHAPTER 8		264
New Feed-Forward Neural Network Flow Pattern Model.....		264
8.1	Introduction	264
8.2	Machine Learning Methodology	264
8.3	Feed-Forward Neural Network Results.....	267
8.3.1	Data Splitting	267
8.3.2	Data Scaling.....	268
8.3.3	Model Cross-Validation.....	269
8.3.4	Feature Selection and Hyperparameter Tuning	270
8.3.5	Final Algorithm Architecture.....	284
8.4	Comparison to Current Study.....	290
8.5	Conclusions	293
CHAPTER 9		294
Conclusions and Recommendations		294
9.1	Conclusions	294
9.1.1	Flow Boiling Patterns	295
9.1.2	Flow Boiling Heat Transfer	295
9.1.3	Flow Boiling Pressure Drop.....	297
9.1.4	Critical Heat Flux.....	298
9.1.5	A Novel Neural Network for Flow Pattern Prediction	298
9.2	Summary of Novel Contributions	299
9.3	Recommendations	301
References.....		303
Appendix A.....		342
Flow Boiling Heat Transfer Correlations		342
Appendix B		346
Flow Boiling Pressure Drop Correlations.....		346

Appendix C	352
Microscale Critical Heat Flux Correlations	352
Appendix D.....	355
75 mm Long Channel Test Section Engineering Drawings.....	355
Appendix E	360
30 mm Long Channel Test Section Engineering Drawings.....	360
Appendix F.....	365
Uncertainty Propagation Analysis	365
Publications.....	372

LIST OF FIGURES

Figure		Page
1.1	Typical pool boiling curve.	2
1.2	Categories of two-phase jet impingement.	3
1.3	Example diagrammatic assembly of microchannel heat sink for an insulated gate bipolar transistor module.	4
1.4	Pumped system loop for micro-evaporator device.	4
1.5	Drawing of microchannel heat sink for cooling integrated circuit chips (taken from [40]).	5
1.6	Water cooled Lenovo ThinkSystem SD650 direct water-cooled server: (a) front view of enclosure with 6 trays; (b) water path in SD650 tray (modified from [47]).	7
1.7	Applications of IGBT modules with their associated voltage and current ratings (taken from Tan et al. [50]).	8
1.8	Diagram of high concentration photovoltaic system for solar energy conversion with microchannel heat sink cooling.	9
1.9	Venn diagram of artificial intelligence, machine learning and deep learning.	10
1.10	Simplified neural network architecture.	11
2.1	Diagram of EDL at the channel wall (adapted from Mehendale et al. [74]).	16
2.2	Drawings of observed flow patterns by Kawaji and Chung [76] in the (a) miniscale and (b) microscale.	17
2.3	Scaled forces per unit area: (a) for HFE-7100; (b) for DI water.	18
2.4	Model of static liquid plug (adapted from Tibiri�� and Ribatski [81]).	20
2.5	Microscale criteria threshold hydraulic diameters for different fluids.	20
2.6	Publications on microscale flow boiling topics per year (Scopus search as of March 2025).	21
2.7	Flow patterns in conventional sized tubes for upwards vertical flows.	23
2.8	Flow patterns in conventional sized tubes for horizontal flows.	23
2.9	Example diagram of typical indirect diabatic test section.	25
2.10	Rivulet flows reported by Barajas and Panton (Modified from [110]).	27
2.11	Flow patterns reported by Triplett et al. [112].	28
2.12	Flow patterns reported by Coleman and Garimella [113].	29

Figure		Page
2.13	Flow pattern transitions with increasing vapour superficial velocity by Holland and Bragg [119] (modified).	31
2.14	Flow patterns reported by Chen for R134a in 1.1 mm internal diameter vertical tube [121].	32
2.15	Flow patterns and their transitions as reported by Revellin et al. [123].	33
2.16	Coalescence phenomena for R134a in a 0.509 mm ID horizontal channel with a 50 mm heated section (taken from [125]).	34
2.17	Bubble nucleation and growth mechanism in triangular microchannels (adapted from [177]).	37
2.18	Predicted bubble departure diameter using correlation by Yang et al. [185].	39
2.19	Bubble growth and confinement in (a) a high aspect ratio microchannel and (b) a low aspect ratio microchannel.	41
2.20	Single vapour bubbles in high aspect ratio microchannels (a) from Widyatama et al. [184] and (b) Özdemir et al. [161].	42
2.21	Intermittent flow patterns reported by Gasche for CO ₂ [187].	45
2.22	Photographs of isolated bubbly flow, confined bubbly flow and slug flow from Martin-Callizo et al. [169].	45
2.23	Photographs of flow patterns from Celata et al. [146] (flow from right to left).	48
2.24	Wispy-annular flow and Annular flow reported by Harirchian and Garimella [143] (modified).	51
2.25	Chen flow pattern map for R134a at 6 bar in a 1.1 mm tube (taken from [198]).	53
2.26	Drawing of force balance on a spherical bubble in which coordinate system moves with bubble (modified from [82]).	55
2.27	Mahmoud and Karayiannis modified Chen flow pattern map for R-245fa (taken from [82]).	56
2.28	Flow pattern map proposed by Revellin and Thome [125] for R-134a in a circular microchannel and an applied heat flux of 50 kW/m ² .	57
2.29	Flow pattern map for FC-77 proposed by Harirchian and Garimella (taken from [202]).	58
2.30	Flow pattern map for FC-77 proposed by Harirchian and Garimella (taken from [203]).	59
2.31	Proposed transition from bubbly/slug to annular flow for FC-72 by Choi et al. (modified from [151]).	60
2.32	Proposed flow pattern map for R-32 in a 0.5 mm internal diameter multiport tube by Jige et al. (modified from [205]).	61

Figure		Page
2.33	Schematics of flow regimes, wall dryout and variation of heat transfer coefficient along uniformly heated channel for; (a) nucleate boiling dominant heat transfer; (b) convective boiling dominant heat transfer (modified from [209])	63
2.34	Gas entrapment criteria: (a) by Bankoff [245] and (b) by Lorenz [246].	67
2.35	Aspect ratio criteria for gas/vapour entrapment by Bankoff [245], Lorenz [246] and Whang and Dhir [248].	68
2.36	Force balance of a vapour bubble at a nucleation site (modified from [253]).	70
2.37	Departure from nucleate boiling CHF mechanism.	84
2.38	Mechanisms that cause the onset of boiling during annular flows: (a) vapour film formation; (b) deposition and evaporation equilibrium; (c) liquid film disruption; (d) dry spot formation; (e) full film dryout (modified from [323]).	86
2.39	Vapour backflow into subcooled liquid (modified from [309]).	87
2.40	Liquid-vapour interfacial forces (modified from [327]).	88
2.41	Conceptualisation of the KNN algorithm.	93
2.42	Typical FFN architecture.	94
2.43	FNN training and prediction process.	95
2.44	Probabilistic Neural Network Architecture (taken from [352]).	96
3.1	Photograph of initial experimental facility in Tower B Building, Brunel of University London.	103
3.2	Schematic diagram of initial experimental facility.	104
3.3	Reservoir tank apparatus.	105
3.4	Krohne Optimass 3000-S01 Coriolis flow meter and Krohne MFC 300 signal converter for mass flowmeters.	106
3.5	Micro-gear pump system: (a) Photograph of Tuthill DGS.19EEETNNOM163 micro gear pump; (b) Photograph of Schneider Electric ATV12H075M2 phase inverter.	107
3.6	Preheater system equipment configuration.	107
3.7	Example RS WCV15E-1 enclosed variable transformer.	109
3.8	LabTech H150-700 water chiller system.	110
3.9	Brazed plate heat exchangers: (a) condenser; (b) sub-cooler.	110
3.10	NI cDAQ-9174 DAQ chassis, NI 9213 DAQ modules and NI 9224 DAQ Module.	111
3.11	Microscope and high-speed camera system.	112
3.12	Exploded CAD diagram of the initial test section design.	113

Figure		Page
3.13	Clear polycarbonate top plate bottom CAD drawing with pressure transducer configuration diagram.	114
3.14	Oxygen-free copper test section: (a) photograph of test section; (b) isometric CAD drawing; (c) thermocouple position diagram.	115
3.15	Configuration used to replace screw immersion pre-heater.	118
3.16	Undesired bubble nucleation in inlet plenum.	119
3.17	Exploded CAD diagram of the long test section design.	120
3.18	CAD drawing of long clear polycarbonate cover plate.	120
3.19	Key dimensions of long clear polycarbonate cover plate.	121
3.20	CAD drawing of long oxygen-free copper test section.	122
3.21	Key dimensions of long oxygen-free copper test section.	122
3.22	Exploded CAD assembly of 30 mm long test section.	124
3.23	Sealing methodology for stainless steel pipework in the test section.	124
3.24	CAD rendering of 30 mm long oxygen-free copper test section.	125
3.25	Key dimensions of short oxygen-free copper test section.	126
3.26	Elektro Automatik EA-PS 9360-30 digital bench power supply.	127
3.27	Pressure drop signal for single-phase liquid flow ($G=1200 \text{ kg/m}^2\text{s}$, $Re=1550$).	127
3.28	FFT for pressure drop signal ($G=1200 \text{ kg/m}^2\text{s}$, $Re=1550$).	128
3.29	FFT for pressure drop signal with gas loaded accumulator ($G=1200 \text{ kg/m}^2\text{s}$, $Re=1550$).	129
3.30	Schematic of final experimental facility.	130
3.31	Surface reconstructions for the 75 mm long channel test section.	132
3.32	Pressure losses inside the test section to be accounted for.	134
3.33	Flow boiling data reduction regions.	137
3.34	Thermocouple calibration procedure apparatus.	143
3.35	Calibration curve for thermocouple T14.	144
3.36	Dead weight gauge tester apparatus: (a) where oil reservoir pressure is lower than reference pressure; (b) where oil reservoir pressure is greater than reference pressure.	145
3.37	Temperature distribution in copper test section for heat loss experiment at a base heat flux of 9.65 kW/m^2 .	151

Figure		Page
3.38	Top plate overall heat transfer coefficients for (left) the 75 mm long channel test section and (right) the 30 mm long test section.	152
3.39	Single-phase Fanning friction factor vs Reynolds number for 75 mm long channel test section.	152
3.40	Nusselt number vs Reynolds number for 75 mm long channel test section.	154
3.41	Single-phase Fanning friction factor vs Reynolds number for 30 mm long channel test section.	155
3.42	Nusselt number vs Reynolds number for 30 mm long channel test section.	156
3.43	Repeatability results for 75 mm long channel test section for an inlet pressure of 1 bar and mass flux of 500 kg/m ² s.	157
3.44	Repeatability results for 30 mm long channel test section for an inlet pressure of 1.5 bar and mass flux of 500 kg/m ² s.	157
4.1	Flow visualisation locations.	160
4.2	Onset of bubble nucleation at channel walls at middle viewing location ($P_{in} = 1.5$ bar, $G = 750$ kg/m ² s, $q''_w = 8.0593$ kW/m ² , $x_{out} = -0.0787$).	160
4.3	Bubble nucleation and departure at the channel outlet in the subcooled region ($P_{in} = 1.5$ bar, $G = 500$ kg/m ² s, $q''_w = 19.79$ kW/m ² , $x_{out} = -0.0072$).	161
4.4	Observed flow patterns for the 75 mm long channel in the saturated region with generic classifications.	162
4.5	Cross section of channel with simultaneous dry spots and bubble nucleation.	163
4.6	Flow pattern maps for 75 mm long test section using mass flux and vapour quality for: (a) 1 bar inlet pressure; (b) 1.5 bar inlet pressure; (c) 2 bar inlet pressure.	165
4.7	Flow pattern maps for 75 mm long test section using alternative coordinates: (a) 1 bar inlet pressure and heat flux; (b) 1 bar inlet pressure and boiling number; (c) 2 bar inlet pressure and heat flux; (d) 2 bar inlet pressure and boiling number flux.	166
4.8	Flow visualisation of transition between bubbly flow and slug flow at outlet for $G = 400$ kg/m ² s, $q''_w = 24.79$ kW/m ² and $P_{in} = 1$ bar.	167
4.9	Bubbly flow visualisation at the middle the 75 mm long channel test section for a boiling number of 0.75 for: (a) a mass flux of 300 kg/m ² s; (b) a mass flux of 750 kg/m ² s.	168
4.10	Flow visualisation of transition between slug flow and churn flow at outlet for $G = 300$ kg/m ² s, $q''_w = 40.43$ kW/m ² and $P_{in} = 1$ bar.	169
4.11	Flow visualisation of transition between churn flow and annular flow at outlet for $G = 500$ kg/m ² s, $q''_w = 78.98$ kW/m ² and $P_{in} = 1$ bar.	171
4.12	Flow pattern maps for 75 mm long test section using superficial velocity coordinates: (a) for 1 bar inlet pressure; (b) for 2 bar inlet pressure.	172

Figure		Page
4.13	Effect of channel inlet pressure on flow patterns in the 75 mm long channel test section.	173
4.14	Flow visualisation in the 75 mm long channel test section for a mass flux of 300 kg/m ² s and heat flux of 19 kW/m ² at: (a) 1 bar and middle location; (b) 2 bar and middle location; (c) 1 bar and outlet location; (d) 2 bar and outlet location.	174
4.15	Observed flow patterns for the 30 mm long channel in the saturated region with generic classifications.	176
4.16	Flow pattern maps for 30 mm long test section using mass flux and vapour quality for: (a) 1.5 bar inlet pressure; (b) 2 bar inlet pressure.	177
4.17	Effect of heated length on flow pattern transition boundaries for the 30 mm and 75 mm channels: (a) inlet pressure of 1.5 bar; (b) inlet pressure of 2 bar	177
4.18	Comparison of experimental flow patterns with flow pattern map proposed by Chen [198]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.	180
4.19	Comparison of experimental flow patterns with flow pattern map proposed by Mahmoud and Karayiannis [82]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.	182
4.20	Force balance acting on bubble with consideration of viscous drag.	183
4.21	Comparison of experimental flow patterns with new semi-analytical bubbly to slug transition: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.	184
4.22	Comparison of experimental flow patterns with flow pattern map proposed by Revellin and Thome [125]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.	186
4.23	Comparison of experimental flow patterns with flow pattern map proposed by Ong and Thome [80]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.	187
4.24	Comparison of experimental flow patterns with flow pattern map proposed by Harirchian and Garimella [202]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.	189
4.25	Comparison of experimental flow patterns with flow pattern map proposed by Harirchian and Garimella [203]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.	190
4.26	Comparison of experimental flow patterns with flow pattern map proposed by Tibiri�� et al. [186]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.	192

Figure		Page
4.27	Comparison of experimental flow patterns with flow pattern map proposed by Choi et al. [151]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.	193
4.28	Comparison of experimental flow patterns with flow pattern map proposed by Jige et al. [205]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.	195
5.1	Flow boiling curves at $Z/Lch = 0.5$ for channel inlet pressures of: (a) 1 bar; (b) 1.5 bar; (c) 2 bar; in the 75 mm long channel test section.	198
5.2	Effective nucleation cavity radii with respect to wall superheat according to Hsu [243].	200
5.3	Local heat transfer coefficient and vapour quality for a mass flux of 400 kg/m ² s: (a) at 1 bar; (b) at 1.5 bar; (c) at 2 bar; in the 75 mm long channel test section.	202
5.4	Local heat transfer coefficient and dimensionless length for a mass flux of 400 kg/m ² s: (a) at 1 bar; (b) at 1.5 bar; (c) at 2 bar; in the 75 mm long channel test section.	203
5.5	Average two-phase coefficient and wall heat flux for a heat flux of 50 kW/m ² : (a) at 1 bar; (b) at 1.5 bar; (c) at 2 bar in the 75 mm long test section.	204
5.6	Local heat transfer coefficient and dimensionless length for a heat flux of 50 kW/m ² : (a) at 1 bar; (b) at 1.5 bar; (c) at 2 bar in the 75 mm long test section.	205
5.7	Effect of mass flux on local wall and fluid temperatures for a heat flux of 100 kW/m ² : (a) at 1 bar; (b) at 1.5 bar; (c) at 2 bar in the 75 mm long test section.	206
5.8	Comparison of average two-phase heat transfer coefficients for a mass flux of 500 kg/m ² s at different channel inlet pressures in the 75 mm long test section.	207
5.9	Comparison of flow boiling curves at $z/Lch = 0.5$, for a mass flux of 500 kg/m ² s at different channel inlet pressures in the 75 mm long test section.	207
5.10	Vapour residual heights in cavities at: (a) time of bubble departure from cavity and (b) at time of maximum liquid penetration depth.	208
5.11	Required cavity depth at a given cavity mouth radius for stable cavity activation for HFE-1700 according to Marto and Rohsenow [399].	210
5.12	Local channel pressure at mass and heat fluxes of 500 kg/m ² s and 130 kw/m ² , respectively.	211
5.13	Flow boiling curves at $Z/Lch = 0.5$ for channel inlet pressures of: (a) 1.5 bar; (b) 2 bar; in the 30 mm long channel test section.	211
5.14	Effect of heated length on average two-phase HTC with wall heat flux at 500 kg/m ² s for; (a) 1.5 bar inlet pressure; (b) 2 bar inlet pressure.	212
5.15	Effect of heated length on average two-phase HTC with outlet vapour quality at 500 kg/m ² s for; (a) 1.5 bar inlet pressure; (b) 2 bar inlet pressure.	213

Figure		Page
5.16	Comparison of experimental two-phase HTC to prediction by Shah [268] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	215
5.17	Comparison of experimental two-phase HTC to prediction by Warriar et al. [269].	216
5.18	Comparison of experimental two-phase HTC to prediction by Lee and Mudawar [266] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	217
5.19	Comparison of experimental two-phase HTC to prediction by Mahmoud and Karayiannis [270] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	218
5.20	Comparison of predicted heat transfer mechanism components from the correlation developed by Mahmoud and Karayiannis [270] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	219
5.21	Comparison of experimental two-phase HTC to prediction by Li and Jia [232] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	220
5.22	Comparison of experimental two-phase HTC to prediction by Thiangtham et al. [149] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	220
5.23	Comparison of experimental two-phase HTC to prediction by Shah [273] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	221
5.24	Comparison of experimental two-phase HTC to prediction by Wen et al. [260] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	222
5.25	Comparison of experimental two-phase HTC to prediction by Zhang et al. [210] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	223
5.26	Comparison of experimental two-phase HTC to prediction by Patra et al [225] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	224
6.1	Two-phase pressure drop vs wall heat flux for channel inlet pressures of; (a) 1 bar; (b) 1.5 bar; (c) 2 bar; in the 75 mm long channel test section.	229
6.2	Two-phase pressure drop vs channel outlet quality for channel inlet pressures of; (a) 1 bar; (b) 1.5 bar; (c) 2 bar; in the 75 mm long channel test section.	230
6.3	Comparison of two-phase pressure drop for varying channel inlet pressures in the 75 mm long channel test section for: (a) 200 kg/m ² s; (b) 500 kg/m ² s; in the 75 mm long channel test section.	231
6.4	Predicted pressure drop components by Gao et al. [298]: (a) acceleration for 200 kg/m ² s; (b) friction for 200 kg/m ² s; (c) acceleration for 500 kg/m ² s; (d) friction for 500 kg/m ² s; in the 75 mm long channel test section.	232
6.5	Predicted acceleration pressure drop component at 500 kg/m ² s and an outlet vapour quality of 0.4: (a) for constant viscosity; (b) for constant densities; in the 75 mm long channel test section.	233
6.6	Predicted acceleration pressure drop component at 500 kg/m ² s and an outlet vapour quality of 0.4: (a) for constant viscosity and surface tension; (b) for constant densities and surface tension; (c) for constant viscosity and densities; in the 75 mm long channel test section.	235

Figure		Page
6.7	Effect of heated length on two-phase pressure drop with outlet vapour quality at 500 kg/m ² s for; (a) 1.5 bar inlet pressure; (b) 2 bar inlet pressure.	236
6.8	Comparison of experimental two-phase pressure drop to prediction by Lockhart and Martinelli [294] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	238
6.9	Comparison of experimental two-phase pressure drop to homogeneous flow model for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	239
6.10	Comparison of acceleration and frictional pressure drop components using the homogeneous flow model for: (a) the 75 mm long channel test section; (b) the 30 mm long channel test section.	240
6.11	Comparison of experimental two-phase pressure drop to prediction by Qu and Mudawar [275] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	240
6.12	Comparison of experimental two-phase pressure drop to prediction by Lee and Garimella [217] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	241
6.13	Comparison of experimental two-phase pressure drop to prediction by Keepaiboon et al. [296] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	242
6.14	Comparison of experimental two-phase pressure drop to prediction by Huang and Thome [297] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	243
6.15	Comparison of experimental two-phase pressure drop to prediction by Markal et al. [298] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	244
6.16	Comparison of experimental two-phase pressure drop to prediction by Gao et al. [299] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	245
6.17	Comparison of experimental two-phase pressure drop to prediction by Moradkhani et al. [301] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	247
6.18	Comparison of experimental two-phase pressure drop to multi-channel heat sink prediction by Al-Zaidi et al. [303] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	248
6.19	Comparison of experimental two-phase pressure drop to vertical tube prediction by Al-Zaidi et al. [303] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.	248
7.1	Boiling curves at $z/L = 0.5$ for the 30 mm long channel test section with CHF indicated and inlet pressure of: (a) 1.5 bar (b) 2 bar.	253
7.2	Average two-phase heat transfer coefficient for the 30 mm long channel test section and inlet pressure of: (a) 1.5 bar (b) 2 bar. Points indicated refer to flow patterns seen in Figure 7.3.	253

Figure		Page
7.3	Flow visualisation at the 30 mm long channel outlet for 400 kg/m ² s mass flux, inlet pressure of 1.5 bar and wall heat fluxes of: (a) 140.72 kW/m ² (b) 204.52 kW/m ² (c) 232.13 kW/m ² .	254
7.4	Outlet flow visualisation during rise of wall temperature at CHF (after 234.13 kW/m ²) for the 30 mm long channel at mass flux of 400 kg/m ² s and an inlet pressure of 1.5 bar.	255
7.5	Outlet annular flow visualisation for the single channel, 400 kg/m ² s mass flux and approximately 200 kW/m ² for inlet pressures of: (a) 1.5 bar (b) 2 bar.	257
7.6	Idealised scenario of annular flow with perfectly circular core cross-section.	258
7.7	Liquid film thickness and interfacial shear stress of idealised scenario of annular flow	259
7.8	Comparison between CHF prediction tools and the CHF results for inlet pressure of: (a) 1.5 bar (b) 2 bar.	260
8.1	Typical FNN architecture.	266
8.2	Multiple FNN model architecture schematic.	266
8.3	Use of data set to train multiple FNN and obtain predictions.	268
8.4	K-Fold methodology schematic in which n folds are used (modified from [420]).	270
8.5	Distribution of flow pattern data points.	271
8.6	Dropout regularisation technique.	274
8.7	Performance of initial bubbly-slug transition boundary FNN with different feature subsets.	275
8.8	Performance of initial slug-churn transition boundary FNN with different feature subsets.	276
8.9	Performance of initial churn-annular transition boundary FNN with different feature subsets.	277
8.10	Accuracy from hidden layer configuration and dropout rate grid search for bubbly-slug FNN model.	281
8.11	Standard deviation from hidden layer configuration and dropout rate grid search for bubbly-slug FNN model.	281
8.12	Accuracy from hidden layer activation function and optimiser type grid search for bubbly to slug FNN model.	282
8.13	Standard deviation from hidden layer activation function and optimiser type grid search for bubbly to slug FNN model.	282
8.14	Accuracy from batch size and epochs grid search for bubbly-slug FNN model.	283
8.15	Standard deviation from batch size and epochs grid search for bubbly to slug FNN model.	283

Figure		Page
8.16	Time from batch size and epochs grid search for bubbly to slug FNN model.	284
8.17	Final bubbly to slug transition boundary FNN architecture.	285
8.18	Final bubbly to slug transition boundary FNN performance.	286
8.19	Final slug to churn transition boundary FNN architecture.	287
8.20	Final slug to churn transition boundary FNN performance.	287
8.21	Final churn to annular transition boundary FNN architecture.	288
8.22	Final churn to annular transition boundary FNN performance.	288
8.23	Final classification FNN architecture.	289
8.24	Final classification FNN performance.	289
8.25	Comparison between FNN model predictions and experimental data: (a) at inlet location of channel; (b) at middle location of channel; (b) at outlet location of channel.	290
8.26	Comparison between FNN model predictions with HFE-7100 removed from the training data set and experimental data: (a) at inlet location of channel; (b) at middle location of channel; (b) at outlet location of channel.	292

LIST OF TABLES

Table		Page
2.1	Forces scaled per unit area by Kandlikar [79].	17
2.2	Reported flow patterns for the bubbly flow pattern group from direct diabatic studies.	36
2.3	Reported flow patterns for the intermittent flow pattern group from direct diabatic studies.	42
2.4	Reported flow patterns for the annular flow pattern group from direct diabatic studies.	48/49
3.1	Thermophysical properties of HFE-7100 and DI water at 1 bar and 25 °C.	117
3.2	Measured test section channel dimensions.	130
3.3	Measured Surface Parameters	131
3.4	Uncertainties of measured parameters.	140
3.5	Propagated uncertainties.	141
3.6	Thermophysical properties of HFE-7100 at the examined inlet conditions.	142
3.7	Experimental parameters.	142
4.1	Experimental range of flow pattern map coordinate parameters.	164
6.1	Thermophysical properties of HFE-7100 at saturation.	233
8.1	Brunel Two-Phase Flow database of high-fidelity experimental data sources.	265
8.2	ANOVA F-test rankings for transition boundary FNNs.	272
8.3	Initial hyperparameters for single FNN model.	273
8.4	Features used to tune and train the bubbly to slug FNN transition model.	276
8.5	Features used to tune and train the slug to churn FNN transition model.	278
8.6	Features used to tune and train the churn to annular FNN transition model.	279
8.7	FNN model hyperparameter tuning variables.	280
8.8	Multiple FNN model hyperparameter tuning variables.	284

NOMENCLATURE

Latin

Bd	Bond Number, $\frac{\text{Gravitational Force}}{\text{Surface Tension Force}}, [(\rho_l - \rho_g)gD_h/\sigma]$, [-]
Bo	Boiling number, $\frac{\text{Vapour generated per heat transfer surface area}}{\text{mass flow rate per cross sectional area}}, [q''/Gi_{ig}]$, [-]
C_d	Drag Coefficient, [-]
C_p	Specific heat capacity, [J/kg]
Co	Confinement number, $\frac{\text{Surface Tension Force}}{\text{Gravitational Force}}, [\sqrt{\sigma/g(\rho_l - \rho_g)/D_h}]$, [-]
CO	Distribution parameter, [-]
D	Diameter, [-]
D_h	Hydraulic diameter, [m]
F	Force, [N]
Fr	Froude number, $\frac{\text{Inertia Force}}{\text{Gravitational Force}}, [U/\sqrt{gD_h}]$, [-]
Fr_{gs}	Superficial vapour Froude number, $[U_{gs}/\sqrt{gD_h}]$, [-]
Fr_{gs}^*	Modified Froude number, $[U_{gs}\sqrt{\rho_g/g(\rho_l - \rho_g)D_h}]$, [-]
Fr_{lo}	Liquid only Froude number, $[G^2/gD_h\rho_l^2]$, [-]
f	Fanning friction factor, [-]
G	Mass flux, [kg/m ² s]
g	Acceleration due to gravity, [m/s ²]

H	Height, [m]
h	Heat transfer coefficient, [W/m ² K]
i	Enthalpy, [J/kg]
i_{fg}	Latent heat of vaporisation, [J/kg]
Ja	Jakob number, $\frac{\text{Sensible heat}}{\text{Latent heat}}, [C_p \Delta T / i_{fg}]$, [-]
Kn	Knudson number, $\frac{\text{Mean Free Path}}{\text{Hydraulic Diameter}}$, [-]
k	Thermal Conductivity, [W/mK]
L	Length, [m]
La	Laplace number, $\frac{\text{Surface tension force}}{\text{Gravitational force}}, [C_p \Delta T / i_{fg}]$, [-]
l_c	Characteristic length scale, $\sqrt{\sigma / g(\rho_l - \rho_v)}$, [m]
$\dot{m}$	Mass flow rate, [kg/s]
Nu	Nusselt number, $\frac{\text{Convection Resistance}}{\text{Conductive Resistance}}, [hD_h / k]$, [-]
P	Pressure, [Pa]
Pr	Prandtl number, $\frac{\text{Momentum diffusivity}}{\text{Thermal diffusivity}}, [C_p \mu / k]$, [-]
Q	Heat, [W]
q''	Heat Flux, [W/m ²]
Re	Reynolds number, $\frac{\text{Inertia Force}}{\text{Viscous Force}}, [\rho U D_h / \mu]$, [-]
Re_{go}	Vapour-only Reynolds number, $[G D_h / \mu_g]$, [-]
Re_{gs}	Superficial vapour Reynolds number, $[G x D_h / \mu_g]$, [-]
Re_{lo}	Liquid-only Reynolds number, $[G D_h / \mu_l]$, [-]

Re_{ls}	Superficial liquid Reynolds number, $[G(1 - x)D_h/\mu_l]$, [-]
R_p	Reduced pressure, $[P_{in}/P_c]$, [-]
r	Radius, [m]
S	Suppression factor, [-]
T	Temperature, [K]
t	Time, [s]
U	Velocity, [m/s]
U_{gs}	Superficial gas or vapour velocity, [m/s]
U_{ls}	Superficial liquid velocity, [m/s]
W	Width, [m]
We	Weber Number, $\frac{Inertia\ Force}{Surface\ Tension\ Force}$, $[\rho U^2 D_h/\sigma]$, [-]
We_{lo}	Liquid-only Weber number, $[G^2 D_h/\rho_l \sigma]$, [-]
We_{ls}	Superficial liquid Weber number, $[G(1 - x)^2 D_h/\rho_l \sigma]$, [-]
We_{go}	Vapour-only Weber number, $[G^2 D_h/\rho_g \sigma]$, [-]
We_{gs}	Superficial vapour Weber number, $[(Gx)^2 D_h/\rho_g \sigma]$, [-]
x	Vapour quality, [-]
x^*	Penetration depth, [m]
ϕ_{pl}	Plenum diameter, [-]

Greek Symbols

α	Void fraction, [-]
β	Aspect ratio, [-] or half the cavity cone angle, [rad]

β^*	Adjusted aspect ratio, $[\max (H_{ch}/W_{ch}, W_{ch}/H_{ch})]$, [-]
ΔP	Pressure drop, [Pa]
ΔT	Temperature difference, [K]
δ	Thermal boundary layer thickness, [-]
ε_{ξ}^z	Phase change number, [-]
θ	Contact angle, [rad]
κ	Thermal diffusivity, $[\text{m}^2/\text{s}]$
μ	Viscosity, $[\text{Pa}\cdot\text{s}]$
v	Specific volume, $[\text{m}^3/\text{kg}]$
ρ	Density, $[\text{kg}/\text{m}^3]$
σ	Surface tension, $[\text{N}/\text{m}]$
ϕ	Two-phase pressure drop multiplier, [-]
X	Martinelli Parameter, $\frac{\text{Liquid frictional pressure drop}}{\text{Gas frictional pressure drop}}, [\sqrt{(dP/dz)_l/(dP/dz)_g}]$, [-]
ψ	Cavity mouth angle, [rad]

Subscript

acc	Acceleration
b	Bubble
$base$	Base
by	Buoyancy
c	Critical
cb	Convective boiling
ch	Channel

<i>col</i>	Column
<i>cu</i>	Copper
<i>d</i>	Departure
<i>du</i>	Unsteady drag
<i>drift</i>	Drift
<i>e</i>	Effective
<i>ent</i>	Entry
<i>exp</i>	Experimental
<i>f</i>	Fluid
<i>fric</i>	Frictional
<i>g</i>	Gas or vapour
<i>gr</i>	Gravity
<i>h</i>	Homogeneous
<i>heat</i>	Heated
<i>i</i>	Inertial
<i>in</i>	Inlet
<i>l</i>	Liquid
<i>loss</i>	Lost
<i>m</i>	Measured
<i>n</i>	Nucleate boiling
<i>out</i>	Outlet
<i>p</i>	Evaporation momentum

<i>pl</i>	Plenum
<i>qs</i>	Quasi-steady
<i>r</i>	Relative
<i>sc</i>	Sudden contraction
<i>se</i>	Sudden expansion
<i>sl</i>	Shear lift
<i>sp</i>	Single-phase
<i>sub</i>	Subcooled
<i>sup</i>	Superheat
<i>th</i>	Thermocouple
<i>top</i>	Top
<i>tp</i>	Two-phase
<i>ts</i>	Top Surface
<i>w</i>	Wall
<i>P</i>	Pressure
μ	Viscous
σ	Surface tension
<i>90</i>	90° change in direction
∞	Ambient

Abbreviations

AF	Annular Flow
AI	Artificial Intelligence

ANN	Artificial Neural Network
ANOVA	Analysis of Variance
AOK	Adaptive Optimal Kernel
BJT	Bipolar Junction Transistor
CAD	Computer Aided Design
CB	Coalescing Bubble
CFD	Computational Fluid Dynamics
CHF	Critical Heat Flux
CNN	Convolutional Neural Network
CPU	Central Processing Unit
DAQ	Data Acquisition System
DI	Deionised
DNB	Departure from Nucleate Boiling
DL	Deep Learning
EDL	Electrical Double Layer
FFT	Fast Fourier Transform
FNN	Feed-Forward Neural Network
GNN	Genetic Neural Network
GPU	Graphics Processing Unit
HCPV	High Concentration Photovoltaic
HFE	Hydrofluoroether
HTC	Heat transfer coefficient

IB	Isolated Bubble
ID	Internal Diameter
IGBT	Insulated Gate Bipolar Transistor
IT	Information Technology
KNN	K-Nearest Neighbours Network
MAE	Mean Absolute Error
ML	Machine Learning
MOFSET	Metal Oxide Semiconductor Field Effect Transistor
ONB	Onset of Nucleate Boiling
PID	Proportional Integral Derivative
PNN	Probabilistic Neural Network
PTFE	Polytetrafluoroethylene
SOM	Self-Organising Map
TFA	Time frequency Analysis

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND AND MOTIVATION

Many engineering applications now require an efficient and reliable method to successfully dissipate large amounts of heat from very small areas [1, 2]. Such applications include Central Processing Units (CPUs), Graphics Processing Units (GPUs), memory cards, data storage devices, Insulated Gate Bipolar Transistors (IGBTs), switch-mode power supplies, laser diode arrays, proton exchange membrane fuel cells and miniature vapour-compression refrigeration systems [3]. Furthermore, the ever-growing concern of climate change has led to the electrification of many systems that conventionally rely on fossil fuels, such as the use of High Concentration Photovoltaic (HCPV) systems for power generation [4–6]. This concern has also nurtured an interest in finding more energy efficient cooling solutions for existing technologies. High-heat flux, energy efficient cooling solutions that have been proposed involve the enhancement of surfaces for single-phase liquid flows, pool boiling, two-phase jet impingement and microscale flow boiling [7].

Enhancing the heat transfer rates in single-phase liquid flows has been investigated to increase heat transfer efficiency. Mousa et al. [8] separated the single-phase enhancement techniques into two categories; namely, passive methods and active methods. Techniques include tube inserts, baffles, fins, surface modification, use of electric fields, magnetic fields and fluid vibration. Both active and passive single-phase enhancement methods have been found to provide improved heat transfer rates, but with significant penalties in pressure drop [9–13]. A significant rise in pressure drop would require that the cooling system pumping power is increased, thus limiting the returns on potential improved system energy consumption. Moreover, the improved heat transfer rates are still lower than other proposed cooling solutions [14].

Pool boiling possesses the advantage of being able to passively dissipate higher heat fluxes than single-phase immersion or forced single-phase convection/internal flow techniques [15]. In pool boiling, the substrate that requires cooling is immersed in liquid and the heat flux generated by the substrate results in boiling of the liquid phase. This dissipation occurs with lower wall superheats that are more capable of keeping temperatures within permissible operating limits. However, limitations on

this technology exist as the boiling regime transitions from nucleate boiling to film boiling at higher heat fluxes, resulting in a sudden and significant reduction in heat transfer rates [16]. A typical boiling curve is presented in Figure 1.1, which plots the substrate wall heat flux against the wall superheat, defined as the difference in temperature between the wall surface and the saturation temperature of the working fluid. The consequent vapour that develops through boiling eventually provides a blanket to the heated surface, resulting in Critical Heat Flux (CHF). The lower thermal conductivity of the vapour phase results in significantly lower heat transfer coefficients and causes the wall temperature to rise significantly.

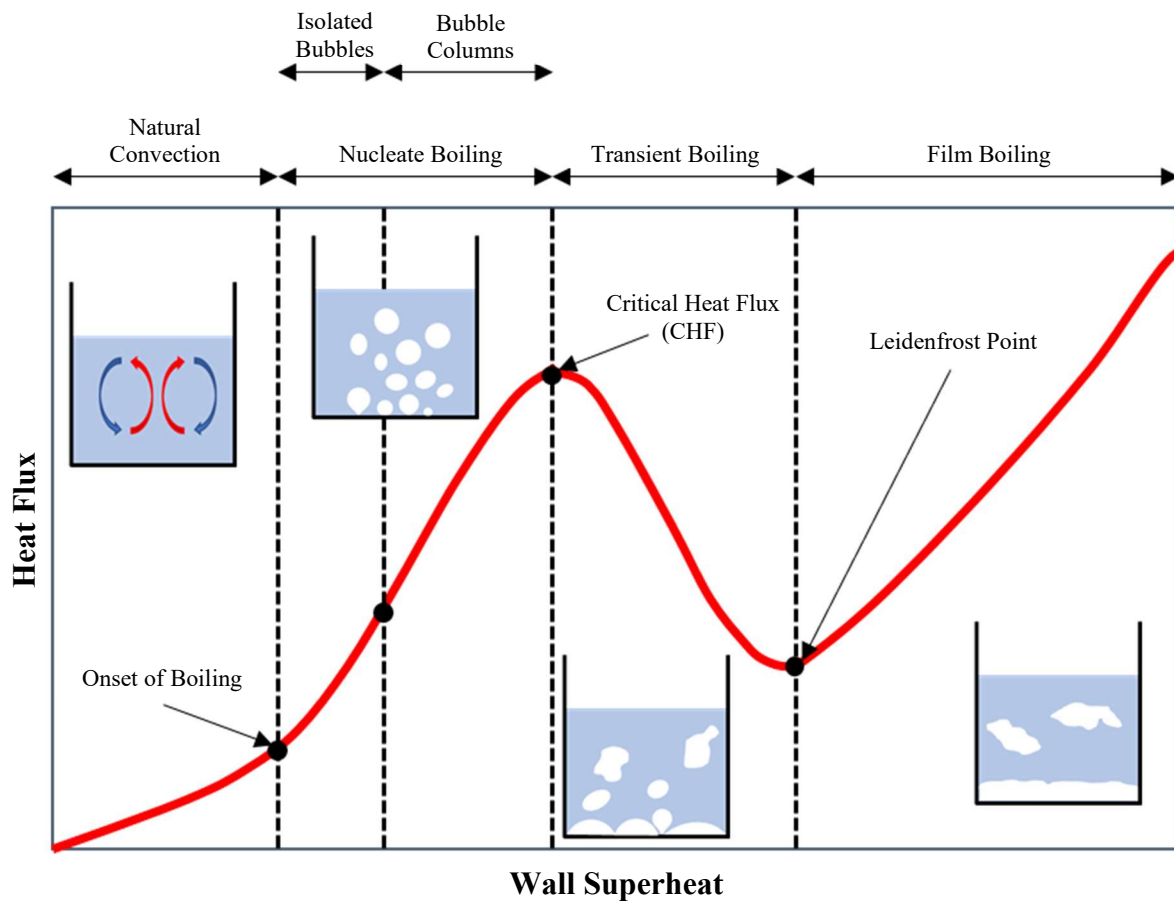


Figure 1.1. Typical pool boiling curve.

Two-phase jet impingement was divided into three categories by Ji et al. [17]; namely, free surface jet boiling, submerged jet boiling and confined jet boiling. The three categories of two-phase jet impingement are shown diagrammatically in Figure 1.2. Confined jet impingement has been found to generally provide the greatest heat transfer rates whilst also maintaining a compact system size [18]. Two-phase jet impingement systems can dissipate very high heat fluxes; however, limitations include complexities in system performance due to the high number of parameters upon which cooling performance is dependent [19]. Furthermore, other issues in implementing two-phase jet impingement

include integrating designs into compact electronic systems and erosion of components due to the high velocities associated with the technique [18, 20].

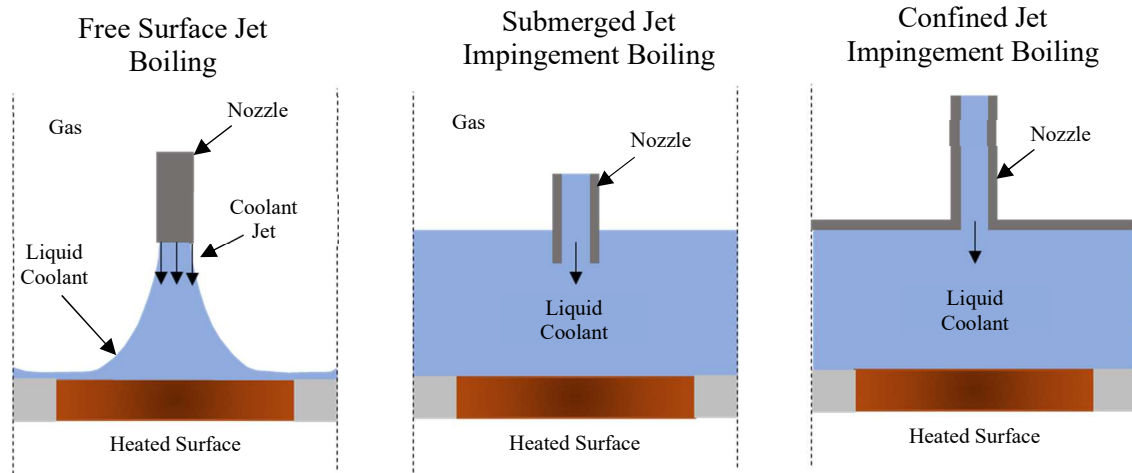


Figure 1.2. Categories of two-phase jet impingement.

Flow boiling in microchannel heat sinks is a promising solution to the heat transfer problem due to the high heat transfer area to flow area ratio, as well as the utilisation of latent heat [3, 21, 22]. This methodology involves attaching the substrate that requires cooling to a microchannel heat sink, where the generated heat flux results in boiling of the flowing liquid phase that enters the heat sink. A resultant two-phase mixture then exits the heat sink. An example diagrammatic assembly of how a microchannel heat sink may be used to cool an IGBT module is shown in Figure 1.3. Despite being a promising solution to the heat transfer problem, many fundamental issues still exist that require elucidation prior to general adoption by industry. These issues include understanding of dominant heat transfer mechanisms, mechanisms of critical heat flux and a lack of universal design criteria prediction tools [23].

Micro evaporators using microscale flow boiling can be used in a pumped loop system, such as the diagram of the system that is shown in Figure 1.4. The system would typically consist of a reservoir, micro-gear pump, the micro-evaporator and a micro-condenser. In such an arrangement, the key parameters required for design are the heat transfer rates in the micro-evaporator and micro-condenser, as well as the pressure drop across these components and, consequently, the entire flow loop. Knowledge of these parameters would ensure that the micro-evaporator design is capable of dissipating the required heat flux, the condenser is capable of supplying the pump with single-phase liquid to avoid cavitation and to ensure the gear pump is capable of providing a sufficient flow rate to the evaporator. Therefore, reliable universal correlations are required to determine the heat transfer and pressure drop across the micro-evaporator.

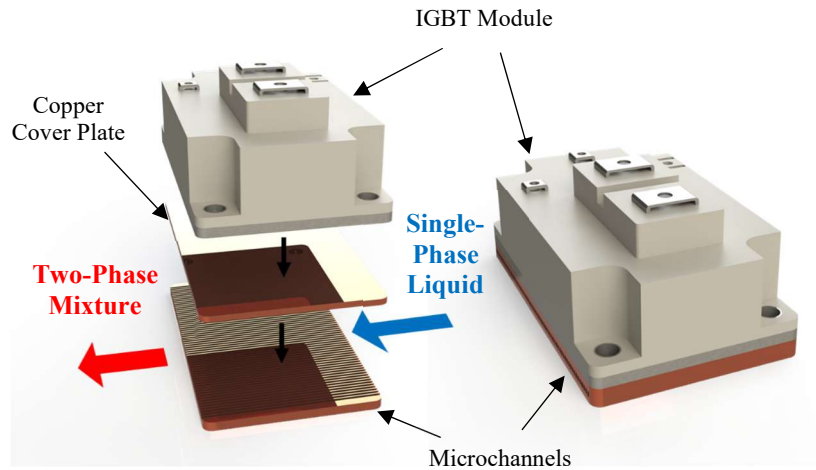


Figure 1.3. Example diagrammatic assembly of microchannel heat sink for an insulated gate bipolar transistor module.

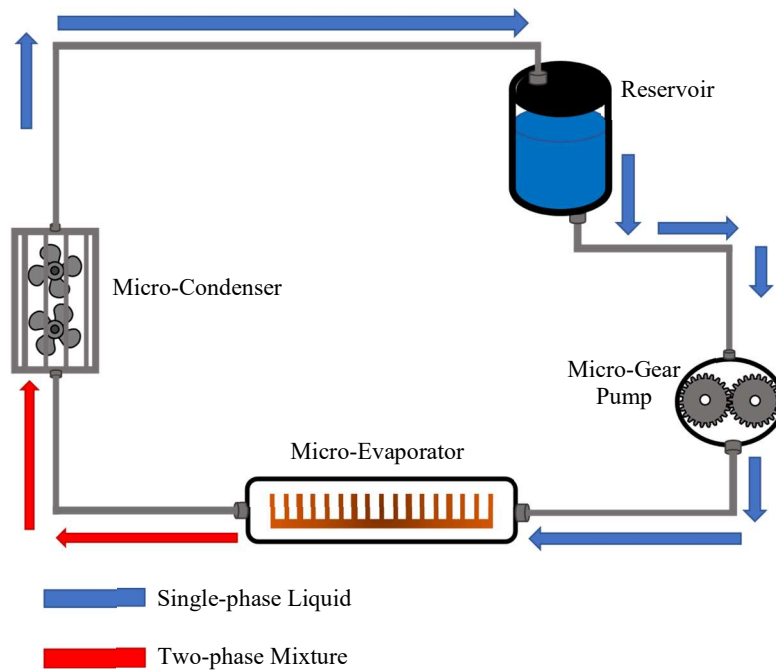


Figure 1.4. Pumped system loop for micro-evaporator device.

Provided that clarification is achieved on key microchannel flow boiling aspects, there is potential for its use in multiple applications. The most common of these is in the Information Technology (IT) industry, where thermal limits are a current barrier to progress [24]. Gordon Moore predicted that the number of transistors in an integrated circuit would double in system-on-chip devices every year in 1965, which was amended to doubling once every two years in 1975 [25]. This exponential growth in the number of transistors results in an exponential growth of computing power, where a proportion of the power is dissipated as heat. Consequently, the high-power density of such systems gives rise to high

heat flux generation. Two examples to show the evolution of heat flux generation in computing system are the IBM System/370 mainframe, introduced in 1970, and the Intel® Core™ i9-12900K processor, introduced in late 2021. The IBM System/370 mainframe produced 32,550 BTU/hr (9545.5 W) of heat with a cooling area of 15575 cm², thus providing a heat flux of 6.1 kW/m² (0.61 W/cm²) [26, 27]. Conversely, the Intel® Core™ i9-12900K processor produces 241 W of heat over a heat dissipation area of 2.2 cm² with a consequent heat flux of 1.12 MW/m² (112 W/cm²), a 18,570% increase in heat flux when compared to the IBM System/370 mainframe [28]. Heat fluxes in the order of 1 MW/m² are similar to that produced by a nuclear blast; although, the permissible operating temperatures of chipsets are significantly lower [29]. Typically, consumer electronics are required to be kept below a maximum operating temperature of 85°C and below 175°C for both silicon carbide-based power modules and gallium nitride semiconductors [30].

Proof that flow boiling in microchannels is a viable solution to cooling heat fluxes in the order of 1 MW/m² exists in many publications, with some of these publications reaching heat fluxes in the order of 10 MW/m² [31–37]. The ability to reach heat fluxes over 10 MW/m² is particularly significant, owing to the fact that semiconductors were predicted to reach heat fluxes up to 4.6 MW/m² by 2025 [38]. However, modern projections for next-generation semiconductor devices, especially wide and ultrawide-bandgap power electronics, indicate power densities and heat fluxes exceeding 10 MW/m² [39]. Not only have microchannel heat sinks been proven to possess the possibility to adequately cool these devices, but they have also shown the potential to provide a solution that will hold longevity in the IT industry. An example assembly of a microchannel heat sink that may be used for cooling an integrated circuit chip is shown in Figure 1.5.

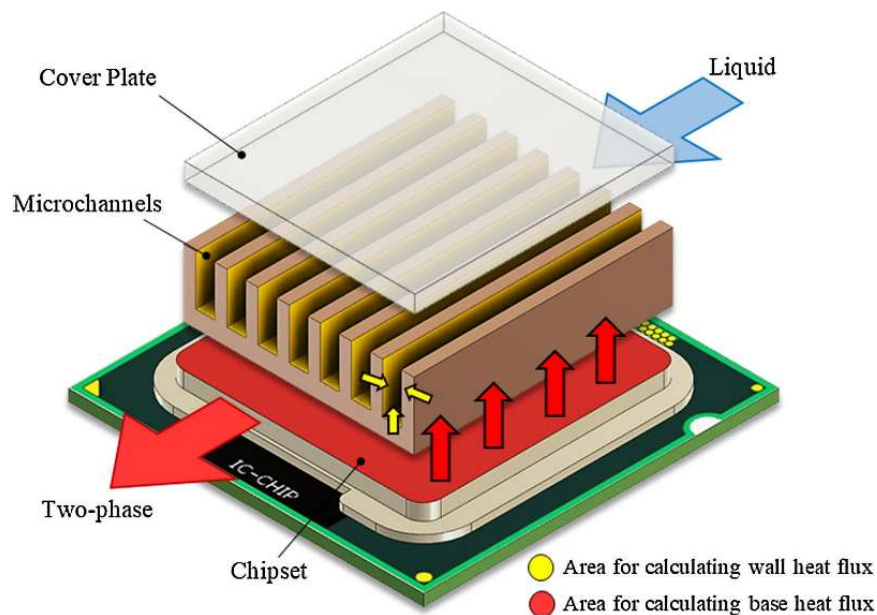


Figure 1.5. Drawing of microchannel heat sink for cooling integrated circuit chips (taken from [40]).

Data centres are an ever-increasing application in the IT industry that have particularly high associated cooling loads. In 2024, global data centres consumed approximately 415 TWh of electricity, around 1.5 % of total global electricity use, and this figure is projected to more than double by 2030 due to continued digitalisation and the expansion of Artificial Intelligence (AI) workloads, potentially approaching nearly 3 % of total global electricity consumption by the end of the decade [41]. Recent breakthroughs in Artificial Intelligence (AI) and, in particular, Artificial Neural Networks (ANNs) have increased the requirements for data centres and lead to predictions that the global carbon emissions from data centres will reach 14% of the total global carbon emissions by 2040 [42]. Energy consumption in data centres comprises of five main areas; namely, building requirements, site power system losses, transmission/ distribution losses, IT equipment and the cooling system. The IT equipment and cooling system are by far the biggest consumers of energy, comprising of 44% and 40% of the total energy consumption, respectively [43]. Hence, the implementation of more energy efficient cooling technologies would significantly reduce the overall consumption of data centres and provide a sizeable reduction in global energy consumption in the future.

Typically, flow boiling solutions require smaller mass flow rates than single-phase flows for a given load due to the exploitation of latent heat. Consequently, significant energy can be saved in relation to the associated pumping loads. The utilisation of flow boiling in the application of data centres would therefore be most beneficial in applications that currently use single-phase liquid flows for cooling. Single-phase liquid cooling is now being used for direct on-server cooling. An example server tray, the Lenovo ThinkSystem SD650, is displayed in Figure 1.6, where Figure 1.6 (a) shows 6 trays in a rack unit and Figure 1.6 (b) illustrates the liquid-cooling configuration of a single tray housing two servers. The CPUs within the server are kept within permissible operating temperatures via water cooled cold plates, where water enters and exits through the inlet and outlet ports on the right side of the image. On-server cooling applications, such as cold plates, could therefore benefit substantially from flow boiling due to the associated reduction in required mass flow rate and, consequently, lower pumping power demands.

IGBTs were introduced in the 1980s as an alternative to bipolar junction transistors (BJTs) and metal-oxide-semiconductor field effect transistors (MOSFETs) for use in power supply and motor control circuits [44]. Typically, BJTs have long switching times, requiring complex circuits and large associated power losses. Conversely, MOSFETs have high switching speeds but possess poor conduction characteristics at high voltages, ultimately also resulting in high switching losses [45]. The IGBT is capable of switching speeds comparable to a MOSFET at high voltages. Figure 1.7 presents example applications in which IGBT modules are utilised with their typical associated voltage and current ratings.

Many modern applications depend heavily on IGBTs; technologies that are essential to the process of global electrification such as smart grids, electric trains and other electric vehicles. However, the

high voltage, quick switching of an IGBT module results in exceptionally high heat generation. As such, generated heat fluxes up to 5 MW/m^2 are imminently expected from IGBT modules [46]. Once again, flow boiling in microchannels has been proven to show capabilities of achieving heat dissipation rates in orders of such magnitudes. It is therefore important that certain aspects of microscale flow boiling are clarified on to enable to further electrification and further decarbonisation of both industrial and commercial systems.

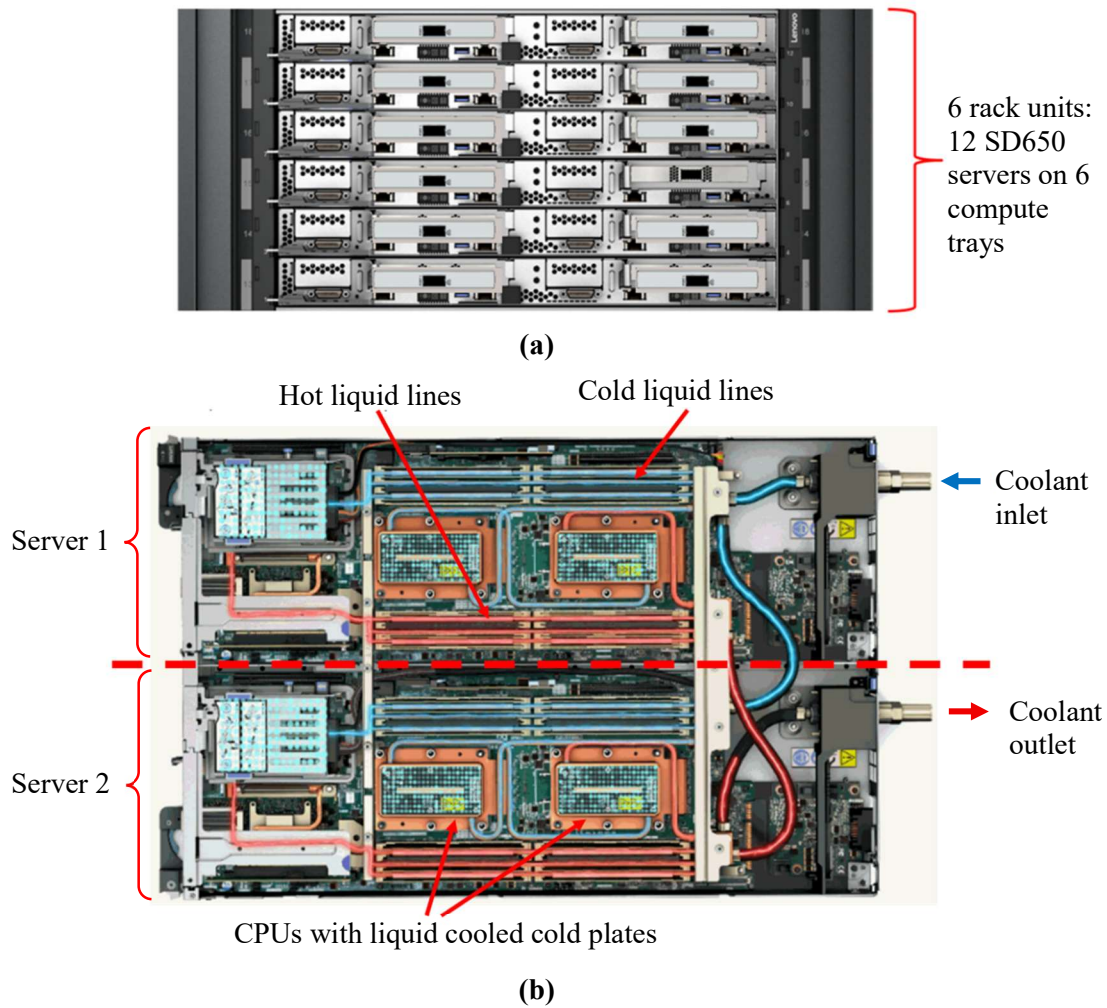


Figure 1.6. Water cooled Lenovo ThinkSystem SD650 direct water-cooled server: (a) front view of enclosure with 6 trays; (b) water path in SD650 tray (modified from [47]).

One of the greatest challenges to reducing global carbon emissions is to drastically reduce the use of fossil fuels for electricity generation. The review of world energy sources published by BP in 2022 showed that fossil fuels accounted for 82% of global energy consumption in 2021 [48]. The actual quantity of fossil fuels burned for energy generation was similar to that of two years prior, despite a higher percentage of fossil fuels being used for global energy consumption in 2019 (83%). This is a result of the ever growing overall global energy demand. The combustion of fossil fuels for the purpose

of energy is fundamentally connected to the total global greenhouse gas and carbon dioxide emissions, which the International Renewable Energy Agency suggests need to be reduced by approximately 70% by the year 2050 if targets set by the Paris agreement are to be met [49]. Therefore, a transition to renewable energy sources will greatly aid in reduction of global carbon emissions.

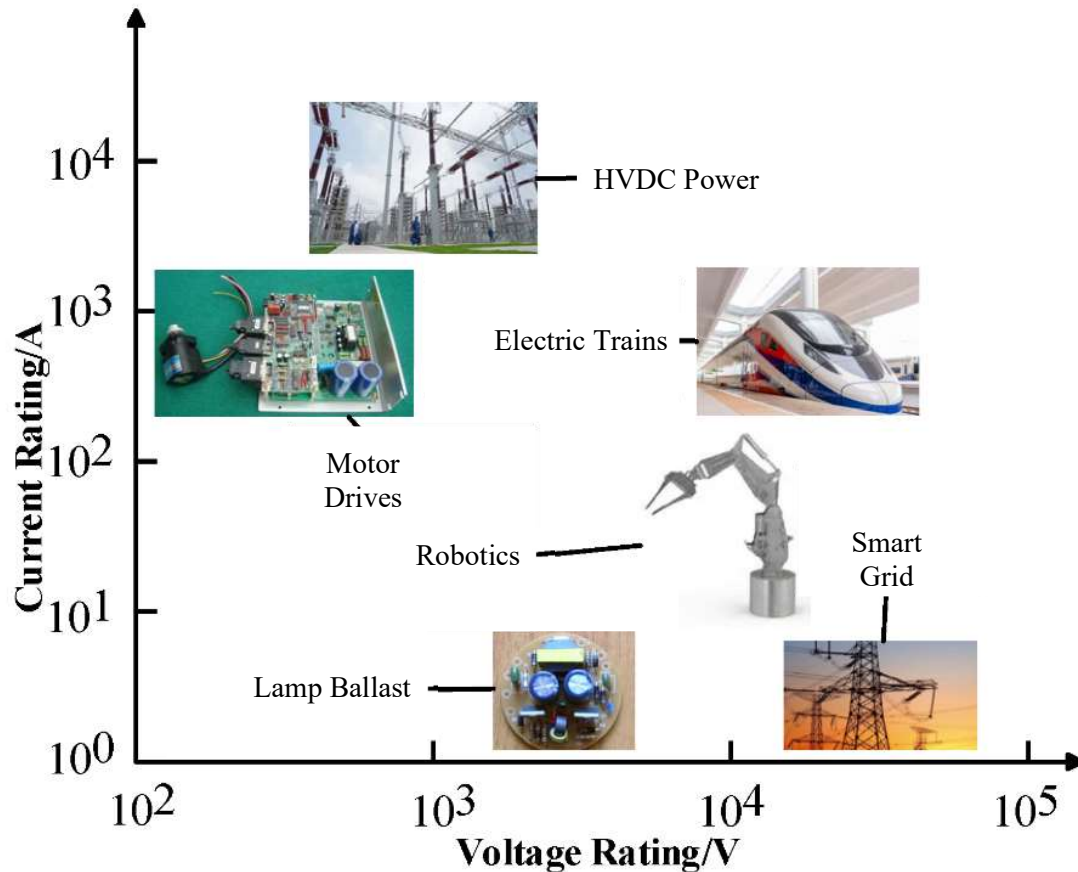


Figure 1.7. Applications of IGBT modules with their associated voltage and current ratings (modified from [50])

Currently, solar photovoltaic systems are the fastest growing contributor to global energy renewable generation; contributing approximately 60% of global renewable energy growth as of May 2022 and 77.8 % in 2025 [51, 52]. HCPV panel systems are emerging as one of the most efficient photovoltaic systems and can currently reach standard efficiencies of 40% [53]. HCPV systems focus the solar rays onto a small cell area through an entrance aperture, unlike conventional photovoltaic systems which simply use the direct solar rays. This concept is shown diagrammatically in Figure 1.8. The process of concentrating the solar rays through a parabolic disk concentrator and aperture may result in very high temperatures generated on the photovoltaic module. This increase in temperature has the potential to drastically reduce the efficiency of such systems. Hence, a sufficient cooling solution is required to utilise the full potential of HCPV systems. Flow boiling in a microchannel heat sinks may serve as a

viable solution and permit the development of such systems and renewable energy sources as a viable solution to reducing global carbon emissions. The solution would be to mount the microchannel heat sink to the photovoltaic module to reduce the module temperature and maintain system efficiency.

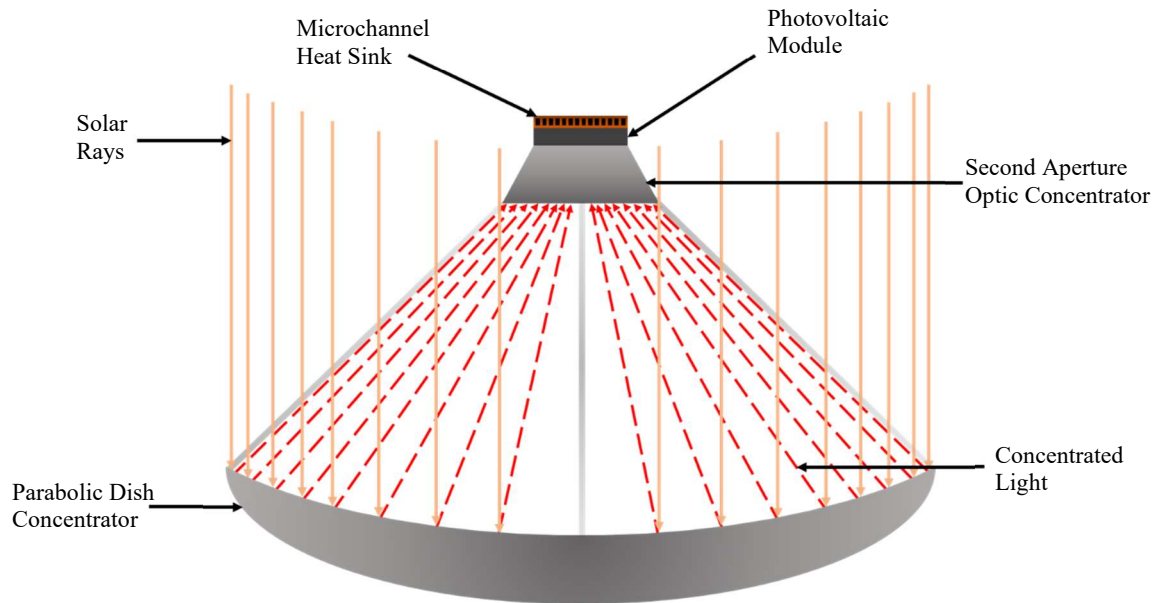
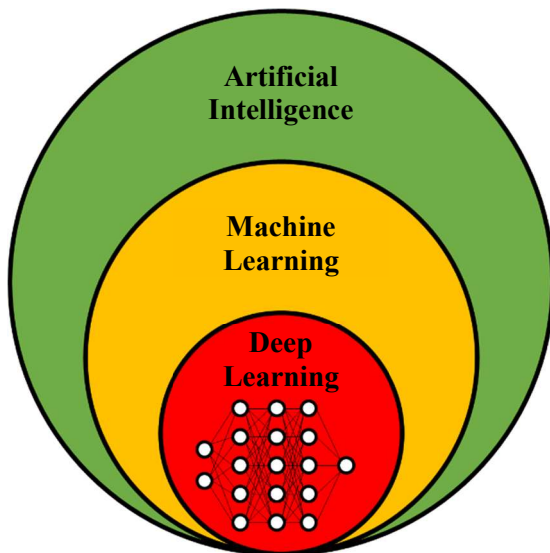


Figure 1.8. Diagram of high concentration photovoltaic system for solar energy conversion with microchannel heat sink cooling.

The increased power of modern computers has enabled Machine Learning (ML) to be used as a tool in many existing industries such as petrochemicals [54–56], weather forecasting [57–60], robotics [61–64] and almost every other aspect of existing technological solutions [65]. This has led to much interest from both academia and the general public; however, it is important to understand what ML is and how it differs from the terms AI and Deep Learning (DL), which are often incorrectly and superficially used interchangeably in all forms of media [66]. Figure 1.9 shows a Venn diagram of artificial intelligence, machine learning and deep learning. This Venn diagram shows that machine learning is a subset of artificial intelligence, whilst deep learning is a subset of machine learning. Beginning with AI, it can be defined as a branch of computer science that relates to the development of computing machines to undertake functions that would conventionally require human levels of intelligence [67]. This is not necessarily a new branch of computer science and has been used for decades. The first subset of AI is machine learning, which focusses on finding patterns in empirical data, i.e. the computer is able to perform the functions without the need for explicit programming [68]. Finally, the subset of machine learning is deep learning, which is defined as a class of machine learning algorithms that are based on artificial neural networks [69].



Artificial Intelligence- A branch of computer science focussing on developing computing machines to complete tasks that require human level intelligence

Machine Learning- A branch of artificial intelligence that completes tasks without the need for explicit programming.

Deep Learning- A branch of machine learning that utilises artificial neural networks.

Figure 1.9. Venn diagram of artificial intelligence, machine learning and deep learning.

Many names have been applied to neural networks, including parallel distributed processing models, ANNs, neurocomputers and connectionist models [70]. Ultimately, a neural network is an assembly of interconnected processing elements referred to as units or nodes. Each processing element is a computational entity that receives one or more inputs, combines them through weighted connections, applies an activation function, and produces an output that is passed to other elements within the network. The function of the processing element takes loose inspiration from the animal neuron, abstracting key features such as signal aggregation and activation [71]. The processing capabilities of the network are stored in the weights that connect nodes, where the weights are modified through an adaptation or learning process in which the network ‘learns’ from training patterns. This concept is displayed in Figure 1.10, which shows a simplified neural network architecture. Input data is fed into the first layer where it is processed through the network with the results fed into the output layers where predictions are made.

Neural networks are capable of identifying complex non-linear patterns within a training data set and making predictions [72]. This differs from ‘classical’ statistical methodologies used to develop correlations in heat transfer, as empirical correlations are typically based on the interdependencies between parameters [73]. The implementation of neural networks therefore creates an advantage as it is not so reliant on the interdependencies between these parameters. Hence, their use may provide a significant design tool and aid in the implementation of flow boiling in the aforementioned applications in this section.

Overall, the motivation of this work is to aid in the understanding of fundamental phenomena that are essential to implement microscale flow boiling in the many applications that require high heat flux and/or energy efficient cooling solutions. As such, a single microchannel test section provides the most

suitable experimental means in which to observe fundamental mechanisms due to the absence of flow maldistribution that can occur in multi-channel heat sinks. The current understanding of such phenomena is reviewed in Chapter 2 and areas in which further understanding is required are identified.

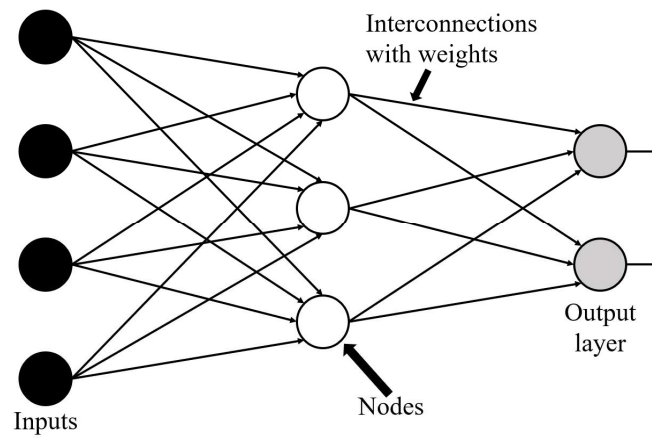


Figure 1.10. Simplified neural network architecture.

The working fluid selected for this study was HFE-7100, a hydrofluoroether dielectric coolant that has gained interest for electronics cooling and microscale boiling applications due to its low toxicity, chemical stability, electrical non-conductivity, and relatively low saturation temperature. In comparison with conventional refrigerants such as R-134a and newer low-global-warming-potential refrigerants such as R-1234ze(E), HFE-7100 generally exhibits lower latent heat and thermal conductivity, but offers advantages in dielectric performance, which makes it particularly advantageous in electronics cooling applications. These aspects make it useful for investigating fundamental microscale flow boiling behaviour in applications where direct-contact cooling and electrical insulation are important considerations.

1.2 RESEARCH OBJECTIVES

Despite substantial progress in the understanding of microscale flow boiling, several important challenges remain unresolved. These include the absence of a universally accepted definition of the microscale, inconsistencies in flow pattern classification, limited agreement among heat transfer and pressure drop correlations and the lack of broadly applicable CHF prediction tools. Furthermore, although newly emerging machine learning tools are being applied to a range of scientific applications, there remains the potential for their use in microscale heat exchanger design.

These unresolved issues present the need for further experimental investigations that utilise flow visualisation with heat transfer, pressure drop, and CHF measurements over a range of operating conditions. In addition, there is a need to assess the possibility of developing machine learning tools in a microscale flow boiling context.

Accordingly, the objectives of the present study into microscale flow boiling are summarised as follows:

1. Investigate the fundamental physical mechanisms that occur during microscale flow boiling using heat transfer and pressure drop measurements. Use a high-speed camera for flow visualisation and investigate the effects of flow pattern geometry on heat transfer rates, pressure drops and critical heat flux.
2. Assess the accuracy of flow pattern boundary transition prediction tools using both new experimental data and existing flow boiling data.
3. Evaluate the effect of important operating parameters on flow pattern transition boundaries, heat transfer rates and pressure drops. Operating parameters include the mass flux, heat flux and inlet pressure.
4. Examine the effect of fluid thermophysical properties on flow pattern transition boundaries, heat transfer rates and pressure drops.
5. Investigate the effect of heated length on flow pattern transition boundaries, heat transfer rates and pressure drops for flow boiling in a single microchannel.
6. Analyse the success of existing heat transfer and pressure drop prediction tools using the new experimental data. Furthermore, to conduct an evaluation of the relevant parameters and methodologies utilised to develop such correlations in relation to the new experimental data.
7. Evaluate CHF mechanisms through flow visualisation, evaluate CHF prediction tools and provide operating limits in which flow boiling in microchannels can be utilised.
8. Develop a new deep learning design tool capable of predicting prevailing flow patterns during microscale flow boiling.

1.3 THESIS STRUCTURE

The thesis structure is as follows:

Chapter 1- Introduction- This introductory chapter presents applications that require high heat flux cooling solutions and the proposed technologies to meet such demands. An argument as to why flow boiling in microchannels is the most promising solution is made and examples of how it can be used in new technologies that reduce global carbon emissions or how it may reduce carbon emissions in existing technologies are presented. Finally, a discussion on how artificial intelligence is pervading many aspects of modern engineering is provided and could be applied to microscale flow boiling. The research objectives of this thesis are then clearly defined.

Chapter 2- Literature Review- The existing literature is explored and critiqued in this chapter, with a particular focus on microscale flow boiling patterns, heat transfer rates and pressure drop. The review is conducted for all microscale flow geometries, i.e. horizontal and vertical tubes, rectangular single channels and multi-channel heat sinks. The effect of fluid thermophysical properties, operating parameters (heat flux and mass flux) and channel geometry are investigated and previous use of machine learning in the application of microscale flow boiling are analysed.

Chapter 3- Experimental Methodology- The initial experimental facility is shown and a systematic explanation of the modifications made to the facility presented. The calibration processes and experimental procedures are clearly described. The design of the microchannel test sections and the evolution of designs are explained. Full data reduction is presented with propagated uncertainty analysis. Finally, single-phase validation and repeatability results are discussed.

Chapter 4- Flow Boiling Patterns- The flow visualisation results from experiments are presented here with explanation of the mechanisms of flow pattern transitions explained. The effect of fluid thermophysical properties, mass flux, heat flux and heated length are presented here. Finally, flow pattern results are plotted on flow pattern maps and data is compared to existing flow pattern correlations.

Chapter 5- Flow Boiling Heat Transfer- The heat transfer performance of the test sections are reported, with analysis beginning at boiling incipience. The effect of fluid thermophysical properties as they vary with channel inlet pressure, mass flux, heat flux and heated length on heat transfer rates is investigated and analysed. The heat transfer mechanisms are considered in relation to the developed interfacial phenomena presented in the previous chapter. The success of existing heat transfer correlations are also examined in this section.

Chapter 6- Flow Boiling Pressure Drop- The pressure drop results are presented and analysed in relation to the flow patterns discussed in Chapter 4. The effect of fluid thermophysical properties, mass

flux, heat flux and heated length on two-phase pressure drop is investigated and the success of existing two-phase pressure drop correlations examined.

Chapter 7- Critical Heat Flux- This chapter focusses on the induced critical heat flux in a test section with shorter length. The mechanisms through which critical heat flux occurs are investigated and existing prediction tools compared to the new experimental data.

Chapter 8- Predicting Flow Patterns Using an Artificial Neural Network- A new artificial neural network is devised to predict flow pattern transitions in microscale flow boiling; developed using a large flow pattern data bank from Brunel University of London. This includes transitions for vertical circular tubes, for horizontal circular tubes, for horizontal single-channels and for horizontal multi-channel heat sinks. The algorithm performance was compared to existing experimental data and new data obtained from the experimental facility described in Chapter 3.

Chapter 9- Conclusions and Recommendations- The key findings of the study are presented in the final chapter. These key findings are then used to form some recommendations on future research into microscale flow boiling and also on design considerations for thermal designers who wish to utilise microscale flow boiling as a cooling solution.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

This chapter presents existing work on flow boiling in the microscale. It begins with existing definitions of the microscale in relation to flow boiling. The three fundamental aspects of microscale flow boiling are presented, i.e. the prevailing flow patterns, the resultant heat transfer and finally, the two-phase pressure drop. Literature on the CHF of microscale heat exchanger devices is then presented and new, novel work that implements the tool of machine learning to flow boiling is also reviewed. Finally, concluding remarks are provided on each of the topics in Section 2.8.

2.2 DEFINING THE MICROSCALE

The discussion of flow boiling in the microscale is further clouded by the lack of a universal definition of the microscale and how this differs from the conventional “macroscale”. Terms such as “small”, “mini” and “micro” have all been used to describe flows where features differ to flow in conventional sized channels. Some publications use several terms to consider such disparate flow characteristics. This further hinders the heat transfer community’s ability to develop suitable prediction models in the microscale due to varying dimensional limits at which correlations are applicable. Several contradictory transition criteria have been presented by researchers as a consequence of the lack of universal microscale definition.

Mehendale et al. [74] provided an early definition of the microscale specifically relating to single-phase liquid flows. The transition was proposed to occur at a sufficiently low hydraulic diameter where an electrical double layer (EDL) may drastically affect heat transfer rates. They argued that the electrostatic charge of the solid surface may cause balancing counterions in the fluid to establish an EDL, even if there are very few ions in the working fluid. The counterions that establish at the channel wall become near immovable due to the surface’s electric field, forming a compact layer approximately 0.5 nm thick. Ions in the diffuse double later, which can be up to several hundred nanometres in thickness, are able to move and cause a streaming current to flow in the same direction as the bulk fluid flow. An accumulation of ions downstream causes a conduction current in the opposing direction and

attracts the liquid, affecting the bulk fluid flow properties and heat transfer rates. This phenomenon is shown diagrammatically in Figure 2.1.

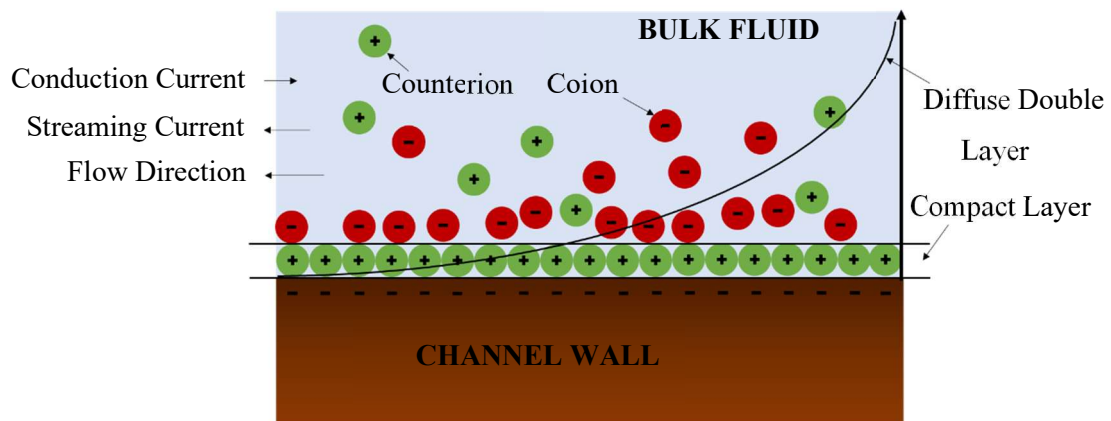


Figure 2.1. Diagram of EDL at the channel wall (adapted from Mehendale et al. [74]).

In conventionally sized “macro” scale channels, such electrostatic forces are deemed insignificant compared to any motion of the bulk fluid. Mehendale et al. [74] proposed that the electrostatic forces would become significant where hydraulic diameters are below $100\ \mu\text{m}$. Consequently, a microscale transition of $<100\ \mu\text{m}$ was suggested with an accompanying meso-scale between $100\ \mu\text{m}$ and $1\ \text{mm}$, a compact scale between $1\ \text{mm}$ and $6\ \text{mm}$ and the conventional scale above $6\ \text{mm}$.

Kandlikar and Grande [75] proposed that the definition of microscale should consider the ease of manufacture of the heat exchanger, operational requirements for cleanliness (i.e. a significantly reduced fouling resistance) and the rarefaction effects of gasses. The Knudsen number, Kn , is defined as the ratio of the molecular mean free path of the working fluid gas phase to the hydraulic diameter of the channel. Kandlikar and Grande proposed that continuum mechanics and no slip assumptions are no longer valid in the slip region of $0.1 > Kn > 0.001$. They argued that general microscale criteria should be considered for all flows, i.e. single-phase liquids, single-phase gasses and two-phase flows, despite the methodology mainly stemming from gas-flow considerations. A microscale criterion was suggested by comparing the molecular mean free path of air, helium, hydrogen and nitrogen at atmospheric pressure to the Knudsen numbers in the slip region, resulting in a microscale criterion of $10\ \mu\text{m} > D_h > 200\ \mu\text{m}$ and an intermediate size of minichannels defined as $200\ \mu\text{m} > D_h > 3\ \text{mm}$. A similarity between the two microscale criteria proposed by Mehendale et al. and Kandlikar and Grande is that they are wholly dependent on the channel hydraulic diameter, i.e. the geometry of the channel.

Kawaji and Chung [76] also presented an approach that only considers the hydraulic diameter of the channel, estimating that the transition from the mini to micro scale occurs at hydraulic diameters between $100\ \mu\text{m}$ and $250\ \mu\text{m}$. However, the definition was wholly through consideration of two-phase

flows. Adiabatic experiments were conducted in which single-phase nitrogen was mixed with single-phase deionised (DI) water and flowed through circular and square channels with hydraulic diameters of $99.6 \mu m$ and $95.6 \mu m$, respectively. The basis for this criterion was the reported alteration in observed flow patterns at lower hydraulic diameters, where there is no longer an appearance of churn flow or annular flow and previously unseen “liquid ring” and “liquid lump” flow patterns are observed. The contrast between the flow patterns observed in Kawaji and Chung’s mini and microscales are displayed in Figure 2.2.

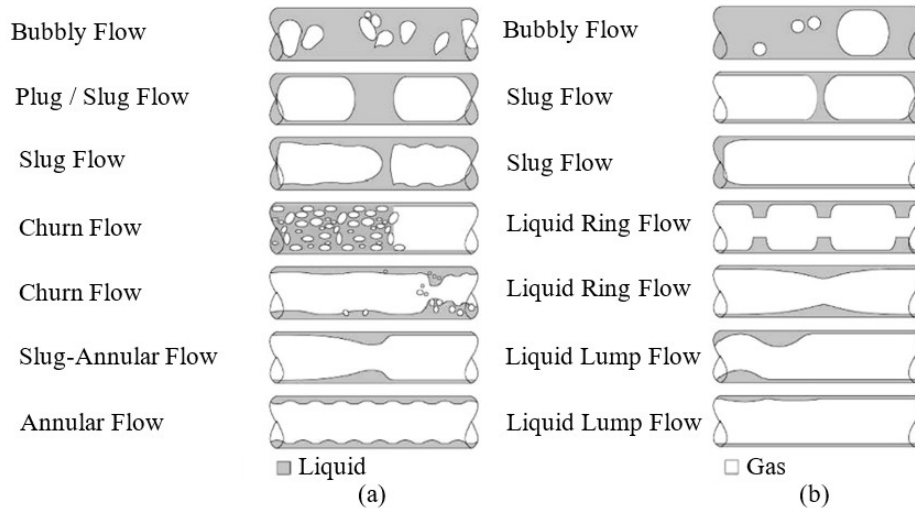


Figure 2.2. Drawings of observed flow patterns by Kawaji and Chung [76] in the (a) miniscale and (b) microscale.

The approach of defining the microscale through an observed change in flow patterns is one taken by other researchers when considering two-phase flows. However, a limitation of the criterion proposed by Kawaji and Chung is the lack of consideration of any fluid thermophysical properties that affect two-phase flow patterns and subsequently, heat transfer and pressure drop characteristics. Kew and Cornwell proposed a microscale transition criterion based on the degree of bubble confinement and the consequent effects on heat transfer characteristics [77]. The confinement number, denoted by Co and defined by Kew and Cornwell in a prior publication [78], was used to quantify the degree of flow confinement. It relates the buoyancy to surface tension forces and is defined in Equation. (2.1), where D_h , σ , g , ρ_l and ρ_g represent the hydraulic diameter, the surface tension, the acceleration due to gravity and the liquid vapour densities, respectively.

$$Co = \frac{1}{D_h} \sqrt{\frac{\sigma}{g(\rho_l - \rho_g)}} \quad (2.1)$$

The buoyancy and surface tension forces, as well as the inertia, viscous and evaporation momentum forces were scaled by Kandlikar [79]. The forces were scaled both per unit characteristic length and by characteristic area, with the latter presented in Table 2.1. Plots of the five scaled forces with respect to unit area and changing hydraulic diameters are displayed in Figure 2.3 for both HFE-7100 and DI water. Plots were developed using a constant mass flow rate of 1 g/s and a wall heat flux of 120 kW/m². Previously undefined parameters in Table 2.1 include the mass flux, G , the liquid phase viscosity, μ_l , the heat applied to the fluid, Q , and the latent heat of vaporisation, i_{fg} .

It is noticeable that the buoyancy force is the only of the five forces to decrease with a decrease in hydraulic diameter. Comparing HFE-7100 to water, the intersection of the buoyancy force with the other forces, in particular the surface tension force, occurs at significantly higher hydraulic diameters for water than it does for HFE-7100. However, it can be seen that the effect of the buoyancy forces become significantly reduced at lower hydraulic diameters. This suggests that the influence of hydraulic diameter on the forces governing two-phase flow is fluid dependent, with buoyancy becoming dominant over surface tension at lower hydraulic diameters for HFE-7100 than for water. Therefore, if microscale behaviour is linked to changes in two-phase flow patterns, criteria defining the microscale should consider the fluid thermophysical properties that govern these forces.

Table 2.1. Forces scaled per unit area by Kandlikar [79].

Inertia Force	Surface Tension Force	Viscous Force	Buoyancy Force	Evaporation Momentum Force
$F_i = \frac{G^2 D_h}{\rho_l}$	$F_\sigma = \sigma$	$F_\mu = \frac{\mu_l G}{\rho_l}$	$F_{by} = (\rho_l - \rho_g) g D_h^2$	$F_p = \frac{\left(\frac{Q}{i_{fg}}\right)^2 D_h}{\rho_l}$

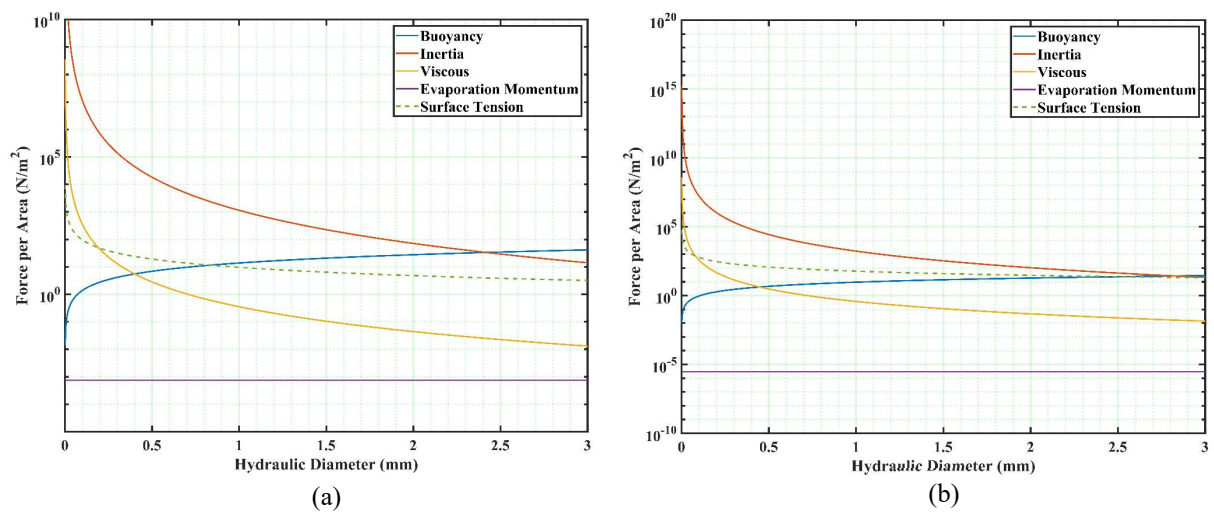


Figure 2.3. Scaled forces per unit area: (a) for HFE-7100; (b) for DI water.

Kew and Cornwell [77] observed that heat transfer correlations for conventional channels poorly predicted heat transfer rates for confinement numbers greater than 0.5 due to the suppression of the buoyancy force and the consequent enhanced role of other forces on flow patterns.

Ong and Thome [80] conducted flow boiling experiments for refrigerants R-134a, R-236fa and R-245fa in channels of 1.03 mm, 2.20 mm and 3.04 mm diameter. A microscale transition criterion was proposed based on flow pattern observations, where only isolated bubble, coalescing bubble and annular flow regimes were observed. The flow patterns of isolated bubble and coalescing bubble depart from the classical designations presented in macroscale flows to eliminate problems with subjectivity and are discussed in Section 2.4 in more detail. The criteria also considered film thickness uniformity for coalescing bubble and annular flows, where a uniform thickness was observed for $Co > 1$, indicating a full suppression of the buoyancy forces and the transition to the microscale. Tibiri   and Ribatski [81] also argued that a uniform film thickness due the suppression of buoyancy forces is a distinctive characteristic of microscale flows and defined the microscale where the scaled gravitational force is less than 5% of the scaled surface tension force, as displayed in Equation (2.2) and Equation (2.3).

$$\frac{F_{by}}{F_{\sigma}} = \frac{(\rho_l - \rho_g)gD_h^2}{\sigma} < 0.05 \quad (2.2)$$

$$D_h < \sqrt{\frac{\sigma}{20(\rho_l - \rho_g)g}} \quad (2.3)$$

Tibiri   and Ribatski [81] also proposed a transition criterion based on the absence of stratified flows in microchannels. A model of a static plug of saturated liquid separated by vapour slugs in a circular channel was considered. They argued that the threshold diameter was at the maximum diameter for which a static liquid plug could exist. At this maximum diameter, it was argued that the hydraulic head pressure, F_p , is in equilibrium with the surface tension force at the triple contact line between the channel wall, the liquid phase and the vapour phase. This scenario is shown in Figure 2.4 and the resultant transition criterion is given in Equation (2.4).

$$D_h = \sqrt{\frac{8\sigma\cos\theta}{g(\rho_l - \rho_g)}} \quad (2.4)$$

The aforementioned microscale transition criteria for DI water, R-134a, FC-72 and HFE-7100 are presented in Figure 2.5. Evident discrepancies between microscale transition definitions can be seen in Figure 2.5, due to the criteria being based on varying phenomenological observations to define the

microscale. The influence of an EDL and the slip condition where continuum mechanics are no longer valid are considerations that should be made for flows in hydraulic diameters below 0.1 mm and 0.2 mm, respectively. The importance of fluid properties is clearly seen in Figure 2.5, since water experiences confinement effects at much larger hydraulic diameters.

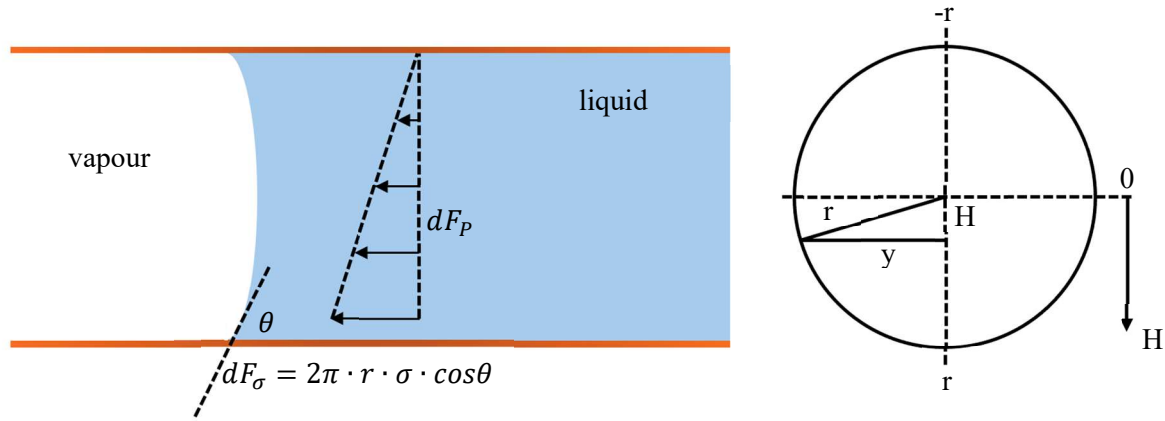


Figure 2.4. Model of static liquid plug (adapted from Tibiriçá and Ribatski [81]).

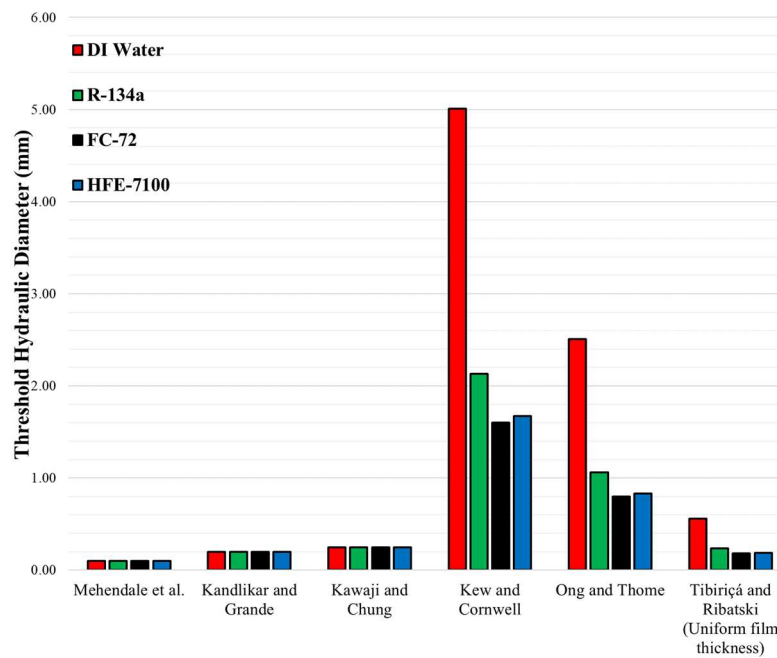


Figure 2.5. Microscale criteria threshold hydraulic diameters for different fluids.

However, in the context of this work, the term "microscale flow" refers specifically to flows in which key aspects of flow boiling, such as flow patterns, heat transfer mechanisms, and pressure drop characteristics, are significantly altered due to the reduction in hydraulic diameter. This definition remains problematic, as existing criteria that account for the influence of geometry on two-phase flow behaviour show considerable disagreement. These discrepancies arise because each criterion is based on differing phenomenological observations that occur at varying hydraulic diameters, reflecting

different dominant mechanisms across the microscale regime. For instance, the criterion by Kawaji and Chung represents a transition from the so-called “miniscale” to the microscale, where stratified flow is already absent, making it conceptually similar to those of Ong and Thome, and Tibiri   and Ribatski, who also base their definitions on the disappearance of stratification. Yet, the differences in transition diameter values among these criteria suggest a need for more comprehensive experimental data across a broader spectrum of fluids and channel geometries to reconcile these definitions.

2.3 FLOW BOILING PATTERNS

As discussed in Chapter 1, the ability to adopt microscale flow boiling in industrial applications requires that predictive tools for heat transfer rates and pressure drops are concluded on. The flow boiling heat transfer rates and pressure drops are fundamentally associated with the geometry of the interface between phases described by flow boiling patterns. However, there are significantly fewer research papers published each year on the topic of microscale flow boiling patterns in comparison to heat transfer rates and pressure drops, despite this intrinsic link. This can be seen in Figure 2.6 which shows the number of publications on the topic of microscale flow boiling this century.

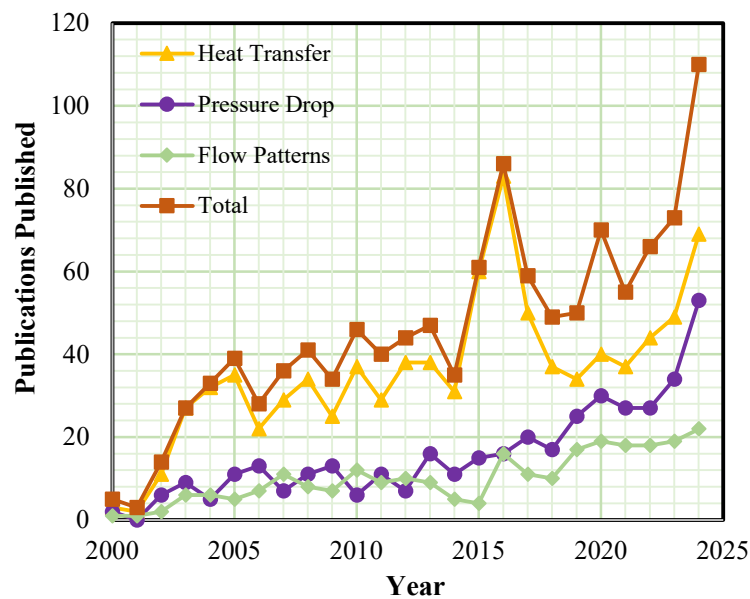


Figure 2.6. Publications on microscale flow boiling topics per year (Scopus search as of March 2025).

The ability to understand both the mechanisms and predict the transitions between the flow boiling patterns will significantly aid in the development of successful heat transfer and pressure drop prediction tools. This is because the clarification of dominant heat transfer and pressure drop mechanisms is limited without an understanding of the prevailing flow patterns and consequently,

correlations will rely solely on semi-empirical fits of data [82]. In spite of this, there are few comprehensive flow pattern maps for microscale flows, where it is agreed that the flow pattern transition models that exist for conventional channels provide poor predictions of transitions in the microscale. Moreover, the small number of current flow pattern maps and models/correlations for determining flow pattern transitions are often limited to particular operating conditions and for a given fluid [21]. These limitations further act as a barrier to their administration as prediction tools in industrial applications, where operation with new and novel fluids may be required.

Attempts to develop universal flow pattern maps are also hindered by an apparent disagreement between the reported flow patterns in the microscale, even when similar operational parameters and fluids are used. Two potential reasons for flow pattern discrepancies include (i) the subjectivity in categorising flow patterns that arises due to visual observation techniques [83] and (ii) the lack of universal terminology that exists when describing flow patterns in the microscale [84]. Therefore, it is important to elucidate generally accepted universal microscale flow pattern designations that would help eliminate the above two issues. It is first important to ascertain what flow patterns are observed in the conventional scale to understand how microscale flow patterns differ.

2.3.1 Flow Patterns in the Conventional/Macro Scale

A wide variety of flow patterns have been identified by various researchers, where flow pattern characteristics have been found to be dependent on fluid thermophysical properties, orientation of flow, channel geometry and whether the flow is diabatic or adiabatic [85–87]. The presence of some reported flow patterns in the microscale is debated, as is if their characteristics deviate substantially from those observed in the conventional scale. As such, it is useful to present clear definitions of flow patterns from older work conducted on macroscale flows as a starting point for discussion.

Flow patterns have been classified in four main groups in the conventional scale [88]. These groups include stratified, bubbly, intermittent and annular flows. Flow regimes in the conventional scale differ depending on flow orientation. The most commonly observed flow patterns that occur inside vertical flows in the conventional scale are shown in Figure 2.7, whilst the flow patterns that occur during horizontal flows in the conventional scale are shown in Figure 2.8, as classified by Collier and Thome [89].

Hewitt provided clear descriptions of the flow patterns shown in Figure 2.7 [90]. Bubbly flow was characterised by a continuous liquid phase where a dispersion of bubbles is present within the liquid. Slug flow was defined as the regime in which the smaller bubbles have coalesced to form larger bubbles with diameters approaching that of the channel. The bubbles that present during slug flow eventually break down and result in an oscillating pattern commonly referred to as churn flow. The flow then

redistributes such that the gas or vapour phase flows through the core of the channel and the liquid phase flows on the channel walls as a film. Some liquid may be entrained in the vapour core. The quantity of liquid phase entrained in the core increases with an increase of liquid phase velocity (assuming a constant vapour or gas velocity). This can result in large liquid droplets in the core and a flow pattern referred to by Collier and Thome [89], Hewitt [90] and Whalley [91] as wispy-annular flow. The mist flow regime was defined by Cheng et al. [87] as a flow pattern in which the vapour velocity is significantly high enough to shear the liquid phase from the wall entirely, eliminating the liquid film.

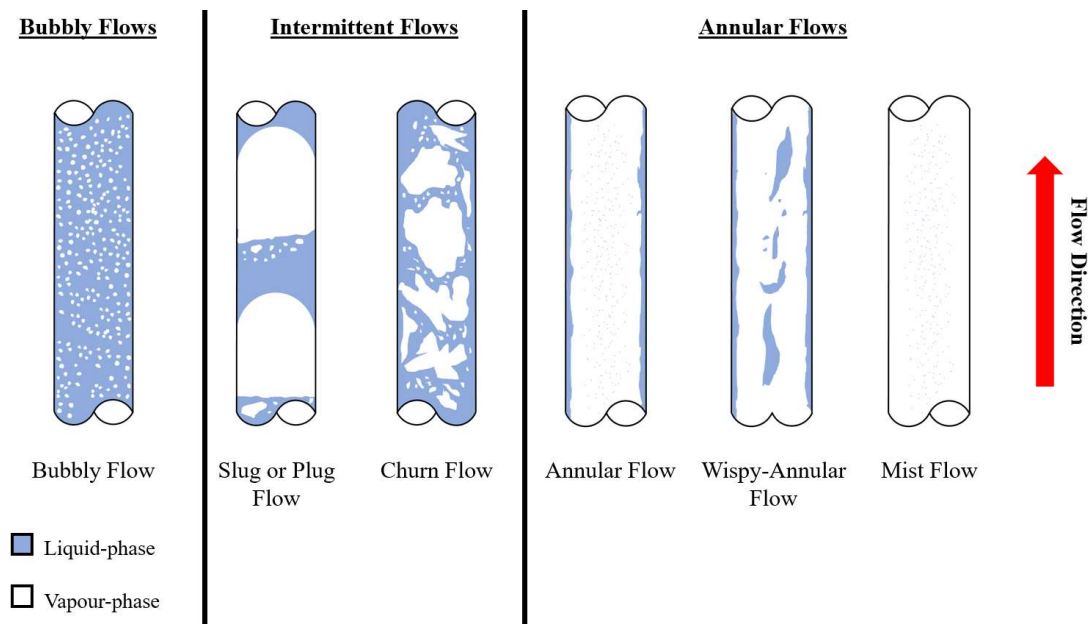


Fig. 2.7. Flow patterns in conventional sized tubes for upwards vertical flows.

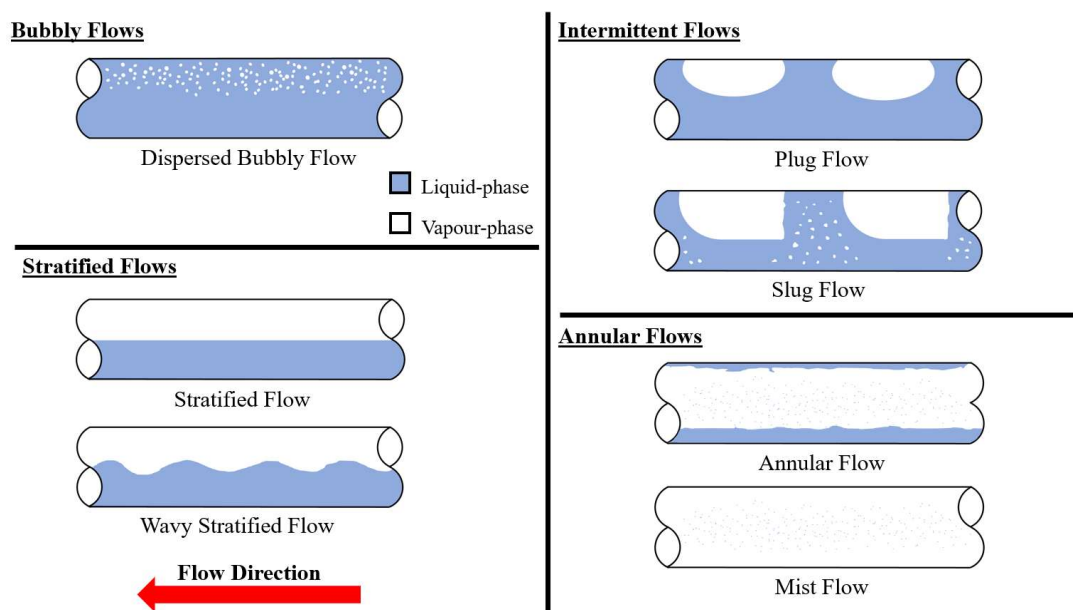


Figure 2.8. Flow patterns in conventional sized tubes for horizontal flows.

Descriptions of the flow patterns present in conventional horizontal flows were also provided by Hewitt [90]. The appearance of stratified flow is the most noticeable difference between the flow pattern diagrams in Figure 2.7 and Figure 2.8. Stratified flow is possible in horizontal channels due to the gravity acting normally to the flow direction and the buoyancy effect acting on the phases resulting in full phase separation. Cheng et al. [87] stated that the wavy stratified flow appears when the gas/vapour velocity is then high enough to shear the liquid at the phase interface of stratified flow. This results in the wavy and disturbed interface observed. Dispersed bubbly flow differs from bubbly flow in that the bubbles are more spherical and travel with a similar phase velocity to the liquid phase. The role of gravity means that these are often distributed at the top of the channel in horizontal flows. Plug flow occurs when bubbles coalesce and the consequent larger bubbles flow along the top wall of the channel. This is differentiated from the slug flow that forms in horizontal tubes, which is a result of the transition from wavy-stratified flow. As the difference in phase velocities increases, the amplitudes of the resultant waves in wavy-stratified flow increase and a liquid slug is formed that flows along the top of the channel. Finally, annular flow remains in horizontal flows; however, the gravitational effects cause a discrepancy in liquid film thickness at different heights of the channel, as seen in Figure 2.8.

Finally, it should be noted here that the term “intermittent flows” has been used to describe the group of flow patterns that lie in between bubbly and annular flows, such as slug, plug and churn flows by some researchers [92–95]. Alternatively, it has been used as a term to interchangeably identify the previously defined slug or plug flows, without the inclusion of churn flow [96–98]. The definition of the term “intermittent flows” shall be further clarified where used in the remainder of this work.

2.3.2 Flow Patterns in the Microscale

2.3.2.1 *Experimental Techniques*

Flow patterns have been studied extensively in macro-scale channels since the 1950's through visual techniques [87]. A number of traditional and novel techniques have been employed to investigate two-phase flow patterns [99]. Traditional techniques include identifying flow patterns using high speed photography or videos. Novel techniques include the use of multi beam gamma densitometry, such as by Tjugum et al. [100], who utilised a configuration in which a fan-beam collimated 500 mCi ^{241}Am radiation source was placed opposite semiconductor radiation detectors with a pipe in between. The measured intensity of the radiation was then able to provide cross sectional information on the phase distributions and consequent flow patterns. Mosdorf et al. [101] used a novel technique in which two-phase flow patterns were identified using transient pressure drop signals at a rate of 1 kHz. X-ray imaging of channel cross sections have also been used in multiple studies as a method of flow pattern identification [102, 103]. A neutron radiography system was implemented by Takenaka et al. [104] to visualise flow patterns, where a proton beam was irradiated to the test section and the neutron flux measured to be converted to a visible image by a scintillation counter.

Ultimately, all two-phase experimental methodologies can be classified into three distinct groups, regardless of flow pattern identification techniques. These groups are as follows:

1. The adiabatic methodology: Two-phase flows are manufactured by injecting a single-phase gas into a single-phase liquid flow. The two-phase mixture may then proceed through a calming section before it is observed in a clear viewing section or through other novel techniques.
2. The indirect diabatic methodology: Single-phase subcooled or saturated liquid is pumped into a heated section of the test facility where flow boiling occurs. The flow then typically enters a clear viewing section that is assumed adiabatic and the flow is visualised. An example of this type of test section is shown diagrammatically in Figure 2.9.
3. The direct diabatic methodology: Single-phase subcooled or saturated liquid is pumped into a test section where the fluid is concurrently heated and visualised.

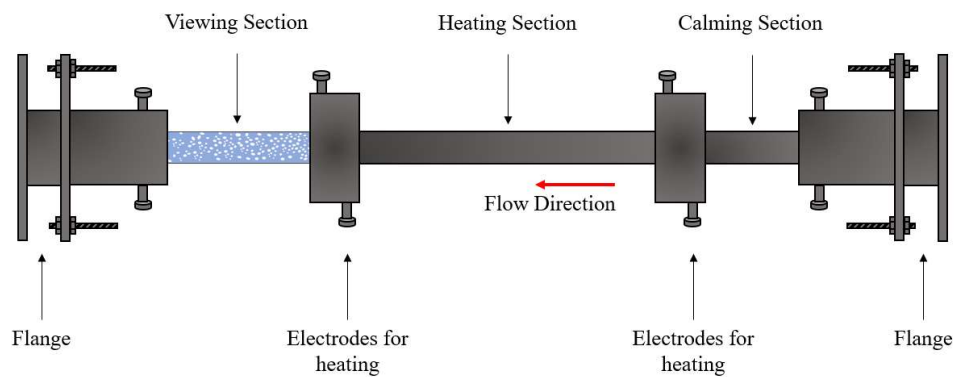


Figure 2.9. Example diagram of typical indirect diabatic test section.

2.3.2.2 *Adiabatic Methodology Studies*

Early studies on two-phase microscale flow patterns typically utilised the adiabatic methodology. One of the earliest studies on microscale two-phase flow patterns was conducted by Suo and Griffith [105] on horizontal capillary tubes. The term ‘capillary tube’ is widely used to describe a tube with a high length to radius ratio [106]. Suo and Griffith [105] studied capillary tubes with internal diameters of 1.28 mm and 1.59 mm for water-nitrogen, heptane-helium and heptane nitrogen mixtures. They observed suppression of stratified flow in the channel and reported only bubbly slug, slug and annular flows.

The bubbly slug flow pattern described by Suo and Griffith [105] was defined as a flow pattern in which the tails of the vapour bubbles would break up and smaller vapour bubbles would fill the space of the liquid slugs. This is similar to the diagram of slug flow presented in Figure 2.8, which shows flow patterns in conventional sized tubes for horizontal flows. Conversely, the reported slug flow included only the presence of vapour bubbles confined by the channel and divided by liquid slugs without the presence of smaller bubbles.

A later study was conducted by Biswas and Greenfield [107], where water-air and distillate-air mixtures in vertical downward flow capillary tubes were investigated using tubes with internal diameters between 0.5 mm and 7.1 mm. They reported dispersed bubbly flow, slug flow, annular flow and stratified flow. Stratified flow was reported to occur at low superficial liquid velocities and low superficial gas velocities. The superficial phase velocities are defined as the mean velocity of the individual phases and are presented in Equation (2.5) and Equation (2.6) for the superficial liquid velocity and the superficial gas velocity, respectively, where x represents the vapour quality.

$$U_{ls} = \frac{G(1-x)}{\rho_l} \quad (2.5)$$

$$U_{gs} = \frac{Gx}{\rho_g} \quad (2.6)$$

The presence of stratified flow appears highly counter-intuitive due to its absence in conventional vertical flows and the buoyancy force being the primary force causing stratified flow. Furthermore, analysis of the scaled forces presented in Figure 2.3 shows that the effect of the buoyancy force diminishes with a reduction in hydraulic diameter and is a primary factor in the definition of microscale.

A similar study was conducted by Galbiati and Andreini [108] for water-air mixtures in vertical capillary tubes between 0.5 mm to 2 mm in diameter and only slug and annular flow patterns were reported. The authors commented on the work conducted by Biswas and Greenfield, arguing that the stratified flow observed by Biswas and Greenfield was a result of an insufficient calming section and hence concluded that the mixing and calming sections were integral to the accurate classification of two-phase flow patterns. This explanation appears reasonable, as the increased effect of the surface tension forces at lower diameters in comparison to the other forces may inhibit mixing of the phases without other destructive forces. This would not be true at higher superficial phase velocities as the increased inertial and viscous forces would be sufficient in magnitude to destroy the phase interface and eliminate separated stratified flows.

Fukano and Kariyasaki [109] also studied water-air mixtures in capillary tubes with internal diameters of 1 mm, 2.4 mm, 4.9 mm and 9 mm for vertical upward flows, vertical downward flows, and horizontal flows. They reported that separated flows (i.e. stratified flows) were not observed in capillary tubes less than 5 mm and consequently, that flow orientation played an insignificant role in determining flow patterns in this scale. They reported bubbly, intermittent and annular flow regimes in all capillary tube orientations. In this context, the intermittent flow regime was defined by the authors as the periodic alternation of gas plugs or slugs and liquid slugs. For horizontal flows in smaller diameter tubes, the intermittent regime replicated the plug and slug flow patterns observed in conventional-sized

vertical channels, as shown in Figure 2.7. In larger diameter tubes, the intermittent flow corresponded to the plug and slug flow observed in conventional horizontal channels, as shown in Figure 2.8.

Barajas and Panton [110] conducted adiabatic experiments using water-air mixtures in a 1.6 mm internal diameter horizontal tube, to assess the effect of tube wettability on the two-phase flow patterns. Wettability was varied by employing tubes manufactured from different materials, namely Pyrex, polyethylene, polyurethane and fluorinated ethylene propylene, while maintaining the same tube diameter and working fluids across all experiments. The wettability of each material was quantified through static contact angle measurements performed on flat samples of the tube materials, with three materials classified as partially wetting (contact angles of 34° , 61° and 74°) and one as partially non-wetting (106°). The reported flow patterns were wavy stratified, plug, slug, annular, bubbly, dispersed bubbly and two new flow patterns which were designated rivulet and multiple rivulet flows. Rivulet flow was defined as a predominantly gas flow with a stream of liquid that flows on the tube surface, not in a straight line but twisting, that may appear on the bottom, middle or top of the tube. Multiple rivulet flow was defined where more than one stream occurred. The reported rivulet and multiple rivulet flows are displayed diagrammatically in Figure 2.10. Both flows replaced the reported wavy stratified flow pattern shown in Figure 2.8 for lower wetting (higher contact angle) surfaces. The experimental facility did not contain a calming section, much like the work conducted by Biswas and Greenfield [107], which may provide an explanation for the appearance of the wavy-stratified and rivulet flows in adiabatic microchannel two-phase studies.

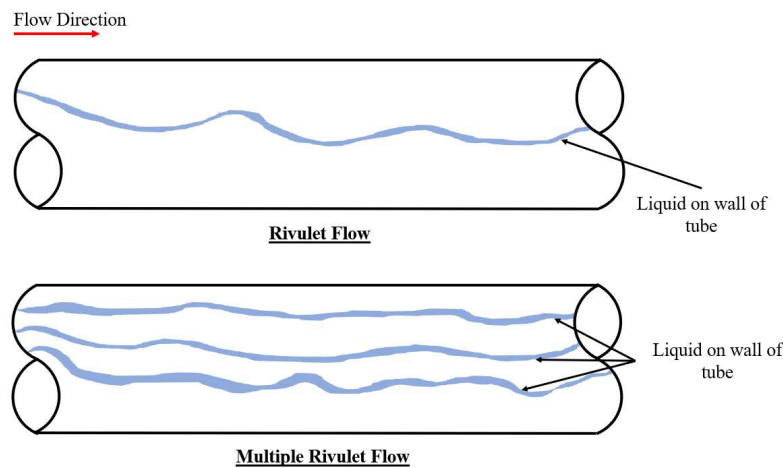


Figure 2.10. Rivulet flows reported by Barajas and Panton (Modified from [110]).

Mishima and Habiki [111] examined air-water flows in vertical capillary tubes between 1 and 4 mm in internal diameter. They reported bubbly, slug, churn, annular and annular-mist (mist) flows; although, differing characteristics of the flow were all categorised under the same flow pattern. For example, what was described as dispersed flow by Biswas and Greenfield [107] and Barajas and Panton [110] was simply categorised under the generic term of bubbly flow. Triplett et al. [112] and Coleman

and Garimella [113] also utilised adiabatic methodologies to study water-air mixtures in horizontal channels between 1.097-1.337 mm and 1.3-5.5 mm, respectively. Triplett et al. [112] reported flow pattern categorisations of bubbly, slug, churn, slug-annular and annular flows, whilst Coleman and Garimella [113] reported bubbly, dispersed, elongated bubble, slug, wavy-stratified, wavy-annular and annular flows. The flow patterns reported by Triplett et al. [112] and Coleman and Garimella [113] are presented in Figure 2.11 and Figure 2.12, respectively. Coleman and Garimella [113] divided the flow patterns into four flow regime categories named stratified, intermittent, annular and dispersed. Figure 2.12 presents the grouping of the flow patterns into the regime categories.

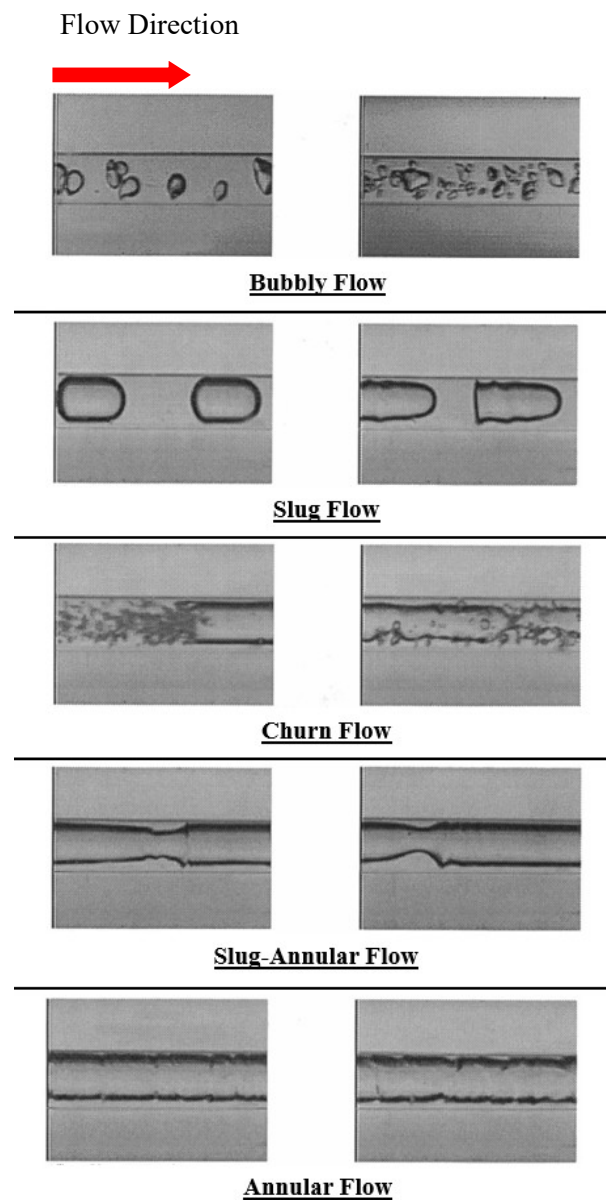


Figure 2.11. Flow patterns reported by Triplett et al. [112].

Triplett et al. [112] reported that the slug-annular flow regime occurred at lower superficial liquid velocities and higher consequent volumetric fluxes, where the volumetric flux is defined as the sum of the superficial velocities, $U_{gs} + U_{ls}$. This led to longer bubbles that merged non-destructively as shown in Figure 2.11. The wavy-annular flow regime reported by Coleman and Garimella [113] only occurred due to a transition from the wavy-stratified flow pattern. The liquid phase was pushed up against the tube walls by the larger gaseous inertia that occurred at higher mass flow rates. The liquid phase then fell downwards as a result of gravity and the resultant flow pattern shown in Figure 2.12 was labelled as wavy-annular flow.

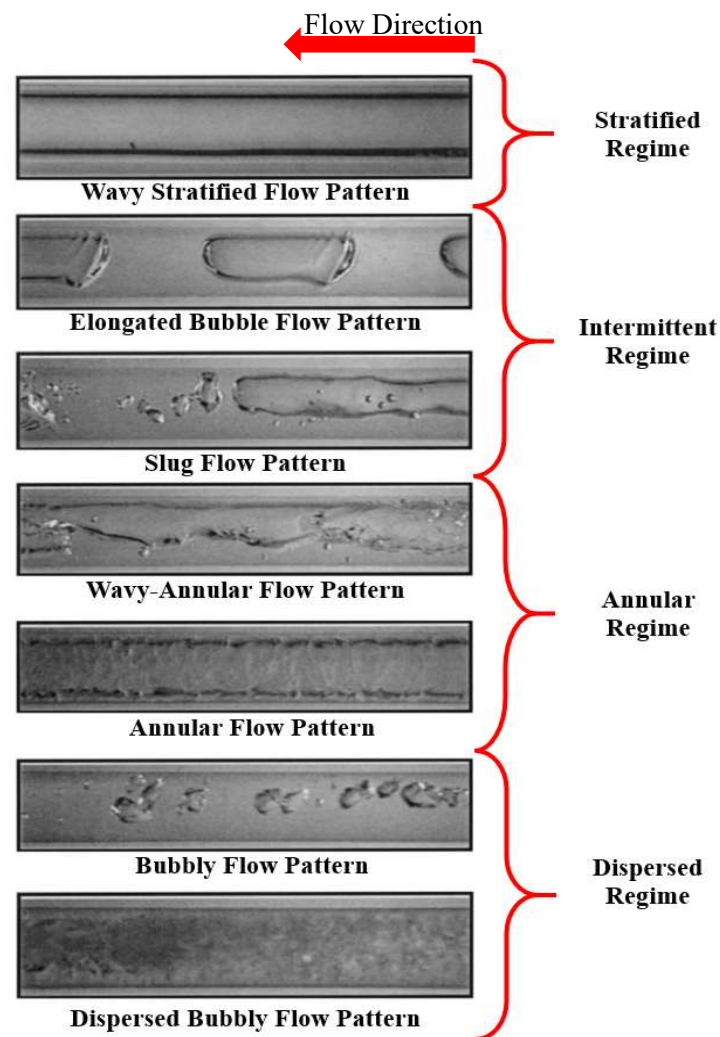


Figure 2.12. Flow patterns reported by Coleman and Garimella [113].

Coleman and Garimella [113] used the term elongated bubble flow but also named plug flow as an alternative designation. However, the definition provided by Coleman and Garimella [113] differs from that proposed by Hewitt [90] in conventional channels. They defined plug flow as a continuous stream of vapour plugs flowing in the liquid phase, where a thin film of liquid surrounds the plug. Some small disturbances may be present in the vapour-liquid interface; however, the vapour bubble remains intact.

This differed from Coleman and Garimella's [113] definition of slug flow, which was defined as the flow pattern that occurs once the vapour bubble has broken apart. Contrary to this, Hewitt [90] defined slug flow in horizontal channels as an intermittent flow pattern that is caused by a transition from the wavy-stratified regime and would therefore not be seen in microchannels.

The terms of slug and plug provide significant inconsistencies between publications when attempting to define both flow patterns. For example, the slug flow reported by Triplett et al. [112] reflected the definition of slug flow provided by Hewitt [90] for vertical conventional channels, i.e. smaller bubbles have coalesced to form larger bubbles with diameters approaching that of the channel. This definition would encompass both the plug flow and slug flows reported by Coleman and Garimella [113]. Triplett et al. [112] noted that the slug flow reported was also referred to as slug by Suo and Griffith [105] but as plug flow by Damianides and Westwater [114] and as bubble-train flow by Thulasidas et al. [115] for other studies involving water-air mixtures. It could therefore be argued that the requirement for clear and universal flow pattern terminology has been apparent for the previous quarter of a century of two-phase microscale research, but remains unaddressed in the present day. As a consequence, the remainder of the review on microscale flow patterns focusses on the propositions of clear generic flow pattern categorisations that can be utilised universally to eliminate the existing issues of subjectivity and a lack of universal terminology. This must include the elucidation of a useful description of the aforementioned plug and slug flows.

Two-phase adiabatic flows can be of particular interest in applications such as gas-oil mixture transportation in pipelines [87]. However, the flow pattern transition mechanisms and phenomena may be dissimilar to the flow patterns observed in cooling applications, due to the differing interfacial phenomena as a result of bubble growth, bubble coalescence, dryout, CHF and other flow instabilities that occur during flow boiling [116]. Unlike flow boiling, adiabatic experiments are conducted with constant gas and liquid flow rates and as such, compressibility effects that may occur for larger vapour qualities that only occur at the end of the channel cannot be observed in adiabatic experiments. A further limitation of this methodology is that it does not account for the effect of heat flux on the two-phase flow patterns and neglects the evaporation momentum force, although its relative scale is considerably smaller than the other forces, as shown in Figure 2.3. Nevertheless, the small role that it plays may show some significance in flow patterns and their resultant heat transfer mechanisms.

2.3.2.3 Indirect Diabatic Methodology Studies

The most applicable methodologies to investigate the flow patterns that occur during the use of flow boiling for cooling applications are those that apply heat and carry out flow visualisation experiments. Yang and Shieh [117] used both the adiabatic and indirect diabatic methodologies to study water-air mixtures and the flow boiling of R-134a in horizontal tubes between 1-3 mm in internal diameter. They

reported dispersed bubbly, bubbly, plug, slug, slug-annular, annular and wavy-stratified flows for water-air mixtures in 2 mm and 3 mm internal diameter (ID) tubes; however, stratified wavy flow was not observed in the 1 mm channel.

Dispersed, bubbly, plug, slug and annular flow regimes were reported for the flow boiling of R-134a, where only 2 mm and 3 mm tubes were studied. They cited the differentiation between plug and slug flow as that outlined by Alves [118] for conventional sized tubes; however, they stated that plug flow was a result of bubbles coalescing and rising to the top of the channel and that slug flow was a result of the amplified waves that occur during the stratified-wavy flow pattern. The transition paths between flow patterns with increasing gas-superficial velocity, as outlined by Alves [118], were drawn by Holland and Bragg [119] (with mist flow referred to as “spray flow”), as shown in Figure 2.13. The differentiation between plug and slug flow as outlined by Alves [118] is reliant on the presence of wavy-stratified flow. Hence, the differentiation between plug and slug flow outlined by Yang and Shieh [117] is not helpful in the distinction between the flow patterns in the microscale.

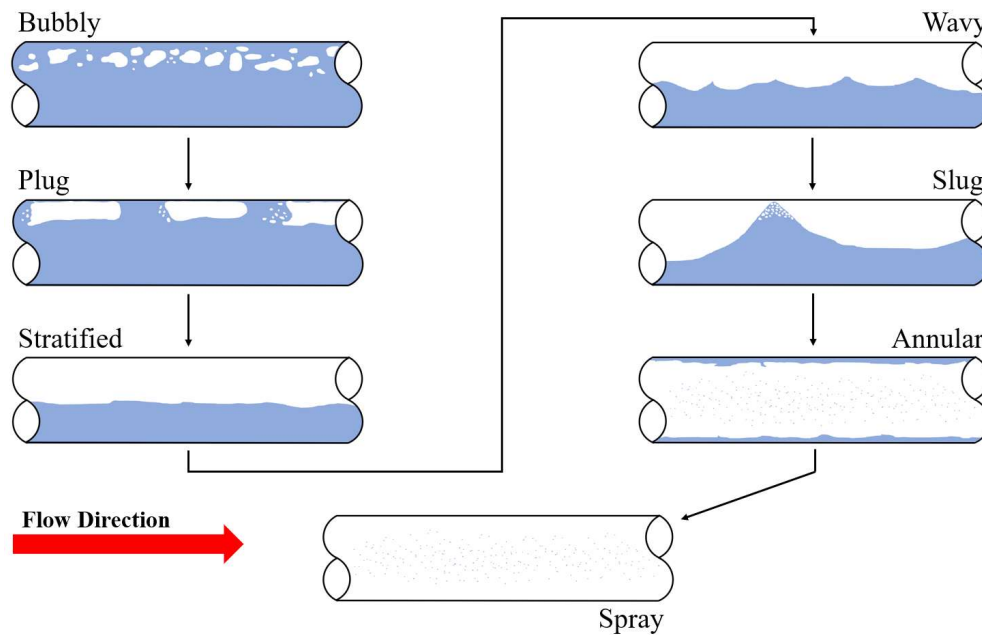


Figure 2.13. Flow pattern transitions with increasing vapour superficial velocity by Holland and Bragg [119] (modified).

Petterson [120] conducted flow boiling experiments of CO₂ in twenty-five horizontal 0.8 mm ID tubes, where heating of the tubes was conducted using a water jacket. Flow patterns were categorised as dispersed flow, intermittent flow, annular flow and droplet flow. Unusually, bubbly flow was also included in the wider categorisation of intermittent flow alongside the plug and slug flow patterns. The droplet flow reported by Petterson [120] resembles the mist flow that has been presented in this review. Stratified flow was not reported for any applied heat flux.

Chen et al. [121] studied flow boiling of R134a in vertical tubes between 1.1 and 4.26 mm in internal diameter. They reported flow patterns of dispersed bubble, bubbly, confined bubble, slug, churn and annular flows. The flow patterns were clearly photographically documented and are displayed in Figure 2.14. The confined bubble flow pattern was said to be analogous to the term plug flow, where both were said to be where the bubble diameter reached that of the tube and it becomes confined by the channel wall. A spherical cap exists for both the top and bottom of the bubble. This differs from the reported slug flow, where the vapour bubbles are more akin to bullet shaped Taylor bubbles, i.e. spherically capped at the nose and flat at the tail [122]. This distinct differentiation between plug and slug flow and the clear flow patterns observed by Chen et al. provide a basis for the proposal of new universal terminology, using what can be described as classical flow pattern categorisations.

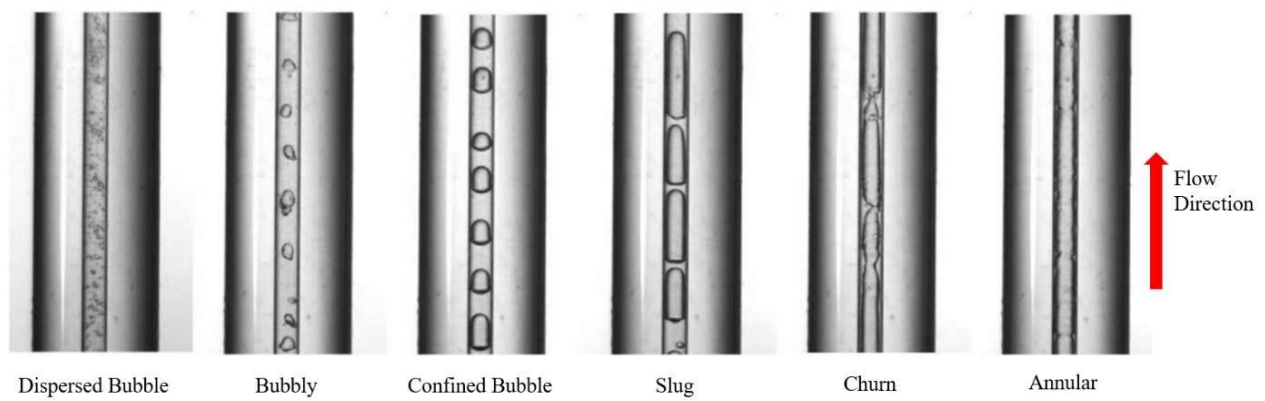


Figure 2.14. Flow patterns reported by Chen for R134a in 1.1 mm internal diameter vertical tube [121].

Revellin et al. [123] also used a similar indirect diabatic technique to study R-134a in a 0.5 mm horizontal channel and recorded clear flow patterns that are shown in Figure 2.15. They reported bubbly, slug, semi-annular and annular flows whilst photographs of the transition between the flow patterns were also photographically reported. The transitional photographs, labelled as bubbly/slug and slug/semi-annular flow in Figure 2.15, display the gradual nature of flow pattern transitions. This gradual nature therefore adds to the subjective nature of categorising flow patterns when visually conducted. Again, discrepancies in terminology emerge when compared to the work by Chen et al. [121] for categorising intermittent flow regimes, the term here referring to slug, plug and churn flows. The label of semi-annular flows can be considered akin to the churn flow pattern reported by Chen et al. [121] whilst the confined bubble flow presented by Chen et al. [121] was categorised as slug flow by Revellin et al. [123]. The added problem of clear transition criteria is therefore evident when determining prevailing flow patterns. Rahim et al. [124] also reported the same flow patterns as Revellin et al. [123] investigating the flow boiling of R-134a and R-245fa in 0.509 mm and 0.790 mm internal diameter horizontal tubes.

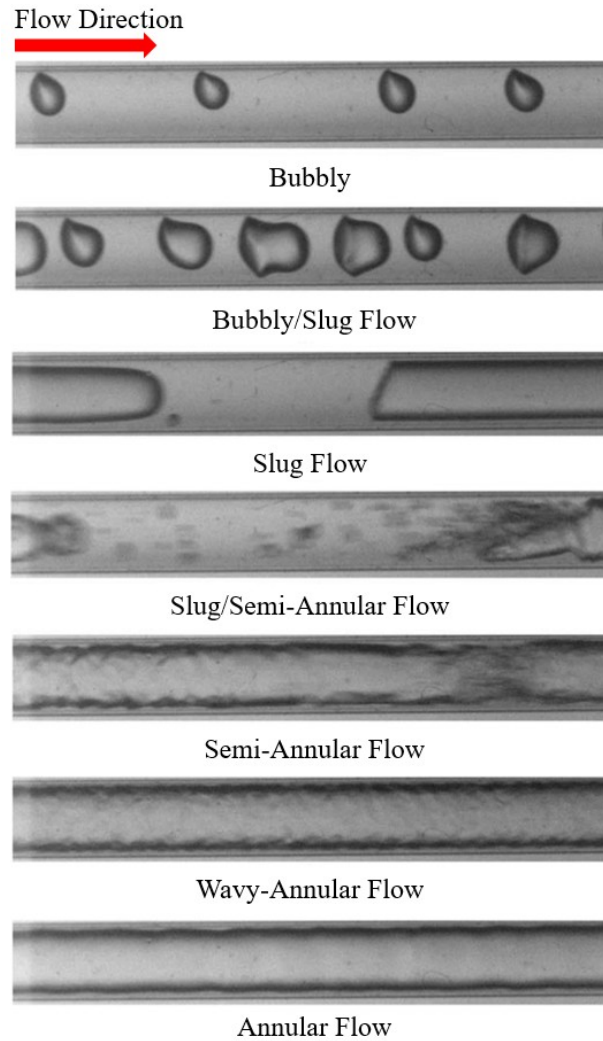


Figure 2.15. Flow patterns and their transitions as reported by Revellin et al. [123].

Revellin and Thome [125] attempted to eliminate the problem of subjectivity by defining flow pattern transitions into a measurable and quantifiable criterion. Indirect diabatic experiments were conducted using R-134a and R-245fa in horizontal tubes between 0.509 mm and 0.790 mm. The same flow patterns described by Revellin et al. [123] were reported; however, a further set of flow pattern designations were utilised to eliminate the subjectivity in categorising flow patterns. New categorisations of Isolated Bubble (IB), Coalescing Bubble (CB) and Annular Flow (AF) patterns were coined. The new set of categorisations were classified based on bubble frequency measurements taken from laser instrumentation. Figure 2.16 shows the coalescence phenomena recorded from the laser instrumentation during the experiments that was used to categorise the flow patterns. The graph of bubble count frequency against thermodynamic vapour quality was used as follows:

IB regime: Occurs from a vapour quality of 0 to the vapour quality of the maximum bubble count frequency shown in Figure 2.16. This phenomenon occurs as the bubble generation frequency increases with heat flux and consequent higher vapour qualities.

CB regime: Occurs from the vapour quality at which the maximum bubble count frequency occurs until the vapour quality at which the bubble count frequency reaches zero. This was shown under some operating conditions to have two distinct gradients/ coalescence rates as displayed in Figure 2.16. They argued that rate 1 indicated small bubbles coalescing to form elongated bubbles, whilst rate 2 indicated coalescence between elongated bubbles and small bubbles.

Annular regime: The annular flow regime was said to occur at the vapour quality where the bubble count frequency reached a zero value. Here there was said to be a constant vapour phase without an intermittent characteristic that was detected by the laser equipment.

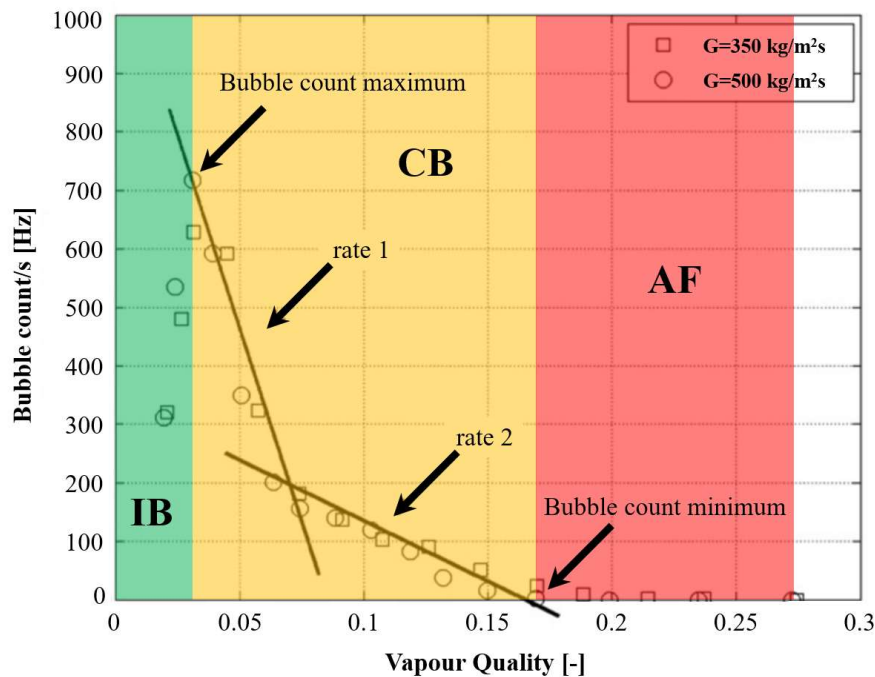


Figure 2.16. Coalescence phenomena for R134a in a 0.509 mm ID horizontal channel with a 50 mm heated section (taken from [125]).

A potential criticism of this technique is the inability to fully capture the relationship between bubble coalescence and bubble generation frequency. The inability to capture this relationship may also result in an ineffective analysis of the connection between interfacial phenomena and heat transfer rates during flow boiling. For example, the bubble count frequency may increase as a result of the bubble generation frequency increasing faster than bubble coalescence frequency with increasing vapour qualities. The resultant bubbles may be confined at this vapour quality, causing what would be considered as classical slug flow although it would still be classified in the IB regime. Hence, the heat transfer would be calculated by considering the classical bubbly flow regime and would result in inaccurate heat transfer predictions. Furthermore, the chaotic phase mixture during churn flow is neglected using this method, which does not possess the same heat transfer characteristics as slug or annular flows.

A further criticism of the overall indirect diabatic methodology is that the viewing only occurs in a theoretically adiabatic section. Hence, the reported flow patterns may not reflect the true flow patterns that occur at the location of the boiling process. For example, many publications have reported the appearance of bubble nucleation in the liquid film that occurs during annular flow [126–129]. Hence, any transition mechanisms deriving from such bubble nucleation cannot be sufficiently observed using the indirect diabatic experimental methodology. Furthermore, the adiabatic condition of the viewing section is an idealistic assumption and, for many fluids and pressures, a degree of heat will be lost to the experiment surroundings. Hence, the outlet thermodynamic vapour quality at the beginning of the viewing section will be greater than that at the viewing section outlet, despite this probably being negligible for most studies.

2.3.2.4 Direct Diabatic Methodology Studies

Criticisms of the indirect diabatic methodologies allude to the fact that the most suitable methodology to understand flow patterns and their transitions during flow boiling cooling applications is implementing the direct diabatic methodology. The review of adiabatic and indirect diabatic studies reveals that stratified flow does not exist in the microscale, other than for adiabatic studies where a settling section was not used. Hence, the flow pattern groups in horizontal and vertical microchannels resembles the groups for vertical conventional channels shown in Figure 2.7, i.e. bubbly, intermittent and annular flow pattern groups, where the term intermittent refers to all flow patterns between bubbly and annular flows. However, discrepancies still exist between publications when reporting flow patterns; both with regards to phenomenological observations and terminology. Consequently, there is a need to clarify why such disparities exist between reported flow patterns and which flow pattern terminologies should be adopted to aid in developing tools for engineering design.

The suggested universal microscale flow pattern terminology made on the basis of this review section will be formed according to (i) whether the flow patterns provide a significant mechanistic alteration to heat transfer rates to aid in the development of engineering design tools and (ii) whether the terminology allows for simple categorisation of flow patterns with minimal subjectivity in classification.

The first criteria for establishing universal flow pattern terminology relies upon the understanding of heat transfer mechanisms so that it can be considered whether the alternative terminologies are useful in identifying mechanistic differences that will aid in developing engineering prediction tools. It is therefore helpful to outline the heat transfer mechanisms that are proposed in microchannels.

It is common knowledge that the heat transfer mechanisms in conventional channels are the nucleate boiling and convective boiling mechanisms. As is true in pool boiling, the bubble departure diameter, bubble generation frequency and active nucleation sites increase with a greater heat flux during flow

boiling, when the nucleate boiling mechanism is present [130]. This increase in nucleation site density and bubble generation frequency results in agitation and greater mixing of the bulk fluid. Consequently, heat transfer rates are dominated by the effect of heat flux during the nucleate boiling regime, with minimal dependency on the mass flux effect.

Conversely, the convective boiling mechanism is said to be dominated by the characteristics of the liquid film, which occurs in some intermittent and, particularly, in annular flows [131]. Heat transfer rates are significantly affected by mass flux and vapour quality due to the reduction of the liquid film thickness. The reduction in film thickness causes a lower thermal resistance and a subsequent enhancement of evaporation from the film to the core [22]. In consequence, heat transfer rates are predominantly affected by the role of mass flux, with little to no reliance on the heat flux [132]. The literature on microscale flow boiling heat transfer mechanisms is discussed in detail in Section 2.4.

2.3.2.4.1 *Bubbly Flow Patterns*

Overall, there appears a general agreement that bubbly flow is the initial flow pattern that occurs at the lower wall heat fluxes. The first flow pattern was commonly referred to as bubbly flow by most authors [128, 129, 133–168]; however, alternative terminologies were used by other authors to describe the same phenomena or to further distinguish between types of bubbly flows [150, 169–175]. These included isolated bubbly flow, rapid boiling, rapid bubbly boiling and discrete bubble flow. Some publications did report an absence of bubbly flows in microchannels altogether [175–184]. The reported bubbly flow pattern terminologies are presented in Table 2.2.

A form of intermittent flow was reported as the first flow pattern by publications in which the bubbly flow was absent. The studies by Xu et al. in [177] and [178] utilised a silicon heat sink with triangular channels to investigate the flow boiling of acetone, where the height of the triangular channels was 212 μm and the maximum width of the channels was 300 μm , resulting in an overall hydraulic diameter of 155.4 μm . An intermittent flow pattern which was reported as “bubbles entrained in liquid plug”, although no bubbly flow pattern in which no confinement existed occurred. This phenomenon can be explained by the observations made by Xu et al. [177] and the triangular cross-sectional geometry of the channel. They stated that bubble nucleation began in all three corners of the channel as shown in Figure 2.17. Figure 2.17 diagrammatically explains the bubble nucleation, growth and coalescence mechanism. The coalescence of the bubbles at the surface then resulted in a single confined bubble. Superheated liquid in a metastable state released energy to the vapour, resulting in a sudden pressure surge within the combined bubble and then caused flow reversal.

Table 2.2. Reported flow patterns for the bubbly flow pattern group from direct diabatic studies.

Reported Flow Pattern	Fluids	Configuration	Hydraulic Diameters	Mass Flux Ranges Studied
			[mm]	[kg/m ² s]
Bubbly	DI Water, FC-77, FC-72, R-134a, R-245fa, ammonia, HFE-7100, R-1234yf, HFE-7200, R-1233zd(E)	Multichannel heat sinks, single rectangular microchannel, circular microchannel, counterflow interconnected heat sink	0.0863 – 1.2	12 - 16765
Isolated bubbly flow	R-134a, R-1234ze(E), R-245fa, DI Water, R-1233zd(E)	Multichannel heat sinks, circular microchannel, single rectangular microchannel	0.294 – 1.33	100-1529
Rapid boiling	DI Water	Single rectangular microchannel	0.19	50 - 583
Rapid bubbly boiling	DI Water	Single rectangular microchannel	0.19	50 - 583
Discreet bubble flow	DI Water	Single rectangular microchannel	0.571	261 - 961
No Reported Bubbly Flows	DI Water, Acetone, CO ₂ , R-1234ze(E), FC-72, HFE-7000, R-1336mzz(Z)	Multichannel heat sinks, single rectangular microchannel, circular microchannel	0.024-0.942	14-1200

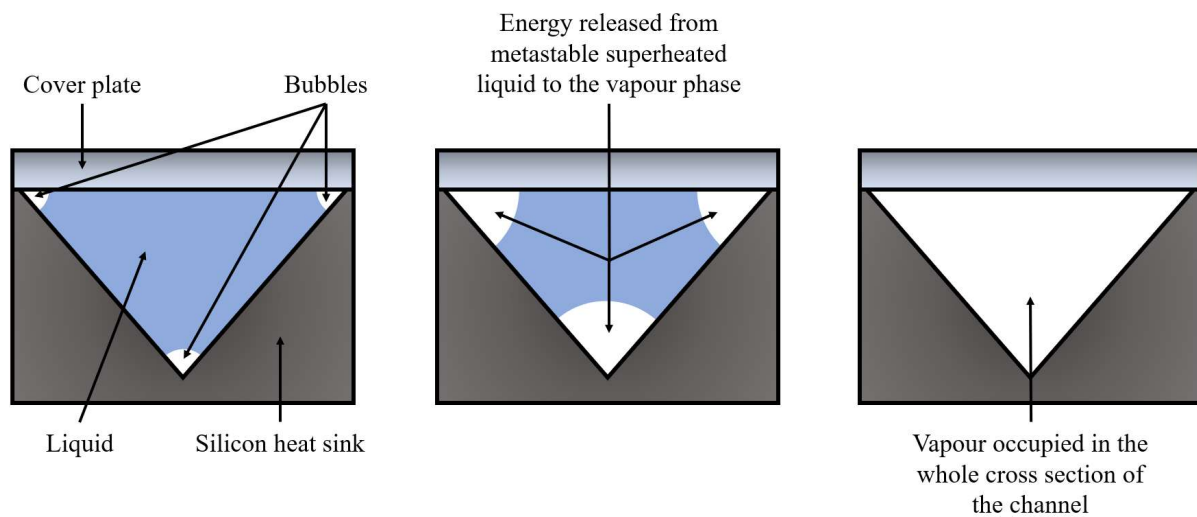


Figure 2.17. Bubble nucleation and growth mechanism in triangular microchannels (adapted from [177]).

The bubble departure diameter has a significant impact on bubble confinement and the potential absence of bubbly flow. Furthermore, there is an even greater chance of bubble coalescence and confinement prior to bubble departure at a high nucleation site density. Yang et al. [185] studied the bubble departure diameter for vertical upward subcooled flow boiling in a narrow rectangular microchannel. DI water was investigated in a channel with a width of 28 mm and 2 mm in height (3.73 mm diameter), with heat and mass flux ranges between 17–289 kW/m² and 122–167 kg/m²s, respectively. The proposed bubble departure diameter prediction correlation is shown in Equation (2.7)

$$D_d = 0.0058 l_c \cdot Ja_{sup}^{1.45} \cdot e^{-0.015 Ja_{sub} - 10^{-4} Re_{lo}} \quad (2.7)$$

In Equation (2.7), l_c represented what was referred to as the characteristic length scale, defined as $l_c = \sqrt{\sigma/g(\rho_l - \rho_g)}$. The two Jakob numbers correspond to the wall superheat and the degree of subcooling, respectively, and are defined in Equations (2.8) and (2.9), where $C_{p,l}$ represents the specific heat of the liquid phase and ΔT_w represents the wall superheat. Finally, Re_{lo} represents the liquid only Reynolds number, where $Re_{lo} = GD_h/\mu_l$.

$$Ja_{sup} = \frac{\rho_l C_{p,l} \Delta T_w}{\rho_g i_{fg}} \quad (2.8)$$

$$Ja_{sub} = \frac{\rho_l C_{p,l} \Delta T_{sub}}{\rho_g i_{fg}} \quad (2.9)$$

Figure 2.18 presents the bubble departure diameters for water, acetone and HFE-7100 for the highest mass fluxes in which bubbly flow was not observed using the correlation provided by Yang et al. [185]. However, the use of Equation (2.10) was dependent on knowledge of the wall superheat, which was calculated using Equation (2.10). The two-phase heat transfer coefficient (HTC), h_{tp} , was calculated using the correlation provided by Mahmoud and Karayiannis [186]. The heat transfer coefficient correlation was selected as it was based on a large amount of data with multiple fluids and channel configurations as well as taking the form of an easily useable single continuous function.

$$\Delta T_w = \frac{h_{tp}}{q''_w} \quad (2.10)$$

The correlation provided was developed using subcooled convective boiling flows; however, it may provide an approximation to the bubble departure size observed during flow boiling in the microscale. Figure 2.18 shows that the bubble departure diameter for the study conducted by Xu et al. [177] was between 660 µm and 1120 µm up to a vapour quality of 0.1. This meant that even with a single bubble, the bubble would become confined by the channel height prior to departure. This is also true for the study conducted by Lee et al. [176], who investigated the flow boiling of DI water in a rectangular microchannel with a width and height of 120 µm and 14 µm, respectively.

The study by Widyatama et al. [184] utilised a high aspect ratio microchannel to investigate flow boiling of HFE-7000 at mass fluxes between 14 kg/m²s and 42 kg/m²s. The aspect ratio, defined in Equation (2.11), represents the ratio of the channel width to channel height. The test section used for the study was designed with an aspect ratio of 10, where the width and height of the channel were 5 mm and 0.5 mm, respectively. Figure 2.18 shows the expected bubble departure size to be between 360 µm and 430 µm according to the correlation by Yang et al. [185], which is smaller than the height of the channel. Despite this, Widyatama et al. [184] reported that slug flow was the first observed flow pattern. The reason for this may be first be that the correlation was proposed for subcooled flow boiling and the bubble departure diameter for saturated boiling may be greater. The second reason may be that the bubble is not only confined by the channel, but also by a liquid film surrounding the bubble.

$$\beta = W_{ch}/H_{ch} \quad (2.11)$$

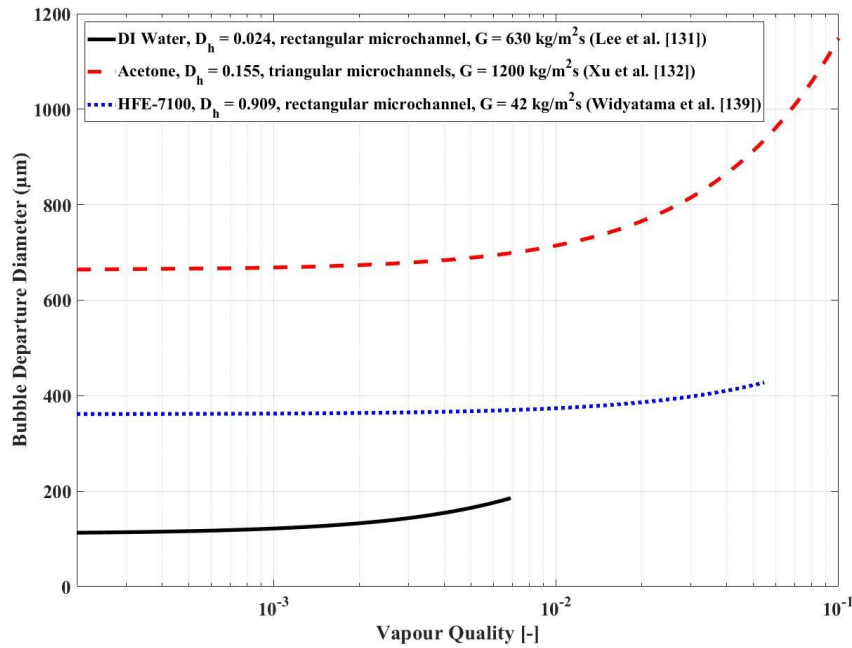


Figure 2.18. Predicted bubble departure diameter using the correlation of Yang et al. [185].

Zhou et al. [179, 182] utilised DI water in a microchannel with an aspect ratio of approximately 10, where the height of the microchannel was 0.52 mm and the width was 5.01 mm in two studies. They also reported an absence of bubbly flows. This differentiation in flow patterns suggests that a further categorisation of the ‘sub-microscale’ may be useful for further distinguishing between flow characteristics at hydraulic diameters smaller than the conventional scale. This is not a new suggestion and was alluded to in the discussion on the work conducted by Kawaji and Chung [76]; however, they suggested the terminology of miniscale and microscale. The sub-microscale would therefore be

influenced by not only the hydraulic diameter and thermophysical properties, but also by aspect ratio, operating parameters and potentially surface morphology upon which the nucleation site density is dependent.

Özdemir et al. [161] investigated the flow boiling of DI water in three rectangular channels with a constant hydraulic diameter of 560 μm but aspect ratios of 0.5, 2.56 and 4.94. Bubbly flow was reported for all aspect ratios; however, the bubbles were said to become quickly confined and more elongated for the low aspect ratio channel ($H_{ch}=0.84$ mm, $W_{ch}=0.42$ mm) which resulted in a very small bubbly flow region. The small bubbly flow region was attributed to confinement effects; however, the same confinement effects were not observed in the high aspect ratio channel ($H_{ch}=0.3367$ mm, $W_{ch}=1.663$ mm).

Provided the same operating parameters are present, it appears unlikely that the bubbles would become confined by a dimension of 0.42 mm and not by a dimension of 0.3367 mm. A potential reason for the lack of reported confinement in high aspect ratio microchannels may be the corner nucleation of the bubble and the viewing of flow patterns from above. Figure 2.19 shows an example diagram of single bubble growth in a low aspect ratio and high aspect ratio channel, where the bubble is confined prior to departure. It should be noted that the sketch is diagrammatic and the actual bubble geometry may differ significantly; however, the diagram is intended to demonstrate that it is more difficult to observe when a bubble is confined by the height of the channel than when it is confined by the width of the channel if the flow patterns were viewed from above. This is because the confinement of a bubble in a single dimension will still permit bubble growth in the remaining two dimensions. Hence, growth can still be observed in two dimensions when a bubble is confined by the height of the channel and the flow patterns are observed from above. This may make it appear that the bubble is not confined in any dimension. Conversely, the bubble will only will be seen to grow along the channel length if a bubble is confined by the width of the channel, potentially making it appear as if the bubble is confined by the entire channel cross section.

The absence or presence of bubbly flow in very high aspect or very low aspect microchannels may also present a disagreement between publications due to a lack of clear definition between bubbly flow and some intermittent flow patterns. For example, Widyatama et al. [184] reported an absence of the bubbly flow pattern in high aspect ratio microchannels and classified the photographs in Figure 2.20 as slug flow, whilst Özdemir et al. detailed the presence of bubbly flow and categorised the photographs in Figure 2.20 as bubbly flow. The vapour bubbles appear to be confined by the height of the channel in both images and hence, are exhibiting the same physical phenomena. However, the discrepancy between the two designations arises because of a confusion as to whether a vapour bubble is required to be confined by both channel cross sectional dimensions as opposed to just one; in this case the height dimension.

Aside from the term bubbly flow, isolated bubbly flow has also been used in publications as a continuation of the work originally conducted by Revellin and Thome [125]. A noticeable omission from the flow patterns presented in Table 2.2 is that of dispersed bubbly flow, which was reported in adiabatic and indirect diabatic studies. However, it is not recommended that the dispersed bubbly flow and bubbly flow are separated as the nucleate mechanism dominates in both. Furthermore, the transition between the two is highly subjective as it is difficult to distinguish how spherical the bubbles are and the degree of slippage between the two phases. Therefore, it is recommended that the generic term of bubbly flow is used to describe all flow patterns in which bubbles exist and are not confined in any dimension or do not exist in a liquid film.

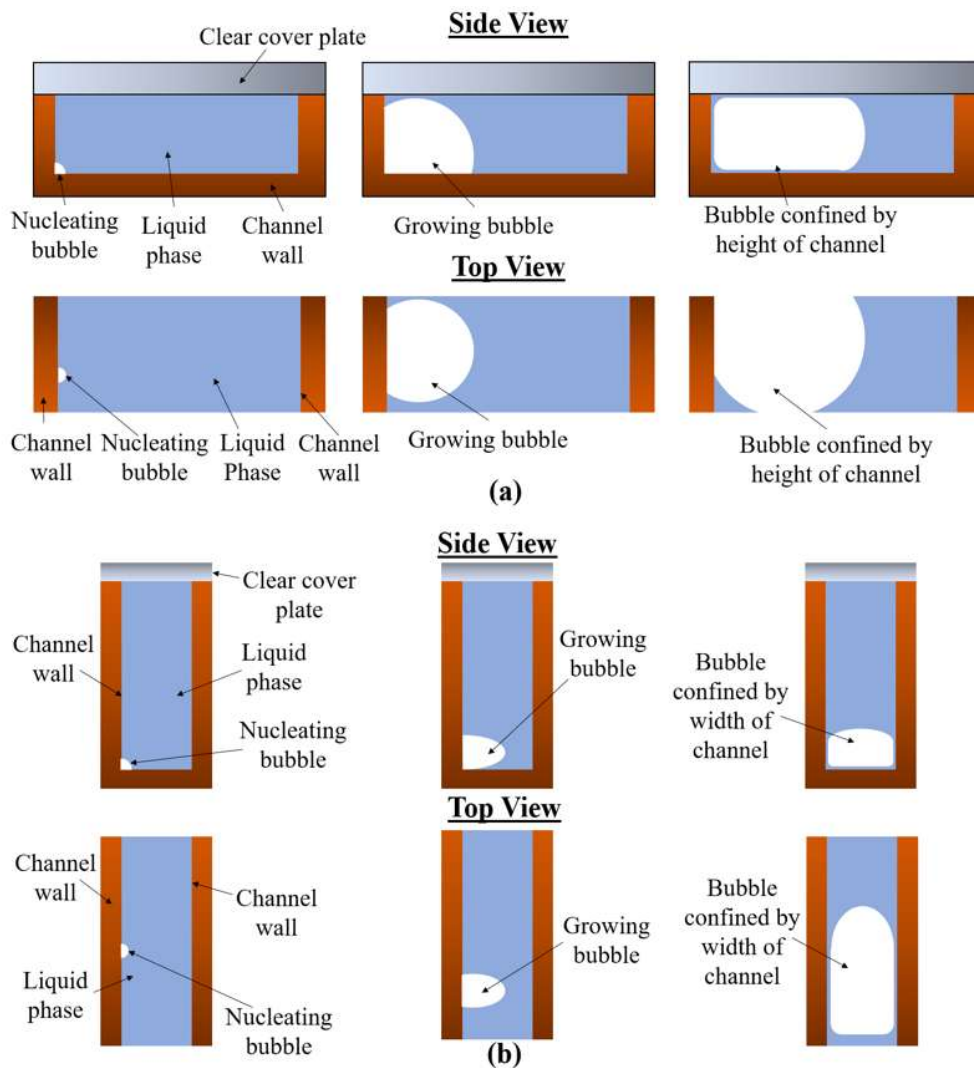


Figure 2.19. Bubble growth and confinement in (a) a high aspect ratio microchannel and (b) a low aspect ratio microchannel.

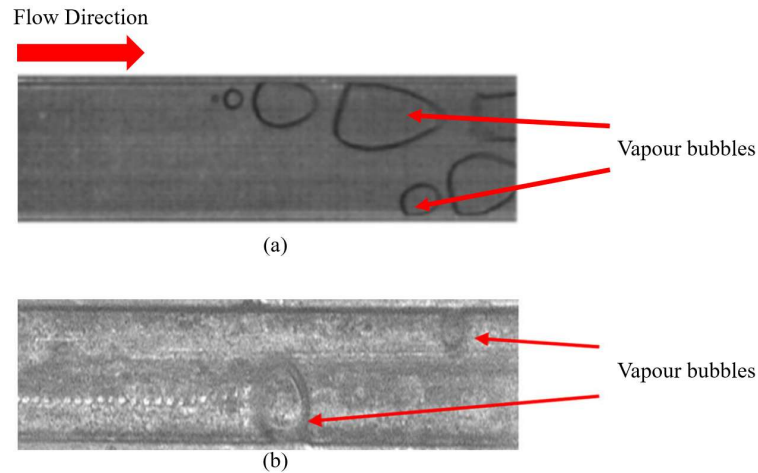


Figure 2.20. Single vapour bubbles in high aspect ratio microchannels (a) from Widyatama et al. [184] and (b) Özdemir et al. [161].

2.3.2.4.2 Intermittent Flow Patterns

As has been previously discussed, subdivisions within the intermittent flow regime shown in Figure 2.7 and Figure 2.8 have provided inconsistent use of terminology between publications. Table 2.3 presents the terminologies used to describe the flow patterns that would be classified in the intermittent flow regimes, i.e. any flow pattern that is not bubbly or annular. The sheer range of designations reported in the literature makes it difficult to disseminate which flow patterns are phenomenologically comparable, although it is necessary for a recommendation of universal flow pattern designations to be used in this study.

Determination of which intermittent flow pattern terminologies are describing the same phenomena and which terminologies may be unnecessarily subdividing flow pattern categorisations is required. The flow pattern transition from bubble flow was defined to occur once a bubble becomes confined in any dimension in the discussion above; hence, it is necessary to determine which physical flow pattern bubble flow transitions to and which terminology should be used.

The most common terminology used to describe the departure from the bubbly flow regime is that of slug flow; although, other publications have used other terminologies to describe a further transition between the bubbly and slug flows. Gasche et al. [187] studied the flow boiling of CO₂ in a single rectangular aluminium microchannel with a hydraulic diameter of 0.8 mm. They reported an absence of bubbly flow but reported plug and slug flows, which are displayed in Figure 2.21, in accordance with the flow pattern designations by Carey [86]. The first image of plug flow, for a mass flux of 58 kg/m²s at a vapour quality of 0.13, appears to follow the same description as that of Chen et al. in which the vapour bubbles possess spherical heads and tails [121]. However, the second image taken at a mass flux of 96 kg/m²s and a vapour quality of 0.25 shows many vapour bubbles with flat tails. It appears that the

only difference between the two flow patterns is based on the length of the vapour bubbles. Li and Hrnjak [180, 181] used the same denotations as Gasche et al. in both reported studies.

Table 2.3. Reported flow patterns for the intermittent flow pattern group from direct diabatic studies.

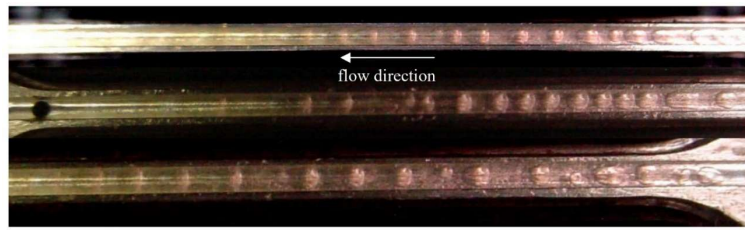
Reported Flow Pattern	Fluids	Configuration	Hydraulic Diameters [mm]	Mass Flux Ranges Studied [kg/m ² s]
Slug	DI Water, CO ₂ , FC-77, R-134a, HFE-7100, FC-72, R-245fa, Ammonia, R-1234ze(E), R-1234yf, HFE-7200, R-1233zd(E), R1336mzz(Z)	Multichannel heat sinks, single rectangular microchannel, circular microchannel, circular microchannel heat sink	0.0863 – 1.2	12 - 16765
Plug	CO ₂ , R-1234ze(E)	Single rectangular microchannel, circular microchannel heat sink	0.643 - 0.736	58 - 235
Elongated Bubble	DI Water, R-134a, FC-72, R-1233zd(E)	Single rectangular microchannel, circular microchannel, multichannel heat sinks	0.088 – 1.33	20 – 2310.9
Confined Bubble	R-134a, DI Water	Circular microchannel, Rectangular microchannel	0.571 - 1.33	20 - 500
Coalescing Bubble	R-134a, R-1234ze(E), R-245fa, FC-72	Multichannel heat sinks	0.12 – 0.5	70 - 1529
Elongated Slug	DI Water, R-1234yf	Multichannel heat sinks, rectangular microchannel	0.108 - 1	100 - 900
Liquid plug/vapour slug	Acetone	Multichannel heat sinks	0.155	50 - 1200
Bubble Slug Entrained in Liquid Plug	Acetone	Multichannel heat sinks	0.155	50 - 1200
Bubbles Entrained in Liquid Plug	Acetone	Multichannel heat sinks	0.155	50 - 1200
Elongated Bubbly/ Slug	DI Water	Circular microchannel, Multichannel heat sinks	0.1 - 1	70 - 900
Confined Slug	FC-72	Multichannel heat sinks	0.096 – 0.707	225 - 1420
Bubbly/ Slug	DI Water	Rectangular microchannel	0.19	50 - 583
Churn	DI Water, FC-77, R-134a, HFE-7100, HFE-7200, R-1233zd(E)	Multichannel heat sinks, circular microchannel, singular microchannel	0.108 – 1.33	14 - 1373
Reversed	DI Water	Rectangular Microchannel	0.571	261 - 961
Transitional Flow with Liquid Ring	R-1234ze(E), R-1336mzz(Z)	Circular Microchannel heat sink	0.643	100 - 200

Table 2.3. continued.

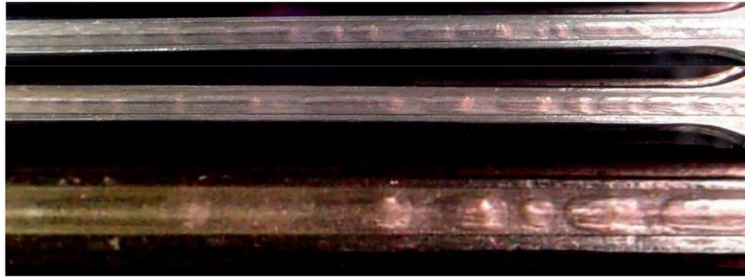
Reported Flow Pattern	Fluids	Configuration	Hydraulic Diameters [mm]	Mass Flux Ranges Studied [kg/m ² s]
Transitional Flow with Rings and Waves	R-1234ze(E), R-1336mzz(Z)	Circular Microchannel heat sink	0.643	100 - 200
Agitated Flow	R-1234yf	Rectangular microchannel	0.5	251.6 – 579.2
No Reported Intermittent Flows	DI Water	Multichannel heat sink	0.186	38.6 – 3523.2

Martin-Callizo et al. [169] studied flow boiling of R-134a in a 1.33 mm internal diameter vertical quartz tube with a homogeneous indium tin oxide coating. The intermittent flow patterns were subdivided into confined bubbly flow, slug flow, churn flow and slug-annular flow in order of increasing heat flux. The initial flow pattern was labelled as isolated bubbly flow, which is akin to the classical bubbly flow designation, and was reported to transition to the confined bubbly flow regime. The photographed isolated bubbly flow, confined bubbly flow and slug flow are presented in Figure 2.22. The key difference between the flow pattern designated as confined bubbly flow and slug flow is the length of the vapour bubbles as both have spherical shaped noses and near flat tails.

Wu and Cheng [133] used the designations of slug and elongated slug flow; however, the elongated slug flow classification described the same phenomena as the slug flow described by Martin-Callizo et al. [169], whilst the slug flow observed by Wu and Cheng [133] would be categorised as confined bubble by Martin-Callizo et al [169]. Yin and Jia [188] used the same designation of confined bubble flow as Martin-Callizo et al. [169] but referred to the slug flow reported by Martin-Callizo et al. [169] as elongated bubble flow. Lv et al. [158] used the terms slug flow and elongated bubbly flow to describe the observed patterns, although little discussion was provided as to the phenomenological differences between the flow patterns. Li et al. [160] used the same designations of slug and elongated slug flows as Wu and Cheng [133]. Finally, Nalbandian et al. [164] also used the terminologies of plug and slug flows to describe the confined bubble and slugs flows as reported by Martin-Callizo et al [169].



Plug Flow ($G = 58 \text{ kg/m}^2\text{s}$, $x = 0.13$)



Plug Flow ($G = 96 \text{ kg/m}^2\text{s}$, $x = 0.25$)



Slug Flow ($G = 58 \text{ kg/m}^2\text{s}$, $x = 0.48$)

Figure 2.21. Intermittent flow patterns reported by Gasche for CO_2 [187].



Isolated bubbly
flow



Confined bubbly
flow



Slug flow

Figure 2.22. Photographs of isolated bubbly flow, confined bubbly flow and slug flow from Martin-Callizo et al. [169].

The use of the many different terminologies to describe the same phenomenological flow patterns makes it difficult to compare publications. Furthermore, the transition between flow patterns is gradual and, whilst plug flow may be characterised by many as vapour bubbles with spherical heads and tails, the transition of the tails to flat Taylor bubbles is a gradual process. The use of this terminology to distinguish between the two plug and slug flow patterns makes it subjective to each researcher as to decide if the tail is sufficiently non-spherical to classify it as slug flow. Additionally, the use of slug flow to describe longer confined vapour bubbles also leaves some subjectivity in deciding which length is sufficient enough to depart from plug flow into the slug flow regime.

Wang and Cheng [166] referred to the categorisation as elongated bubble/slug, acknowledging the consistent interchangeable terminology within the literature. The same terminology was also used by Ajith et al. [159] when describing the blanket slug flow categorisation. Only one publication did not explicitly define a flow pattern categorisation to describe the blanket definition of slug flow; however, the presence of a vapour slug was reported by Wang et al. [141] and thus, no publication reported that a vapour bubble confined by the channel geometry was not present in microchannels. Hence, it is recommended that the generic term slug flow is used as a blanket categorisation for any phenomena in which a single stable vapour bubble is confined by any dimension of the channel.

Further flow pattern categorisations have also been utilised to describe scenarios in which vapour bubbles are confined by the channel geometry. The terms liquid plug/vapour slug flow, bubble slug entrained in liquid plug flow and bubbles entrained in liquid plug flow were used in two separate publications by the same group [177, 178]. In this case, the terms were utilised to describe transient phase interfacial geometry phenomena that occurs for the same operational parameters and location. A single steady state categorisation for a single set of operating parameters and geometry is essential for engineering design, although knowledge of the transient characteristics of the categorisation is also of great importance [189]. For example, significant pressure drop fluctuations that may want to be avoided have been observed for slug flows [190]. However, the knowledge required to avoid such fluctuations would be the fluid properties, geometry and operating parameters, i.e. heat flux and mass flux, that result in the blanket ‘steady-state’ slug flow. Therefore, it is not necessary to report the transient effects of flow patterns when categorising them.

The transition from slug flow as the heat flux and/or vapour quality increases is a further discrepancy between publications for intermittent flow pattern categorisations seen in Figure 2.7. Many publications reported that the previously defined blanket slug flow transitions into a form of annular flow. However, multiple alternative publications reported an intermediate flow pattern; most commonly a reported churn flow, although differing flow patterns were reported by other authors. Hence, elucidation is required to decipher whether or not an intermediate flow pattern is observed for microscale flow boiling configurations and also whether the identification of the flow pattern is essential for engineering design.

Costa-Patry et al. [171] utilised the same bubble count frequency methodology as Revellin and Thome [125] to determine the prevailing flow patterns and as such, any intermediate flow pattern categorisation between the coalescing bubble regime and annular regime would not be classified. Some publications even reported the omission of a transitional flow pattern between slug and annular flows and stated that a transitional region did exist between the two flow patterns. Sobierska et al. [139] studied flow boiling of DI water in a single copper 1.2 mm hydraulic diameter rectangular microchannel. Whilst only bubbly, slug and annular flow patterns were reported, it was stressed that a transition region existed between the slug and annular regime.

Celata et al. [146] investigated flow boiling of FC-72 in a circular Pyrex microchannel and explicitly stated that the appearance of churn flow did not exist. Instead, they argued that a transitional flow pattern occurred in which long Taylor bubbles were momentarily separated by liquid slugs. This flow pattern was referred to by Celata et al. [146] as Slug/Annular flow. Once again, this appears to be a problem with universal terminology as opposed to phenomenological differences. The flow patterns observed by Celata et al. [146] are displayed in Figure 2.23 and the slug/annular flow pattern would be classified under the blanket term of slug as per the previous discussion. However, the photograph of the annular flow regime appears to show two different flow pattern phenomena. The right-hand side of the image, i.e. the upstream section of the channel with a lower vapour quality, does not appear to have the clearly defined vapour core surrounded by a liquid film that is synonymous with annular flow. Instead, it appears that there is still a large mixing of the phases that is typically associated with churn flow. However, the left of the image, i.e. the downstream section of the channel with a higher vapour quality, clearly shows the developed liquid film and vapour core. Once again, this inconsistency appears to be a product of a lack of universal terminology and classifications.

There were no other publications outlined in Table 2.3 that claimed churn flow did not occur, rather it was not discussed or dismissed as a transitional flow pattern. As discussed previously, transitional categorisations in which there is no benefit to engineering design may be best neglected. Hence, it must be discerned whether it is beneficial to distinguish between slug and churn flow or churn to annular flow, based on mechanistic phenomena and the significance to heat transfer and pressure drop. The most prominent mechanistic methodology for calculating heat transfer during slug flow is the three-zone model proposed by Thome et al. [191] that has since been modified by other authors. Modifications have been made to account for intermittent dryout of the vapour bubble tail caused by flow instabilities [192], coefficients to account for different film thicknesses [193] and bubble coalescence [194]. The appearance of churn flow is defined by a chaotic mixture of phases; however, the three-zone model considers a distinct liquid slug and elongated bubble region. Therefore, current mechanistic heat transfer models for slug flow are not applicable to the categorisation of churn flow.

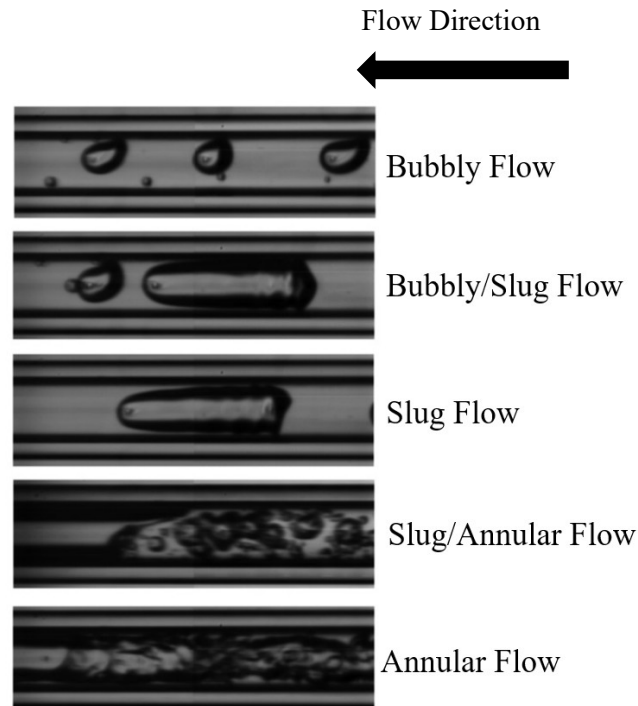


Figure 2.23. Photographs of flow patterns from Celata et al. [146] (flow from right to left).

Mechanistic heat transfer models for the annular flow regime have also been proposed in the literature. Patel et al. [195] developed a mechanistic heat transfer model by assuming one-dimensional radial conduction across a liquid film thickness that varies along the channel, permitting a calculation of the heat transfer coefficient at various lengths along the channel. Li and Hrnjak [196] developed a mechanistic model where it was assumed that annular flow is a system with a vapour core and liquid film, in which the liquid film is relatively uniform. The overarching similarities between the two mechanistic approaches are that the models are primarily based on the presence of a liquid film. As such, the chaotic liquid-vapour interface that occurs during churn flow cannot be mechanistically considered using these models. In fact, it may be that the random and chaotic nature of churn flow may make it very difficult to develop a suitable mechanistic model to determine heat transfer and pressure drop rates in this regime. Thus, more empirical tools may be required to develop prediction tools in the churn flow regime. Therefore, it can be considered an important flow pattern and should not be dismissed as only the transition between slug and annular flows.

2.3.2.4.3 *Annular Flow Patterns*

Annular flow was observed for all studies in which the heat flux was increased sufficiently that the flow transitioned from slug or churn flows. Further subdivisions of annular flow have been utilised, such as wispy-annular flow, semi-annular flow, slug-annular flow, wavy annular flow and smooth

annular flow, which is often also simply referred to as annular flow. Table 2.4 presents the terminology used to describe different flow patterns that would be classified in the annular flow group. It must also be ascertained as to whether there exist phenomenological differences between these subdivisions, whether such subdivisions provide ease in visual differentiation for classification and whether the phenomenological differences provide significantly different heat transfer and pressure drop characteristics.

Table 2.4. Reported flow patterns for the annular flow pattern group from direct diabatic studies.

Reported Flow Pattern	Fluids	Configuration	Hydraulic Diameters [mm]	Mass Flux Ranges Studied [kg/m ² s]
Annular	DI Water, Acetone, CO ₂ , R-134a, HFE-7100, FC-72, R-1234ze(E), R-245fa, Ammonia, R-1234yf, HFE-7200, HFE-7000	Rectangular microchannel, microchannel heat sink, Triangular channel heat sink, Circular microchannel	0.024 – 1.33	14 – 16765
Wispy-Annular	FC-77, DI Water	Multichannel heat sink, rectangular microchannel	0.096 - 0.707	261 - 961
Semi-Annular	DI Water, R-134a, R-1234yf	Circular microchannel, multichannel heat sink	0.155 – 1	70 - 900
Wavy-Annular	DI Water, R-134a, FC-72, R-1234yf	Multichannel heat sink	0.12 – 0.781	100 – 1373
Slug-Annular	R-134a	Circular Microchannel	1.33	100 – 500
Churn/Confined Annular	FC-72	Multichannel Heat Sink	0.096 – 0.707	225 - 1420
Churn/Wispy-Annular	FC-72	Multichannel Heat Sink	0.096 – 0.707	225 - 1420
Churn/Annular	FC-72, R-1233zd(E)	Multichannel Heat Sink	0.096 – 0.707	225 - 1420
Bubbly/Annular Alternating	DI Water	Rectangular microchannel	0.19	50 - 583
Annular Flow with Liquid Film	R-1234ze(E)	Circular microchannel heat sink	0.643	100 - 200
Annular Flow with Waves	R-1234ze(E)	Circular microchannel heat sink	0.643	100 - 200
Throat Annular	R-1234yf	Rectangular microchannel	0.5	251.6 – 579.2
Ring	R-134a	Rectangular microchannel	0.805	713 - 1629
Annular Slug	DI Water	Rectangular microchannel	0.4	668.95

Table 2.4. continued.

Reported Flow Pattern	Fluids	Configuration	Hydraulic Diameters [mm]	Mass Flux Ranges Studied [kg/m ² s]
Annular Slug	DI Water	Rectangular microchannel	0.4	668.95
Mist	DI Water, R-134a, HFE-7100, FC-72	Rectangular microchannel, circular microchannel	0.155 – 1.33	38.6 – 16765
Inverted Annular	DI Water	Multichannel heat sink	0.2174 – 0.4194	340 - 1373
Dryout Flow	R-1234yf	Rectangular microchannel	0.5	251.6 – 579.2
No Reported Annular Flows	DI Water	Multichannel heat sink, rectangular microchannel	0.333 – 0.94	12 - 500

The wispy-annular flow regime was reported in two studies by Harirchian and Garimella [135, 143] to differentiate from another reported annular flow pattern. The wispy-annular flow and annular flow regimes reported by Harirchian and Garimella [135, 143] are presented in Figure 2.24. The presence of irregular liquid droplets entrained in the vapour core, bubbles nucleating in the liquid film and a thicker liquid film was used to distinguish wispy-annular flow from typical annular flow. The irregularity of the liquid droplets entrained in the vapour core presents the same issues as the differentiation between dispersed bubbly flow and bubbly flow. The criteria of the sphericity difference of the droplets between wispy-annular flow and annular flow becomes highly subjective and difficult to measure. This can also be true for the liquid film thickness, although measurements are easier to conduct.

The greatest potential difference between the two subdivided categorisations is the presence of nucleation in the liquid film when considering heat transfer characteristics. This is not a phenomenon only observed by Harirchian and Garimella [135, 143]. Lee et al. [129] noted nucleation in the liquid film during annular flow when studying flow boiling of HFE-7200 in a microchannel heat sink. Alam et al. [126] and Al-Zaidi et al. [154] also reported nucleating bubbles in the liquid film for HFE-7100 in microchannel heat sinks. Borhani and Thome [197] also observed this phenomenon for FC-77 in a micro evaporator. The differentiation between annular flows in which nucleation is present or not in the liquid film may be significant due to the heat transfer mechanisms that occur and the role that certain operating parameters have on heat transfer rates.

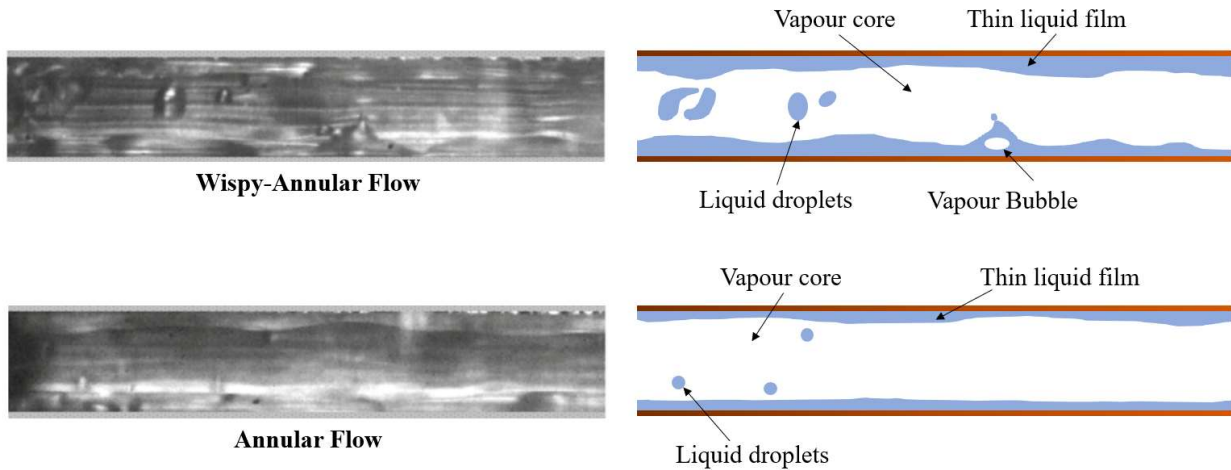


Figure 2.24. Wispy-annular flow and Annular flow reported by Harirchian and Garimella [143] (modified).

The presence of nucleation in the film may significantly enhance heat transfer due to increased agitation in the liquid phase. This effect would be further enhanced with increasing heat flux and the added resultant nucleation sites and bubble departure rate. However, the mass flux is typically the most significant operating condition that effects heat transfer rates in the annular flow regime due to increased convective effects and additional shear stress at the liquid vapour interface. If the enhancement of heat transfer in the liquid film due to bubble departure is insignificant in comparison to the convective effects then the need to distinguish between the wispy-annular and annular flow regime becomes redundant. The current state of microscale flow boiling research remains inconclusive in this matter, as many discrepancies still exist regarding the dominant heat transfer mechanisms. As such, it may not be beneficial to separate the annular flow regime into wispy-annular flow and annular flow with the requirement that future mechanistic models for annular flows allow for consideration of potential nucleation in the liquid film.

The slug-annular flow pattern was proposed by Martin-Callizo et al. [169] and differed from annular flow in that it resulted from the collision of neighbouring slugs that lead to a wavy-annular flow pattern. Hence, the phenomenon can be considered the same as the wavy-annular flow pattern that was reported by other authors. The primary problem that differentiating between wavy-annular and annular flow presents is the subjectivity in defining the transition. For example, it is documented that the transition between flow patterns is a gradual process. In the case of the transition between wavy-annular flow and annular flow, the amplitude of the waves in the liquid film will reduce with an increase in heat flux and the thickness of the film will reduce. Hence, the classification of annular flow will occur when it is deemed that the wave amplitude is sufficiently low. Again, this provides the opportunity for subjectivity

when defining flow patterns. As such, it is recommended that the wavy-annular and annular flow (or smooth annular flow) regimes are classified under the generic classification of annular flow.

The final flow pattern that is reported at the highest heat fluxes has been labelled as mist flow [141, 144, 166, 167, 169], inverted annular flow [145] and dryout flow [160]. This flow pattern refers to the phenomenon in which the entire channel wall is covered in a vapour blanket with liquid droplets running through the centre of the channel, thus drying out the channel wall. As a result, this flow pattern provides the lowest heat transfer rates and the dry-out of the wall can result in CHF. This causes a sudden dramatic rise in wall temperatures. As such, it was not reported by many researchers as experiments are typically terminated once this heat flux is reached for a given mass flux, fluid and geometry. In terms of engineering design, it is important to predict the CHF; however, its consideration in heat transfer and pressure drop is not a barrier to industry adoption.

2.3.3 Microscale Flow Pattern Prediction Tools

Many flow pattern prediction and transition tools have been proposed in the literature, often for a given geometry, fluid and operating conditions. Such prediction tools are typically presented using flow pattern maps with transition boundary lines separating the flow pattern categorisation regions. Although attempts to develop a universal flow pattern map have been made, the maps are only valid for a given set of operating conditions. Various coordinate systems have been proposed; however, these can be separated into three main groups [87, 116]:

Phase velocity flow pattern maps: The co-ordinates of the map are the liquid and gas superficial phase velocities or superficial phase mass fluxes.

Two-phase homogeneous model flow pattern maps: Parameters from the homogeneous flow model are used as co-ordinates. The most commonly used parameters are mass flux and vapour quality; however, the total Froude number, defined as the ratio of inertial forces to gravitational forces based on the phase mixture velocity and density, and void fraction have also been used.

Non-dimensional flow pattern maps: The flow pattern map co-ordinates are non-dimensional parameters relating to the phase parameters. This section reviews various previously developed microscale flow maps, which address different configurations such as both circular and non-circular geometries, horizontal or vertical flows and single or multiple channels.

Chen [198] developed detailed flow pattern maps based on indirect diabatic flow boiling experiments with R-134a at 6 bar, 10 bar and 14 bar in upwards vertical tubes. Mass fluxes between 50-6200 kg/m²s and vapour qualities between 0.0005 and 0.9 were studied in tubes with internal diameters of 1.1 mm, 2.01 mm 2.88 mm and 4.26 mm. The heated part of the test section was manufactured using stainless

steel and the flow visualised through a borosilicate glass tube downstream. The flow pattern maps developed by Chen utilised the superficial liquid and gas velocities, where the flow pattern map for R134a at 6 bar in a 1.1 mm tube is shown in Figure 2.25.

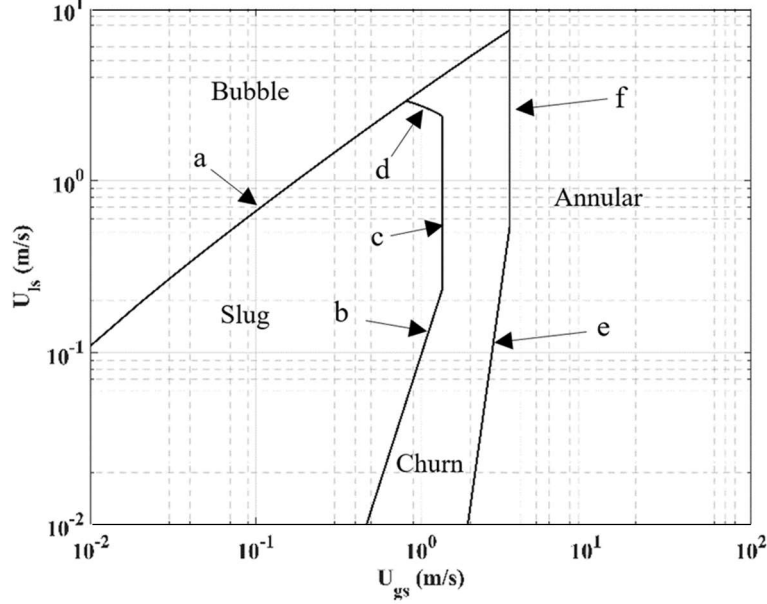


Figure 2.25. Chen flow pattern map for R134a at 6 bar in a 1.1 mm tube (taken from [198]).

The bubbly-slug and bubbly-churn boundary line, denoted by the letter a in Figure 2.25, was proposed by Chen to occur when the void fraction, α , reached a critical value, α_c . Equation (2.7) shows the void fraction in terms of the superficial velocities, the distribution parameter, CO and the drift velocity U_{drift} . Chen proposed that the critical void fraction is a function of the superficial velocities, as seen in Equation (2.8) where C_1 and C_2 are empirical constants. Chen found the empirical constants by plotting the void fraction at which the transition occurs to the homogeneous velocity, $U_h = U_{gs} + U_{ls}$, and assigned the values of $C_1 = 0.138$ and $C_2 = 0.344$.

$$\alpha = \frac{U_{gs}}{CO \cdot (U_{gs} + U_{ls}) + U_{drift}} \quad (2.7)$$

$$\alpha_c = C_1 (U_{gs} + U_{ls})^{C_2} \quad (2.8)$$

Chen also used measured values for the distribution parameter and drift velocity. However, Karayiannis et al. [199] universalised the transition correlation by suggesting Equation (2.9) and Equation (2.10) that were cited by Mishima and Ishii [200].

$$CO = 1.2 - 0.2 \sqrt{\frac{\rho_g}{\rho_l}} \quad (2.9)$$

$$U_d = 0.35 \sqrt{\frac{gD_h(\rho_l - \rho_g)}{\rho_l}} \quad (2.10)$$

Chen et al. proposed three boundary lines to describe the transition between slug and churn flow, accounting for three individual mechanisms through which the transition occurs. All boundary lines were obtained through empirical fits of data. The first transition, represented by line b in Figure 2.25 and Equation (2.11), occurs due to bubble elongation at low liquid superficial velocity. In equation 2.25, Re_{ls} , We_{gs} and Fr_{gs}^* represent the superficial liquid superficial Reynolds number, the superficial vapour Weber number and modified Froude number, respectively. At a certain bubble length, the surface tension force becomes relatively insufficient to maintain the liquid-vapour interface and the interface becomes unstable, resulting in a transition to chaotic churn flow.

$$Re_{ls} = 81.08 We_{gs}^{1.626} Fr_{gs}^{*-0.267} \quad (2.11)$$

The second transition criteria, corresponding to line c in Figure 2.25 and Equation (2.12), was proposed to occur due to deformation of the slug nose and tail as a result of turbulent flow at the tail of the slug. The slug will then distort, once again making the liquid-vapour interface unstable, at the higher superficial gas velocities.

$$U_{gs} = 587.1 \left(\frac{\mu_l}{\rho_g D_h} \right)^{1.447} \left(\frac{\rho_g D_h}{\sigma} \right)^{0.937} \quad (2.12)$$

The final slug to churn transition criteria, line d in Figure 2.25 and Equation (2.13), happens due to the liquid-vapour interface becoming unstable due to high liquid superficial velocities. Here, the high inertia forces created by the increased liquid superficial velocity result in an inability of the bubble to maintain its shape. Chen proposed that this transition line is dependent on the Fanning friction factor, f in Equation (2.14), where the correlation advanced by Jayanti and Hewitt [201] was cited.

$$U_h = 2.75 \sqrt{\frac{\sigma}{f \rho_l D_h}} \quad (2.13)$$

$$f = 0.046 Re_h^{-0.2} \quad (2.14)$$

The final boundary proposed by Chen, the churn to annular transition, was obtained by empirical fits of data to two lines, e and f shown in Figure 2.25. Lines e and f are represented by Equation (2.15) and Equation (2.16), respectively.

$$We_{ls} = 1.567 \times 10^{-17} (Fr_{gs} Re_{gs})^{3.41} \quad (2.15)$$

$$Fr_{gs} Re_{gs} = 3.119 \times 10^5 \quad (2.16)$$

Mahmoud and Karayiannis [82] modified the Chen flow pattern map by altering the bubbly to slug transition line. This modification was made to account for the effect of flow inertia, thermophysical fluid properties and channel size. This transition was developed analytically through the force balance on a spherical bubble, where gravity and viscous forces were neglected. This case is represented in Figure 2.26, where the bubble becomes confined when the bubble diameter, D_b , is the same diameter as the tube. At this condition the critical void fraction becomes 0.67 when the relative velocity Weber number is 4. A comparison between the two flow pattern maps was presented by Mahmoud and Karayiannis for data collected for R-245fa at 1.85 bar, which is shown in Figure 2.27.

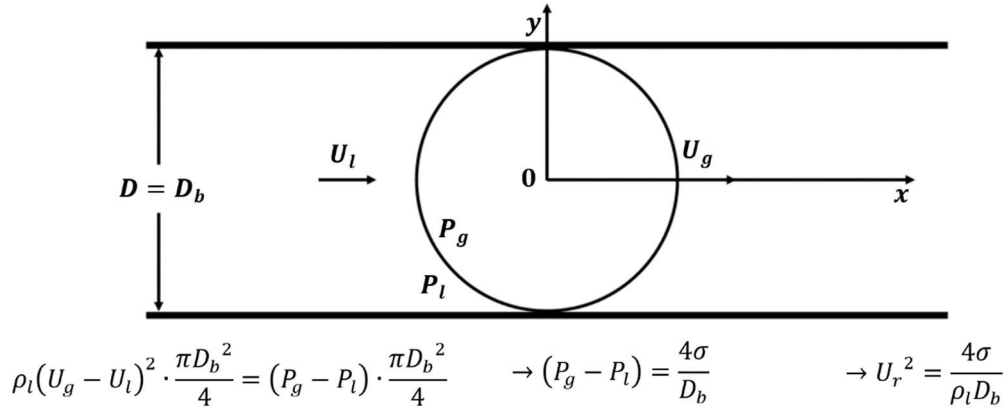


Figure 2.26. Drawing of force balance on a spherical bubble in which coordinate system moves with bubble (modified from [82]).

A diabatic flow pattern map was proposed by Revellin and Thome [125] based on experimental data using R134a and R245fa in horizontal tubes between 0.509 mm and 0.79 mm inner diameter and the previously explained bubble count frequency approach in Section 2.3.2.3. The flow pattern map was presented using the coordinates of mass flux and vapour quality. The isolated bubble to coalescing bubble transition boundary and the coalescing bubble to annular flow transition boundary are presented in Equation (2.17) and Equation (2.18), respectively. The inclusion of the boiling number, Bo , is noticeable in Equation (2.17). This accounts for heat flux effects and the consequent evaporation momentum force on the transition boundary. As such, Revellin and Thome [125] proposed that each flow pattern map is plotted for a given heat flux. The resultant proposed flow pattern map for R-134a

in a 70 mm long, 0.5 mm circular inner diameter tube at a heat flux of 50 kW/m² is presented in Figure 2.28. In Figure 2.28, the black line labelled as the lower band of experimental mass fluxes denotes the minimum mass flux for which experimental data were obtained. Below this limit, the flow pattern transition boundaries are extrapolated from the experimentally observed trends at higher mass fluxes.

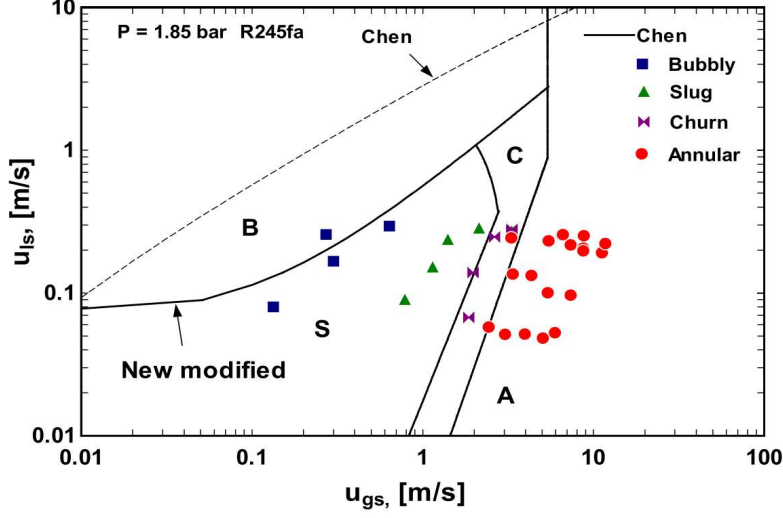


Figure 2.27. Mahmoud and Karayiannis modified Chen flow pattern map for R-245fa (taken from [82]).

$$x = 0.763 \left(\frac{Re_{lo} Bo}{We_{go}} \right)^{0.41} \quad (2.17)$$

$$x = 0.00014 Re_{lo}^{1.47} We_{lo}^{-1.23} \quad (2.18)$$

Ong and Thome [80] used the same experimental technique to classify flow patterns in 1.03 mm, 2.20 mm and 3.04 mm channel for refrigerants R-134a, R-236fa and R-245fa. They noted four flow regimes, namely; the isolated bubble regime, the coalescing bubble regime, the wavy-annular flow regime and the annular flow regime. For confinement numbers lower than 0.34, a fourth plug-slug regime was introduced to account for a long vapour bubble, surrounded by a slug that exhibited significant buoyancy effects. The Ong and Thome transition boundaries for the isolated bubble to coalescing bubble transition boundary and the coalescing bubble to annular flow transition boundary are presented in Equations (2.19) and (2.20), respectively.

$$x = 0.36 Co^{0.2} \left(\frac{\mu_g}{\mu_l} \right)^{0.65} \left(\frac{\rho_g}{\rho_l} \right)^{0.9} Re_{go}^{0.75} Bo^{0.25} We_{lo}^{-0.91} \quad (2.19)$$

$$x = 0.047 Co^{0.05} \left(\frac{\mu_g}{\mu_l} \right)^{0.7} \left(\frac{\rho_g}{\rho_l} \right)^{0.6} Re_{go}^{0.8} We_{lo}^{-0.91} \quad (2.20)$$

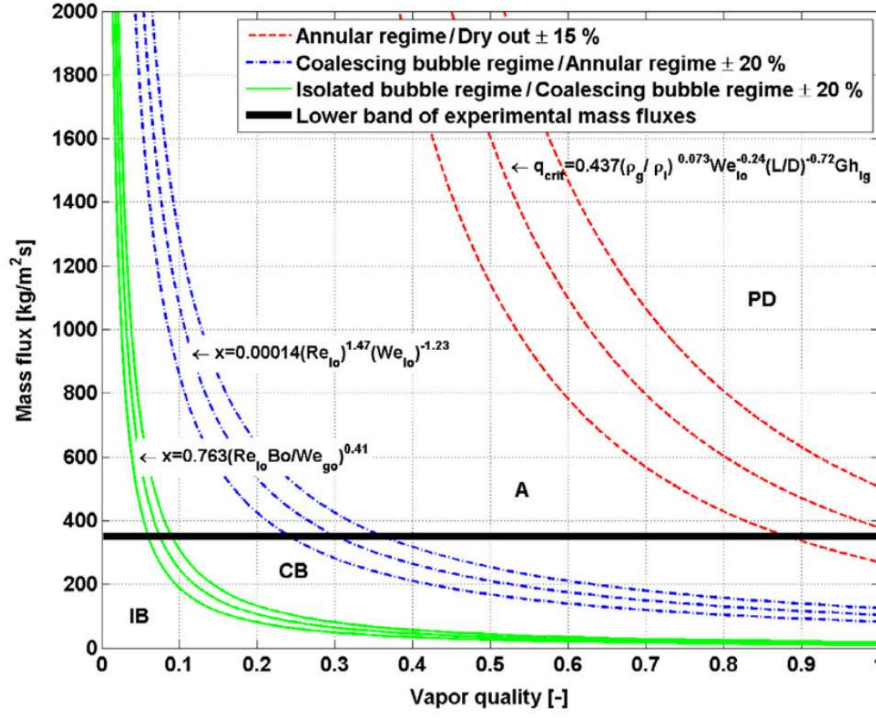


Figure 2.28. Flow pattern map proposed by Revellin and Thome [125] for R-134a in a circular microchannel and an applied heat flux of 50 kW/m².

Harirchian and Garimella [202] developed an initial flow pattern map based on flow boiling experiments in which FC-77 was utilised as the working fluid in a multichannel heat sink. The authors noted classical flow pattern categorisations of bubbly, slug, churn, wispy-annular and annular flows; however, they proposed that these flow patterns be grouped into two new categories. The two new groups were defined as the confined and unconfined regions, where the confined region included confined slug and annular flows, i.e. any flow pattern where a liquid film is present to confine the vapour phase. A non-dimensional flow pattern map was utilised in which the product of the liquid only Reynolds number and the square root of the Bond number, $Re_{lo} \cdot \sqrt{Bd}$, was used as the abscissa and the product of the Reynolds number and the boiling number used as the ordinate, $Re_{lo} \cdot Bo$. The resultant flow pattern map is displayed in Figure 2.29 for FC-77. The confinement criterion is presented in Equation (2.21), where $Re_{lo} \cdot \sqrt{Bd}$ was denoted as the convective-confinement number.

$$Re_{lo} \sqrt{Bd} = 160 \quad (2.21)$$

A second transition boundary was added to the map to establish the transition to annular flows. This empirical fit is given in Equation (2.22).

$$Bo = 0.017(Bd^{0.4}Re_{lo}^{-0.3}) \quad (2.22)$$

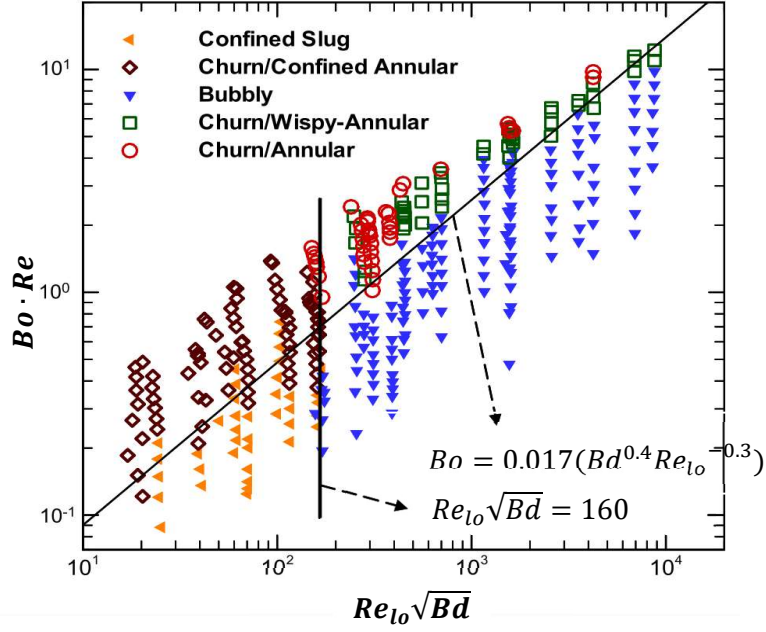


Figure 2.29. Flow pattern map for FC-77 proposed by Harirchian and Garimella (taken from [202]).

The same authors later modified the flow pattern map for the purpose of developing flow-regime based modelling of heat transfer and pressure drop [203]. The map was modified to use the phase change number, originally introduced by Shah et al. [204]. The phase change number is defined as the product of the frequency of vapour generation and the fluid particle residence time. This number can be reduced to Equation (2.23).

$$\varepsilon \xi = Bo \frac{L_{heat}}{D_h} \cdot \frac{\rho_l - \rho_g}{\rho_g} \quad (2.23)$$

Flow pattern measurements and categorisations were taken at various lengths along the channels and consequently, a new flow pattern map was developed using the new co-ordinates. This flow pattern map is shown in Figure 2.30 where a new criterion for the transition to annular flows replaced Equation (2.22), whilst Equation (2.21) remained. The new transition criterion is presented in Equation (2.24).

$$\varepsilon \xi = 96.65 (Re_{lo} \sqrt{Bd})^{-0.258} \quad (2.24)$$

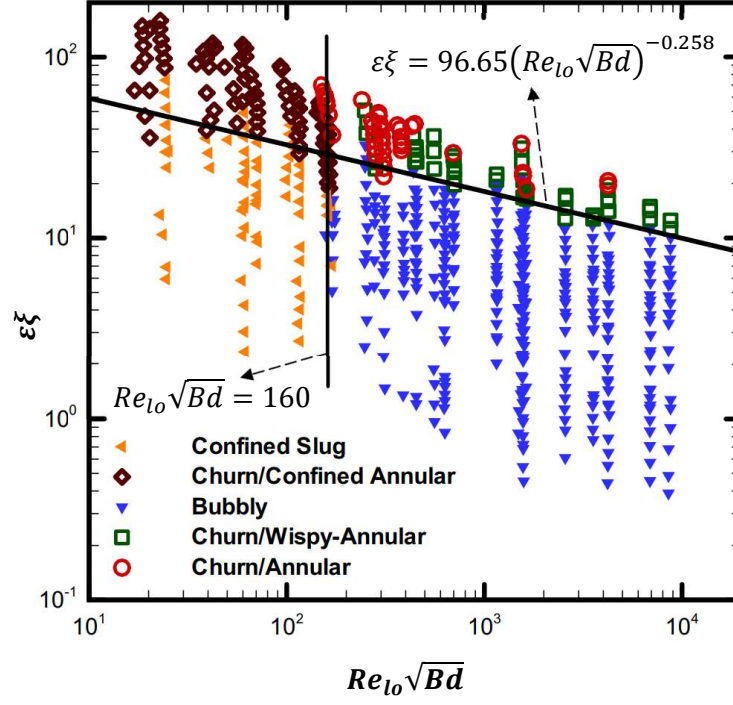


Figure 2.30. Flow pattern map for FC-77 proposed by Harirchian and Garimella (taken from [203]).

Tibiriça et al. [186] developed a complete set of correlations for the CHF, flow pattern transitions, heat transfer coefficients frictional pressure drop, void fraction, slip ratio and liquid entrainment fraction for flow boiling in microchannels. The correlations were required to be simple and easily implemented, which was defined by the authors as (a) a single continuous function with few dimensionless numbers (b) easy to linearise and (c) formatted so that the inverse function can be obtained analytically. Four correlations were developed for the bubble to slug, slug to churn, churn to annular and annular to dry out transitions, with a separate data bank used for each transition boundary line. The four banks included data for R-134a, R-236fa, R-245fa, R-1234ze(E), R-32, R-152a and nitrogen using either the indirect or direct diabatic experimental technique. However, much of the data included air-water mixtures obtained through the adiabatic experimental technique. Hydraulic diameters ranged between 0.25 mm and 3.04 mm. The transition boundaries for the bubble to slug, slug to churn and churn to annular are displayed in Equation (2.25), Equation (2.26) and Equation (2.27), respectively.

$$x = \left[1 + 1.268 \left(\frac{\rho_l}{\rho_g} \right)^{2.793} Re_{lo}^{0.3283} \left(\frac{\mu_l}{\mu_g} \right)^{-3.383} \right]^{-1} \quad (2.25)$$

$$x = \left[1 + 0.0721 \left(\frac{\rho_l}{\rho_g} \right)^{0.7215} Re_{lo}^{0.7864} La^{-0.3003} \right]^{-1} \quad (2.26)$$

$$x = \left[1 + 10^{-2.1765} \left(\frac{\rho_l}{\rho_g} \right)^{0.6534} Re_{lo}^{0.4444} Fr_{lo}^{0.2416} \right]^{-1} \quad (2.27)$$

Choi et al. [151] developed a flow pattern transition boundary between bubbly or slug flows to annular flows for the purpose of identifying the transition between the nucleate boiling heat transfer mechanism and the convective boiling heat transfer mechanism when studying FC-72 as the working fluid in a microchannel heat sink. This transition was plotted on a flow pattern map where the liquid-only Weber number was utilised as the ordinate and thermodynamic vapour quality was used as the abscissa. The resultant transition is displayed in Equation (2.28), whilst the proposed transition for FC-72 is shown in Figure 2.31.

$$We_{lo} = 0.021x^{-2.458} \quad (2.28)$$

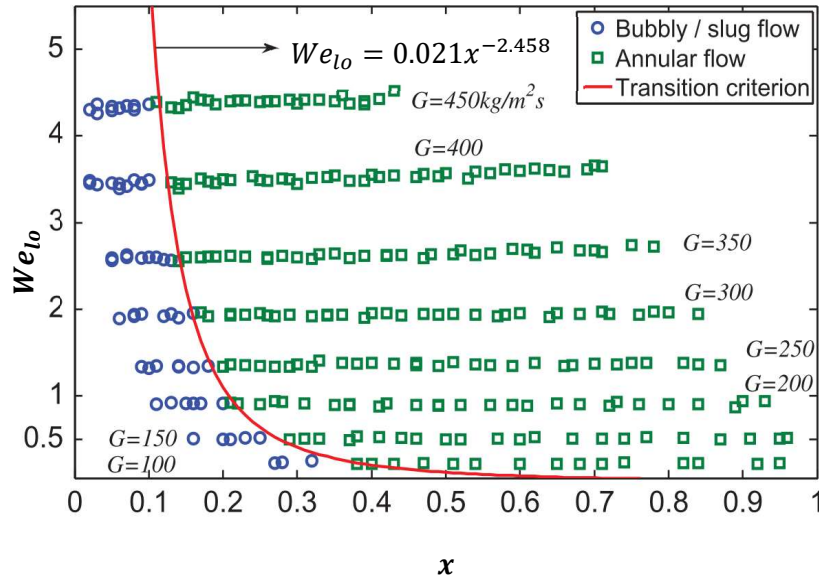


Figure 2.31. Proposed transition from bubbly/slug to annular flow for FC-72 by Choi et al. (modified from [151]).

Two empirical transition boundaries were proposed by Jige et al. [205] for R-32 in multiport tubes. Transition boundaries were suggested for the transitions between intermittent and transitional flow regimes, given in Equation (2.29), and transitional to annular flow pattern regimes in Equation (2.30). It was found that any existing correlations did not accurately reflect confinement effects as two hydraulic diameters were investigated. Hence, the use of the confinement number was included in the transition boundary correlations. The proposed flow pattern map for R-32 in 0.5 mm internal diameter multiport tubes is shown in Figure 2.32.

$$\frac{We_{lo}}{Co} = 2.3 \times 10^{-4} \left(\frac{We_{go}}{Co} \right)^{3.7} \quad (2.29)$$

$$\frac{We_{go}}{Co} = 75 \quad (2.30)$$

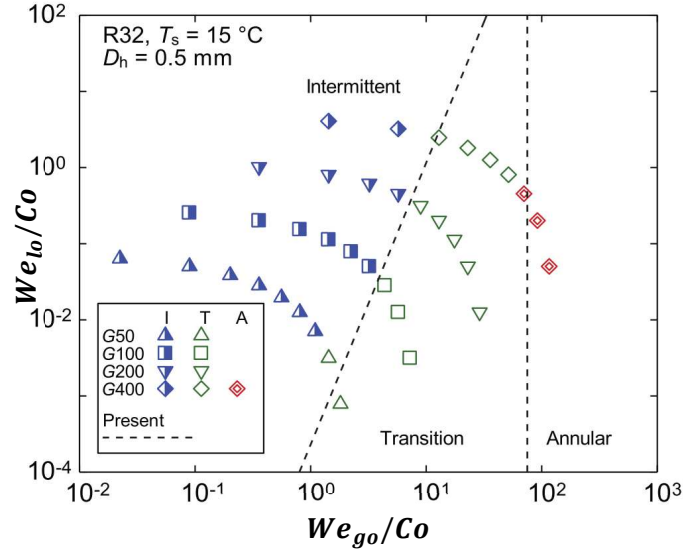


Figure 2.32. Proposed flow pattern map for R-32 in a 0.5 mm internal diameter multiport tube by Jige et al. (modified from [205]).

2.4 MICROSCALE FLOW BOILING HEAT TRANSFER

As discussed in the introductory chapter, the implementation of microscale flow boiling as a solution to a wide range of industrial applications relies on the accurate prediction of heat transfer coefficients. This ensures that the microscale heat exchanger is capable of maintaining suitable temperatures for the equipment that it is cooling. Accurate modelling of heat transfer coefficients is dependent on an understanding of the fundamental mechanisms of microscale flow boiling heat transfer, which is discussed in Section 2.4.1. The parametric effects on microscale flow boiling heat transfer is discussed in Section 2.4.2. Finally, existing microscale flow boiling heat transfer coefficient prediction tools are presented in Section 2.4.3.

2.4.1 Flow Boiling Heat Transfer Mechanisms

As previously mentioned, flow boiling heat transfer in large or conventionally sized channels is primarily driven by two dominant mechanisms: nucleate boiling and convective boiling. Nucleate boiling tends to dominate at low to moderate vapour qualities, typically associated with bubbly, slug, or churn flow regimes [206]. The enhancement in heat transfer during nucleate boiling arises from the rapid evaporation of the thin liquid layer beneath a growing bubble, as well as from the intense mixing

caused by bubble movement after detachment from the heated surface. According to principles derived from pool boiling, increases in heat flux lead to greater bubble departure sizes, higher bubble formation frequencies, and a broader range of active nucleation site diameters. As such, the traditional method of identifying nucleate boiling involves observing increases in the heat transfer coefficient with rising heat flux, while remaining relatively unaffected by changes in mass flux and vapour quality.

Conversely, at higher vapour qualities where the flow assumes an annular pattern, nucleation activity is significantly diminished and convective boiling becomes the predominant mechanism. This process closely resembles single-phase forced convection, albeit with an added enhancement resulting from higher vapour velocities and continued evaporation of the liquid film along the wall. The standard criterion for identifying convective boiling is based on the response of the heat transfer coefficient to variations in mass flux and vapour quality, typically increasing both, while showing little or no dependence on heat flux.

In microscale flow boiling, there does not exist the general agreement on the dominant heat transfer mechanisms [154]. Kim and Mudawar [7] proposed that nucleate boiling heat transfer is the dominant mechanism when bubbly and slug flow occupy a substantial portion of the channel. As the flow develops and nucleate boiling becomes progressively suppressed, the heat transfer coefficient steadily declines. This scenario is presented in Figure 5.33(a), which shows the local heat transfer coefficient with length. In contrast, Figure 5.33(b) illustrates a scenario where convective boiling is the dominant heat transfer mechanism. This occurs when annular flow prevails over a large portion of the channel length. Here, the heat transfer coefficient increases progressively along the channel, primarily due to the gradual thinning of the annular liquid film. A critical operational constraint for both nucleate and convective boiling is the dryout of this film. However, due to the asymmetric nature of the film's formation and depletion, localised dry spots begin to emerge at the point known as dryout incipience.

A final proposed heat transfer mechanism is that of thin film evaporation that occurs during the slug or annular flow regimes [207]. This process involves the evaporation of the liquid film that remains between the channel wall and the vapour core. Heat transfer during the thin film evaporation mechanism is related to the thermal conduction across the liquid film and is therefore strongly linked to the thickness of the film itself [208].

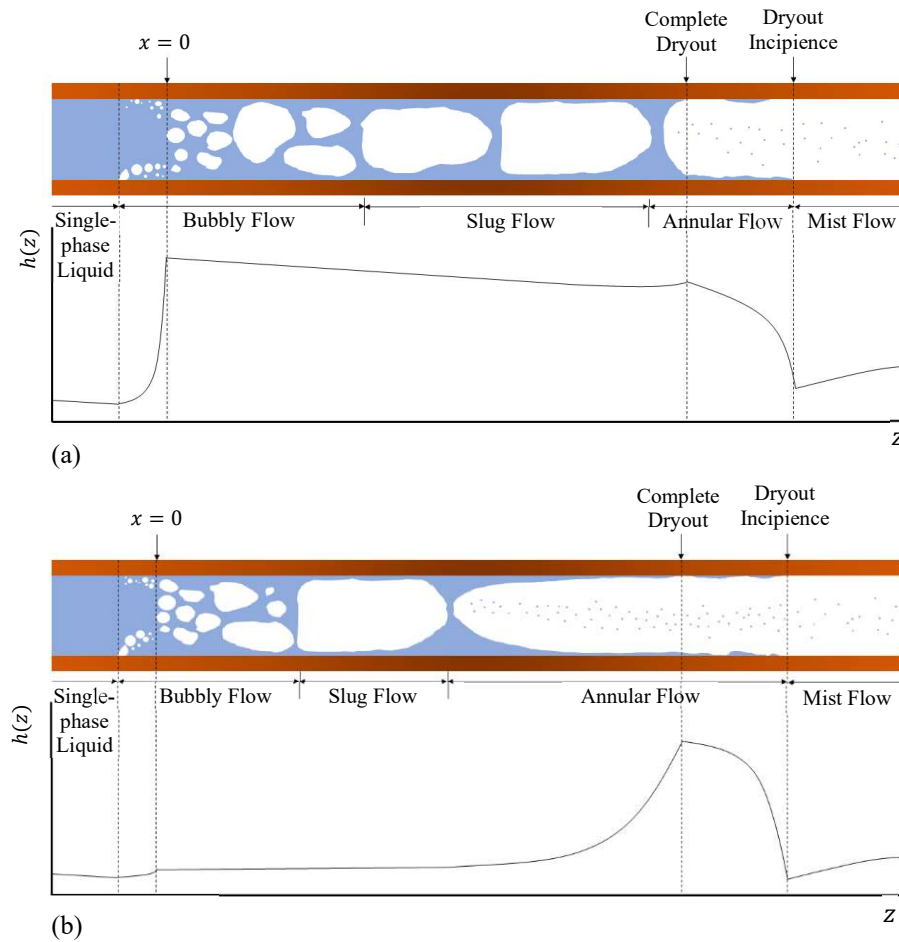


Figure 2.33. Schematics of flow regimes, wall dryout and variation of heat transfer coefficient along uniformly heated channel for; (a) nucleate boiling dominant heat transfer; (b) convective boiling dominant heat transfer (modified from [209])

2.4.2 Parametric Effects on Microscale Flow Boiling Heat Transfer

A thorough understanding of the influence of key parameters on both the development of flow boiling heat transfer mechanisms and the heat transfer performance within each mechanism is essential for the advancement of accurate predictive models. Critical parameters include heat flux, mass flux, system pressure, channel geometry, and surface morphology, all of which significantly affect the onset, evolution, and dominance of various boiling mechanisms. This section presents a detailed examination of these parametric effects and their subsequent effects on flow boiling heat transfer characteristics.

2.4.2.1 Effect of Heat Flux

Considerable disagreement exists among microscale flow boiling studies regarding the influence of heat flux on two-phase heat transfer coefficients. Numerous investigations have reported an increase in heat transfer coefficients with rising heat flux in microchannels [152, 210–213], a trend that is

commonly attributed to the presence and contribution of nucleate boiling within the flow. The increased number of nucleation sites and consequent rise in bubble departure frequency then results in the observed increased heat transfer coefficients [154].

There are a small number of publications that have reported a decrease in heat transfer coefficients with increasing heat flux. Markal et al. [214] observed significant flow reversal, a mechanism that can cause a significant loss in micro-scale heat exchanger thermal performance and arises due to rapid bubble growth, manifold geometry, boiling incipience location, certain surface characteristics or channel geometry [206]. They reported that the increase in heat flux significantly enhanced any flow reversal effects and resulted in a significant decrease in heat transfer coefficient due to bubble nucleation suppression. Other studies [215, 216] reported that the heat transfer coefficient declined sharply with rising heat flux after boiling commenced, but then had no impact at higher heat fluxes.

Other researchers reported that heat transfer coefficients increased with heat flux, but that the effect of heat flux became negligible at higher heat fluxes [217, 218]. Sun et al. [219] found that heat transfer coefficient increased with heat flux for HFE-7100 in a 180 μm microchannel in the nucleate boiling zone. Outside of the nucleate boiling zone, increased heat flux led to increased slug coalescence and annular flows, where nucleate boiling was suppressed and no heat flux effect was observed. Costa-Patry et al. [220] observed a comparable trend and attributed it to two opposing mechanisms: the enhanced evaporation rate at elevated heat fluxes, which tends to raise the overall heat transfer coefficient (based on local wall heat flux in the heat sink), and the occurrence of intermittent localised dryout. As heat flux increases, the frequency of these dryout events also rises, counteracting the benefits of increased evaporation and leading to a reduction in heat transfer performance. Alternatively, some researches [221, 222] have reported no effect of heat flux on the two-phase heat transfer coefficient, citing that the convective heat transfer mechanism dominates in microchannels.

Finally, a large number of researchers have reported that an increase in heat flux significantly reduced the two-phase heat transfer coefficient at higher heat fluxes [223–226]. Publications by Yu et al. [224] and Zhang et al. [227] observed that the two-phase heat transfer coefficient increased with heat flux up to a maximum; however, it then suddenly deteriorated. This was attributed to dry-out phenomena. However, prior to dry-out, the nucleate boiling mechanism was reported to dominate. Conversely, whilst Patra et al. [225] also observed the rapid decline due to dry-out at higher heat fluxes, they reported similar trends to Sun et al. [219] prior to dryout, where heat flux only affected two-phase heat transfer coefficients at lower heat fluxes.

2.4.2.2 *Effect of Mass Flux*

The influence of mass flux on flow boiling heat transfer in microchannels remains an area of ongoing investigation and, in many cases, debate. While nucleate boiling is widely acknowledged to be relatively

insensitive to mass flux variations, experimental evidence suggests that in microchannel systems, especially under conditions favouring convective boiling, mass flux can have a significant impact on heat transfer performance.

In many experimental studies, the enhancement of the heat transfer coefficient with increasing mass flux has been observed predominantly in convective boiling regimes. For instance, Qu and Mudawar [221] noted that in microchannel arrays, the heat transfer coefficient increased with mass flux due to enhanced droplet entrainment and deposition in the liquid film during annular flow. This trend was echoed by Sumith et al. [228], who observed a similar increase in heat transfer with mass flux in annular and slug-annular flow regimes, attributing the enhancement to the evaporation of thin liquid films. Their analysis identified a threshold vapour superficial velocity that marked a transition from nucleate to convective boiling, beyond which mass flux exerted a more pronounced influence. Further reinforcing this distinction, Lee and Lee [229] demonstrated that in rectangular microchannels with low aspect ratios, the heat transfer coefficient increased significantly with mass flux and vapour quality, while the effect of heat flux remained minimal. This suggested that convective mechanisms, rather than nucleate boiling, were dominant, with mass flux playing a critical role in maintaining liquid film integrity and enhancing evaporation rates.

Other works have continued to confirm that mass flux dependence is particularly marked in flow regimes where convective boiling dominates, such as annular or churn flow. Harirchian and Garimella [230] reported that heat transfer coefficients became increasingly sensitive to mass flux only after the suppression of nucleation at the channel walls. Higher mass velocities under these conditions enhanced wall wetting and delayed dry-out, thereby improving thermal performance. Similarly, Huang et al. [231] found that increasing mass flux in silicon microchannel heat sinks shifted the transition to annular flow to lower vapour qualities, magnifying the role of convective heat transfer. This trend has been supported by multiple studies, including those by Li and Jia [232], where flow pattern maps predicted increased dominance of convective boiling mechanisms with rising mass flux.

However, the mass flux effect is not universal across all flow regimes or operating conditions. In systems operating predominantly in the nucleate boiling regime, especially at low heat fluxes, the influence of mass flux tends to be negligible. Early studies such as those by Lazarek and Black [233] and Bao et al. [234] concluded that the heat transfer coefficient was predominantly governed by heat flux, with minimal sensitivity to changes in mass flux or vapour quality. This aligns with the classical understanding that nucleate boiling is largely heat flux driven.

In contrast, at elevated heat fluxes or higher vapour qualities where convective contributions increase, mass flux plays a more prominent role. Agostini et al. [235] found that in microchannels using R-245fa and R-236fa, the mass flux effect became significant only at medium to high heat fluxes, with the exact onset dependent on fluid properties and flow dynamics. Such observations point to the need

for considering both fluid type and operating regime when evaluating mass flux sensitivity. Nevertheless, the relationship between mass flux and heat transfer is not always monotonic. Several researchers have observed that beyond certain mass velocity thresholds, further increases may lead to diminished thermal performance. For example, Costa-Patry et al. [220] reported a decline in heat transfer coefficients at higher mass fluxes due to enhanced bubble coalescence and intermittent dry-out, particularly in the low vapour quality range. This finding challenges the notion that higher mass fluxes perpetually improve heat transfer performance if it has an effect at all.

Interestingly, several studies found weak or no dependence on mass flux [154, 236–238]. Bertsch et al. [239] even reported negative correlations between mass flux and heat transfer coefficient under certain experimental conditions. In some cases, higher mass flux led to lower vapour qualities at the exit, which indirectly affected heat transfer rates by altering the thermal boundary conditions at the channel wall.

2.4.2.3 Effect of System Pressure

Many studies have found an increase in heat transfer coefficients with system pressure [213, 238, 240, 241]. Lee et al. [241] investigated the effect of system pressure on the flow boiling characteristics of HFE-7200 for pressures between 1 and 2 bar in a heat sink with forty-four 0.48 mm hydraulic diameter microchannels at a constant mass flux of 200 kg/m²s. They found that increased system pressure increased the two-phase heat transfer coefficients due to increased bubble generation frequency and nucleation site density, where it was found that bubble departure diameter also decreased. A recent study by Al-Zaidi et al. [242] also found that nucleation site density increased with system pressure when conducting a study of the flow boiling of HFE-7100 in a micro-gap heat exchanger with a width of 20 mm and height of 1 mm, where bubble departure diameter was also found to decrease with increased system pressure. Both studies attributed these phenomena to the Hsu [243] correlation, which determines the required wall superheat for a cavity to become active.

In fact, much of the explanation of the effect of system pressure on heat transfer rates relies on an understanding of vapour entrapment inside a cavity and the conditions at which the cavity then becomes active. This is because boiling incipience and a large amount of the heat transfer that occurs during boiling on metallic surfaces have been proven to be a result of the nucleation of vapour bubbles from vapour embryos in surface cavities [244]. There is broad consensus that, before nucleation can take place, gas or vapour nuclei must first become trapped within surface cavities.

Various gas/vapour entrapment criteria have been suggested. Bankoff [245] and Lorenz [246] proposed a gas entrapment criterion for a conical cavity based on the displacement of the gas by the advancing liquid front towards the cavity. Both agreed that the three-phase contact angle, θ , must be greater than the cavity cone angle, 2β , for gas entrapment, as shown in Figure 2.34. This criterion

assumed that the contact angle was constant, despite the advancing liquid. The same criterion was also proposed by Tong et al. [247]; however, they argued that the dynamic advancing contact angle should be used instead.

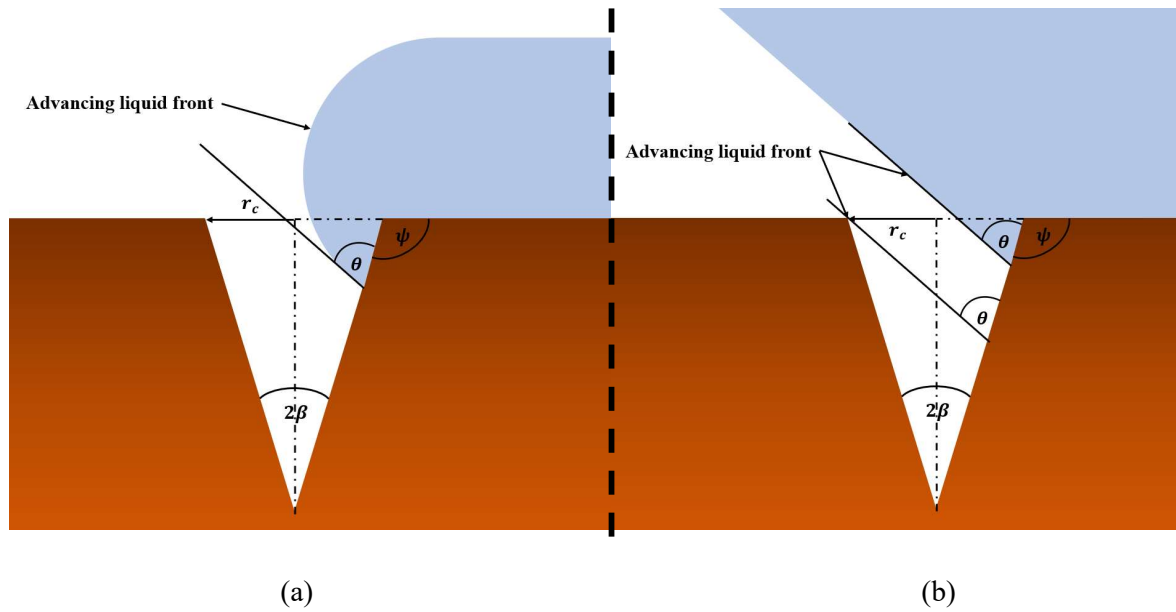


Figure. 2.34. Gas entrapment criteria: (a) by Bankoff [245] and (b) by Lorenz [246].

Some researchers have argued that gas or vapour entrapment is not possible in highly wetting fluids, i.e. those with a contact angle exceeding 90° . Wang and Dhir [248], through minimisation of the Helmholtz free energy, proposed that for entrapment to occur, the static contact angle must be greater than the cavity mouth angle, represented as ψ in Figure 2.34. In the case of a conical cavity this condition becomes $\theta > 90^\circ + \beta$. Building on this, Winterton [249] suggested that it is the advancing contact angle, as opposed to the static one, that must satisfy Wang and Dhir's [248] criterion. Their reasoning was grounded in the principles of mechanical and thermal equilibrium, particularly under subcooled conditions ($\Delta T_w \ll 0$), where the liquid-vapour interface within the cavity must be concave on the liquid side. The relationship between static contact angle and the necessary cavity aspect ratio for vapour entrapment is illustrated in Figure 2.35, where L_c represents the cavity depth.

Shi et al. [250] argued that bubble generation only emerges at individual cavities on the surface. They argued that if only a small volume of vapour is trapped in the cavity, then large superheats are required to activate the cavity. Whilst the criterion proposed by Wang and Dhir [248] and Winterton [249] in Figure 2.35 suggests that high wetting fluids, such as refrigerants, cannot facilitate vapour entrapment, all proposed criteria suggest that low wetting fluids, i.e. fluids with higher contact angles such as water, trap vapour for a wider range of cavities.

Provided that vapour entrapment has occurred within a potential cavity, certain conditions must then be met for a cavity to become active and for boiling incipience to occur. Two primary categories of models have been developed to predict the superheat at which boiling begins: those based on phase stability and those centred around the superheated boundary layer. Both modelling approaches typically start by examining the mechanical and thermodynamic equilibrium of a spherical vapour embryo within a cavity. This analysis generally involves the application of the Young–Laplace equation in conjunction with the Clausius–Clapeyron relation.

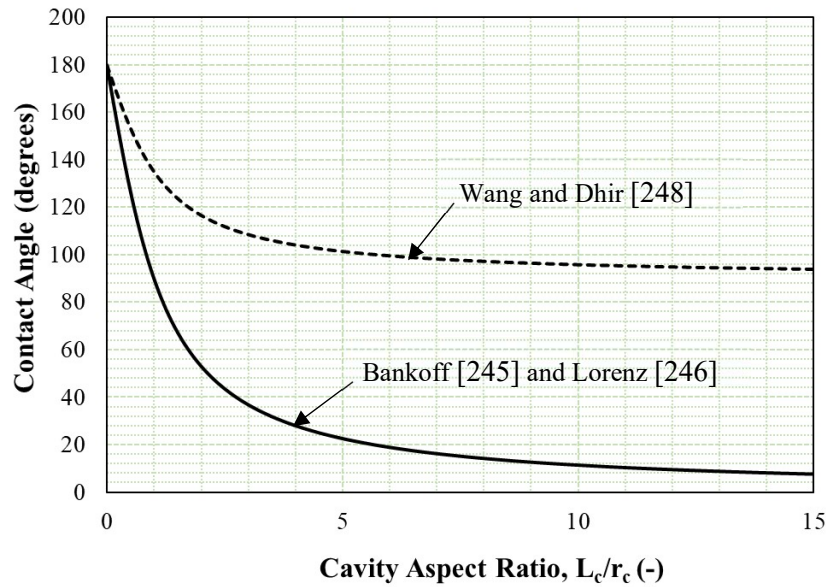


Figure. 2.35. Aspect ratio criteria for gas/vapour entrapment by Bankoff [245], Lorenz [246] and Whang and Dhir [248].

Phase stability models consider the conditions under which a liquid remains in a metastable state, where liquid is superheated beyond the thermodynamic equilibrium saturation temperature, but the mechanical equilibrium condition is still met. It is proposed that boiling incipience occurs when the liquid then transitions to an unstable state. Conversely, superheated boundary layer-based models consider the thermal boundary layer at the heated wall and suggest that nucleation occurs when the temperature of the liquid in the boundary layer is equal to or greater than that of the vapour trapped inside the bubble. Hsu [243] predicted the wall superheat required to activate a cavity containing trapped vapour using the superheated boundary layer-based incipience model, proposing the quadratic equation presented in Equation (2.32), below. Lee et al. [241] and Al-Zaidi et al. [242] applied the fluid thermophysical properties of HFE-7200 and HFE-7100 to Equation (2.31) for a range of pressures, finding that Hsu [243] predicts a wider range of active cavities at higher system pressures. In Equation (2.31), previously undefined parameters include the degree of subcooling, T_{sub} , the saturation temperature T_{sat} and the thermal boundary thickness layer, δ_{th} .

$$r_c(max, min) = \frac{\delta_{th}}{2} \cdot \frac{\sin\theta\Delta T_w}{(1+\cos\theta)(\Delta T_w+\Delta T_{sub})} \left[1 \pm \sqrt{1 - \frac{8(1+\cos\theta)\sigma T_{sat}(T_w+T_{sub})}{\delta_{th}\rho_g i_{fg} \Delta T_w^2}} \right] \quad (2.31)$$

Euh et al. [251] investigated bubble departure frequency for low-pressure subcooled boiling flows of water in a 19.81 mm diameter conventional tube and also found an increase of bubble departure frequency with a rise in system pressure. They described this phenomenon by linking bubble departure frequency to periodic wall temperature variation, which was originally proposed by Podowski et al. [252]. They argued that when a bubble nucleates, the wall superheat drives vapour formation, briefly cooling the surface to the saturation temperature. As the bubble departs and single-phase convection becomes dominant, the wall temperature rises until it reaches the nucleation threshold again, resulting in a cyclic temperature pattern, where the rate of rise and peak wall temperature determine the bubble waiting time. They linked the bubble waiting time to the required wall superheat for nucleation, which, if lower, will reduce the bubble waiting time as the surface does not require as long to heat up. They derived Equation (2.32) by considering the mechanical equilibrium of a spherical vapour nucleus, the Clausius Clapeyron equation and the ideal gas law. It can be seen from Equation (2.32) that an increase in vapour density decreases the wall superheat and, consequently, reduces the waiting time, thereby increasing the bubble departure frequency.

$$\Delta T_{sat} = \frac{2\sigma T_{sat}}{r_c \rho_g i_{fg}} \quad (2.32)$$

Euh et al. [251] also linked the bubble growth time to the effect of system pressure based on the force balance developed by Situ et al. [253] and Klausner et al. [254] and shown in Figure 2.36. They stated that the forces acting on a departing bubble at the nucleation site are the unsteady drag force, F_{du} , gravity, F_{gr} , pressure force, F_p , quasi-steady force, F_{qs} , surface tension force, F_σ and shear lift force, F_{sl} . For clarification, the unsteady drag force acting on a bubble results from its interaction with the surrounding fluid, particularly when there are changes in the bubbles motion [255]. This is particularly true during flow boiling at the stage of bubble growth. The pressure force is a result of the pressure differential between the vapour phase inside the bubble and the surrounding liquid [256]. The quasi-steady force represents the drag experienced by a moving bubble when it is assumed that the surrounding fluid adjusts instantaneously to the bubble's motion. This approach treats the bubble as if it were travelling at a constant velocity through a steady flow, despite the presence of unsteady effects in reality [257]. Finally, the shear lift force is a result of the liquid velocity changes in position which occurs in the boundary layer, where the bubble nucleates. The bubble then experiences a lateral force from the surrounding fluid, acting perpendicular to its primary direction of motion [258]. Situ et al. [253] proposed that a bubble lifts from the surface when the force balance shown in Equation (2.33) is violated.

$$\sum F_y = F_{\sigma y} + F_{duy} + F_p + F_{gr} + F_{qs} \quad (2.33)$$

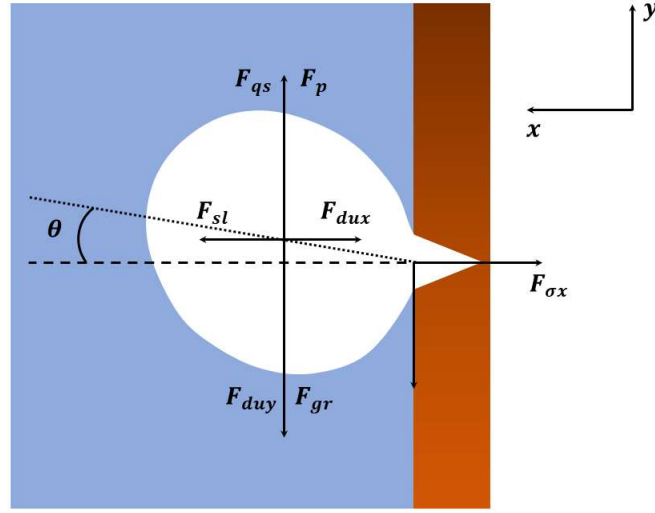


Figure 2.36. Force balance of a vapour bubble at a nucleation site (modified from [253]).

Situ et al. [253] also proposed an equation for the unsteady drag force, also referred to as the bubble growth force, which is shown in Equation (2.34) and Equation (2.35), where C is a constant and θ is the contact angle shown in Figure 2.36. Euh et al. [251] showed that the unsteady drag force reduced with increased system pressure, due to an increase in density of the vapour phase. They argued that in order to maintain force equilibrium, both buoyancy and quasi-steady drag forces must be diminished, leading to a smaller bubble departure size. This reduction in departure size shortens the bubble growth time, as the departure size serves as a boundary condition in the energy equation governing bubble growth.

$$F_{duy} = -C\kappa_l^2 \rho_l J a_e^4 \sin \theta \quad (2.34)$$

$$J a_e \equiv \frac{\rho_l C_{p,l} (T_w - T_{sat})}{\rho_g i_{fg}} S \quad (2.35)$$

Huo et al. [259] investigated flow boiling in small diameter vertical tubes ranging between 2.01 mm and 4.26 mm internal diameter using R-134a as the working fluid. They observed an increase in heat transfer coefficient across all vapour qualities for pressures between 8 and 12 bar. They proposed that this could possibly be due to the change in latent heat of vaporisation. Xu et al. [240] also investigated the flow boiling of R-134a in tubes, using copper pipes with inner diameters of 1.002 mm, 2.168 mm and 4.065 mm. They also reported an increase of heat transfer coefficient with increasing system pressure and attributed this to the decrease in surface tension. However, they argued that the increase

in vapour to liquid density ratio is not beneficial to the two-phase heat transfer coefficient but the effect was not as significant as the impact of reduced surface tension. This is contrary to the work conducted by Euh et al. [251] who proposed that the increased vapour density increases bubble nucleation frequency.

Despite many researchers observing a rise in two-phase heat transfer coefficients with system pressure, other studies have reported no significant system pressure effect or that the pressure effect is only evident for certain operating parameters. Wen et al. [260], for instance, examined flow boiling in serrated fin minichannels and reported that raising the pressure from 2.9 to 4.5 bar led to improved heat transfer coefficients when the vapour quality remained below approximately 0.5. This enhancement was attributed to an increase in bubble generation activity. Wen et al. [260] also suggested that the increase in heat transfer was due to a decrease in latent heat of vaporisation, citing that that more vapour is generated at lower wall superheats which results in a greater number of bubbles. A potential criticism of this explanation is that the generation of more vapour alone does not necessarily equate to a greater bubble generation frequency. For example, if the departing bubbles were larger, then a greater mass of vapour would still be produced but the bubble departure frequency would remain the same.

Karayiannis et al. [261] investigated the impact of system pressure on flow boiling in a 1.1 mm diameter tube and found that heat transfer performance was enhanced only at low vapour qualities, specifically, when the vapour quality was below 0.3. Increasing the pressure from 6 bar to 10 bar led to a noticeable rise in heat transfer coefficients below this threshold. However, this improvement did not extend to higher vapour qualities, where no significant change was observed. The enhancement at low vapour quality was linked to reduced surface tension at elevated pressures, which promotes smaller bubble departure sizes.

In contrast to studies reporting improved heat transfer at elevated pressures, Saisorn et al. [262] observed a significant decline in heat transfer coefficients during R-134a flow boiling in a 1.75 mm diameter tube, when the system pressure was increased from 8 to 10 bar. They attributed this reduction to the combined effects of lower latent heat of vaporisation and decreased liquid viscosity at higher pressures. These conditions were believed to produce thinner liquid films along the tube walls, which are more susceptible to rupture. The resulting local dryout reduced surface wetting and was believed to ultimately lead to diminished heat transfer performance.

2.4.2.4 Effect of Heated Length

The effect of heated length on two-phase heat transfer rates is a relatively less investigated parameter in flow boiling. This is surprising as it has been proposed as a potential reason for the discrepancies between results for different publications in which flow boiling results are presented [261]. Karayiannis et al. [261] investigated the effect of heated length on flow boiling for upwards vertical flow of R-134a

in a 1.1 mm inner diameter tube. They found that length of the heated section had a notable influence on the local heat transfer behaviour. This was attributed to the fact that the relationship between the heat transfer coefficient, heat flux, and vapour quality varied depending on the heated length. The heat transfer coefficient remained relatively insensitive to changes in vapour quality in the shortest tube tested but generally increased with rising heat flux unless dry-out occurred. However, as the heated length increased, the coefficient began to show a clear upward trend with vapour quality beyond a value of approximately 0.3, while the impact of heat flux diminished in the high-quality region.

Özdemir [263] investigated the effect of heated length on the flow boiling of water in test sections comprising of a single rectangular microchannel. The test sections were manufactured to have nominally identical surface characteristics, using the same machining parameters, although small, unavoidable variations in surface condition may have been present. Both test sections had a constant hydraulic diameter of 0.56 mm and an aspect ratio of 2.56, with heated lengths of 62 mm and 86.8 mm, respectively. The shorter channel was found to consistently exhibit higher flow boiling heat transfer coefficients. This behaviour was attributed to factors such as enhanced nucleation activity (potentially influenced by surface characteristics), the requirement for higher heat flux to achieve equivalent exit quality, and the increased pressure drop in the longer channel, which led to lower local saturation temperatures and reduced heat transfer performance.

2.4.3 Existing Heat Transfer Prediction Tools

Existing microscale flow boiling heat transfer mechanisms have been proposed, often based on the parametric effects discussed above. As previously discussed, the ability to accurately predict two-phase heat transfer coefficients will aid in the implementation of microscale flow boiling as an industrial cooling solution in regards to two aspects:

- a) Assurance that the microscale heat exchanger is correctly sized and is capable of dissipating the required heat.
- b) Assurance that the temperature of the equipment to be cooled is kept within permissible operating limits.

Current approaches for predicting heat transfer during flow boiling in both conventional and microscale channels can be broadly categorised into four main groups [264]:

1. Superposition models: These models determine the overall two-phase heat transfer coefficient, h_{tp} , by combining the individual contributions from nucleate boiling, h_{nb} , and convective boiling, h_{cb} , heat transfer mechanisms. An example of this approach is the correlation developed by Chen [265] for macroscale vertical tubes. The general structure of superposition models is represented by Equation (2.36).

$$h_{tp} = h_{nb} + h_{cb} \quad (2.36)$$

2. Enhancement models: In this approach, the two-phase heat transfer coefficient is treated as an enhancement of the single-phase heat transfer coefficient, h_{sp} , typically through the use of an empirically derived enhancement factor, E . This methodology assumes that phase change enhances convective heat transfer beyond that of single-phase flow. A representative example is the correlation proposed by Lee and Mudawar [266]. The general form of enhancement models is given in Equation (2.37).

$$h_{tp} = E h_{sp} \quad (2.37)$$

3. Asymptotic models: These models share a similar structure with superposition models but incorporate an asymptotic index (n) to account for the diminishing contribution of either nucleate or convective boiling under certain conditions. This modification allows for a smoother transition between the dominant heat transfer mechanisms. A prominent example of this modelling approach is the correlation developed by Kim and Mudawar [267]. The general form of asymptotic models is shown in Equation (2.38).

$$h_{tp}^n = h_{nb}^n + h_{cb}^n \quad (2.38)$$

4. Dimensionless models: In this category, the two-phase heat transfer coefficient is expressed as a function of key dimensionless parameters, such as Reynolds, Prandtl, Boiling, and Weber numbers, often derived through empirical fitting of experimental data. These models aim to capture the underlying physics in a generalised form, enabling broader applicability across different operating conditions. The typical form of dimensionless models is presented in Equation (2.34). The application of dimensionless models is especially prevalent in recent literature, with a recent example presented by Zhang et al. [210].

$$h_{tp} = f(Bo, Re, Bd, We, X, \dots) \frac{k_l}{D_h} \quad (2.39)$$

The flow boiling heat transfer correlations compared to in this study are presented in Appendix A. Shah [268] introduced a correlation that selects the higher value between the nucleate and convective boiling heat transfer coefficients. In this formulation, the nucleate boiling component is expressed as a function of the Boiling number, while the convective boiling component depends on the Convection number. This correlation demonstrated validity across a wide range of experimental conditions, encompassing various fluids, including water, R-11, R-12, R-22, and R-113, covering both vertical and horizontal channels with diameters between 6 and 25.4 mm.

Warrier et al. [269] conducted an experimental study on flow boiling of FC-84 within horizontal rectangular multi-microchannel heat sinks. The device, fabricated from aluminium, featured five parallel channels each with a hydraulic diameter of 0.75 mm. Their experiments were performed under heat fluxes up to 59.9 kW/m² and mass fluxes ranging from 557 kg/m²s to 1600 kg/m²s. The researchers incorporated both the Boiling number and vapour quality into their newly developed correlation. Additionally, the correlation included a single-phase heat transfer coefficient tailored for laminar, fully developed flow. This correlation was validated for vapour qualities between 0.03 and 0.55, and Boiling numbers ranging from 0.00027 to 0.00089.

Lee and Mudawar [266] conducted an experimental investigation into flow boiling heat transfer using two fluids, water and R-134a, in horizontal parallel rectangular microchannels. These channels, constructed from oxygen-free copper, had a hydraulic diameter of 0.348 mm. The study covered heat fluxes ranging from 159 kW/m² to 938 kW/m² and mass fluxes between 127 kg/m²s and 654 kg/m²s. They developed three distinct correlations corresponding to different vapour quality intervals: $x < 0.05$, $0.05 < x < 0.55$ and $x > 0.55$. Their results indicated that nucleate boiling was the dominant heat transfer mechanism at vapour qualities below 0.05, whereas convective boiling prevailed at moderate to high vapour qualities. The proposed correlations express the heat transfer coefficient as a function of the liquid Weber number, Boiling number, Martinelli parameter, and the single-phase heat transfer coefficient.

Mahmoud and Karayiannis [270] developed an empirical correlation for flow boiling of R-134a in vertical stainless-steel tubes with diameters between 0.52 mm and 4.26 mm. Their correlation was formulated using an extensive dataset of 5,152 points, covering heat fluxes from 1.7 kW/m² to 158 kW/m² and mass fluxes ranging from 100 kg/m² to 700 kg/m². They introduced a novel experimental enhancement factor inspired by the method originally proposed by Chen [271]. Additionally, their two-phase heat transfer coefficient incorporated the pool boiling correlation developed by Cooper [272] within its structure.

Li and Jia [232] proposed two correlations for saturated flow boiling in horizontal rectangular multi-microchannels: one targeting the nucleate boiling dominant regime and the other addressing the convective boiling regime. The study used R-134a as the working fluid within three-sided heated multi-microchannels having a hydraulic diameter of 0.5 mm. Experiments were carried out at heat fluxes reaching up to 802.12 kW/m² and mass fluxes ranging between 373.33 kg/m² and 1244.44 kg/m².

Thiangtham et al. [149] conducted an experimental study on flow boiling heat transfer of R-134a in horizontal multi-microchannels. The test section comprised of twenty-seven rectangular microchannels fabricated from oxygen-free copper, each with a hydraulic diameter of 0.421 mm. Their experiments spanned heat fluxes up to 3127 kW/m², mass fluxes between 150 kg/m² and 600 kg/m² and outlet vapour qualities ranging from 0.05 to 0.92. The authors highlighted the significant influence of flow patterns

on heat transfer performance. At lower heat fluxes, when bubbly and slug flow regimes prevailed, the heat transfer coefficient was observed to increase with heat flux, indicating nucleate boiling as the dominant mechanism. Conversely, at higher heat fluxes, the heat transfer coefficient rose with mass flux during wavy-annular and annular flow regimes, suggesting convective boiling dominance. However, beyond wall heat fluxes of 80 kW/m^2 , the heat transfer coefficient declined due to partial dryout. To capture these effects, an empirical correlation for flow boiling heat transfer was proposed, incorporating dimensionless parameters such as the Boiling number, liquid Weber number, and liquid Reynolds number.

Shah [273] conducted a comparison of an extensive dataset against their earlier correlation from 1982. This dataset encompassed various channel geometries, including both partially and fully heated channels with hydraulic diameters of 3 mm or less. While the original correlation provided reasonable predictions for most of the data, it tended to underpredict heat transfer coefficients at low flow rates. This discrepancy was attributed to the influence of surface tension forces, which dominate at low flow velocities, whereas inertial forces prevail at higher flow rates. To characterise this balance, the Weber number was proposed, representing the ratio of inertial to surface tension forces, noting that surface tension effects are dominant when the Weber number is below 100. Additionally, it was observed that the Weber number decreases with increasing Boiling number. Accordingly, a revised correlation was developed that integrates both the Weber and Boiling numbers, introducing a modification factor to adjust the 1982 model. This updated correlation was validated against a database of 4,852 data points, covering 31 different fluids, a variety of channel shapes, orientations (vertical and horizontal), and heating conditions (fully or partially heated channels) with diameters ranging from 0.38 to 27.1 mm and mass fluxes from 15 to $2,437 \text{ kg/m}^2\text{s}$. The correlation demonstrated good predictive accuracy, achieving a mean absolute error (MAE) of 18.6%.

Zhang et al. [210] conducted a visualisation-based experimental study to investigate the influence of hydraulic diameter on flow boiling heat transfer in rectangular microchannels using DI water as the working fluid. Experiments were performed across six microchannels with hydraulic diameters ranging from 0.25 mm to 1.5 mm, under heat fluxes from 203 to 880 kW/m^2 and mass fluxes between 200 $\text{kg/m}^2\text{s}$ and 600 $\text{kg/m}^2\text{s}$. A new dimensionless model was proposed for the two-phase heat transfer coefficient, formulated through a Spearman rank correlation analysis and incorporating dimensionless groups including the Reynolds number, Weber number, Boiling number, and vapour quality.

Patra et al. [225] developed a new empirical correlation for saturated flow boiling heat transfer in a microchannel evaporator using acetone and methanol as working fluids. The aluminium test section consisted of twenty-nine square microchannels, each measuring $0.5 \text{ mm} \times 0.5 \text{ mm}$ in cross-section and 45 mm in length. Experiments were conducted over heat fluxes ranging from 30 kW/m^2 to 640 kW/m^2 and mass fluxes between 88 $\text{kg/m}^2\text{s}$ and 988 $\text{kg/m}^2\text{s}$. A universal two-phase heat transfer prediction tool was proposed, using the boiling number, liquid only Weber number and liquid only Froude number as

a single-phase heat transfer enhancement multiplier. However, a fluid specific parameter was required which was only provided for acetone and methanol.

The correlations presented can be seen to differ substantially in their underlying assumptions, governing parameters, and ranges of applicability. While many models report good agreement with the specific datasets from which they were derived, their predictive accuracy often deteriorates when applied outside those conditions, particularly across different fluids, channel geometries, surface characteristics, and flow regimes. The strong dependence of heat transfer on factors such as vapour quality, flow pattern, surface tension effects, and channel scale highlights the complex and highly coupled nature of microscale flow boiling. Consequently, there remains no universally accepted correlation capable of reliably predicting heat transfer performance across the broad range of operating conditions relevant to practical microscale cooling applications. This motivates continued experimental investigation and comparative assessment of existing models, such as that undertaken in the present study, to improve understanding and guide the development of more robust predictive tools.

2.5 MICROSCALE FLOW BOILING PRESSURE DROP

Understanding two-phase pressure drop in microchannels is critical for their successful implementation in industrial applications as it directly impacts system efficiency, reliability, and design feasibility. Two-phase flow introduces complex pressure dynamics due to rapid phase changes, high interfacial shear, and variable flow regimes. Excessive pressure drops can lead to flow maldistribution, pump oversizing, reduced cooling performance, and, in severe cases, flow reversal or dry-out. Moreover, accurate prediction of pressure drop is essential for optimal system design, ensuring that microchannels are neither under or over-engineered, which would otherwise lead to inefficiencies or unnecessary cost. This section outlines the reported parametric effects on two-phase pressure drop and presents existing two-phase pressure drop prediction tools and models that were developed for both conventional and microscale channels.

2.5.1.1 *Effect of Heat Flux*

The influence of heat flux on pressure drop in microchannel flow boiling systems is well established, yet far from straightforward. While it is generally accepted that increasing heat flux tends to elevate two-phase pressure drop, the extent and nature of this effect are highly sensitive to channel geometry, flow regime, fluid properties, and thermal boundary conditions.

In most cases, increasing heat flux results in a rise in pressure drop due to enhanced vapour generation. These bubbles expand rapidly and disrupt the liquid flow, increasing the local void fraction

and introducing substantial shear and interfacial resistance. This phenomenon has been widely observed in both single and multi-channel configurations. For example, in a study using DI water for flow boiling in stainless-steel tubes with diameters ranging from 1.05 to 2.44 mm, Tong et al. [274] found that pressure drop increased consistently with heat flux, attributing this to the intensified evaporation and resultant vapour acceleration in the tube. Similar results were reported by Lee and Garimella [217], who also studied DI water in horizontal multi-microchannels and observed that the pressure drop increased rapidly with heat flux due to accelerated bubble growth and coalescence near the channel walls.

This heat-flux-driven increase in pressure drop is not only due to higher vapour quality but also to the flow regime transitions that heat flux often induces. As the boiling process moves from bubbly to slug or annular flow with increasing thermal input, the nature of two-phase interaction changes significantly. The shift to annular flow, in particular, is known to produce high shear stresses between the vapour core and the surrounding liquid film, thereby amplifying frictional losses. Qu and Mudawar [275], although focusing on heat transfer, associated annular flow regimes with high channel pressure drops due to these mechanisms. Costa-Patry et al. [276] demonstrated that R-245fa showed a steeper increase in pressure drop with vapour quality than R-236fa, attributing this to the higher volumetric expansion of vapour at a given heat flux.

However, it is not just the flow pattern regime that has an effect; the channel geometry also moderates how heat flux impacts pressure drop. Expanding channels, as studied by Balasubramanian et al. [277], showed reduced two-phase pressure drop at higher heat fluxes compared to straight channels, due to downstream deceleration of vapour resulting from increased cross-sectional area. Conversely, microchannels with embedded grooves or cavities, such as those used by Zeng et al. [278], experienced higher pressure drops as heat flux increased. The added surface area and channel complexity enhanced bubble nucleation, but also introduced additional flow resistance and liquid retention zones that amplified pressure losses.

Yet, these general trends are not universal. Several studies have reported a limit or even a reversal of the pressure drop response at high heat fluxes. Lee and Mudawar [279] observed that beyond a certain thermal input, pressure drop in microchannels levelled off or slightly decreased. This was attributed to the complete vaporisation of the fluid downstream, which eliminated the two-phase mixture and thus reduced frictional pressure losses in that region. This counterintuitive result suggests that under certain conditions, particularly at high vapour qualities or with sufficiently long heated lengths, the system may transition to a predominantly single-phase vapour regime, where pressure drop stabilises or even declines with further heat input. However, when complete vaporisation has occurred this would no longer be classified as two-phase flow, but rather single-phase gas or vapour flow.

Another important consideration is the role of flow stability and confinement effects. In channels with high confinement, vapour bubbles are more restricted in their movement, leading to elevated local

pressures when heat flux increases. Soupremanien et al. [280], using Forane-365HX as the working fluid in two channels with the same hydraulic diameter but different aspect ratios, found that the lower aspect ratio channel, with greater lateral confinement, exhibited significantly higher pressure drops at elevated heat flux, attributed to enhanced bubble confinement and greater wall interaction. This contrasts with findings from Singh et al. [281], who observed a non-monotonic dependence of pressure drop on aspect ratio under similar conditions.

Experimental observations are not entirely aligned when it comes to very small diameters or high heat flux extremes. Xu et al. [240], investigating R-134a in tubes of 1.002 to 4.065 mm diameter, found the frictional pressure drop remained almost unchanged with increasing heat flux, without offering a clear explanation for the anomaly. A similar finding was reported by Tibiriçá and Ribatski [282], where heat flux appeared to have no significant effect on two-phase pressure drop in vertical tubes using R-245fa, despite variations in mass flux and vapour quality. These findings suggest that under specific combinations of fluid properties, orientation, and operational parameters, the conventional pressure drop–heat flux relationship may be masked by other dominant mechanisms, such as mass flux or inlet subcooling.

Some systems have shown that pressure drop initially increases with heat flux but later decreases due to a shift in flow structure. Yin et al. [283] observed in open microchannels with DI water that the pressure drop first increased with heat flux as bubbly flow transitioned to slug flow. However, with the onset of stratified flow at higher heat flux, the pressure drop declined, owing to reduced interfacial friction between the phases. However, as per the previously reviewed flow pattern literature, the presence of stratified flow would render this irrelevant to the conversation of microscale flows.

2.5.1.2 Effect of Mass Flux

Most studies examining flow boiling pressure drops have observed a strong positive correlation between increasing mass flux and pressure losses. For instance, Tong et al. [274] and Karayiannis et al. [284] observed that increasing mass flux led to greater pressure drops in stainless-steel tubes and vertical channels due to rising shear and reduced stabilisation time for vapour bubbles. In heat sink studies, Harirchian and Garimella [230] found that the pressure drop for FC-77 increased significantly with mass flux in silicon microchannels of varying width, rising from 11 kPa to 24 kPa as mass flux increased and channel width decreased. This relationship held for other fluids as well: Agostini et al. [285] reported pressure drops exceeding 110 kPa for R-236fa and 120 kPa for R-245fa at high mass fluxes, with the variation attributed to differences in fluid density and manifold design. Similarly, Roldão et al. [286] noted a strong pressure drop increase with mass flux for R-123, particularly due to frictional forces in circular microchannels.

One of the primary drivers of this relationship is the interplay between frictional and accelerational pressure drop components. As shown by Zhang et al. [227], higher mass velocity enhances both components, especially in systems where high vapour generation creates significant shear between phases. This behaviour is further amplified in narrow or high-aspect-ratio geometries. In microchannels with hydrophobic walls, such as those tested by Sun et al. [219], increased vapour generation at higher mass flux led to even steeper pressure gradients due to reduced liquid film stability and increased vapour hold-up. These conditions accelerate transitions to flow regimes like annular or churn flow, both of which are known to generate substantial pressure fluctuations.

Interestingly, while the dominant trend supports increased pressure drop with mass flux, there are exceptions where higher mass fluxes produce more stable flow and can even reduce pressure losses. In a study by Deng et al. [287], higher mass fluxes resulted in lower vapour quality for a given heat flux, effectively shortening the boiling length and reducing the pressure drop over the same heated region. Likewise, Zhang et al. [210] found that elevated mass fluxes suppressed bubble backflow and expansion in DI water systems, thereby stabilising flow and lowering pressure spikes. This stabilisation effect was also reported by Widyatama et al. [226], who showed that higher mass flux reduced bubble residence time and chaotic fluctuations in HFE-7000 flow, maintaining a lower average pressure drop.

Some studies also reveal a non-monotonic behaviour. Markal et al. [288] observed that total pressure drop initially decreased with mass flux before increasing again at higher values. This was explained by the fact that stronger inertial forces at higher flow rates could overcome upstream resistance caused by vapour momentum during bubble nucleation, effectively reducing the net pressure drop despite higher frictional losses. A similar balance between frictional and acceleration terms was suggested by Chen and Garimella [135], who found that pressure drop remained relatively insensitive to mass flux in FC-77 microchannels, likely due to competing effects between the two components.

2.5.2 Existing Pressure Drop Prediction Tools

Reliable prediction of flow boiling pressure drop in microchannels is a critical aspect of designing pumped two-phase cooling systems. Accurately estimating pressure losses within the microchannel heat sink, together with those occurring in the rest of the flow loop, enables appropriate pump selection and facilitates optimisation of the system's power consumption and overall performance. Existing analytical and empirical pressure drop correlations have been proposed for both conventional channels and microchannels.

The pressure drop for two-phase flow can be separated into three components, namely the frictional pressure drop component, ΔP_{fric} , the acceleration pressure drop component, ΔP_{acc} , and the gravitational pressure drop component ΔP_{gr} . The two-phase pressure drop components are presented in Equation (2.40). In flow boiling, the frictional pressure drop arises from two primary sources: the shear

interaction between the fluid and the channel walls as it moves through the heated passage, and the interfacial friction between the liquid and vapour phases within the flow [289]. The acceleration pressure drop occurs as a result of the substantial density reduction accompanying liquid-to-vapour phase change. This density variation leads to an increase in flow velocity, causing a change in momentum and, consequently, a drop in pressure along the channel [290]. Finally, the gravitational pressure drop simply refers to the head loss due to changes in height, i.e. there is no gravitational pressure drop in horizontal channels.

$$\Delta P_{tp} = \Delta P_{fric} + \Delta P_{acc} + \Delta P_{gr} \quad (2.40)$$

The prediction of two-phase pressure drop is typically approached using one of two correlation forms: the homogeneous flow model or the separated flow model. These approaches differ fundamentally in how they characterise the flow. The homogeneous model simplifies the two-phase mixture by treating it as a single-phase fluid with averaged properties, applying a unified velocity and friction factor. In contrast, the separated flow model recognises the distinct motion of the liquid and vapour phases, necessitating individual phase velocities and a detailed treatment of void fraction and interfacial shear. The primary differences between the two approaches therefore centre on their treatment of void fraction and frictional resistance. The correlations examined in this study are presented in full in Appendix B.

In the homogeneous flow model, it is assumed that the liquid and vapour phases are completely mixed and travel at the same velocity along the channel, meaning the slip ratio is equal to 1 [89]. This simplification is most suitable for dispersed bubbly flow regimes, where the velocity difference between the vapour bubbles and the surrounding liquid is relatively small. Under this model, the frictional pressure drop is not calculated separately for each phase. Instead, it is described using a two-phase friction factor which depends on the two-phase Reynolds number. The Reynolds number itself is determined using the effective viscosity of the mixture which reflects the combined flow characteristics of both phases. Qu and Mudawar [275] later used 0.003 for the two-phase friction factor for use in the homogeneous flow model. Various two-phase viscosity models have been proposed by other researchers for use in the homogeneous flow model [291–293]. The acceleration pressure drop in the homogeneous flow model is a function of the homogeneous void fraction, which is presented in Appendix B.

The separated flow model does not assume that the liquid and vapour phases have the same velocity and that consequently, there is slippage between phases. In separated flow models, the frictional pressure drop in two-phase flow is commonly represented by scaling the single-phase liquid pressure drop using a two-phase multiplier. The first two-phase multiplier was proposed by Lockhart and Martinelli [294] for conventional channels, which was expressed as the ratio of the frictional pressure gradients of the two phases. A relationship between the two-phase multiplier and the Martinelli

parameter, defined in Equation (2.41), was later recommended by Chisholm [295]. The relationship is presented in Equation (2.42), where the value of the Chisholm constant, C , is dependent on whether the liquid and vapour phases are in laminar or turbulent regimes.

$$X = \sqrt{\frac{\left(\frac{dP}{dz}\right)_l}{\left(\frac{dP}{dz}\right)_g}} \quad (2.41)$$

$$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \quad (2.42)$$

Qu and Mudawar [275] carried out an experimental study on hydrodynamic instability and flow boiling pressure drop in horizontal, rectangular multi-microchannels. The test setup featured a copper heat sink containing 21 parallel channels, each with a hydraulic diameter of 0.35 mm. Water was used as the working fluid, with mass fluxes ranging from 135 kg/m²s to 400 kg/m²s. Based on their findings, the authors developed a two-phase pressure drop correlation that accounts for the effects of channel dimensions and mass flux. To improve the predictive capability of the model, the Chisholm constant was adjusted using experimental data. Lee and Garimella [217] investigated flow boiling in horizontal, rectangular multi-microchannels using water as the working fluid. The experimental setup involved a silicon heat sink with channels having hydraulic diameters between 0.162 mm and 0.571 mm. To enhance the accuracy of two-phase pressure drop predictions, they also modified the Chisholm constant by expressing it as a function of both mass flux and hydraulic diameter.

In another experimental study, Keepaiboon et al. [296] examined the flow boiling pressure drop of R-134a in a horizontal, single rectangular microchannel with a hydraulic diameter of 0.68 mm. The experiments were conducted across a heat flux range of 7.63 kW/m² to 49.46 kW/m² and mass fluxes varying from 600 kg/m²s to 1400 kg/m²s. From their results, the authors developed a correlation incorporating a coefficient formulated to account for the effects of the liquid-only Reynolds number, the confinement number, and the Martinelli parameter.

Huang and Thome [297] investigated the flow boiling pressure drop characteristics of three refrigerants, R-1233zd(E), R-245fa, and R-236fa, within horizontal square multi-microchannels. The experimental setup featured a silicon heat sink containing 67 parallel microchannels, each with a hydraulic diameter of 0.1 mm. A wide range of operating conditions was examined, with heat fluxes varying from 200 to 640 kW/m² and mass fluxes between 1250 and 2750 kg/m²s. The researchers introduced a revised Chisholm constant that incorporated key flow parameters, specifically the liquid-only Reynolds number and the superficial vapour Reynolds number, to improve the accuracy of pressure drop predictions.

Markal et al. [298] developed two pressure drop correlations tailored for low mass flux conditions, derived from an experimental investigation using de-ionised water in horizontal rectangular multi-microchannels. The experiments were conducted over a heat flux range of 36 kW/m^2 to 121.8 kW/m^2 and mass fluxes between $51 \text{ kg/m}^2\text{s}$ and $324 \text{ kg/m}^2\text{s}$. A total of nine silicon heat sinks were employed, each with varying aspect ratios from 0.37 to 5 and hydraulic diameters ranging from 0.1 mm to 0.25 mm. The first correlation focused on frictional pressure drop, where the Chisholm constant was reformulated to incorporate the influence of several parameters: liquid-only Reynolds number, aspect ratio, Boiling number, Weber number, hydraulic diameter, and the two-phase flow length. For flows characterised by bubble confinement, a separate dimensionless correlation was introduced to estimate the total pressure drop more accurately.

Gao et al. [299] investigated two-phase flow of ammonia in horizontal stainless-steel tubes, focusing on frictional pressure drop behaviour. The primary experiments were conducted using tubes with an 8 mm inner diameter, across a mass flux range of $20 \text{ kg/m}^2\text{s}$ to $100 \text{ kg/m}^2\text{s}$ and saturation temperatures from $-15.8 \text{ }^\circ\text{C}$ to $4.6 \text{ }^\circ\text{C}$. To expand the scope of their analysis, data from previous experiments using 4 mm diameter tubes were also included. The study began by evaluating the accuracy of existing pressure drop correlations against the full dataset. Among those tested, the Müller-Steinhagen and Heck [300] correlation provided the most accurate predictions, with 80% of the experimental data falling within a $\pm 30\%$ error band. Based on this strong performance, the authors used it as a foundation for developing a new frictional pressure drop correlation. The proposed model introduced the Reynolds number and Bond number to better capture the influence of fluid properties and tube dimensions. To assess its general applicability, the correlation was validated using 951 data points drawn from eleven independent sources. This validation dataset covered a wide range of operating conditions, including various working fluids, both horizontal and vertical flow orientations, tube diameters from 0.81 mm to 14 mm, mass fluxes between 50 and $800 \text{ kg/m}^2\text{s}$, and vapour qualities ranging from 0 to 1.

Moradkhani et al. [301] developed a comprehensive empirical model to predict saturated flow boiling frictional pressure drop across a broad spectrum of channel sizes, from macro-scale to mini- and micro-scale. Their method utilised genetic programming to formulate an explicit two-phase multiplier, building on a modified version of the separated flow model. Genetic programming is an artificial intelligence technique that uses evolutionary, stochastic, and population-based search methods to automatically generate programs or functions that best fit a given dataset [302]. Unlike traditional optimisation algorithms, genetic programming simultaneously evolves both the structure and parameters of linear or nonlinear functions without assuming any predefined model form.

The model was both trained and validated using an extensive database consisting of 6,021 experimental data points gathered from 42 different studies. This dataset included a wide variety of working fluids and covered channel hydraulic diameters ranging from 0.0695 mm to 14 mm, as well as mass fluxes from $50 \text{ kg/m}^2\text{s}$ to $1,504 \text{ kg/m}^2\text{s}$. The genetic programming algorithm incorporated several

key input parameters: the ratio of Reynolds numbers, Bond number, exit vapour quality, pressure drop ratio, and reduced pressure.

More recently, Al-Zaidi et al. [303] formulated two correlations aimed at predicting microscale pressure drops, drawing on an extensive database comprising data from vertical tubes and horizontal heat sinks. The dataset included multiple fluids, R-134a, R-245fa, HFE-7100, and HFE-7200, covering a range of hydraulic diameters between 0.46 mm and 4.26 mm, heated lengths from 20 mm to 500 mm, system pressures spanning 1 to 14 bar, mass fluxes from 50 to 700 kg/m²·s, and wall heat fluxes reaching up to 234 kW/m². Their correlations focused on estimating two-phase multipliers, which showed a strong correlation with the Boiling number, while also incorporating the Martinelli parameter, superficial Reynolds number, and reduced pressure as influential variables.

A substantial proportion of the pressure drop correlations proposed for microscale flow boiling are based on separated flow formulations in which the two-phase multiplier is derived from the Lockhart–Martinelli framework, with predictive accuracy improved through modification of the Chisholm constant. While this approach often yields good agreement with the specific experimental datasets used for model development, the Chisholm constant is frequently adjusted or empirically fitted to account for particular channel geometries, fluids, flow regimes, and operating conditions. As a result, the physical generality of such correlations may be limited, and their applicability to alternative microscale configurations or broader operating ranges remains uncertain. This reliance on dataset-specific tuning highlights the continued challenge of developing pressure drop models that are both accurate and transferable, and underscores the need for further experimental validation and the development of correlations with more robust physical foundations for microscale two-phase flows.

2.6 CRITICAL HEAT FLUX

The implementation of flow boiling in microchannels is dependent on knowledge of the operational limits for which it can be used. The CHF defines the maximum heat transfer rate beyond which a further increase in heat input causes a sharp deterioration in thermal performance [304]. At this threshold, the coolant can no longer maintain adequate wetting of the heated surface, leading to the development of a vapour layer that acts as a thermal insulator. Once CHF is exceeded, the surface temperature can rise rapidly, posing a serious risk of thermal damage or failure.

The CHF in both pool boiling and flow boiling arises from a sharp decline in local heat transfer, caused by the inadequate contact between the liquid and the heated surface, which promotes the formation of an insulating vapour layer [2]. In the case of pool boiling, heat transfer initially occurs by natural convection, without the generation of vapour bubbles. As the wall temperature increases beyond the point of nucleation, bubbles begin to appear at active sites on the heated surface [305]. With further

increases in wall superheat, bubble generation and detachment become more vigorous, improving heat transfer through latent heat uptake and enhanced fluid mixing. At higher intensities, bubbles merge and form vertical jets or columns [306]. Ultimately, the surface reaches CHF, at which stage a continuous vapour film develops, drastically diminishing heat transfer and initiating the transition to film boiling [307]. This vapour layer can cause a rapid rise in wall temperature, posing a severe threat to the structural integrity of the surface and potentially leading to catastrophic system failure.

In flow boiling, CHF is similarly governed by the loss of liquid-to-wall contact, but here it is often associated with dryout, a condition in which regions of the channel wall are no longer wetted by the liquid phase [308]. Although the absence of liquid contact is the underlying cause, different models have been suggested to explain the precise mechanisms leading to dryout. The following sections examine the various CHF mechanisms proposed in the literature, as well as the correlations developed for predicting CHF in flow boiling.

2.6.1 Critical Heat Flux Mechanisms

Reported CHF mechanisms are generally classified into two groups: subcooled CHF mechanisms and saturated CHF mechanisms [309]. One such subcooled mechanism, described by Kim and Mudawar [310] and also noted in other studies [311–314], is termed the departure from nucleate boiling (DNB). In this scenario, a vapour layer develops along the heated channel walls while liquid continues to pass through the channel core, as shown in Figure 2.37. The formation of this vapour film triggers a sharp rise in wall temperature, signalling a severe reduction in heat transfer efficiency. DNB typically arises in strongly subcooled flows subjected to high mass flux and elevated heat flux conditions.

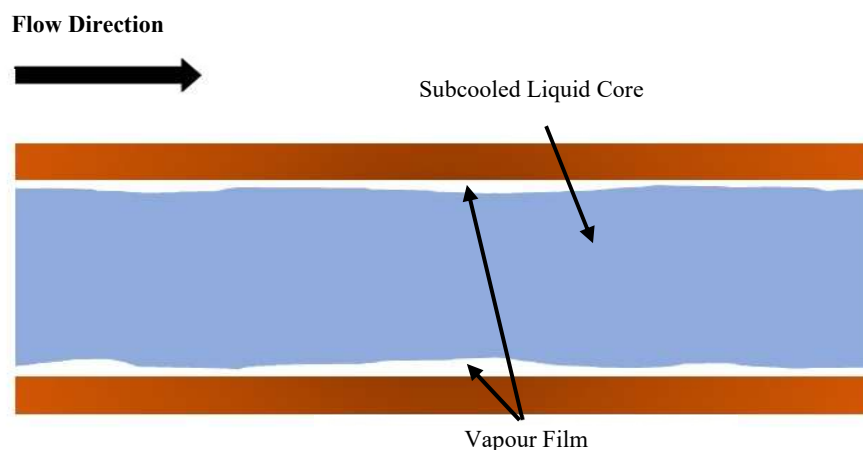


Figure 2.37. Departure from nucleate boiling CHF mechanism.

Another frequently cited subcooled CHF mechanism is bubble congestion (or overcrowding), described in several studies [315–317]. In this case, the boundary layer adjacent to the heated wall

becomes saturated with vapour bubbles. As the number of bubbles increases, detachment is inhibited and dry spots form on the surface. Once released into the cooler bulk flow, many bubbles condense; the resulting cycle of rapid growth and collapse produces significant disturbances in the boundary layer [318]. Moreover, the excessive bubble generation impedes liquid rewetting, as accumulated bubbles physically block contact between the coolant and the heated wall [319], ultimately precipitating CHF.

A further subcooled CHF mechanism was introduced by Kirby et al. [320], centred on the presence and stability of a dry macro-layer beneath vapour bubbles. Their model, based on the flow boiling of water at 1.72 bar, suggests that once the wall reaches a critical temperature, the thin liquid film beneath the bubbles evaporates more quickly than it can be replenished. This produces dry patches directly under the bubbles, severely reducing local heat transfer and initiating CHF.

Tong and Hewitt [321] proposed a more general dryout mechanism, relevant to both subcooled and saturated boiling. Here, when the wall heat flux becomes excessive, the thin liquid film separating the heated surface from an adjacent vapour slug is entirely vaporised. The resulting dry patches drive a rapid rise in wall temperature, leading to CHF. Extending this idea, Liu et al. [322] developed a mechanistic model in which detached bubbles coalesce into extended vapour blankets. The growth of these blankets depends on the departure size of individual bubbles, limited by surrounding vapour structures. Downstream, the blankets fragment under Helmholtz instabilities as the liquid sub-layer beneath them continues to evaporate. CHF is reached once this sub-layer is fully depleted, leaving sections of the surface directly exposed to vapour.

In saturated flow boiling, dryout beneath vapour slugs has been proposed as a possible CHF trigger. However, the dominant mechanism is generally associated with the annular flow regime, where CHF often occurs first. Even relatively low heat fluxes can induce CHF in this regime, provided the heated length is sufficiently long [323]. Annular flow has therefore been the focus of extensive study, since it is both common under saturated boiling conditions and characterised by high void fractions and elevated vapour quality, both of which heighten the likelihood of dryout.

Hewitt [85] identified five distinct dryout mechanisms within the annular flow regime, each capable of triggering CHF under specific conditions. These mechanisms, illustrated in Figure 2.38, are outlined below:

- a) Vapour film: Very high heat fluxes result in a vapour blanket occurring at the channel wall, surrounded by the liquid film. This typically appears at very high heat fluxes with lower vapour qualities.
- (b) Deposition and evaporation equilibrium: The CHF condition arises when the evaporation rate of liquid droplets at the heater wall exceeds the deposition rate that occurs at the end of annular flows into the mist flow regime.

(c) Disruption of the liquid film: The liquid film is broken by an unstable surface wave, causing the lower part of the wave to contact the heater wall and dry out.

(d) Dry spot formation: A dry patch forms in the middle of the liquid films, leading to a partial increase in wall temperature. The effect of this dry patch then causes more to form and eventually results in CHF.

(e) Full film dryout: CHF occurs when liquid droplets escape from the film and are carried away by the vapour stream, or when the film completely evaporates.

A CHF mechanism unique to microchannel systems is the phenomenon of early-onset dryout. This arises from the combined effects of vigorous vapour generation and flow reversal, both of which hinder the consistent supply of liquid to the heated surface. As vapour production increases, part of the vapour stream may reverse into the inlet plenum, where it mixes with the incoming subcooled liquid. This interaction elevates the liquid temperature towards saturation before it re-enters the microchannels, thereby reducing its cooling effectiveness. The diminished thermal margin accelerates the onset of dryout. This mechanism, illustrated in Figure 2.39, has been documented by several researchers [309, 324].

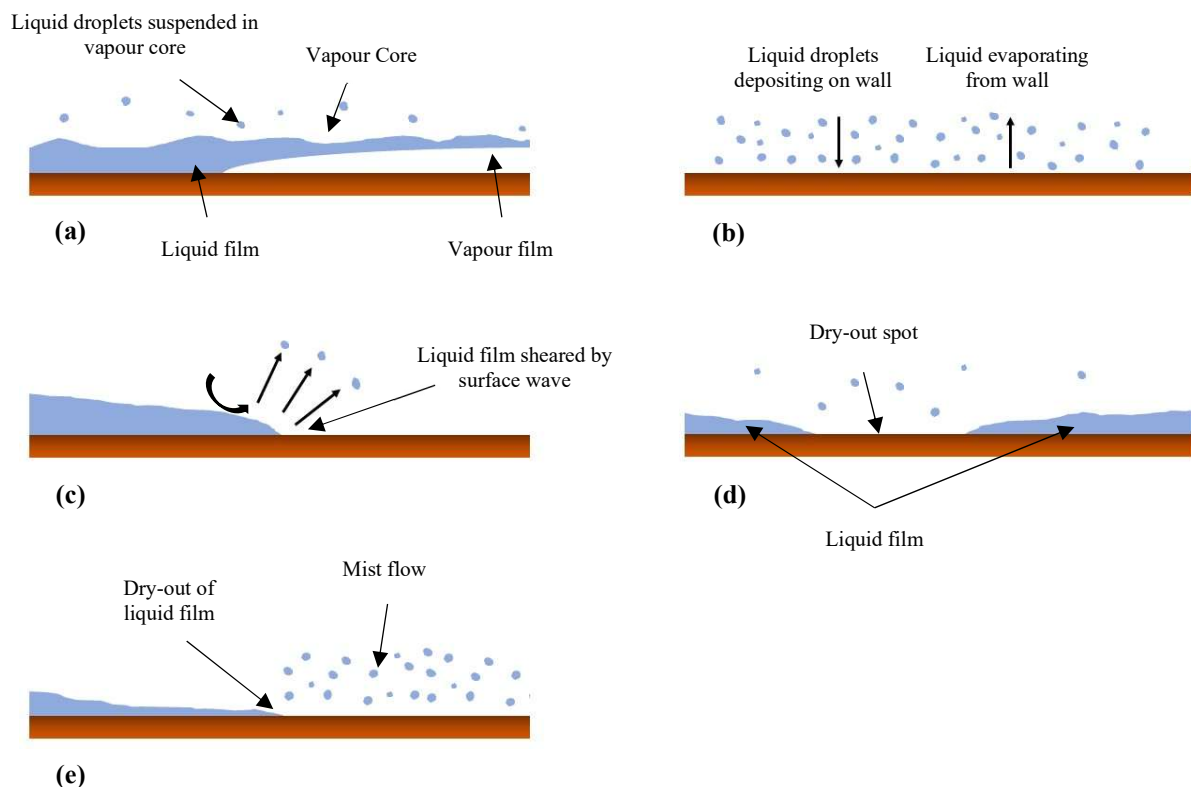


Figure 2.38. Mechanisms that cause the onset of boiling during annular flows: (a) vapour film formation; (b) deposition and evaporation equilibrium; (c) liquid film disruption; (d) dry spot formation; (e) full film dryout (modified from [323]).

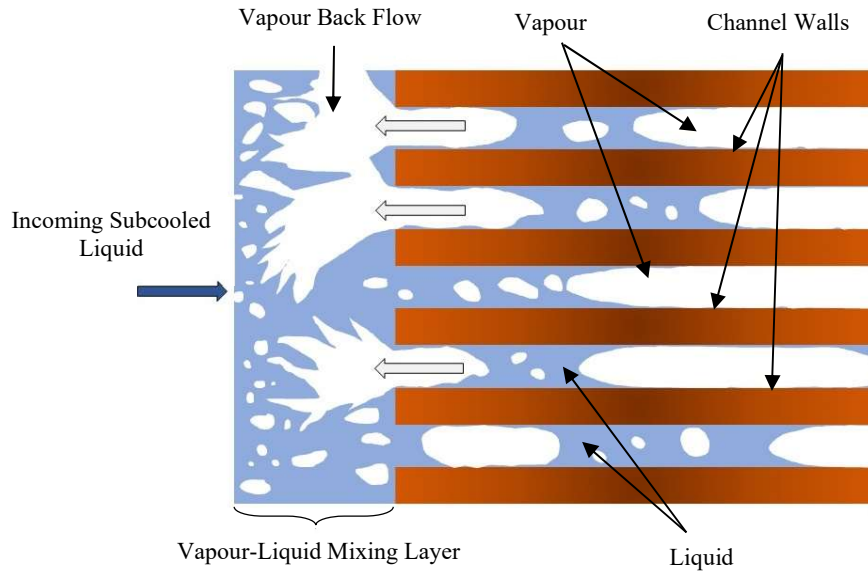


Figure 2.39. Vapour backflow into subcooled liquid (modified from [309]).

2.6.2 Microscale Critical Heat Flux Prediction Tools

Prediction tools for CHF have been developed for both macroscale and microscale flows, which are presented in Appendix C. These correlations reflect the influence of operating conditions on heat transfer mechanisms and, consequently, the control parameters proposed by different researchers. In conventional channels, CHF is strongly influenced by mass flux, inlet subcooling, inlet pressure, channel dimensions (diameter and length), and working fluid properties [31]. In micro-scale channels, additional factors such as flow reversal (linked to system instabilities), surface microstructures, and flow maldistribution in multi-channel systems also play a role. The following review discusses the effects of mass flux, inlet subcooling, inlet pressure, fluid properties, and channel geometry on CHF.

Bowers and Mudawar [325] developed a CHF correlation for flow boiling of R-113 in circular multi-channel heat sinks with inner diameters of 0.51 mm and 2.54 mm and a length of 10 mm. Their experiments covered inlet subcooling levels of 10–32 K and mass fluxes of 31–480 kg/m²s. Under low flow conditions, they observed superheated outlet states and reported that inlet subcooling had no measurable effect on CHF. By contrast, mass flux was found to exert a strong influence. Their correlation, based on an empirical fit of the data, employs the liquid only Weber number, We_{lo} , and the channel length-to-diameter ratio, suggesting that CHF increases with both mass flux and the latent heat of vaporisation.

Qu and Mudawar [309] later investigated CHF for DI water in a microchannel heat sink comprising 21 parallel channels of dimensions 0.215 mm x 0.821 mm and length 44.8 mm. Mass fluxes of 86–368 kg/m²s were tested with inlet temperatures of 30 °C and 60 °C, corresponding to subcooling levels of 70 K and 40 K, respectively. As CHF was approached, they observed flow instabilities that induced

vapour flow reversal into the inlet plenum. This mixing raised the inlet fluid temperature, effectively eliminating the imposed subcooling. Consequently, no difference in CHF was observed between the two inlet temperatures. CHF was nevertheless found to increase with mass flux. From these findings, a universal microchannel correlation was proposed by combining their data with that of Bowers and Mudawar [325]. The resulting model retains the same functional dependencies but incorporates the vapour-to-liquid density ratio, implying that CHF rises as the density ratio decreases.

Kuan [326] examined CHF and heat transfer in flow boiling using R-123 and water within a heat sink consisting of six parallel microchannels of dimensions $1.054 \text{ mm} \times 0.157 \text{ mm}$ and length 63.5 mm . Dryout was observed within the annular flow regime, with CHF initiation linked to the appearance of localised dry patches. Based on experimental results, a new empirical correlation was proposed, which identifies mass flux, latent heat of vaporisation, and the length-to-diameter ratio as the only governing parameters, with other thermophysical properties deemed negligible.

Kuan and Kandlikar [327] proposed a semi-empirical CHF correlation based on flow boiling experiments with water and R-123 in six rectangular multi-channels (1.054 mm width, 0.157 mm height). Their model was developed using a force balance framework, accounting for the ratio of evaporation momentum to surface tension forces, with inertia effects also included (Figure. 2.40). As illustrated, this balance is influenced by the three-phase contact angle. An empirical constant, C , was determined during calibration, with reported values of 0.002679 for both fluids combined, 0.002492 for water, and 0.003139 for R-123. The study showed that CHF increases with mass flux (represented by the Weber number), surface tension, three-phase contact angle, and latent heat of vaporisation. Receding contact angles were also measured on copper surfaces using high-speed photography, yielding values of 45° for water and 5° for R-123.

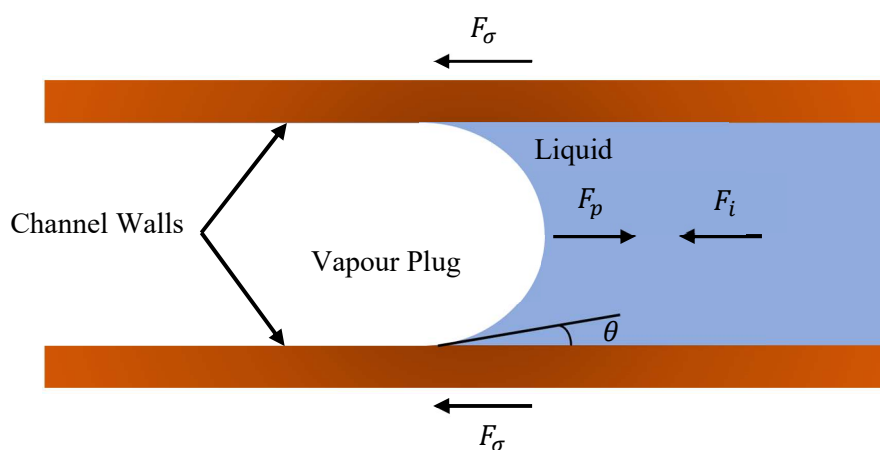


Figure. 2.40. Liquid-vapour interfacial forces (modified from [327]).

Koşar and Peles [31] developed a CHF correlation from experiments using R-123 in a microchannel heat sink comprising of five parallel channels, each 10 mm long, 0.2 mm wide, and 0.264 mm high. Their results indicated that CHF was governed by dryout. They observed that CHF initially increases with system pressure, reaching a maximum before declining at higher pressures. This behaviour was attributed to competing mechanisms: on one hand, reduced liquid to vapour density ratio and enhanced inlet subcooling tend to raise CHF; on the other, increased pressure diminishes both surface tension and latent heat, thereby reducing CHF. The net effect produces a non-monotonic pressure dependence, with CHF peaking at an intermediate pressure. In addition, CHF was found to increase with higher mass fluxes and with lower exit vapour qualities. To account for these effects, they proposed a correlation incorporating exit quality, mass flux, latent heat, and the ratio of exit pressure to the fluid's critical pressure.

Ong and Thome [208] developed an empirical CHF correlation using experimental data for R-134a, R-236fa, and R-245fa in single horizontal channels with inner diameters of 1.03 mm, 2.20 mm, and 3.04 mm and heated lengths of 180 to 540 mm. Their dataset was supplemented with results from Wojtan et al. [328] for R-134a and R-245fa in microchannel tubes (0.5 mm and 0.8 mm diameter, 20 to 70 mm length), as well as additional measurements by Park [329] for R-134a, R-236fa, and R-245fa in channels of 0.35 mm and 0.88 mm diameter.

In addition to developing a new correlation, several parametric effects were identified. Channel size was found to have little influence on CHF at mass fluxes below 500 kg/m²s; however, above this threshold, CHF began to diverge, with the 1.03 mm hydraulic diameter channel producing the highest values for both R-134a and R-236fa. This trend was attributed to confinement effects, where surface tension and viscosity dominate over gravitational and inertial forces in micro-scale geometries. CHF was also observed to decrease with increasing system pressure at constant mass flux, owing to the reduction in latent heat at higher pressures. Additionally, comparisons of CHF among R-134a, R-236fa, and R-245fa at a saturation temperature of 31°C showed that R-245fa exhibited the highest CHF, followed by R-134a, while R-236fa had the lowest. These trends closely correlate with the latent heat values of the fluids, with R-245fa having the highest latent heat and R-236fa the lowest. They also reported that the inlet subcooling was insignificant on the CHF results, while a clear mass flux effect was found.

Anwar et al. [330] investigated dryout in vertical upward tubes using a range of refrigerants, namely R-134a, R-1234yf, R-152a, R-22, R-245fa, R-290, and R-600a. The tubes had internal diameters of 0.64 mm to 1.70 mm and lengths of 213 mm to 245 mm. Their results showed that the heat flux required to initiate dryout increased with mass flux, since greater heat input was needed to completely evaporate the liquid and reach critical vapour quality. Interestingly, dryout heat flux was found to be largely independent of saturation temperature (system pressure). This was explained by two counteracting effects: increasing saturation temperature reduces the liquid to vapour density ratio, which stabilises the

liquid film and raises dryout heat flux, but it also decreases the latent heat of vaporisation, which would lower the required heat input. The opposing influences balanced out, resulting in negligible net variation across the tested range. In addition, larger tube diameters produced higher dryout heat fluxes, attributed to greater mass flow rates and reduced wall shear stresses that helped maintain the liquid film. Based on these observations, a correlation was proposed that depended only on the hydraulic diameter to heated length ratio, mass flux, and latent heat, with other thermophysical effects (e.g. surface tension or density ratios) assumed negligible.

Tibiriça et al. [186] developed a CHF correlation through empirical fitting of a broad databank covering water, R-113, R-12, R-134a, R-123, nitrogen, R-236fa, R-245fa, and R-1234ze, in single channels with diameters of 0.24 mm to 6.92 mm and mass fluxes between 23.6 kg/m²s and 8800 kg/m²s. Their analysis did not attempt to resolve underlying mechanisms but produced a correlation predicting that CHF increases with mass flux and latent heat, while decreasing with Laplace number, Weber number, and liquid to vapour density ratio.

In a subsequent study, Tibiriça et al. [331] examined CHF in saturated flow boiling of R-134a and R-245fa within a horizontal stainless-steel tube of 2.2 mm diameter and heated lengths between 154 mm and 361 mm. The operating range included mass fluxes of 100 kg/m²s to 1500 kg/m²s and exit saturation temperatures of 25 °C, 31 °C, and 35 °C. They found that CHF increased with inlet subcooling and mass flux, as greater heat input was required to reach a given vapour quality under such conditions. Conversely, CHF decreased with increasing channel length and saturation temperature. The length effect was attributed to reduced heat transfer area in shorter channels, requiring higher flux to reach the same vapour quality, while the pressure effect was linked to reduced latent heat at higher saturation temperatures. Finally, CHF was consistently higher for R-245fa compared with R-134a, owing to the larger latent heat of the former.

The reviewed studies highlight that CHF in microchannels is dictated by a complex interaction of thermophysical properties, flow conditions, and channel geometry. Among these factors, mass flux consistently emerges as the dominant parameter, with higher values generally producing increased CHF. The role of inlet subcooling, however, appears less consistent: several studies reported negligible influence [208, 309, 325], whereas others observed a clear positive effect, with CHF rising as subcooling increased [331].

The influence of pressure is particularly complex and varies across studies: (i) little to no effect when the opposing influences of liquid-to-vapour density ratio and latent heat cancel out [330]; (ii) a negative effect due to the reduction of latent heat at elevated pressures [208, 331]; and (iii) a non-monotonic response, with CHF initially increasing and then decreasing as competing effects of density ratio, surface tension, and latent heat take hold [31]. Geometric factors also play a significant role. Larger channel diameters were associated with higher CHF in some cases [330], whereas other studies reported

negligible or even inverse effects depending on the mass flux range [208]. Shorter heated lengths were generally found to promote higher CHF values [331]. Additional control parameters have also been introduced in specific models, including the receding contact angle (θ) and average density ($\bar{\rho}$), as incorporated by Kuan and Kandlikar [327]. Overall, these findings demonstrate that no single parameter universally governs CHF behaviour. Instead, CHF arises from the combined effects of multiple, often competing mechanisms, underscoring the need for robust multi-parameter correlations that can reliably account for these interactions under varying operating conditions.

2.7 FLOW BOILING AND ARTIFICIAL INTELLIGENCE

As discussed in Chapter 1, the increase in widely available computing power has permitted the field of artificial intelligence to be adopted in a wide range of applications. A short discussion on the difference between artificial intelligence, machine learning and deep learning was also presented, as well as a definition of ANNs. However, it is important to further understand the types of machine learning for the purpose of this section. These can be separated into four main categories, namely:

- (i) Supervised learning: Algorithms are trained using labelled datasets, i.e. datasets in which both inputs and known outputs are present [332]. The algorithm is then trained to map input data to the correct output data, such that when it is fed new unseen input data, it is capable of providing correct outputs.
- (ii) Unsupervised learning: Algorithms are provided with unlabelled datasets, i.e. datasets without output values, and find hidden patterns and structures in the data [333]. An unsupervised model will then group unseen data into categories that were detected during training with the unlabelled datasets.
- (iii) Semi-supervised learning: Algorithms are trained with a mixture of both labelled and unlabelled data, usually with a much larger percentage of unlabelled data [334]. This is often used when labelled data is not widely available and the model will benefit from the identifiable patterns in the unlabelled data.
- (iv) Reinforcement learning: An algorithm is trained through a numerical reward signal [335]. The algorithm then adapts based on this signal, such that it is continually learning with the environment.

This subsection of the literature review focusses on studies in which artificial intelligence has been used in the topic of flow boiling, but in particular, the topic of microscale flow boiling.

2.7.1 Flow Patterns

Much of the most recent AI work on flow boiling patterns focusses on flow pattern recognition and classification. Zaborowska et al. [336] identified and categorised flow patterns for water in a 1 mm internal diameter circular microchannel using measured pressure drop oscillations. The recurrence of pressure fluctuations was then fed into a Kohonen network [337], also known as a Self-Organising Map (SOM), which is a type of unsupervised neural network used primarily for two-dimensional visualisation of higher-dimensional data. The SOM then presented average hit histograms with local maximums. The number of local maximums presented then determined the number of flow patterns identifiable within the flow. This was then cross referenced with flow visualisation data. The algorithm was able to successfully identify liquid flow, liquid flow with small vapour bubbles, confined bubble flow, wavy-annular flow and vapour slug flow.

Chu et al. [338] investigated boiling flow pattern recognition in a narrow rectangular channel with a cross-sectional size of 30 mm × 1.2 mm and a length of 650 mm, using DI water as the working fluid. They developed a method to identify flow boiling patterns based on transient pressure signals, rather than relying on video or direct visual observation. First a Time-Frequency Analysis (TFA) was conducted on the pressure signal using the Adaptive Optimal Kernel (AOK) Algorithm. The AOK algorithm converted the pressure-time signal into a two-dimensional time-frequency spectrogram, which shows the strength of the signal at a specific time and frequency. A convolutional neural network (CNN) was then trained with the time-frequency spectrograms as inputs and flow pattern labels, taken from flow visualisation, as the outputs. A CNN is a specialised form of deep learning architecture designed to interpret data with a spatial or visual structure, including images, signal maps, and spectrograms [339]. At its core, the CNN applies signal filters over the image and extracts spatial features. The network is then trained to highlight important patterns of the image in filtered layers. The CNN was used to identify dispersed bubble, bubble-slug, plug-churn and annular like flows.

CNNs have been widely used in other studies for two-phase flow pattern image identification. Xu and Tang [340] used a CNN to identify air-water flow patterns in a 12.7 mm inner diameter tube with a length of 0.89 m and were able to identify bubbly, slug, churn and annular flows. OuYang [341] et al. used a variation of a CNN to classify flow patterns for air-water flows in a 20 mm internal diameter Plexiglass tube. Schepperle et al. [342] used a CNN to classify flow patterns for the flow boiling of DI water in a 0.5 mm x 1.5 mm microchannel. Paz et al. [343] implemented a CNN model to classify flow patterns for water, ethylene glycol-water mixture, and an HFE in a 10 mm x 25 mm rectangular channel at inclinations of 0°, 45°, 90° and 135°. Moiz et al. [344] used an ANN to identify flow patterns, whilst Dias et al. [345] implemented a K-nearest neighbours (KNN) algorithm to identify flow patterns.

The KNN algorithm is a simple yet effective machine learning method used for both classification and regression tasks [346]. It simply stores the entire training dataset without processing it in advance,

i.e. it does not require training. When a new input needs to be evaluated, the algorithm calculates the distance between this point and all stored data points, identifying the k closest examples, where k is a user-defined number. For classification problems, the most frequent category among these k neighbours is assigned to the new input. An example is shown in Figure 2. 41, where the scenario includes 3 classifications and two input parameters, x_1 and x_2 .

Whilst the work on flow pattern identification may aid in eliminating the issues of subjectivity when categorising flow patterns, it is unable to predict flow patterns from a given set of operating conditions which is necessary to designing flow boiling cooling systems. However, there has been some work that utilises the power of machine learning for this purpose. Scalabrin et al. [347] noted that the heat transfer dynamics that occur during flow boiling follow the development of the interaction between liquid and vapour phases as the vapour fraction increases in 2005. As such, they developed a Feed Forward Neural Network (FNN) that aimed to predict two-phase heat transfer rates based on the flow patterns in tubes between 7.7 mm and 14 mm in diameter, i.e. the conventional scale. A function was therefore built into the FNN that modelled the prevalent flow pattern for the given operating and fluid parameters and compared to an existing flow pattern transition prediction model proposed by Thome and Hajal [348] for horizontal tubes. The FNN model performed comparably to the existing correlation; although only 46% of the experimental data points for slug flow were correctly estimated and a fairly small data bank was used.

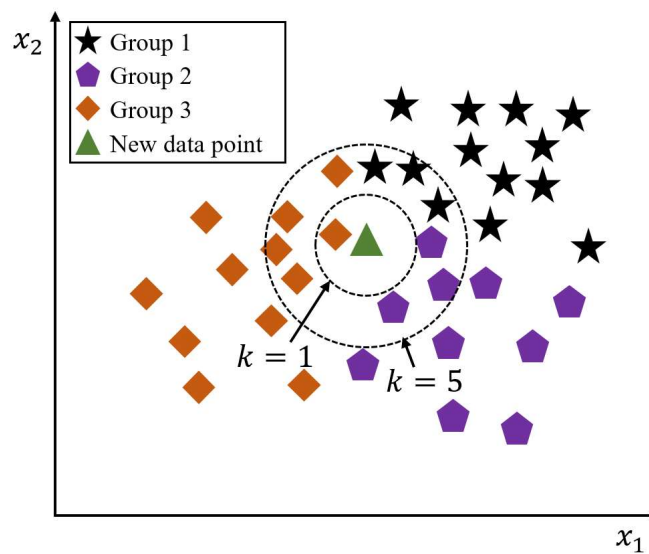


Figure 2.41. Conceptualisation of the KNN algorithm.

FNNs are one of the most commonly implemented type of artificial neural networks to obtain forecast predictions in multiple applications with great accuracy [349]. The plethora of benefits that FNNs pose result in their commonplace in an abundance of applications. The first of these benefits is that it does not require explicit programming and is able to “learn” from the provided examples [350].

As a consequence, FNNs are also highly generalisable, further permitting their use in many applications. FNNs are capable of recognising and adjusting to complex and non-linear patterns within a data set and are therefore able to make predictions for patterns in data sets that are alike but not indistinguishable from those that it has been trained with.

Typical FNN architecture is presented in Figure 2.42 and consists of a series of interconnected layers that include a single input layer, a number of hidden layers and a single output layer. The function of the input layer is to receive the raw data from a data set which is modified as it propagates through the hidden layers over the weighted internodal connections. The nodes in the hidden layers apply an activation function that introduces non-linearity to the system and increases the model's ability to capture complex patterns. The final layer then creates a classification or prediction depending on the input features and the transformations that the model has conducted. The neurons inside the network perform weighted summations and apply the activation function to allow non-linear mapping to the problem. The model is trained by adjusting the internodal weights through backpropagation which optimises the network to make accurate predictions. This process is represented in the flow diagram in Figure. 2.43.

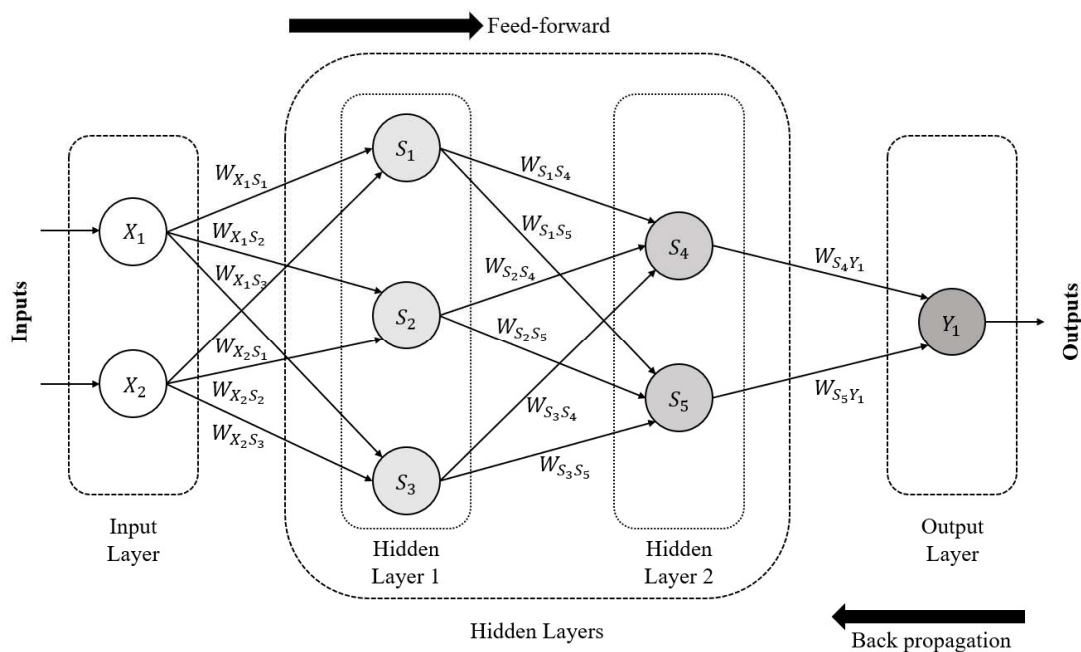


Figure 2.42. Typical FFN architecture.

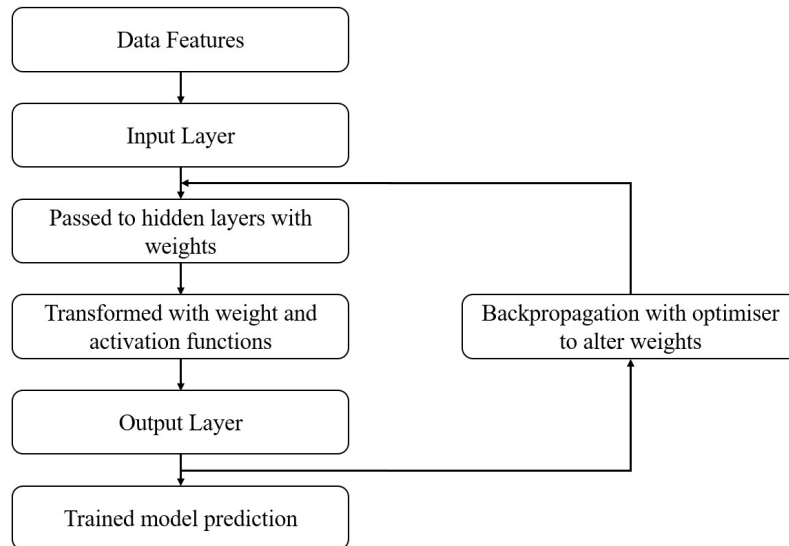


Figure 2.43. FNN training and prediction process.

Guo et al. [351] used a three-layer back propagation neural network to predict the resultant flow patterns in the water wall tubes of a steam boiler. The input vectors of pressure, temperature and vapour quality were used to obtain the flow pattern categorisations from the neural network. Performance accuracies of 95.24% were reported. The limitation of this model is that only 100 data points were used for each flow pattern and the data was compared to a computational fluid dynamics (CFD) software model of the steam boiler tubes, as opposed to experimental data. Furthermore, the primary purpose was to predict flow patterns for water and therefore did not require that an extensive range of thermophysical fluid properties were investigated to form a universal prediction tool.

Timung and Mandal [352] developed a Probabilistic Neural Network (PNN) to determine gas-liquid flow patterns in micro-tubes using nitrogen-water and air-water flow pattern data. A PNN is a type of supervised artificial neural network that classifies input data based on statistical principles. It consists of four layers: an input layer, a pattern (or hidden) layer, a summation (or class) layer, and an output layer, as shown in Fig. 2.44. The PNN algorithm used superficial velocities, channel inclination and fluid thermophysical properties such as density, viscosity and surface tension. The model was then compared to existing flow pattern prediction tools by Brauner [353] and Akbar et al. [354] and was found to perform well for both air-water and nitrogen-air flows. This work was unable to consider the potential effects of heat flux and the resultant evaporation momentum force due the utilisation of adiabatic flow pattern data. Additionally, it should be noted that the performance of the algorithm derived from a split of the overall data set. Thus, the algorithm was not tested against a new data set previously unseen by the algorithm during training but rather new data from the existing data set that was seen during training of the network.

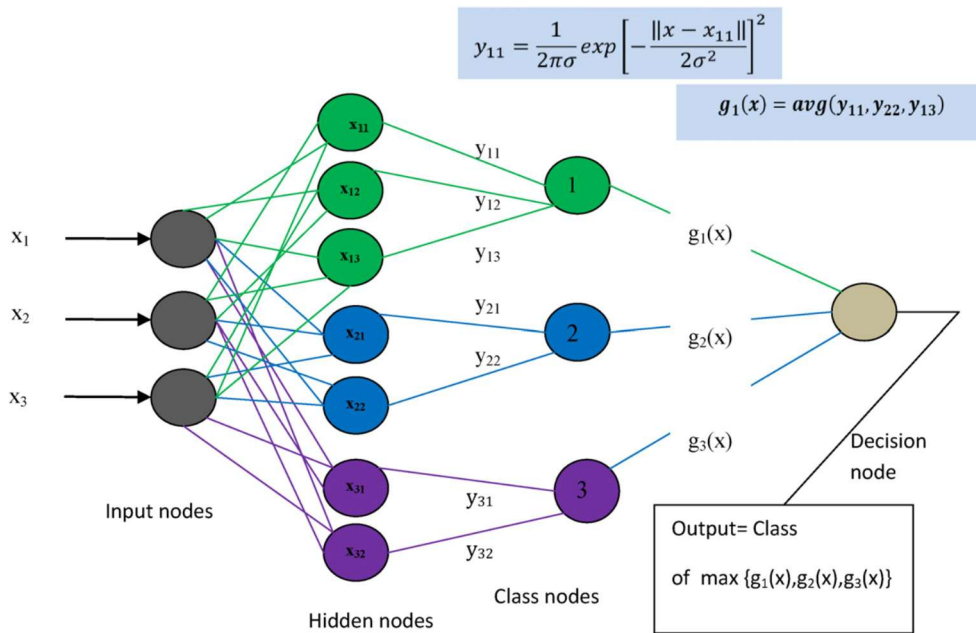


Figure 2.44. Probabilistic Neural Network Architecture (taken from [352]).

In summary, recent advances in the use of AI in flow boiling have leveraged a range of machine learning techniques, particularly convolutional and feed-forward neural networks, to automate classification and reduce subjectivity. These approaches have shown promising accuracy in recognising flow regimes across various experimental setups and working fluids. However, two key limitations persist across the current body of work. First, while classification models are increasingly sophisticated, there remains a notable absence of predictive models capable of forecasting microscale flow patterns directly from operating conditions, a critical requirement for the design and optimisation of compact two-phase cooling systems. Second, many models are validated using subsets of the same data used for training, limiting the ability to assess whether the model has genuinely learned the underlying physics or is merely interpolating within the training data. These gaps must be addressed to fully unlock the potential of machine learning in predictive flow boiling analysis and ensure physically robust generalisation to new operating conditions and geometries.

2.7.2 Other Applications in Flow Boiling

2.7.2.1 Heat Transfer Coefficient

Much of the existing work using artificial intelligence in the field of microscale flow boiling has been conducted on predicting heat transfer coefficients. Parveen [355] et al. implemented an ANN and a support vector regression-based model to predict flow boiling heat transfer coefficients for R-600a in 1.1 mm internal diameter channel. Zhang et al. [356] compared the performance of two machine learning algorithms for predicting flow boiling heat transfer of 8 refrigerants in channels ranging from

0.84 mm to 6 mm. The two algorithms were a back-propagation neural network, such as an FNN, and a genetic neural network (GNN). A GNN combines the learning capability of artificial neural networks (ANNs) with genetic algorithms, that in effect train the network. Zhang et al. found that the GNN outperformed the back-propagation neural network and existing correlations with an RMS% of 17.16%.

Qiu et al. [357] developed an ANN model trained on a dataset of over 16,000 experimental data points from 50 studies covering 16 working fluids and a wide range of mini/micro-channel conditions. Their final model, using only dimensionless input parameters and a multi-layer network architecture, achieved a mean absolute error (MAE) of 14.3 %, outperforming universal correlations in both accuracy and generalisability across different fluids and channel geometries. Agustiarini et al. [358] also used an ANN to predict flow boiling heat transfer of R-32 in a 0.969 mm multiport tube, finding an MAE of only 6.35 %. The same group then applied a similar methodology to R-1234yf in a later study [359], again finding that the algorithm performed significantly better than existing correlations with an MAE of 7.12 %.

Chen et al. [360] assembled a comprehensive database of 2,500 experimental data points for saturated flow boiling using seven refrigerants and channels ranging from 1 to 7 mm in diameter. A new empirical correlation was proposed and shown to outperform previous models, achieving an MAE of 11.28% overall, with stronger performance in convective boiling regimes (MAE = 10.18%) than nucleate boiling regimes (MAE = 14.6%). In addition, they developed an ANN model trained on 70% of the dataset and tested on 30%, which predicted heat transfer with an MAE of 4.5% and $R^2 = 0.984$. The ANN also predicted two-phase pressure drop with high accuracy ($R^2 = 0.994$, MAE = 7.39%), significantly improving upon existing correlations.

Galicía et al. [361] compiled a comprehensive dataset of over 20,000 experimental observations covering a wide range of refrigerants, heat exchanger geometries, and operating conditions. They trained 27 different machine learning models, including regression trees, support vector machines, and neural networks, to predict both heat transfer coefficients and pressure drops for flow boiling and condensation.

Regression trees are a supervised machine learning method used to predict numerical values [362]. They work by splitting data into smaller and smaller groups, based on input variables, using simple "yes/no" rules at each step. At each split, the algorithm chooses the variable and cut-off that best reduce the error in the data. Once the data is split enough, each group (called a leaf) gives a prediction, usually the average of the experimental values in that group. The bi-layered neural network was found to achieve the best performance, with an R^2 of 0.94 and an MAE% of 1.04 % for heat transfer coefficient prediction, and an R^2 of 0.97 % and MAE of 0.65 for pressure drop prediction.

2.7.2.2 *Pressure Drop*

While much of the current machine learning research focuses on predicting heat transfer coefficients these methods also hold promise for modelling pressure drop. Montañez-Barrera et al. [363] introduced a novel approach called Correlated-Informed Neural Networks, which integrates physical correlations into the architecture of an artificial neural network. The model was trained on experimental data from zeotropic mixtures in microchannels and the pressure drop model by Sun and Mishima [364]. The model achieved a mean relative error of 6%, significantly outperforming the traditional correlation methods alone. This hybrid approach demonstrates enhanced accuracy and generalisability, especially in cases involving limited experimental data or new refrigerant mixtures.

Building on this trend, Zheng et al. [365] developed a multi-layer ANN optimised using a genetic algorithm to predict subcooled boiling pressure drops in microchannels under high heat flux conditions. Trained on more than 1,000 experimental data points, the GA-ANNs model identified an optimal set of dimensionless parameters and achieved a mean absolute error of 4.89% with an R^2 of 0.9971. The same parameter set was subsequently used to derive a new empirical correlation, offering enhanced prediction accuracy over existing models across a wide range of operating conditions and channel dimensions. Whilst ANNs are often considered “black-box” models, they can provide powerful insights by enabling feature selection, i.e. highlighting the most influential input parameters for pressure drop, heat transfer and flow pattern prediction.

In a novel application of deep learning, Xiao and Zhang [366] developed a CNN model for predicting two-phase pressure drops during flow boiling in mini-channels subjected to ultrasound fields. Their method incorporates time-frequency representations of pressure signals, generated via wavelet transform, to capture non-stationary fluctuations induced by ultrasonic excitation. CNN model achieved a mean absolute error of 5.6% and score of 94.84%, significantly outperforming traditional correlations under ultrasonic field conditions.

2.8 CONCLUSIONS

This review has examined the current body of work on flow boiling in the microscale, with emphasis on definitions of the microscale, prevailing flow patterns, heat transfer behaviour, pressure drop characteristics, critical heat flux, and more recent developments incorporating artificial intelligence techniques. The literature reveals that while there is broad recognition of the unique physical phenomena occurring at reduced hydraulic diameters, there is still no universal agreement on where the microscale boundary lies. Competing criteria, whether geometric, force-based, or phenomenological, result in significant variation, and this lack of consensus hinders the establishment of widely applicable predictive methods.

Research into flow patterns demonstrates that the current terminology is largely inadequate in describing microscale boiling, with discrepancies not only in the reported regimes but also in the terminology used. This lack of clarity has a direct bearing on attempts to correlate heat transfer and pressure drop, since both are strongly linked to the prevailing interfacial geometry. Although some progress has been made towards mechanistic models that account for confined bubble dynamics and film behaviour, most correlations remain semi-empirical and limited in scope. Likewise, studies on pressure drop and critical heat flux point to important influences from channel geometry, aspect ratio and surface condition, yet general predictive tools remain elusive.

More recently, machine learning has emerged as a potentially transformative tool for addressing these gaps. Early applications have shown encouraging results in predicting heat transfer coefficients and critical heat flux across a wider parameter space than traditional methods. However, the reliability of such models is contingent upon the availability of large, diverse, and high-quality data sets, a resource that is currently lacking in microscale boiling research.

The field has developed a substantial foundation of experimental and theoretical work, but significant challenges persist in establishing consensus definitions, unifying flow pattern classification, and producing predictive models with wide applicability. Progress will depend on coordinated efforts to standardise terminology and methodologies, to extend experimental investigations to underexplored fluids and geometries, and to integrate emerging data-driven techniques with physical modelling. Only by addressing these issues can the promise of microscale flow boiling be fully realised in practical engineering applications.

In addition to the above conclusions, a further detailed summary of each aspect of the literature review is addressed below.

2.8.1 Defining the Microscale

1. There is no single accepted definition of the microscale, with proposed thresholds varying according to geometry, flow characteristics, or physical force balances.
2. Some definitions emphasise very small hydraulic diameters where surface or electrokinetic effects become significant, while others focus on conditions where conventional continuum assumptions begin to break down.
3. Alternative approaches identify the microscale by changes in flow behaviour, such as the disappearance of stratified flow or the onset of confined bubble dynamics.
4. Reported thresholds span a wide range, from tens of micrometres to around a millimetre, reflecting strong dependence on fluid properties, operating conditions, and experimental methods.

5. Overall, the lack of consensus creates difficulties in developing broadly applicable predictive tools for microscale boiling.

2.8.2 Flow Boiling Patterns

1. Flow boiling patterns are central to understanding microscale heat transfer and pressure drop, yet fewer studies exist compared with other topics such as correlations or heat flux limits.
2. Conventional flow pattern maps developed for larger channels do not accurately predict transitions at the microscale, where interfacial behaviour is significantly altered.
3. Reported flow regimes in small channels are often inconsistent, with disagreements both in which patterns occur and in how they are named or categorised.
4. Stratified flow, common in macroscale horizontal channels, is generally absent at the microscale due to the reduced influence of buoyancy.
5. Bubbly, slug, churn, and annular patterns form the main groups of microscale boiling flows, though their precise boundaries remain debated.
6. A major difficulty lies in the subjectivity of visual classification methods and the lack of universally accepted terminology across different studies.
7. The transitional nature of many regimes further complicates classification, as bubbles may gradually evolve into slugs or annular films without a sharp dividing line.
8. Without clear and consistent flow pattern designations, it remains challenging to link mechanistic heat transfer models to observed behaviour.
9. Developing a universal grouping of microscale flow boiling patterns is therefore a critical step towards creating reliable predictive tools for engineering applications.

2.8.3 Heat Transfer

1. Heat transfer in microscale boiling is governed by mechanisms that differ from those in conventional channels due to confinement effects and altered interfacial behaviour.
2. Two primary mechanisms are recognised: nucleate boiling, where bubble activity dominates heat transfer at lower vapour qualities, and convective boiling, where thin film evaporation becomes more important at higher vapour qualities.
3. In the nucleate regime, bubble departure diameter, nucleation site density, and bubble generation frequency strongly influence the local mixing and resultant heat transfer rates.
4. The convective mechanism is linked to the thinning of the liquid film surrounding the vapour core, making heat transfer highly sensitive to mass flux and vapour quality.
5. Transitions between nucleate and convective dominance are gradual and may overlap depending on fluid properties, channel geometry, and applied heat flux.

6. Discrepancies between reported studies reflect differences in identifying which mechanism is active, particularly in intermittent regimes such as slug or churn flows.
7. Many correlations developed for macroscale channels fail to capture these mechanisms in small passages, leading to under- or over-prediction of heat transfer coefficients.
8. Mechanistic models that explicitly account for bubble confinement, film thickness variation, and interfacial instabilities are more promising but remain limited in validation.
9. A unified understanding of when and how each mechanism dominates is essential for reliable predictive modelling and for the design of practical microscale boiling systems.

2.8.4 Pressure Drop

1. Pressure drop in microscale boiling channels is a critical parameter, as it directly influences system stability and pumping requirements.
2. Reported data shows significant variation, with measurements often higher than predictions from conventional correlations, reflecting the altered interfacial dynamics at small scales.
3. Flow regime plays a major role, with slug and annular flows typically associated with large pressure fluctuations and oscillations.
4. Geometry, aspect ratio, and surface characteristics can strongly affect frictional and acceleration components of pressure drop in microscale channels.
5. Despite many experimental investigations, a broadly reliable correlation remains elusive, as existing models tend to be highly case-specific.

2.8.5 Critical Heat Flux

1. CHF in microscale channels is a key limiting factor for the safe and efficient operation of boiling-based cooling systems.
2. CHF has been found to be highly sensitive to channel geometry, aspect ratio, surface condition, and fluid type, leading to large variations between studies.
3. Conventional CHF correlations developed for larger channels generally fail to predict results in the microscale, underlining the need for new, scale-specific models.
4. Emerging approaches that incorporate mechanistic understanding of confined bubble dynamics and thin-film dryout offer promise but require broader validation.
5. Overall, CHF remains insufficiently characterised across fluids and geometries, limiting its use as a reliable design parameter in microscale applications.

2.8.6 Flow Boiling and Artificial Intelligence

1. Machine learning has recently been introduced as a tool to improve prediction of microscale boiling heat transfer and critical heat flux.
2. Data-driven models have shown the potential to capture complex, nonlinear relationships between operating conditions, geometry, and thermal performance that conventional correlations struggle to represent.
3. Early studies indicate that such approaches can achieve higher accuracy and broader applicability than traditional methods, provided sufficient data are available.
4. A key limitation is the current scarcity of large, high-quality experimental datasets, which restricts the robustness and generalisability of machine learning models.
5. Integrating physical insight with data-driven methods may provide the most effective path forward, combining interpretability with predictive power.

CHAPTER 3

EXPERIMENTAL METHODOLOGY

3.1 INTRODUCTION

This chapter begins with a description of the initial apparatus and provides a systematic explanation of the evolution of the experimental facility. The design methodologies and manufacturing of the various test sections are presented separately in Section 3.3. A validation of the experimental facility is provided and experimental procedures outlined. Finally, a full data reduction and uncertainty propagation analysis is presented.

3.2 INITIAL EXPERIMENTAL FACILITY

An existing experimental apparatus was situated in Laboratory 011a in Tower B Building, Brunel University of London, assembled by Korniliou and Karayiannis [367]. A photographic image of the initial experimental facility is presented in Figure 3.1 whilst a schematic of the experimental facility is depicted in Figure 3.2. The schematic drawing includes representations of both the main flow and separate cooling closed flow loops as well as a simplification of electrical connections.

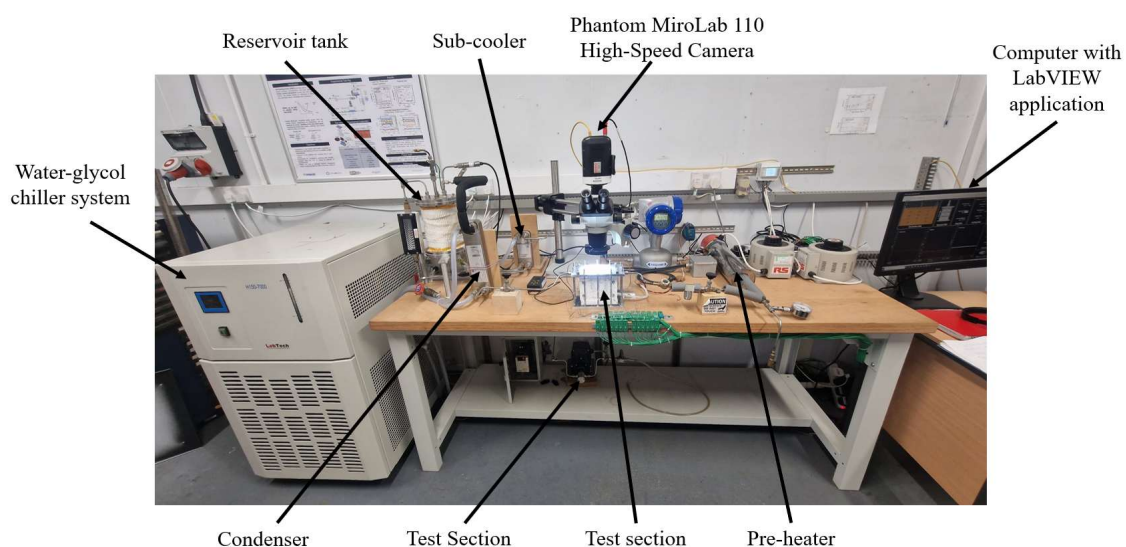


Figure 3.1. Photograph of initial experimental facility in Tower B Building, Brunel University of London.

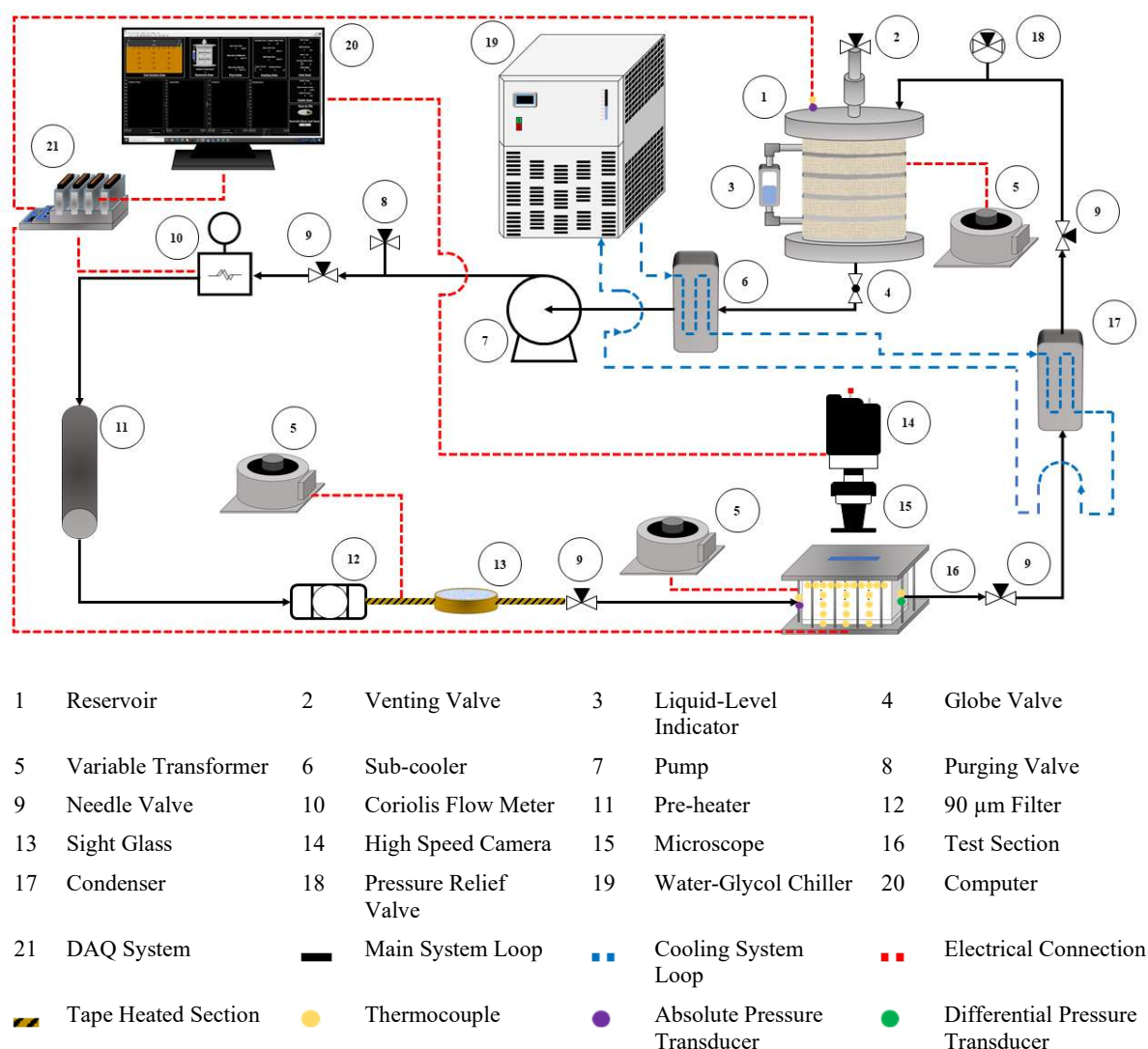


Figure 3.2. Schematic diagram of initial experimental facility.

The main closed flow loop consisted of a reservoir tank, a sub-cooler, a micro gear pump, a Coriolis flow meter, pre-heater, filter, sight glass, test section and condenser. The cooling loop comprised of the sub-cooler and condenser, with cooling water and glycol being circulated through the components via a circulation chiller. Finally, all instrumentation, i.e. absolute/differential pressure transducers, thermocouples and flow meter, were connected to a data acquisition system with the exception of the high-speed camera and microscope. The majority of the instrumentation was located in the test section, where various temperature and inlet, outlet and differential pressure measurements were recorded. However, further pressure and temperature measurements were also taken from the reservoir tank. Instrumentation located in the reservoir tank allowed the fluid in the main closed flow loop to be degassed as well as providing data for the control of system pressure inside the test section. HFE-7100 was used as the working fluid in the main closed flow loop in this study.

3.2.1 Main Closed Flow Loop

3.2.1.1 Reservoir Tank

The reservoir tank is photographed in Figure 3.3. It consisted of a rolled stainless-steel cylindrical shell with two flat blind flanges at either end. A liquid level indicator was inserted with ports on both the top and bottom flange that permitted observation of the liquid, which was essential for observing long-term leaks in the system and to ensure a certain liquid level is maintained during the degassing process. A tape heater was wrapped around the reservoir tank to heat the internal refrigerant so as to control the system pressure. The tape heater was controlled via a RS WCV15E-1 enclosed variable transformer, further described in Section 3.2.1.8. A pressure relief valve set at 6 bar was inserted on the reservoir return channel for safety.

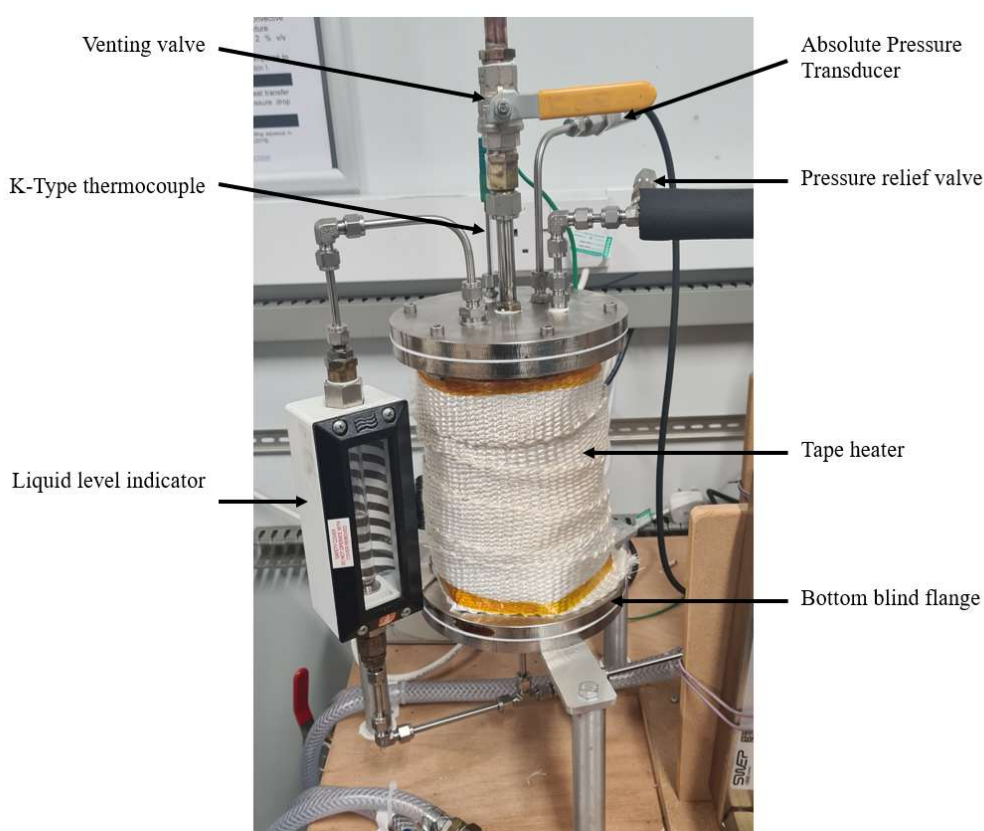


Figure 3.3. Reservoir tank apparatus.

The venting valve was used to attach a vacuum pump to the reservoir, which was used to both insert the fluid and to degas the fluid. The importing of the fluid was conducted by first placing the reservoir under a vacuum, using the vacuum pump. The valve was then closed and a line to the fluid vessel was attached. The valve was then opened and the pressure differential was utilised to push the liquid into the reservoir. An Omega PXM409-007BAI absolute pressure transducer and K-type thermocouple were inserted in the top flange of the reservoir tank to obtain the vapour pressure and temperatures. This

allowed the fluid temperature and pressure to be compared to the saturation curve and ensure the fluid has been correctly degassed.

3.2.1.2 Coriolis Flow Meter

A Krohne Optimass 3000-S01 Coriolis flow meter with a single stainless-steel tube system was used to measure the liquid mass flow rate from the pump with an accuracy of $\pm 0.1\%$. The Coriolis flow meter was manufactured to provide a flow rate of up to 6 kg/hr with a maximum design process temperature and pressure of 150 °C and 30 barg, respectively. A Krohne MFC 300 signal converter for mass flowmeters was used to supply a voltage of 100-230 V ac and provide an output signal of 4-20 mA. Requirements of the Data Acquisition System (DAQ) include that a voltage signal of ± 10 V is used. Hence, a 500 Ω shunt resistor was inserted to obtain a signal value between 0-10 V. The Krohne Optimass 3000-S01 Coriolis flow meter and Krohne MFC 300 signal converter for mass flowmeters are photographed in Figure 3.4. The calibration of the Coriolis flow meter was conducted by the manufacturer (Krohne), prior to the commencement of experimentation.

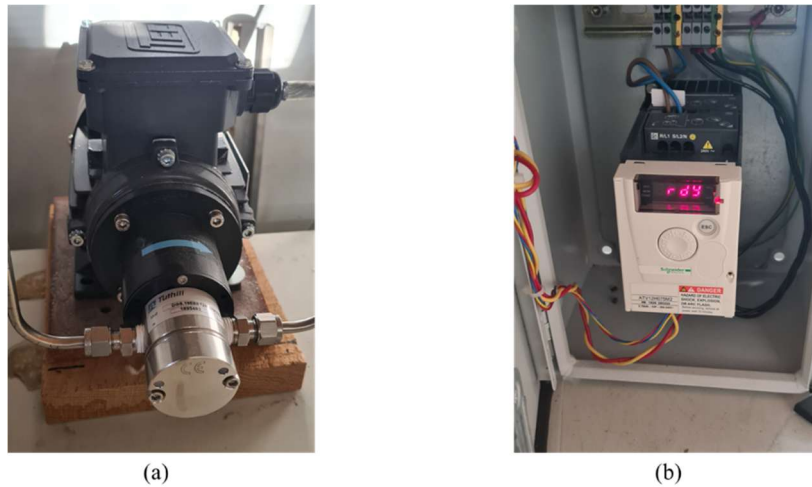


Figure 3.4. Krohne Optimass 3000-S01 Coriolis flow meter and Krohne MFC 300 signal converter for mass flowmeters.

3.2.1.3 Gear Pump

A Tuthill DGS.19EEETNNOM163 micro gear pump was used to pump the working fluid through the main closed flow loop. The pump possessed a maximum rotational speed, volume flow rate and differential pressure of 500 RPM, 38 L/h and 17.2 bar, respectively. The pump, shown in Figure 3.5(a), was certified for maximum operating temperatures and pressures of 176 °C and 35.5 bara. The AC pump motor was controlled using a Schneider Electric ATV12H075M2 0.75kW phase inverter drive,

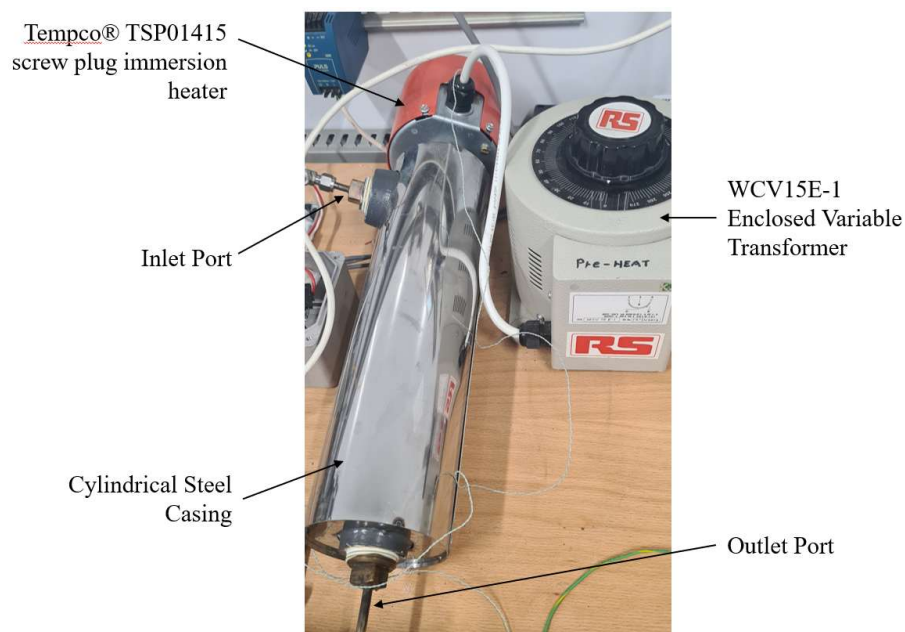
depicted in Figure 3.5(b). The phase inverter was used to set the flow rate in the main flow loop to the desired experimental conditions.



3.5. Micro-gear pump system: (a) Photograph of Tuthill DGS.19EEETNNOM163 micro gear pump; (b) Photograph of Schneider Electric ATV12H075M2 phase inverter.

3.2.1.4 Preheater

A 2.5 kW capacity Tempco® TSP01415 screw plug immersion heater, inserted into a stainless-steel cylindrical casing, was used to pre-heat the working fluid prior to entering the test section. This pre-heater, displayed in Figure 3.6, allowed for the degree of test section inlet fluid sub-cooling to be set. The screw plug immersion heater power was controlled using one of the enclosed variable transformers presented in section 3.2.1.8.



3.6. Preheater system equipment configuration.

3.2.1.5 *Thermocouples*

Twenty-one K-type thermocouples were used to monitor temperature distributions within the test section, with a further two to measure the fluid inlet and outlet temperatures and a final thermocouple in the reservoir tank. All thermocouples in the test section had a length and diameter of 150 mm and 0.5 mm, respectively. K-type thermocouples have the capability of measuring between -200 °C and 1250 °C [127]. The thermocouples were calibrated using the apparatus and methodology outlined in Section 3.5.

3.2.1.6 *Pressure Transducers*

Two Omega PXM409-007BAI absolute pressure transducers were used to measure both the reservoir tank and test section inlet pressures. An excitation voltage of 7 V to 35 V DC was used to provide an output voltage of 0-10 V, corresponding to a measuring range of 0-7 bar. Both absolute pressure transducers were calibrated using the apparatus and methodology outlined in section 3.4.1.2.

A differential pressure transducer was used to measure the pressure gradient between the channel inlet and outlet plenums. Three separate differential transducers were selected for this purpose, each with a different measurement range. The first differential pressure transducer, an Omega PXM409-350HDWU10V, had a measurement range of 0-0.35 bar. The second differential pressure transducer, an Omega PXM409-001BDWU10V, had a measurement range of 0-1 bar and was used to measure differential pressures between 0.35-1 bar. The final differential transducer, an Omega PXM409-3.5BDWUI, had a measurement range of 0-3.5 bar to measure pressure drops in the range of 1-3.5 bar. All differential transducers also used an excitation voltage between 7 V and 35 V DC to provide an output voltage of 0-10 V across each of the measurement ranges. The differential pressure transducers were calibrated by the manufacturer.

3.2.1.7 *Enclosed Variable Transformers (Variac)*

Three separate RS WCV15E-1 enclosed variable transformers were used to control the heat supplied to: (1) the reservoir tank tape heater (2) the pre-heater and (3) test section cartridge heaters. The first variac that controlled the heat supplied to the reservoir tank tape heater was used to both degas the working fluid and to set the system pressure. The second variac that varied the heat supplied to the pre-heater controlled the degree of subcooling at the test section inlet. Finally, the third variac was used to alter the heat flux applied to the microchannel wall. All variacs were identical and possessed a power rating of 4050 W and maximum output current of 15 A. An example Variac is depicted in Figure 3.7, where the variac in the image was used to control the reservoir tank tape heat.



Figure 3.7. Example RS WCV15E-1 enclosed variable transformer.

3.2.2 Auxiliary Cooling Loop

An auxiliary cooling loop was utilised to remove both sensible and latent heat from the main experimental flow loop. A water-glycol mixture was used as the working fluid and was stored in a chiller system. A pump in the chiller system cycled the mixture into a condenser and sub-cooler before entering back into the chiller system.

3.2.2.1 Water Chiller

The LabTech H150-700 water chiller used to provide the water-glycol mixture to the auxiliary cooling loop is shown photographically in Figure 3.8. The chiller contained a 22-litre reservoir and a circulating pump, capable of providing a maximum volume flow rate of 14 l/min at 4.1 bar system pressure. Fluid outlet temperatures between $-35\text{ }^{\circ}\text{C}$ and $8\text{ }^{\circ}\text{C}$ could be achieved using a vapour-compression refrigeration cycle with R-404A. Overall, the system was capable of providing a cooling capacity of 7000 W in an environment with an ambient temperature of $25\text{ }^{\circ}\text{C}$. The water outlet pressure and temperature were controlled through the use of a touch-screen on the front panel of the water chiller.

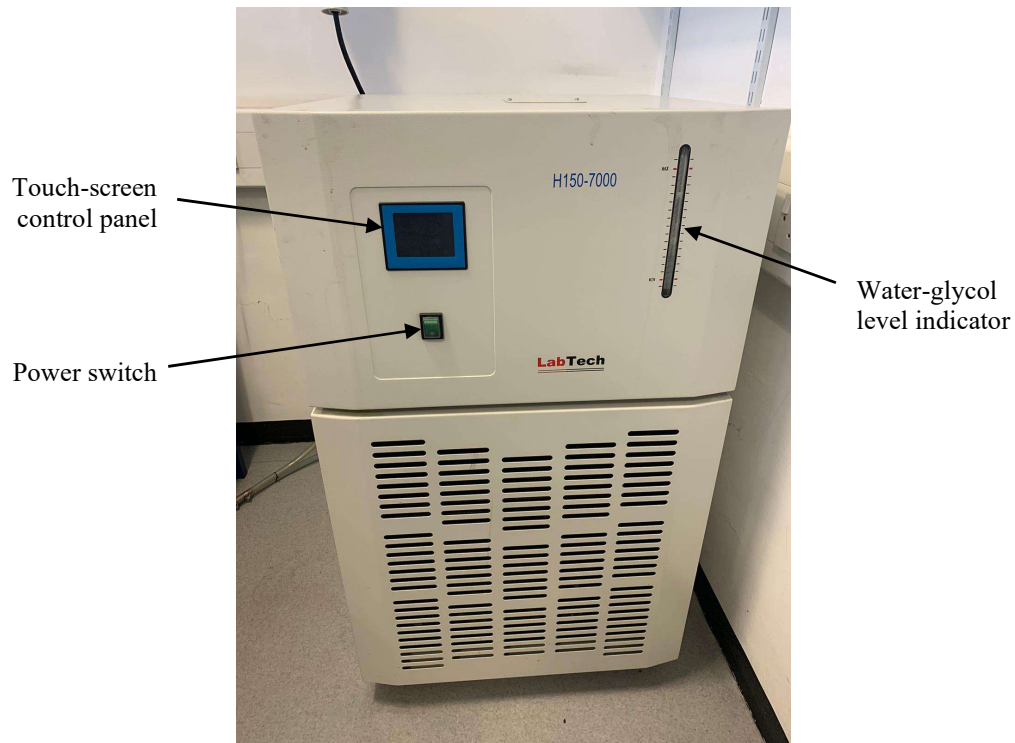


Figure 3.8. LabTech H150-700 water chiller system.

3.2.2.2 *Condenser and Sub-cooler*

Two brazed plate heat exchangers were used as a condenser and a sub-cooler in both flow loops. Both heat exchangers are shown in Figure 3.9. The condenser was placed after the test section and before the reservoir to condense the main loop working fluid before being recycled into the reservoir. The sub-cooler was placed upstream of the micro-gear pump to both ensure that the fluid is fully condensed and that there is sufficient subcooling, such that any cavitation of the gear pump is avoided.

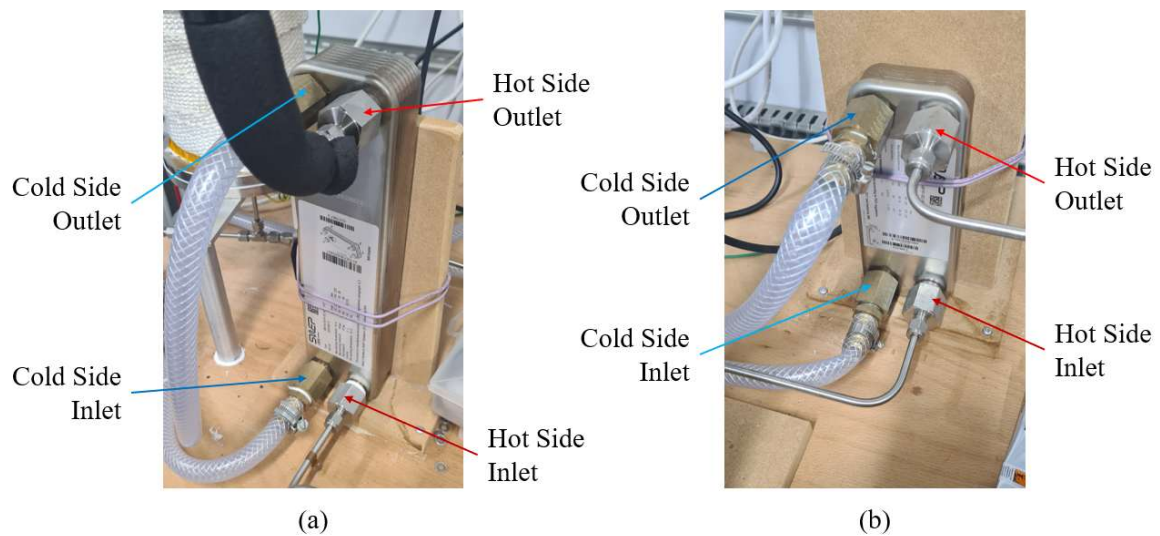


Figure 3.9. Brazed plate heat exchangers: (a) condenser; (b) sub-cooler.

3.2.3 Data Acquisition System (DAQ)

Three separate DAQ modules were inserted into a NI cDAQ-9174 DAQ chassis for data collection. The first two were NI 9213 DAQ modules, which are spring terminal modules designed for thermocouple connection with a voltage measurement range of ± 78.125 mV. NI 9213 DAQ modules are designed for high accuracy thermocouples and possess a resolution of 0.02 K and have a maximum sampling rate of 1 kHz, as per the manufacturer's supplied data. All thermocouples were connected to either of these DAQ modules. The third DAQ module was used to record pressure and flow rate measurements at inputs of ± 10 V. The NI-9224 DAQ module used was capable of the same sampling rate of 1 kHz. Therefore, temperature measurements were sampled at 1 kHz, whilst flow rate and pressure measurements were also sampled at 1 kHz. The assembled DAQ system is displayed in Figure 3.10.

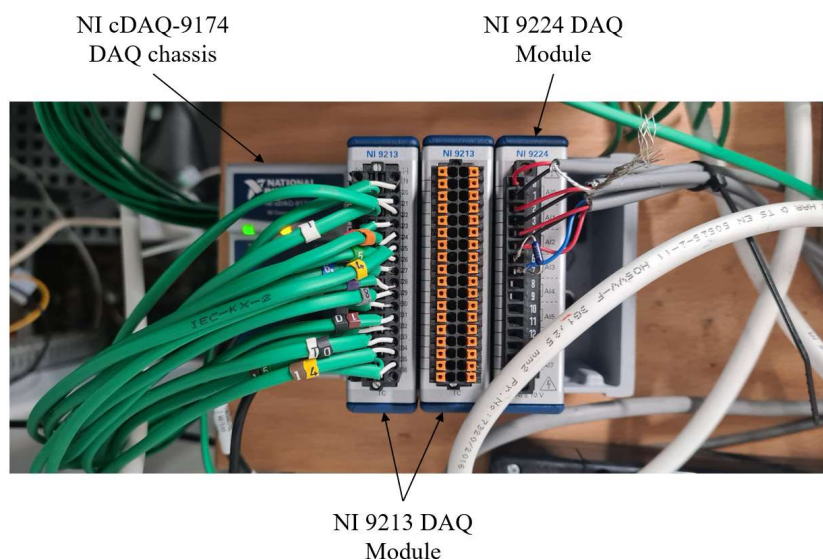


Figure 3.10. NI cDAQ-9174 DAQ chassis, NI 9213 DAQ modules and NI 9224 DAQ Module.

3.2.4 High-Speed Camera and Microscope

A Phantom MiroLab 110 digital high-speed, high resolution camera was used to obtain videos and images of the microchannel. The MiroLab 110 high-speed camera was attached to a moveable stand to allow for positioning of the camera around the test section so that different sections can be analysed, for example, the inlet and outlet plenums as well as different locations along the channel. The digital camera is capable of providing videos with a frame rate of 6000 frames per second and a resolution of 512x512 pixels. The camera was also mounted to a Huvitz HSZ-645TR microscope with a lighting system, providing sufficient illumination and magnification to observe flow patterns in the microchannel. A further fibre-optic microscope lighting system was also used to increase the intensity

of the light to allow for lower exposure times and greater frames recorded per second, whilst maintaining image clarity. The full system with all components is shown in the photograph displayed in Figure 3.11.

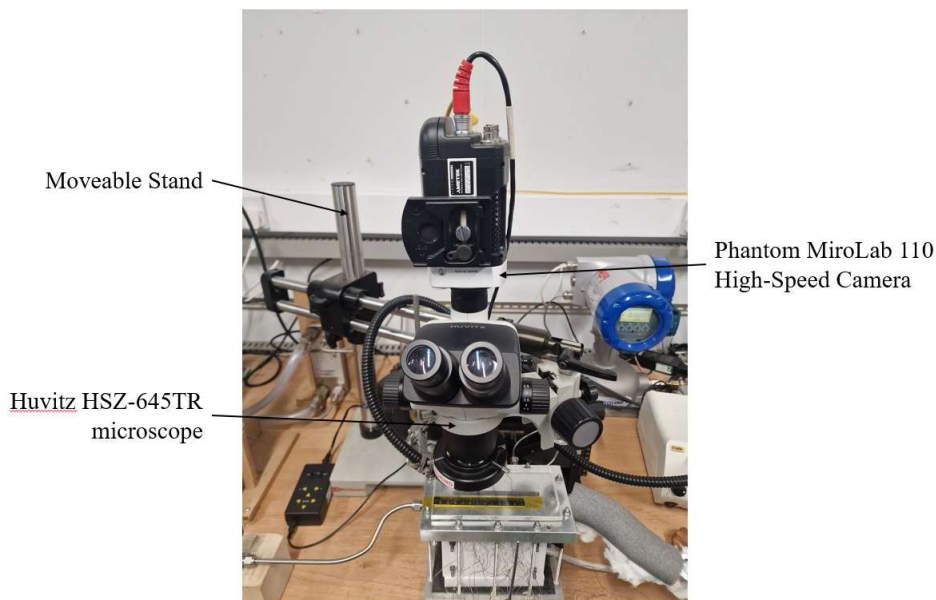


Figure 3.11. Microscope and high-speed camera system.

3.2.5 Software

NI LabView 2020 graphical programming software for tests and measurements was used to record all data taken from instruments such as the flow meter, thermocouples and pressure transducers during experimentation. This includes any facility validation experiments. Only the raw measurements, i.e. temperatures, mass flow rates, time and pressures, were obtained using NI LabView 2020. All data reduction required to obtain derived parameters, i.e. heat transfer coefficients, applied heat flux, fluid temperatures (as these cannot be measured directly in the channel), and graph creation was possible using MATLAB programming software. Fluid thermophysical property curves were extracted from Engineering Equation Solver and written into a MATLAB function script for ease of use in data reduction. Images obtained using the microscope and high-speed camera to study flow patterns were imported from video using Phantom Camera Control Software.

3.2.6 Initial Test Section Design

An initial single microchannel copper test section was designed for the present study using four main component groups. The groups included a clear polycarbonate viewing plate, an oxygen-free copper test section, Polytetrafluoroethylene (PTFE) housing and the aluminium clamping plates. The components are shown in the exploded computer aided design (CAD) drawing presented in Figure 3.12.

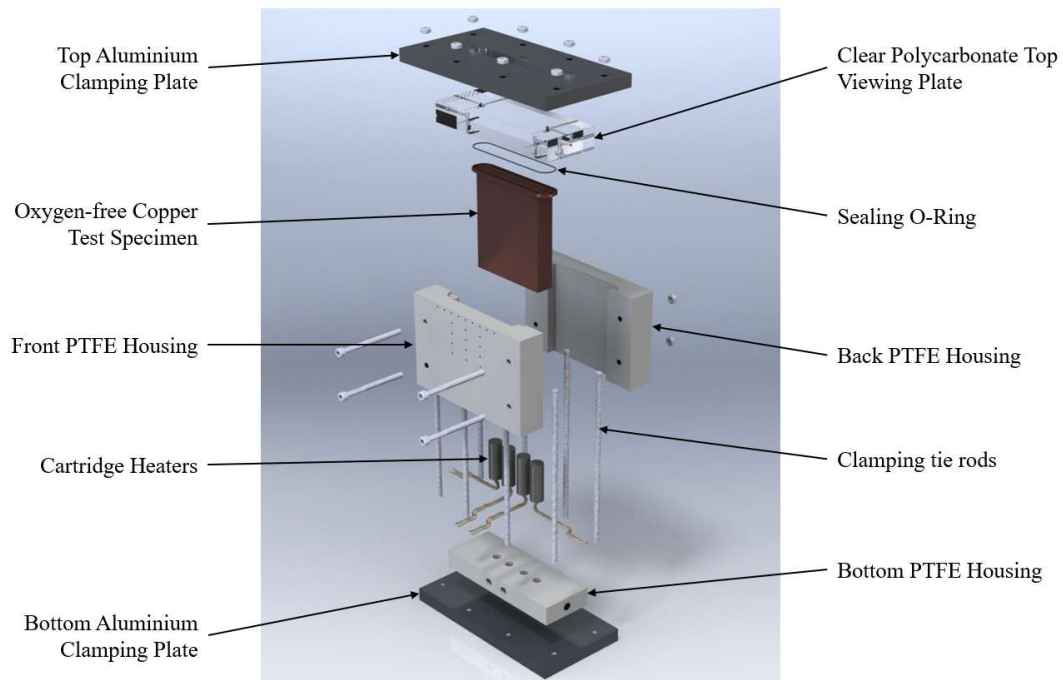


Figure 3.12. Exploded CAD diagram of the initial test section designed for the current study.

The polycarbonate top plate both allowed for flow visualisation and sealed the system with the aid of an o-ring. The plate also contained the top half of the inlet and outlet plenums, two 3 mm drilled holes into each plenum and a single 0.6 mm hole into each plenum. The two 3 mm holes in each plenum were drilled to take pressure measurements in the plenums and to allow flow in or out of the inlet or outlet plenums, respectively. The single 0.6 mm holes were drilled to insert the k-type thermocouples and take inlet and outlet temperature measurements. This configuration is explained diagrammatically in Figure 3.13 which also shows how the pressure drop between the plenums was measured.

The PTFE housing was designed to both house the oxygen-free copper test section and to limit heat loss. PTFE was chosen for this purpose due to its low thermal conductivity. The housing was manufactured in three parts; namely, the front PTFE housing, the back PTFE housing and the bottom PTFE housing. The front PTFE housing and back PTFE housing was manufactured in two parts and then bolted together for ease of manufacture and to reduce manufacturing costs. Twenty-one further holes were drilled into the front housing block for twenty-one K-type thermocouples to pass through the housing block and enter the oxygen-free copper test specimen. The PTFE base also housed the cartridge heaters and their connecting wires. The aluminium clamping plates secured the test section to negate any leakage and a slot was cut into the aluminium top plate to facilitate flow visualisation.

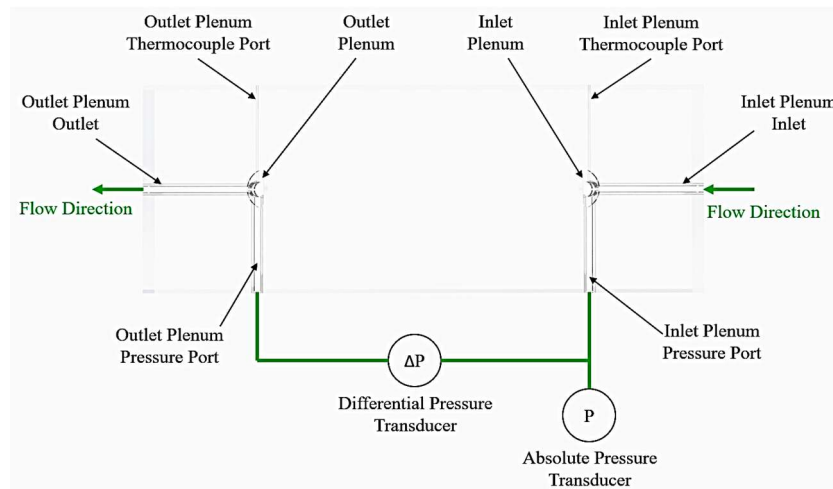


Figure 3.13. Clear polycarbonate top plate bottom CAD drawing with pressure transducer configuration diagram.

The oxygen-free copper test section is displayed in Figure 3.14, which was manufactured in the Precision Manufacturing Centre at The University of Nottingham using a Hurco VM1 CNC machine. The test specimen consisted of four 12.5 mm holes drilled 40 mm into the bottom to insert the four 250W cartridge heaters, providing a total heating power of 1 kW. Twenty-one 1 mm holes were drilled into the side of the specimen at a depth of 4 mm and then twenty-one 0.6 mm holes were drilled into the specimen, concentrically to the 1 mm holes, at a depth of 8 mm. These holes allowed the K-type thermocouples to be inserted with the tip located in a plane in line with the middle of the channel. The arrangement of these thermocouples is displayed in Figure 3.14 (c), where the top row of thermocouples was located 1.5 mm below the channel wall. The horizontal row of thermocouples was used to calculate the channel wall bottom temperatures, whilst the vertical columns were used to calculate the temperature distribution and hence heat flux perpendicular to the flow. The full data reduction is discussed later in this section; however, a one-dimensional heat flux was assumed in the y-direction. The one-dimensional assumption was confirmed by ensuring the heat flux was consistent across the three vertical thermocouple columns.

Three main features were machined into the copper test section; namely, the o-ring groove, the plenums and the single microchannel. An o-ring was sized according to the AS568F standard [368] to fit in the o-ring groove and compress against the clear polycarbonate plate, thus sealing the test section. Viton was selected as the o-ring material due to its high-wearing and relatively non-reacting properties. The plenums and channel were machined using a 0.5 mm end mill with a rotation speed of 8000 rpm and a feed rate of 200 mm/min. The channel was designed with a 0.75 mm x 0.75 mm square cross section and a length of 75mm. The total dimensions of the assembled test section were 175 mm x 90 mm x 160.5 mm, corresponding to a volume of 2528 cm³.

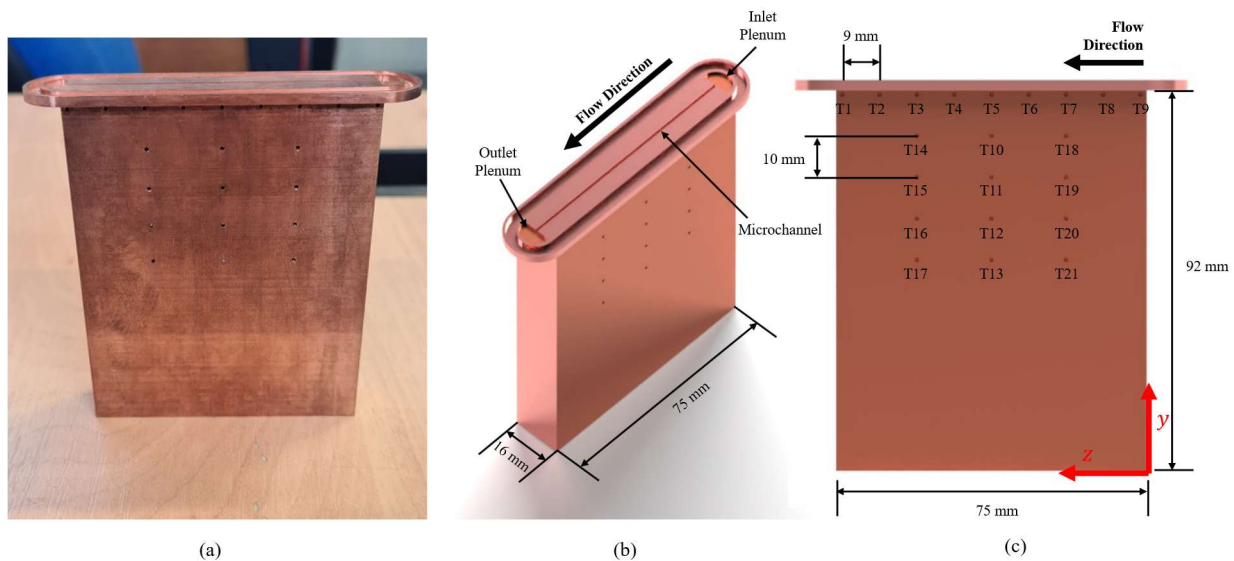


Figure 3.14. Oxygen-free copper test section: (a) photograph of test section; (b) isometric CAD drawing; (c) thermocouple position diagram.

3.2.7 Fluid Selection

The selection of an appropriate coolant is critical in thermal management studies, particularly in electronics cooling applications where safety, compatibility, and thermal performance are paramount. In this work, HFE-7100 (methoxy-nonafluorobutane, $C_4F_9OCH_3$), a dielectric fluid produced by 3M as part of their Novec range, was chosen as the working fluid. When this research commenced, HFE-7100 was widely accepted in the scientific community for its use in flow boiling applications due to its combination of desirable thermal and environmental properties. These included non-flammability, low toxicity, excellent material compatibility, zero ozone depletion potential (ODP), relatively low global warming potential (GWP of 320 over 100 years), and a short atmospheric lifetime (4.1 years) [369]. These characteristics made it a strong candidate for safe, environmentally responsible, and effective electronics cooling, especially in precision experiments where electrical insulation and thermal stability are essential.

In 2025, HFE-7100 was discontinued due to tightening environmental regulations on fluorinated greenhouse gases. This regulatory development was not foreseeable at the time of experimental design and system commissioning. Nonetheless, the experimental value of the fluid remains significant for several reasons. First, a large body of prior research exists for the fluid within the research group, including studies using multi-channel heat sinks [128, 303, 370, 371], providing a strong foundation for comparison and benchmarking. The ability to compare directly with previously published data, including work by close collaborators using similar equipment, strengthens the relevance and contextualization of the current results. This continuity supports validation and cross-referencing of single-channel results with multi-channel findings under consistent thermophysical conditions. This

then allows for the research team to build on this for future understanding of the complex phenomena that prevail in flow boiling in microchannels and make novel contributions that be relevant for other similar fluids.

Furthermore, although discontinued for widespread commercial use, HFE-7100 remains an ideal fluid for controlled academic environments where environmental release is strictly limited, and volumes used are small. In this context, the fluid's prior widespread adoption and well-documented performance characteristics offer practical advantages in data interpretation, model validation, and reproducibility [372–376]. Experimental data obtained using HFE-7100 are still highly valuable for advancing the understanding of fundamental flow boiling mechanisms in dielectric liquids. Moreover, insights from this research are likely to be transferable to future low-GWP alternatives that share similar thermophysical behaviour, helping bridge the transition from legacy to next-generation coolants.

3.3 MODIFICATIONS TO EXISTING FACILITY

The existing experimental facility located in Tower B at Brunel University of London was previously used to investigate the flow boiling of DI water in a single microchannel. The use of a different fluid as well as interesting initial results and observations meant that certain modifications needed to be made to the existing facility. This section outlines the modifications that were made for the purpose of investigating the flow boiling of HFE-7100 in a single copper microchannel.

3.3.1 Replacing the Pre-Heater

Significant difficulties arose when attempting to reach the desired degree of inlet subcooling in the test section. The desired degree of inlet subcooling for all experiments was approximately 10 K. A recurring problem present during initial experiment attempts was the appearance of vapour bubbles in the sight glass upstream of the test section as the degree of subcooling decreased, i.e. the inlet conditions reached closer to saturation. It was believed that the cause of these vapour bubbles was due to the pre-heater which was designed for the previous experimental configuration with water. The thermophysical properties of HFE-7100 and DI water are presented in Table 3.1 to help explain the problem. Noticeable in the table is the significant difference between specific heat and latent heat values of the two fluids, with DI water possessing much larger values.

The initial pre-heater presented in Section 3.2.1.4 was designed with a 2.5 kW capacity with a screw plug heater in a cylindrical shell with an internal diameter of 100 mm. The mass flow rates required to achieve mass fluxes in the microchannel between 50 kg/m²s to 8000 kg/m²s are 0.02813 g/s to 4.5 g/s. Hence, the average flow rate of the fluid was likely low enough that the heat transfer was driven nearly

entirely by natural convection with minimal forced convective effects. Furthermore, a sight glass and filter were placed in between the test section and the pre-heater where heat could be lost by the fluid. Hence, the preheater was required to heat the fluid to a subcooling lower than 10 K and hence closer to saturation to achieve the desired subcooling at the channel inlet. This required that the wall temperature of the pre-heater was likely greater than the saturation temperature of the fluid; however, this was not a problem for water as it requires a significantly higher wall superheat than HFE-7100 to initiate boiling. However, the wall superheats were high enough to initiate boiling in the fluid when HFE-7100 was used in the facility and vapour bubbles were consequently observed at the test section inlet.

Table 3.1. Thermophysical properties of HFE-7100 and DI water at 1 bar and 25 °C.

Fluid	Thermal Conductivity ($Wm^{-1}K^{-1}$)	Specific Heat at Constant Pressure ($Jkg^{-1}K^{-1}$)	Latent Heat of Vaporisation ($kJkg^{-1}$)	Surface Tension (Nm^{-1})	Density (kgm^{-3})
HFE-7100	0.06883	1.2	128.8	0.01281	1516
DI Water	0.6065	4.181	2442	0.07197	997

A solution was implemented to both increase the forced convective heat transfer to improve pre-heater heat transfer performance, thus reducing the required wall superheat. The solution was also designed to minimise the heat loss between the pre-heater and the test section. The simple solution was to replace the pre-heater with the 3 mm ID stainless steel piping used elsewhere in the test facility. A tape heater was then wrapped round the piping all the way up to the test section such that the fluid is being heated until it enters the test section inlet. The new configuration is displayed in Figure 3.15. Insulation was also placed over the tape heater to reduce heat loss from the tape heater. The tape heater was connected to the variac previously used to control the pre-heater.

3.3.2 Change of Test Section Design

Further problems arose when using the initial test section design. The first problem arose due to the cross-sectional area of the copper test section base. Section 3.7 summarises the utilised data reduction; however, it is necessary to note in this section that the applied heat flux was calculated using the temperature gradient in the base of the copper test section. The temperature gradient was low due to the cross-sectional area of the copper base in relation to that of the channel. Consequently, the thermocouple uncertainty was significant in comparison to the variation in temperatures between the test section base locations.

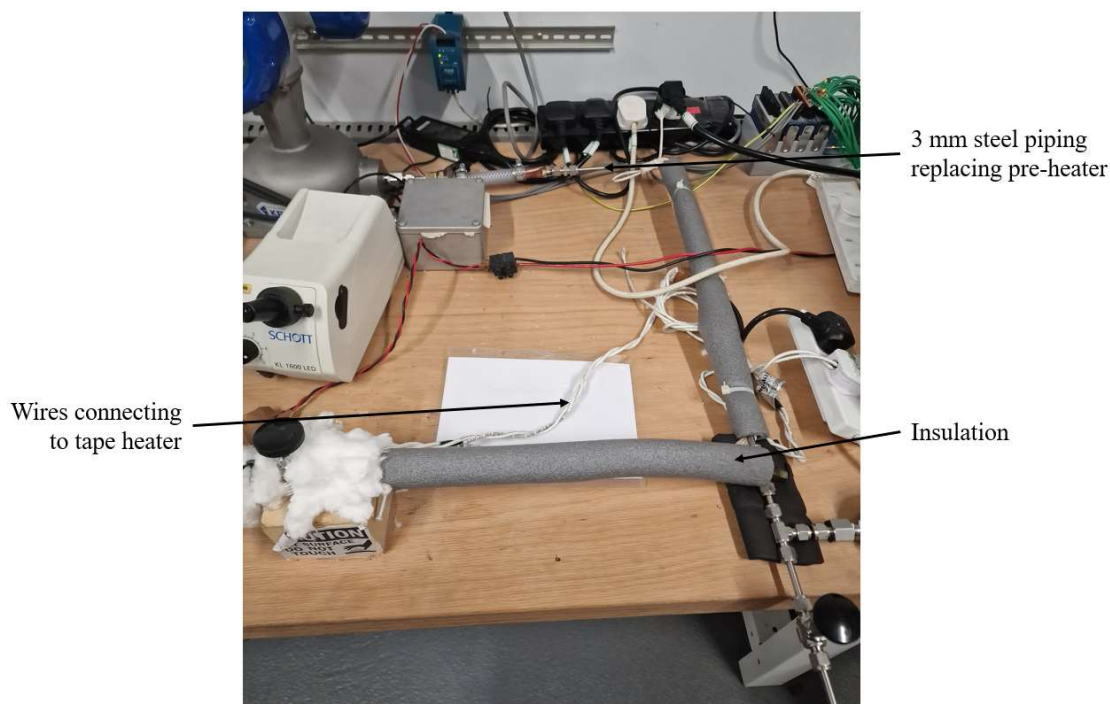


Figure 3.15. Configuration used to replace screw immersion pre-heater.

The other problem in the test section design was due to the plenum design. The fluid entered the clear polycarbonate plate through 3 mm drilled holes before then entering the semi-circular plenum with a 10 mm diameter. The fluid then flowed from a 10 mm semi-circular plenum at right angles into the 0.75 mm x 0.75 mm square channel cross section. The design of the plenum meant that much of the fluid was held in the corners of the plenum and resulted in bubble nucleation in the plenums prior to fluid entering the channel. This phenomenon is shown in Figure 3.16, although the image is not entirely clear as the polycarbonate plate was not designed to observe the plenums. The result was that only a form of annular flow was observed in the channel and that all boiling was suppressed due to the reduced film thickness.

A new test section was designed to increase the temperature gradient for a given heat flux and to eliminate bubble nucleation in the inlet plenum. However, the current study aimed to both investigate flow boiling patterns and critical heat flux mechanisms. The study of a full range of flow patterns required that a full range of vapour qualities was observed, i.e. 0-1, and that critical heat flux was not reached. This required that the test section was sufficiently long that the full range of vapour qualities could be observed. This meant that the critical heat flux could not be reached in the same test section. Therefore, two test sections were designed and manufactured; one to investigate flow patterns for a full range of vapour qualities and the other to reach critical heat flux conditions. This also allowed for the effect of heated length to be examined on flow patterns, heat transfer rates and pressure to be assessed.

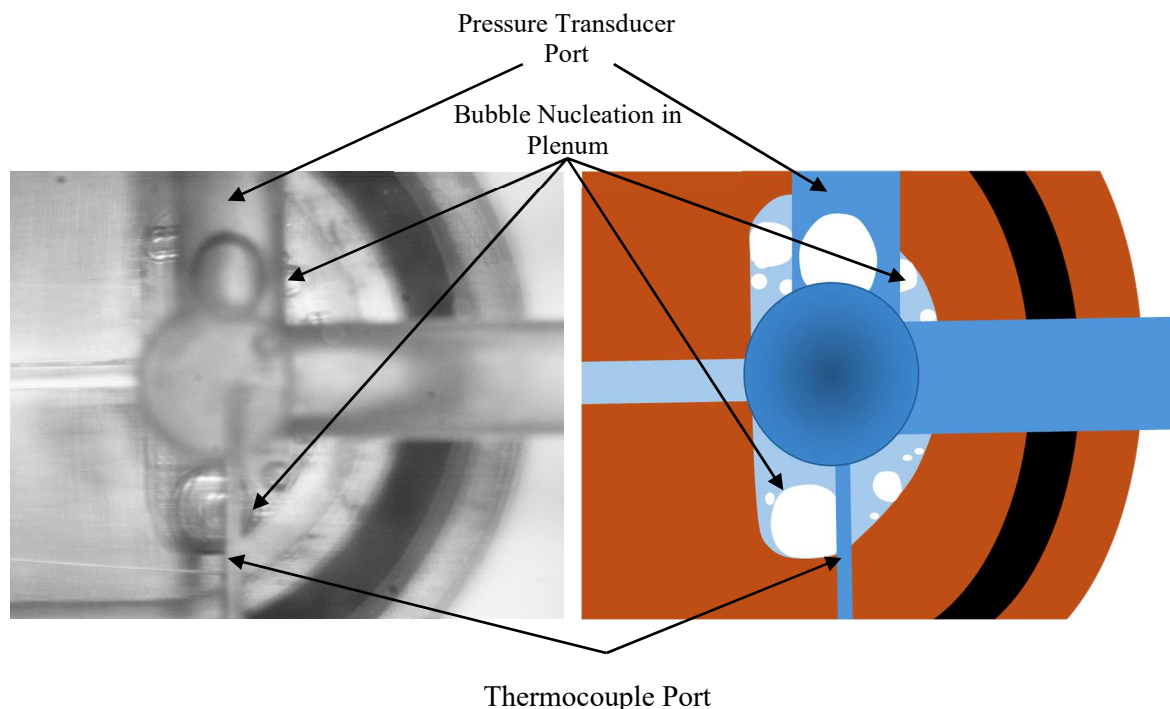


Figure 3.16. Undesired bubble nucleation in inlet plenum.

3.3.2.1 75 mm Long Test Section Design

The design of the new test section was similar to the previous one in terms of components, i.e. a clear polycarbonate cover plate, aluminium clamping plates, PTFE housing and an oxygen free copper test section. An exploded CAD drawing of the design is shown in Figure 3.17. Full technical drawings of the copper test section, the PTFE base and the clear polycarbonate cover plate are presented in Appendix D. The functions of each component remained the same with the main difference in the new design lying in oxygen-free copper test section and the clear polycarbonate top viewing plate.

A CAD drawing of the clear polycarbonate top viewing plate showing the key design features is presented in Figure 3.18 and the dimensions of the clear polycarbonate cover plate are displayed in Figure 3.19. Again, a part of each plenum was manufactured in the polycarbonate plate; however, this was achieved by drilling two 2 mm diameter holes into the plate in the new design. Each plenum then possessed an inlet or outlet port, a thermocouple insert port and a pressure transducer port.

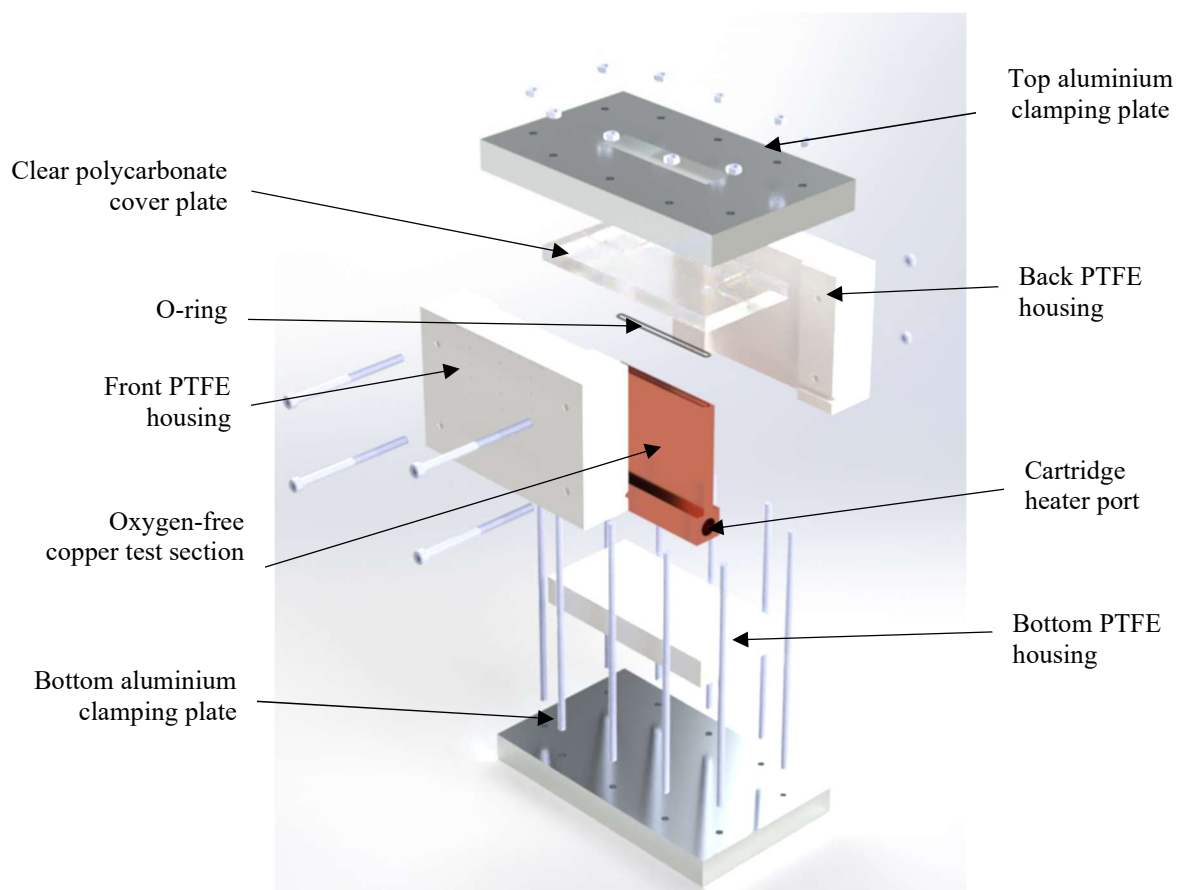


Figure 3.17. Exploded CAD diagram of the long test section design.

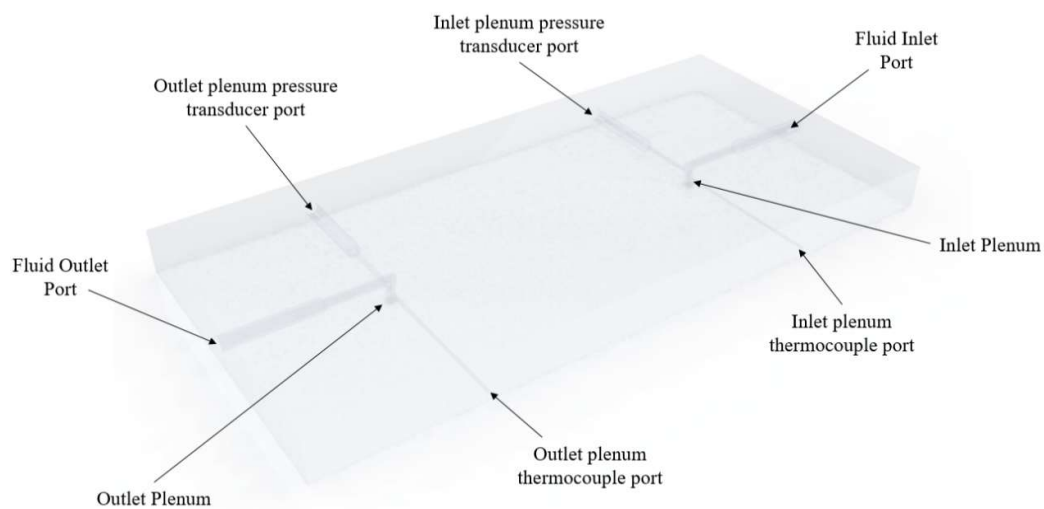


Figure 3.18. CAD drawing of long clear polycarbonate cover plate.

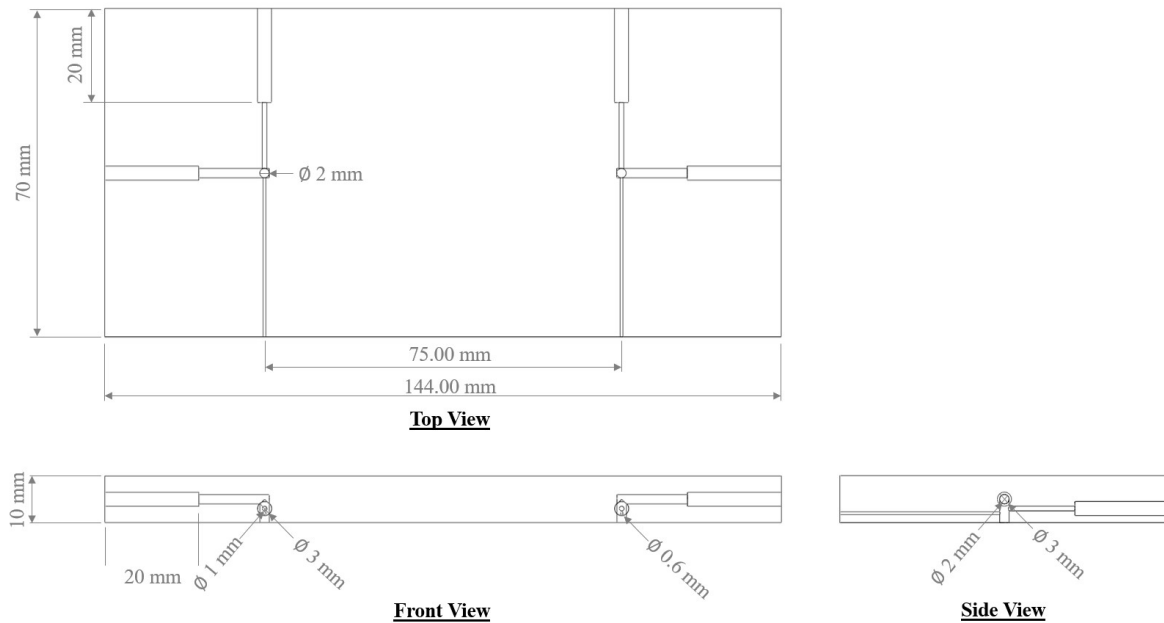


Figure 3.19. Key dimensions of long clear polycarbonate cover plate.

The fluid inlet port was redesigned from a single 3 mm drilled hole into a new design where a 3 mm hole was drilled into the plate for 20 mm, before a 2 mm diameter hole was drilled into the plenum. This design feature was implemented to reduce the possibility of flashing at the channel inlet. The stainless-steel tubing inserted into the test section possessed an outer diameter of 3 mm with an internal diameter of 2 mm, thus eliminating any change in flow area that may cause a sudden change in pressure drop. A similar design was utilised for the transducer ports, where a 3 mm hole was drilled 20 mm into the rear of the plate but a 1 mm hole was drilled into the plenum. A 1 mm hole was used in this situation due to the increased allowance in manufacturing tolerance.

Finally, a 0.6 mm hole was drilled from the front side of the plate into the plenum to insert the tip of the thermocouples into the plenums. All ports, i.e. inlet/outlet, thermocouple and transducer ports were sealed using silicon.

The long oxygen-free copper test section was also manufactured in the Precision Manufacturing Centre at The University of Nottingham using a Hurco VM1 CNC machine. A CAD diagram of the oxygen-free copper test section for the long test section design is presented in Figure 3.20 and the key dimensions are displayed in Figure 3.21. The first alteration to the design was to reduce the number of cartridge heaters from four 250 W heaters to a single 500 W cartridge heater with a horizontal orientation. The reason for this was two-fold: (1) cartridge heaters are prone to burnout and a single cartridge heater would make replacing it easier and (2) a 75 mm heater orientated horizontally would provide a more uniform heat flux to the channel wall. Hence, a 12.5 mm diameter through hole was drilled into the base of the test section which possessed a width and height of 20 mm.

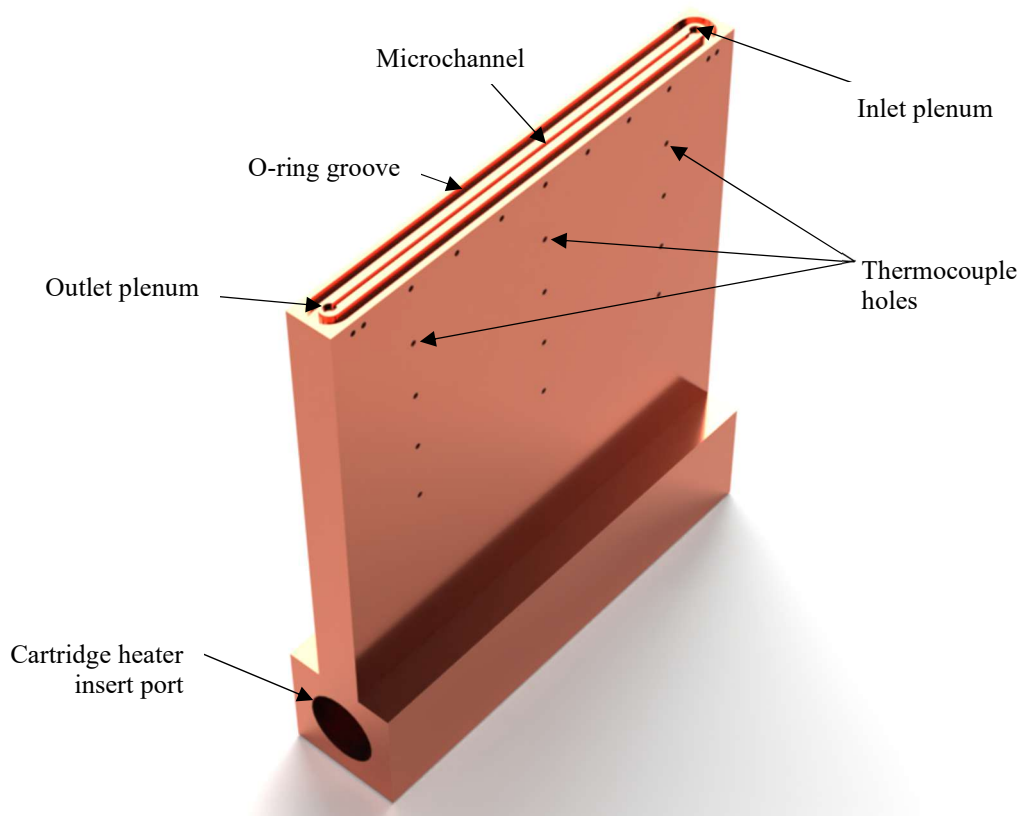


Figure 3.20. CAD drawing of long oxygen-free copper test section.

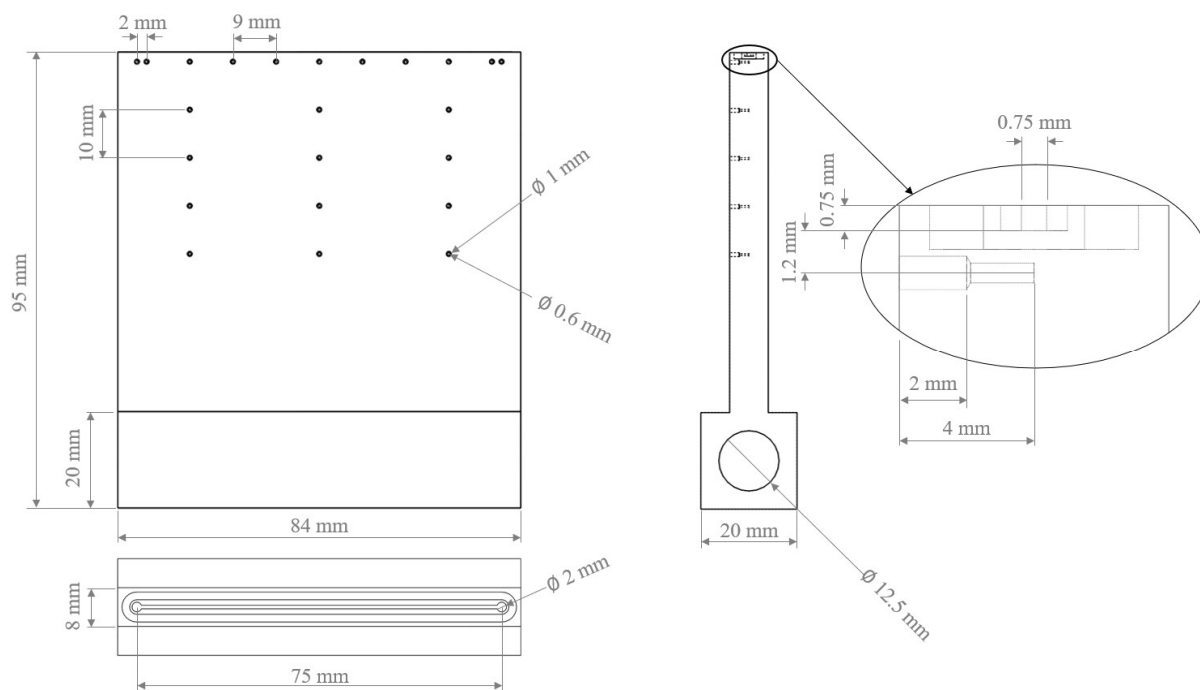


Figure 3.21. Key dimensions of long oxygen-free copper test section.

The area of the oxygen-free copper base was also modified in the new design, where it was significantly reduced. The initial test section base width and length were 16 mm and 75 mm, respectively. The new test section was designed with a width of 8 mm and a length of 84 mm, reducing the cross-sectional area from 600 mm² to 336 mm². This would allow a greater temperature gradient for a given heat flux and therefore reduce the associated error in measuring the heat flux, as is made clear in Section 3.5.3 on uncertainty propagation analysis.

Thermocouple holes were drilled into the side of the test section, first with 1 mm diameter holes drilled 2 mm into the front face and then 0.6 mm diameter holes were drilled 4 mm into the front face. This allowed the thermocouple tips to be located directly underneath the centre of the channel in the axial direction. The plenums milled into the test section were also reduced to 2 mm in diameter, as mentioned when discussing the long test section design clear viewing plate. They were milled directly into the copper in this design, which meant that further consideration was required for the heat lost through the plenums. This was because the overhang on the initial design was in the order of a few millimetres and was consequently prone to bending and warping. This resulted in difficulty sealing the test section.

The microchannel was designed with same dimensions, i.e. a square 0.75 mm x 0.75 mm cross. However, circular plenums were incorporated at a length of 75 mm apart. This resulted in a designed channel length of 74.15 mm.

3.3.2.2 30 mm Long Channel Test Section Design

A second shorter test section was designed and manufactured for two purposes. The first purpose was to investigate the effect of heated length on flow patterns, heat transfer rates and pressure drop. The second reason for the shorter test section was to investigate critical heat flux in a single microchannel. Thus, a new test section was designed with a 30 mm channel length based on the previous design but with some modifications. A rendered CAD assembly drawing of the 30 mm long channel test section is displayed in Figure 3.22 and the full engineering drawings shown in Appendix E.

The first modification to the design was the manner in which the pipework was sealed into the test section. Sealing problems were an initial obstacle with the first design, as the stainless-steel pipework and thermocouples were sealed against the polycarbonate cover plate using an epoxy resin. The epoxy resin would take time to cure and would often fail pressure tests that were conducted with nitrogen prior to degassing of the facility to ensure the test section and facility were hermetically sealed. The point of failure was found where the stainless-steel pipework entered the polycarbonate cover plate. This meant that a further layer of epoxy resin would be applied to fix the leak and an additional waiting time would occur before the facility could be pressure tested once more, then degassed and prepared for experiments. The new design was inspired by pipework flanges and was sealed with an O-ring acting

as a gasket against the polycarbonate block. This is shown in Figure 3.23, where the left side shows the fully assembled sealed pipework in the polycarbonate and the right side displays the exploded view of the components.

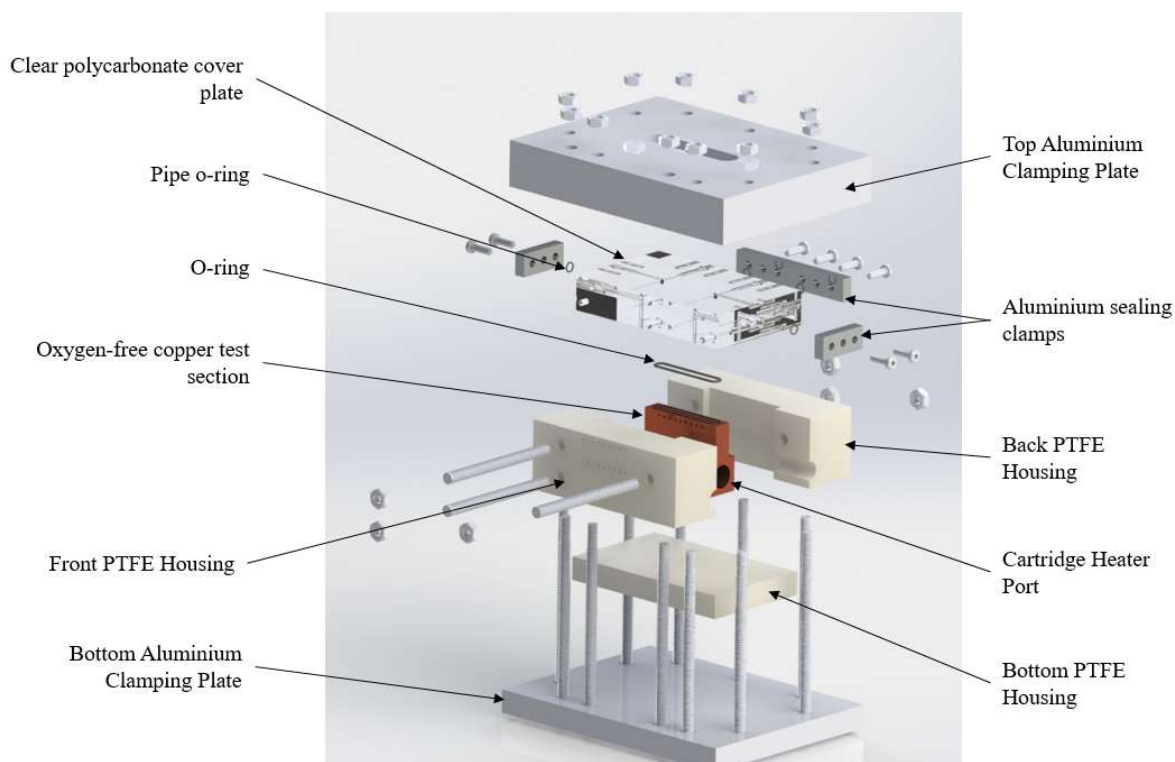


Figure 3.22. Exploded CAD assembly of 30 mm long test section.



Figure 3.23. Sealing methodology for stainless steel pipework in the test section.

As shown in Figure 3.23, the 3 mm outer diameter pipework was fed through the aluminium sealing plate and a 3 mm internal diameter, 1 mm thick O-ring was placed over the end of the pipework. The pipework was then slid into the polycarbonate cover plate and the O-ring guided into a 5.2 mm diameter

and 0.8 mm deep O-ring groove that allowed for the compression and expansion of the O-ring to form a hermetic seal at the joint. This was then fastened into place with two M3 hex screws inserted into the two M3 tapped holes in the polycarbonate cover plate.

The second modification to the design was in the oxygen-free copper block itself. Firstly, the height of the test section was reduced from 95 mm to 30 mm. As the channel reaches CHF, the end portion of the channel would be prone to dry-out and significant drops in heat transfer coefficients would occur. If the channel is too far away from the cartridge heater, there is potential for the heat to redistribute in the copper block towards the region of the channel where the heat transfer coefficients are the greatest. Section 3.5 presents the data reduction for the experiments, where a uniform heat flux is assumed. Therefore, to achieve this assumption, the channel was moved significantly closer to the cartridge heater but still with enough height that the heat flux could be measured in the channel.

The thermocouple arrangement was also altered in the second test section, where only two rows of thermocouples were used. The shorter oxygen-free copper test section and the key dimensions, including thermocouple arrangement are shown in Figure 3.24 and Figure 3.25, respectively. However, nine columns of thermocouples were now used to evaluate the heat flux across the length of the channel. This provided a greater assurance that the uniform heat flux assumption was valid as the heat flux could be assessed at nine different axial distances.

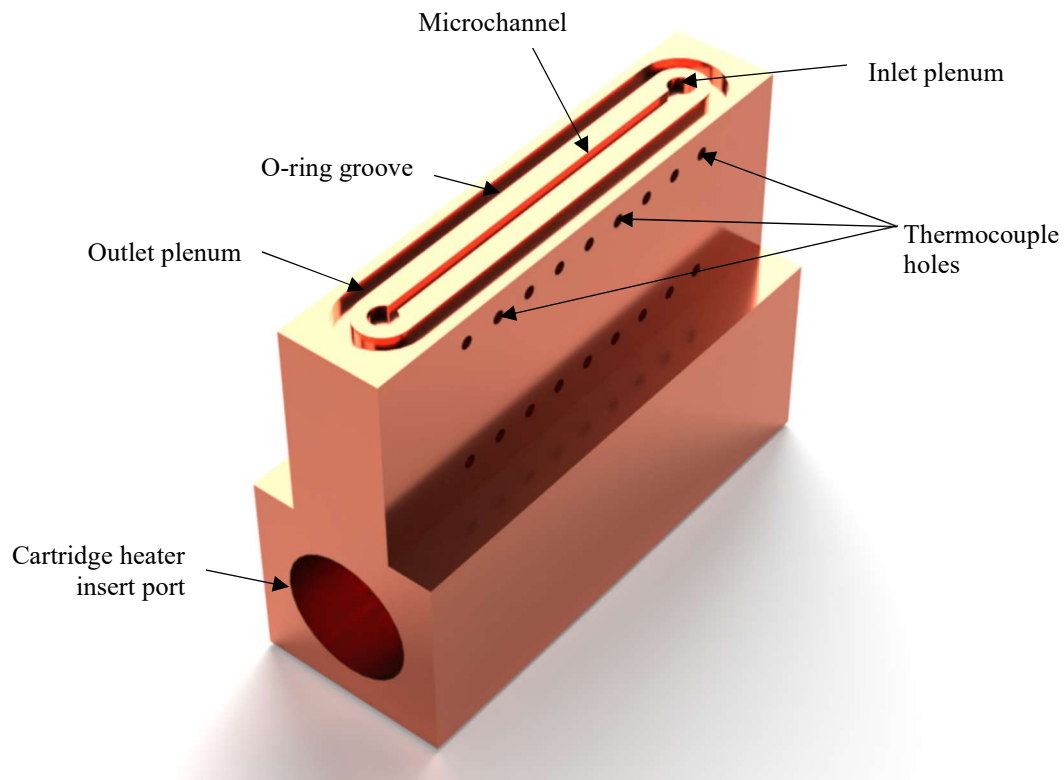


Figure 3.24. CAD rendering of 30 mm long oxygen-free copper test section.

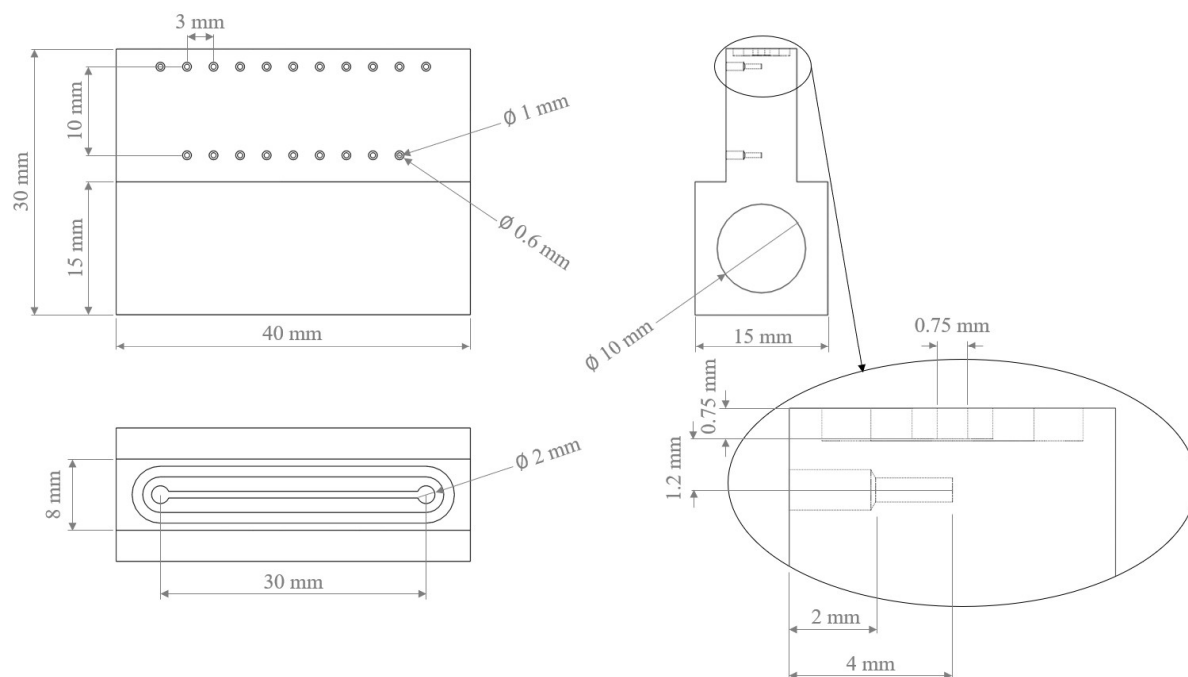


Figure 3.25. Key dimensions of short oxygen-free copper test section.

Aside from the aforementioned changes, it was ensured that the design did not alter the flow path of the working fluid, i.e. the channel and plena design were unchanged apart from the length. This ensured that any consideration of pressure drop between the plena and the channel would be consistent between the two test sections and that they only variable for comparison was the heated length.

3.3.3 Replacing Variac for Test Section

The heating of the test section was initially conducted through one of the RS WCV15E-1 enclosed variacs. However, the power range between 0 and 4050 W was significantly too large to investigate flow boiling in a single 0.75 mm hydraulic diameter channel where HFE-7100 was used as the working fluid. Furthermore, the design of the enclosed variacs means that the input power to the test section is difficult to control, due to the discrete nature of the variac windings.

The enclosed variable transformer to the test section was replaced by an Elektro Automatik EA-PS 9360-30 digital bench power supply, shown in Fig. 3.26. Whilst this also possessed a very high output power range of 3 kW, it was capable of high levels of control at lower powers. Consequently, the voltages and Amperes could each be controlled in steps of 0.1 V and 0.1 A, respectively. This allowed for an input power resolution of 0.01 W, which was more suitable for the study of flow boiling in a single microchannel.



Figure 3.26. Elektro Automatik EA-PS 9360-30 digital bench power supply.

3.3.4 Inclusion of Gas-Loaded Accumulator

Attempts were conducted to validate the facility using single-phase flows, where experimental results were compared to existing literature correlations for Fanning friction factor and Nusselt number. However, significant pressure fluctuations were observed during the initial experiments for laminar single-phase flows. An example pressure signal for a mass flux of $1200 \text{ kg/m}^2\text{s}$ is presented in Figure 3.27. The mass flux of $1200 \text{ kg/m}^2\text{s}$ corresponded to a laminar Reynolds number of 1550. Thus, the significant pressure fluctuations observed in Figure 3.26 are not typical of such flow rates.

A fast Fourier transform (FFT) of the signal was conducted to identify the pressure drop frequency components and the potential cause of the pressure fluctuations, which is presented in Figure 3.28. It was identified that the largest contributor was the rotation speed of the pump, corresponding to a frequency of 19.85 Hz. Furthermore, it was also identified that many of the other frequencies were simply harmonics of the pump rotational speed.

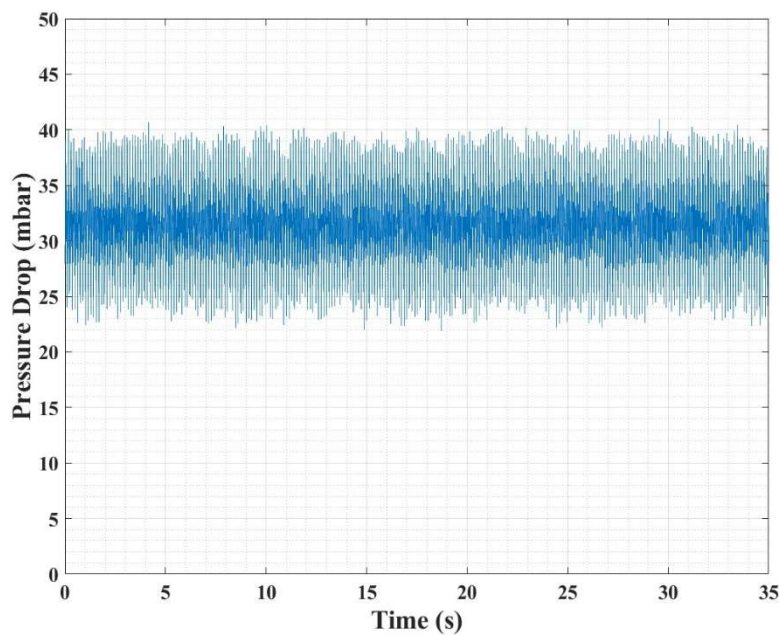


Figure 3.27. Pressure drop signal for single-phase liquid flow ($G=1200 \text{ kg/m}^2\text{s}$, $Re=1550$).

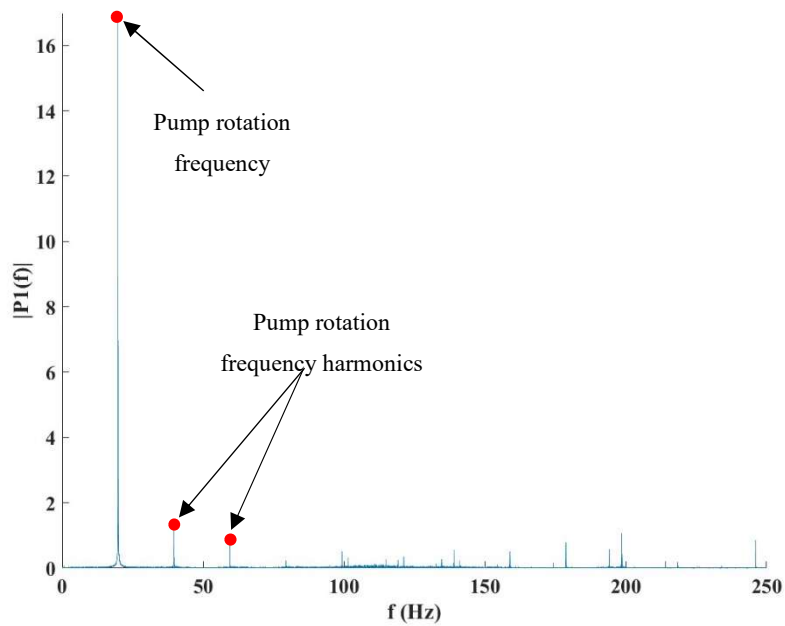


Figure 3.28. FFT for pressure drop signal ($G=1200 \text{ kg/m}^2\text{s}$, $Re=1550$).

A gas loaded accumulator was placed after the pump in the main closed flow loop to account for any flow and pressure drop disturbances caused by the pump. This was found to eliminate the signal coming from the pump; however, a small contribution still remained at 246.146 Hz . The resulting FFT of the pressure signal for a mass flux of $1200 \text{ kg/m}^2\text{s}$ after the inclusion of the gas loaded accumulator is displayed in Figure 3.29. This contribution to the pressure drop frequency matched that of the Coriolis flow meter. Consequently, the gas loaded accumulator was placed after the Coriolis flow meter to eliminate all signal contributions. However, the placement of the gas loaded accumulator after the Coriolis flow meter resulted in a reappearance of the contributions from the gear pump. Therefore, the gas loaded accumulator was placed after the pump and a flexi hose was included at the outlet of the Coriolis flow meter. This configuration was found to eliminate any pressure drop fluctuations.

A schematic of the final experimental facility that includes the change of pre-heater, test section heating, inclusion of gas loaded accumulator and flex-hose is presented in Figure 3.30. This configuration was used for the remainder of the study.

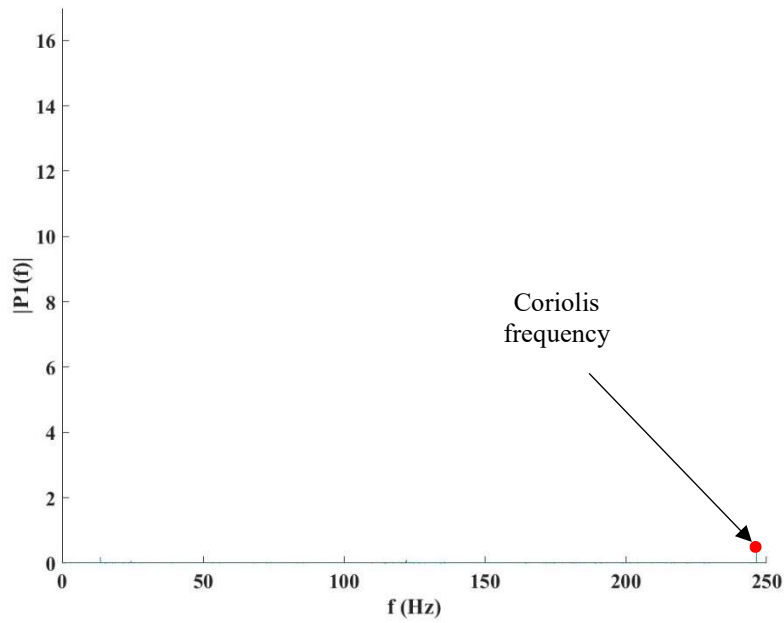


Figure 3.29. FFT for pressure drop signal with gas loaded accumulator ($G=1200 \text{ kg/m}^2\text{s}$, $Re=1550$).

3.4 SURFACE ANALYSIS

Surface characteristics have been found to play a significant role in flow boiling heat transfer rates [377, 378]. Whilst the surface parameters were attempted to be kept consistent for both test sections and were therefore not considered in this study, accurate reporting of the surface characteristics is necessary for those who wish to compare new experimental data to the current work. Firstly, a Zeiss O-inspect 01-442 multi-sensor measurement machine was used to find the average channel height, H_{ch} , channel width, W_{ch} , and channel length, L_{ch} , for both the 75 mm long and 30 mm long channel test sections. The channel height and width were measured at each location with an accuracy of $\pm 1 \text{ } \mu\text{m}$. The channel measurement results are presented in Table 3.2, where the mean width, mean height and length were used in the data reduction in Section 3.5.

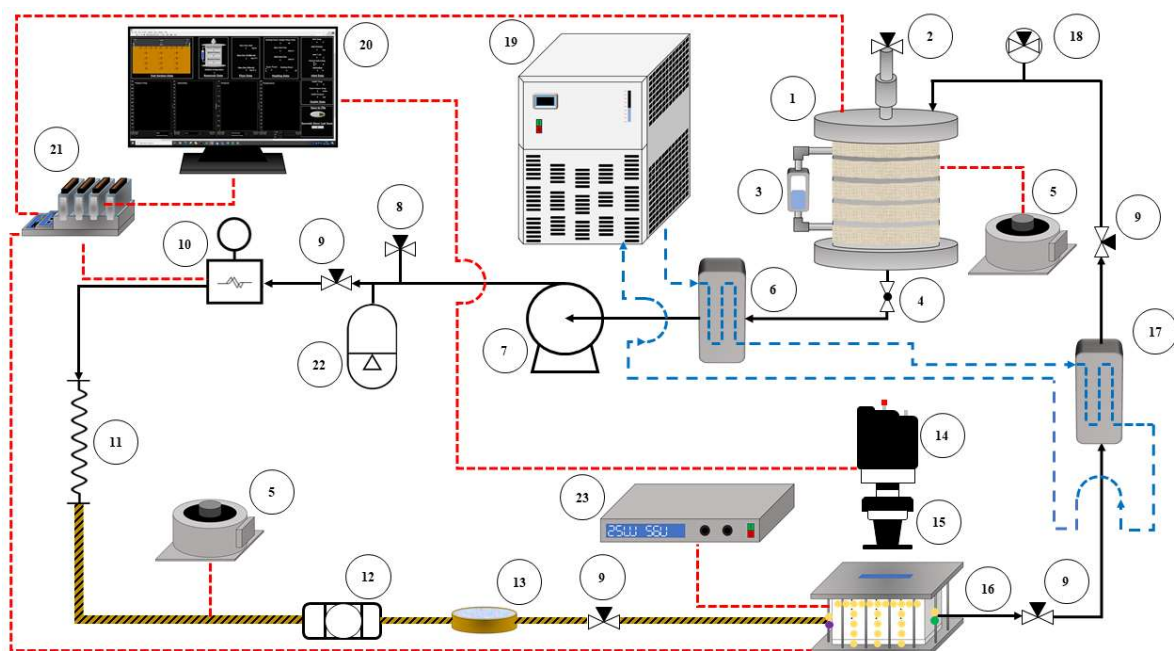


Figure 3.30. Schematic of final experimental facility.

Table 3.2. Measured test section channel dimensions.

Test Section	Inlet	Middle	Outlet	Inlet	Middle	Outlet	Mean	Mean	Length
	Width	Width	Width	Height	Height	Height	Width	Height	
	(μm)	(μm)	(μm)	(μm)	(μm)	(μm)	(μm)	(μm)	(mm)
75 mm long Channel	752.00	755.62	751.59	808.96	814.11	810.12	753.07	811.06	74.22
30 mm long channel	758.24	756.74	756.28	814.23	813.78	813.48	757.09	813.83	29.15

A Bruker NP Flex 3D non-contact, white light optical profiling machine was used to carry out any roughness analysis. The Bruker NP Flex 3D uses white light interferometry to obtain 3D areal data of the given sample area and has a vertical resolution of <0.15 nm. Three separate areas were spot checked in the same manner as the measurements for the channel dimensions. Three 0.63 mm x 0.48 mm samples were taken, one at each channel location. The sample surface areal data obtained by the Bruker NP Flex 3D was then processed using Mountains® 8 Smile-View Map software, where surface reconstructions could also be processed. Reconstruction of the channel surface allowed for measurement of surface parameters such as the surface roughness, Kurtosis, maximum peak height, root mean square height, skewness, maximum pit height and maximum height. The obtained surface parameters from the reconstructions are presented in Table 3.3, whilst the reconstruction for the 75 mm long channel test section is presented in Figure 3.31. Significant variation in the measured values and in the overall surface structure was observed. Clear cavities are apparent at the inlet, and cutting marks were also evident across the surface, as can be seen in the figure.

Table 3.3. Measured Surface Parameters

Surface Parameter	Mean 75 mm long channel test section value	Mean 30 mm long channel test section value	Units
Sa	173	148	μm
Sku	11.2	14.6	
Sp	1.93	1.80	μm
Sq	0.216	0.197	μm
Ssk	-0.620	-0.533	
Sv	-3.92	-4.74	μm
Sz	5.84	6.54	μm

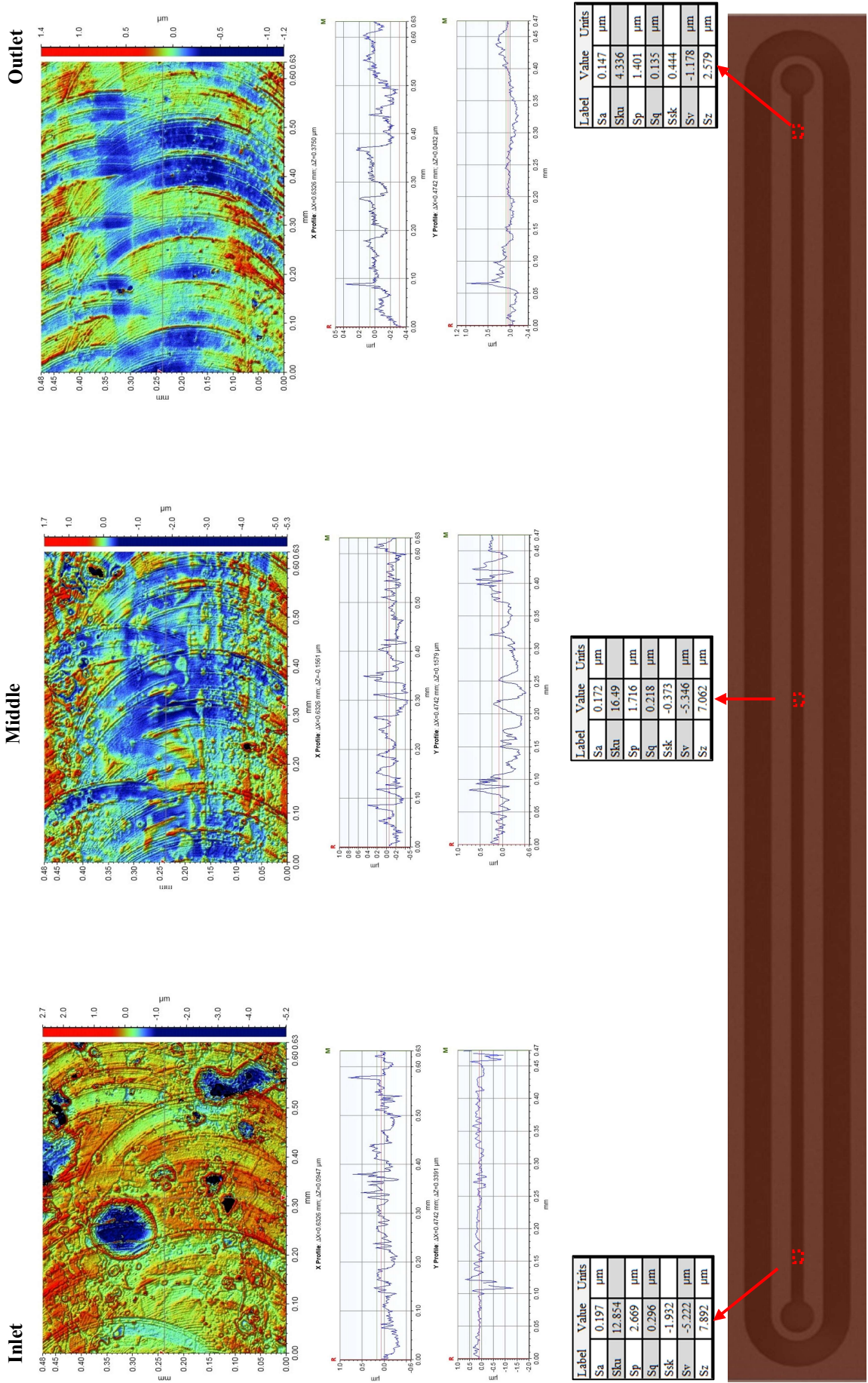


Figure 3.31. Surface reconstructions for the 75 mm long channel test section.

3.5 DATA REDUCTION

This subsection presents the data reduction that was required to calculate the pressure drop, heat transfer coefficient, vapour quality and other parameters required to plot experimental flow pattern maps. The single-phase data reduction that was used to validate the experimental facility is presented first. The experimental Fanning friction factor and average Nusselt numbers were calculated in this subsection and then compared to existing literature correlations in Section 3.8. The data reduction used for flow boiling experiments is then presented. Finally, the uncertainty analysis is discussed.

3.5.1 Single-phase Data Reduction

The pressure drop across the entire channel was calculated to account for the change in pressures between the channel and the plenums. This was calculated as such:

$$\Delta P_{ch} = \Delta P_{meas} - (\Delta P_{in,90} + \Delta P_{sc} + \Delta P_{se} + \Delta P_{out,90}) \quad (3.1)$$

where ΔP_{ch} and ΔP_{meas} are the actual pressure drop along the channel and the measured pressure drop between the plenums, respectively. $\Delta P_{in,90}$ and $\Delta P_{out,90}$ represent the pressure drop as a result of the flow direction change in the inlet and outlet plenums, respectively. ΔP_{sc} and ΔP_{se} represent the pressure drop due to the sudden contraction from the inlet plenum to the channel and the sudden expansion from the channel to the outlet plenum, respectively. This concept is shown diagrammatically in Figure 3.32. The pressure drop components relating to the inlet and outlet plenums were calculated from Equation (3.2) to Equation (3.5). Equations (3.2) and (3.3) were proposed by Remsburg [379], whilst the pressure losses due to a sudden contraction and a sudden expansion were proposed by Kays and London [380].

$$\Delta P_{in,90} = \frac{K_{90} G_{pl}^2}{2\rho_l} \quad (3.2)$$

$$\Delta P_{out,90} = \frac{K_{90} G_{pl}^2}{2\rho_l} \quad (3.3)$$

$$\Delta P_{sc} = \frac{G_{ch}^2}{2\rho_l} [(1 - \beta_{pl}^2) + K_c] \quad (3.4)$$

$$\Delta P_{se} = \frac{G_{ch}^2}{2\rho_l} [(1 - \beta_{pl}^2) + K_e] \quad (3.5)$$

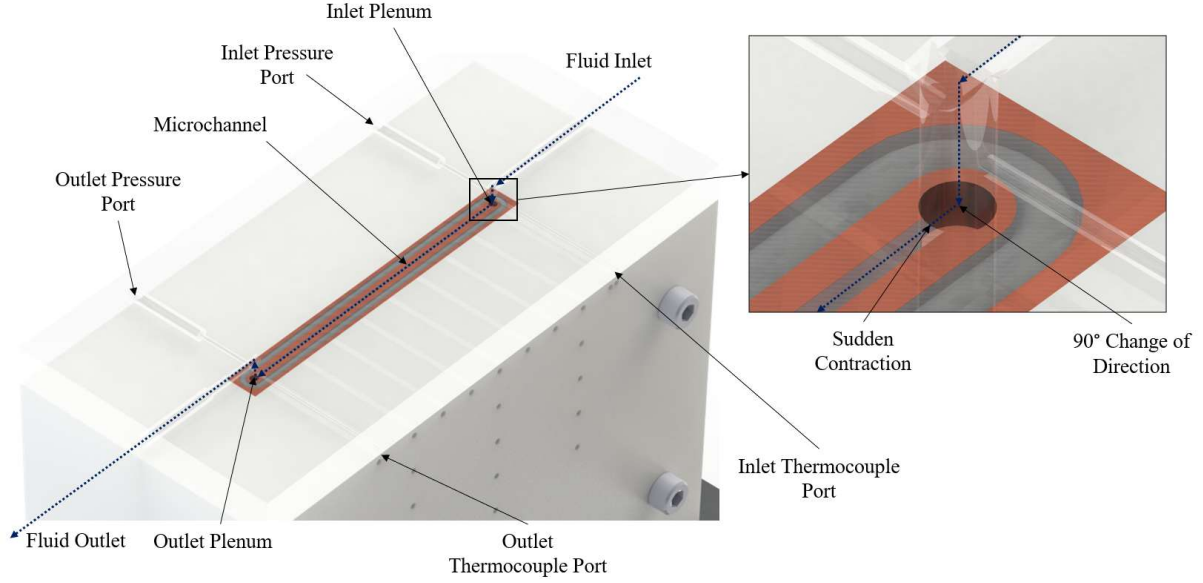


Figure 3.32. Pressure losses inside the test section to be accounted for.

K_{90} represents the loss coefficient for 90° changes in flow direction, which was proposed by Phillips [381] to be 1.2 and ρ_l represents the liquid density. K_c and K_e represent the coefficients for a sudden contraction and a sudden expansion, respectively. These values were extracted from the figure provided by Kays and London [380] and were found to be 1.133 for sudden contractions and 1 for sudden expansions. These values were dependent on the flow regime (laminar or turbulent) and plenum to channel area ratios. All relevant fluid thermophysical properties were found using Engineering Equation Solver software. G_{pl} , and G_{ch} represent the mass flux in the plenum and channel mass flux, respectively. G_{pl} , and G_{ch} are presented in Equations (3.6) and (3.7), respectively; where $\dot{m}$ represents the mass flow rate, H_{ch} represents the channel height, W_{ch} represents the channel width and ϕ_{pl} represents the diameter of the plenums.

$$G_{pl} = \frac{4\dot{m}}{\pi\phi_{pl}^2} \quad (3.6)$$

$$G_{ch} = \frac{\dot{m}}{H_{ch}W_{ch}} \quad (3.7)$$

Finally, β_{pl} represents the flow area ratio between the plenum and the microchannel as shown in Equation (3.8).

$$\beta_{pl} = \frac{H_{ch}W_{ch}}{H_{ch}\phi_{pl}} \quad (3.8)$$

The experimental pressure drop was used to calculate the experimental Fanning friction factor as follows:

$$f_{exp} = \frac{\Delta P_{ch} D_h \rho_l}{2 L_{ch} G_{ch}^2} \quad (3.9)$$

where D_h and L_{ch} are the channel hydraulic diameter and the channel length, respectively. The hydraulic diameter of the channel was calculated using Equation (3.10).

$$D_h = \frac{2 H_{ch} W_{ch}}{(H_{ch} + W_{ch})} \quad (3.10)$$

The heat flux applied to the channel was calculated by measuring the temperature distribution in the test section and accounting for the heat loss from the top of the test section. The methodology through which this was estimated is presented in Section 3.9. A one-dimensional heat flux was assumed in the copper block and the base heat flux was calculated using Equation (3.11).

$$q''_{base} = -k_{cu} \left. \frac{dT}{dy} \right|_{y=0} \quad (3.11)$$

where k_{cu} represents the conductivity of the copper test section and $\left. \frac{dT}{dy} \right|_{y=0}$ is the vertical temperature gradient inside the copper test section. The heat flux applied to the channel wall was then calculated by assuming that any heat that was not lost through the top of the test section was applied to the channel. This is represented in Equation (3.12), where A_{base} , $A_{pl,heat}$ and Q_{loss} represent the test section base area, the plenum area and the heat loss through the top of the test section, respectively. A uniform heat flux was assumed.

$$q''_w = \frac{q''_{base} A_{base} - Q_{loss}}{(2 H_{ch} + W_{ch}) L_{ch} + A_{pl,heat}} \quad (3.12)$$

The local wall temperature was calculated as follows:

$$T_w(z) = T_{th}(z) - \frac{q''_{base} y}{k_{cu}} \quad (3.13)$$

where T_{th} is the temperature of the thermocouple located below the channel, y is the distance between the bottom channel wall and the thermocouple location and z is the axial distance along the channel.

To account for the heat lost to the fluid through the plenums, the temperature of the fluid into the channel was calculated using Equation (3.14).

$$T_{in} = T_{in,meas} + \frac{q''_w A_{pl,heat}}{\dot{m} C_{p,l}} \quad (3.14)$$

The local fluid temperature was calculated using Equation (3.15), where $C_{p,l}$ is the specific heat capacity of the liquid phase at constant pressure.

$$T_f(z) = T_{in} + \frac{q''_w (2H_{ch} + W_{ch})z}{\dot{m} C_{p,l}} \quad (3.15)$$

The local heat transfer coefficient was then calculated as follows:

$$h_{sp}(z) = \frac{q''_w}{T_w(z) - T_f(z)} \quad (3.16)$$

The average heat transfer coefficient and the average Nusselt numbers were then calculated using Equation (3.17) and Equation (3.18), respectively. $\bar{k}_l$ represents the average thermal conductivity of the liquid phase.

$$\bar{h}_{sp} = \frac{1}{L_{ch}} \int_0^{L_{ch}} h_{sp}(z) dz \quad (3.17)$$

$$\overline{Nu} = \frac{\bar{h}_{sp} D_h}{\bar{k}_l} \quad (3.18)$$

3.5.2 Flow Boiling Data Reduction

Ideally, saturated liquid would enter the channel to purely study flow boiling inside the channel; however, it is impractical to reach perfect saturated liquid conditions. Therefore, an inlet subcooling approximately equal to 10 K was implemented. It should be clarified here that the value of T_{in} was kept at a constant 10 K below saturation, as opposed to the value of $T_{in,meas}$. This resulted in the channel possessing both a subcooled section and a saturated two-phase section as shown in Figure 3.18.

The first step in the flow boiling data reduction was to determine the length of the single-phase subcooled region as follows:

$$L_{sub} = \frac{\dot{m}C_{p,l}(T_{sat,L_{sub}} - T_{in})}{q''_w(2H_{ch} + W_{ch})} \quad (3.19)$$

where $T_{sat,L_{sub}}$ is the saturation temperature at the location where the thermodynamic vapour quality is zero. This is dependent on the pressure at this location which was calculated using Equation (3.20).

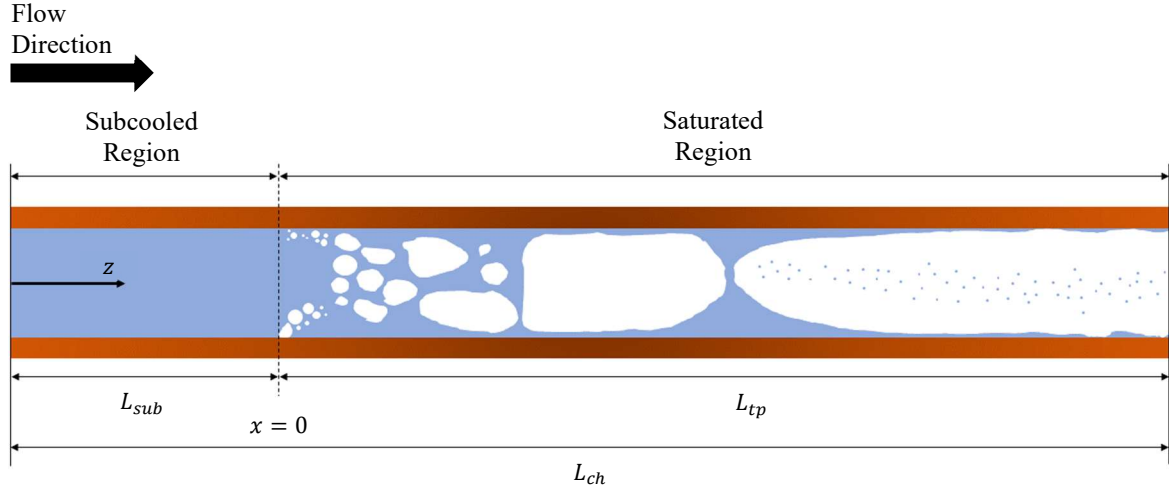


Figure 3.33. Flow boiling data reduction regions.

$$P_{sat,L_{sub}} = P_{in} - \frac{2fG_{ch}^2L_{sub}}{\rho_l D_h} \quad (3.20)$$

The friction factor used in Equation (3.20) was that proposed by Shah and London [382], which is presented in Equation (3.21). The fully developed Poiseuille number, $f_{FD}Re$, and the subcooled dimensionless length, L^* , were calculated using Equation (3.22) and (3.23), respectively. β^* is a parameter similar to the channel aspect ratio. In this case, it is the ratio of the shortest length to the largest length and always retains a value between 0 and 1. K_∞ and C are the dimensionless pressure drop number and dimensionless correction factor proposed by Shah and London, respectively. The values for K_∞ and C were obtained from a table provided by Shah and London [382] and were found to be 1.43 and 0.00029, respectively.

$$f = \frac{3.44}{Re\sqrt{L^*}} + \frac{f_{FD}Re + \frac{K_\infty}{4L^*} - \frac{3.44}{\sqrt{L^*}}}{Re\left(1 + \frac{C}{L^{*-2}}\right)} \quad (3.21)$$

$$L^* = \frac{L_{sub}}{ReD_h} \quad (3.22)$$

$$f_{FD}Re = 24(1 - 1.355\beta^* + 1.946\beta^{*2} - 1.7012\beta^{*3} + 0.9564\beta^{*4} - 0.2537\beta^{*5}) \quad (3.23)$$

Equation (3.20) is also dependent on the channel inlet pressure, which was calculated as:

$$P_{in} = P_{in,meas} - P_{in,90} - P_{in,sc} \quad (3.24)$$

The subcooled region length could then be used to calculate the saturated region length using Equation (3.25).

$$L_{tp} = L_{ch} - L_{sub} \quad (3.25)$$

The subcooled region length was then also used to calculate the two-phase pressure drop in the channel, as shown in Equation (3.26).

$$\Delta P_{tp} = \Delta P_{ch} - \frac{2fG_{ch}^2 L_{sub}}{\rho_l D_h} \quad (3.26)$$

However, the pressure losses in the outlet plenum presented in Equation (3.1) required consideration of the two-phase nature of the flow. An equation proposed by Chisholm [383] was used to account for the sudden expansion between the microchannel and the plenum as follows:

$$\Delta P_{se} = \frac{G_{ch}^2}{\rho_l} \left[\frac{1}{\beta_{pl}} \left(1 - \frac{1}{\beta_{pl}} \right) \right] \left[\frac{(1 - x_{out})^2}{1 - \alpha_{out}} + \left(\frac{\rho_l x_{out}^2}{\rho_g \alpha_{out}} \right) \right] \quad (3.27)$$

where x_{out} and α_{out} are the channel outlet vapour quality and channel outlet void fraction, respectively. The local vapour quality was calculated using Equation (3.28), where $i(z)$, $i_l(z)$ and $i_{fg}(z)$ represent the local specific enthalpy, the saturated liquid enthalpy and the enthalpy of vaporisation, respectively.

$$x(z) = \frac{i(z) - i_l(z)}{i_{fg}(z)} \quad (3.28)$$

The local specific enthalpy was found using the energy balance presented in Equation (3.29), where i_{in} is the channel inlet enthalpy. The wall heat flux was evaluated using Equation (3.12), as in the single-phase data reduction.

$$i(z) = i_{in} + \frac{q''_w(2H_{ch} + W_{ch})z}{\dot{m}} \quad (3.29)$$

The local void fraction was calculated using the local superficial liquid velocity, $U_{ls}(z)$, and the local superficial vapour velocity, $U_{gs}(z)$, from the correlation provided by Kawahara et al. below [384].

$$\alpha(z) = \frac{0.03 \sqrt{\frac{U_{gs}(z)}{U_{gs}(z) + U_{ls}(z)}}}{1 - 0.97 \sqrt{\frac{U_{gs}(z)}{U_{gs}(z) + U_{ls}(z)}}} \quad (3.30)$$

The liquid and vapour superficial velocities were calculated using Equation (3.31) and Equation (3.32), respectively.

$$U_{ls}(z) = \frac{G(1 - x(z))}{\rho_l} \quad (3.31)$$

$$U_{gs}(z) = \frac{Gx(z)}{\rho_g} \quad (3.32)$$

The pressure loss due to a change of direction of the two-phase flow in the outlet plenum was calculated using Equation (3.33) and Equation (3.34), as proposed by Chisholm [385]. B is a dimensionless constant proposed by Chisholm, r_{90} is the bend radius in the plenum and ϕ_{pl} is the diameter of the plenum.

$$\Delta P_{out,90} = \frac{G_{ch}^2 K_{90}}{2\rho_l} \left\{ 1 + \left[\left(\frac{\rho_l}{\rho_g} - 1 \right) (Bx_{out} - Bx_{out}^2 + x_{out}^2) \right] \right\} \quad (3.33)$$

$$B = 1 + \frac{2.2}{K_{90} \left(2 + \frac{r_{90}}{\phi_p} \right)} \quad (3.34)$$

The channel wall surface temperature was calculated using Equation (3.13) whilst the local bulk fluid temperature in the sub-cooled region was calculated as follows:

$$T_l(z) = T_{in} + \frac{q''_w (2H_{ch} + W_{ch})z}{\dot{m}c_{p_l}} \quad (3.35)$$

The local fluid temperature in the saturated region, $T_f(z)$, was found from the corresponding saturation temperature to the local pressure, which was calculated using Equation (3.36).

$$P_{sat}(z) = P_{sat,L_{sub}} - \left(\frac{z - L_{sub}}{L_{ch} - L_{sub}} \right) \Delta P_{tp} \quad (3.36)$$

The fluid properties required for calculating Equation (3.26) through to Equation (3.36) required that the data reduction was performed iteratively until a converged solution was found.

The local heat transfer coefficient in the single-phase region was still calculated using Equation (3.15) whilst the local two-phase heat transfer coefficient in the saturated region was calculated using Equation (3.37).

$$h_{tp}(z) = \frac{q''_w}{T_w(z) - T_f(z)} \quad (3.37)$$

Finally, the average two-phase heat transfer coefficient was calculated using Equation (3.38).

$$\overline{h_{tp}} = \frac{1}{L_{tp}} \int_0^{L_{tp}} h_{tp}(z) dz \quad (3.38)$$

The data reduction for both single-phase and two-phase flow experiments present measured and propagated experimental uncertainties. The measurement uncertainties are presented in Table 3.4, whilst the propagated uncertainties are presented in Table 3.5. The process of determining the uncertainty propagation was conducted using the methodology outlined in Coleman and Steele [386] and is presented in detail in Appendix F.

Table 3.4. Uncertainties of measured parameters.

Variable	Measuring Device	Uncertainty	Source of Uncertainty
Channel Height	CMM Machine	$\pm 1.84 - 2.16 \mu\text{m}$	Manufacturer
Channel Width	CMM Machine	$\pm 1.64 - 1.8 \mu\text{m}$	Manufacturer
Channel Length	Digital Vernier Callipers	$\pm 0.01 \text{ mm}$	Manufacturer
Copper Base Dimensions	Digital Vernier Callipers	$\pm 0.01 \text{ mm}$	Manufacturer
Mass Flow Rate	Krohne Optimas 3300C	$\pm 0.1\%$	Manufacturer (Krohne)
Differential Pressure Transducer	Omega	$\pm 0.08\%$	Manufacturer (Omega)
Inlet Pressure Transducer	Omega	$\pm 0.038 - 0.41 \text{ kPa}$	Calibration
Temperatures	K-Type Thermocouple	$\pm 0.011 - 0.039 \text{ K}$	Calibration

Table 3.5. Propagated uncertainties.

Experimental Parameter	Uncertainty
Mass flux	$\pm 0.032 - 0.57 \%$
Fanning friction factor	$\pm 2.44 - 11.61 \%$
Heat flux	$\pm 2.39 - 4.10 \%$
Average Nusselt number	$\pm 4.37 - 10.6 \%$
Channel pressure drop	$\pm 0.12 - 0.62 \%$
Local heat transfer coefficient	$\pm 3.97 - 7.54 \%$
Local vapour quality	$\pm 1.99 - 8.56 \%$
Average heat transfer coefficient	$\pm 5.31 - 9.82 \%$

3.6 EXPERIMENTAL PARAMETERS

The experimental parameters in this study varied depending on which test section was being investigated due to the purpose of the experiment and practical considerations; however, all experiments used an inlet subcooling of 10 K. This ensured that there was no nucleation in the plenum or the pipework of the channel. Channel inlet pressures of 1 bar, 1.5 bar and 2 bar were investigated for the 75 mm long channel test section, whilst pressures of 1.5 bar and 2 bar were investigated in the 30 mm long test section. The thermophysical properties of HFE-7100 at the studied channel inlet pressures are displayed in Table 3.6. The change of channel inlet pressures not only allowed for the effect of pressure on flow patterns, heat transfer coefficients and pressure drop to be evaluated, but enables the consequent effect of changing thermophysical properties on these aforementioned flow boiling aspects to be assessed.

However, a limitation of the experimental facility was the absence of an active cooling method inside the reservoir tank. This meant that the temperature inside the reservoir tank could only approach the ambient temperature as it relied solely on cooling by natural convection to the surroundings. Therefore, the pressure at the channel inlet could not be reduced any further than was permitted by the ambient temperature. The pressure drop for high heat fluxes caused a significant rise in the channel inlet pressure during attempts to complete CHF experiments at 1 bar inlet pressure using the 30 mm long channel test section. This raised the inlet pressure to over 1 bar and could not be reduced further without an active cooling method in the reservoir tank. Therefore, the experimental results for 1 bar in the 30 mm long channel test section were omitted.

Table 3.6. Thermophysical properties of HFE-7100 at the examined inlet conditions.

P_{in} [bar]	T_{sat} [°C]	ρ_l [kg/m ³]	ρ_g [kg/m ³]	i_{lg} [kJ/kg]	μ_l [μPa s]	μ_g [μPa s]	σ [N/m]
1	60.67	1420	9.58	115.7	393.7	19.84	0.0096
1.5	73.35	1383	14.13	111.3	341.6	20.62	0.0085
2	83.08	1353	18.66	107.8	312.2	21.22	0.0077

Mass fluxes of 100 kg/m²s, 200 kg/m²s, 300 kg/m²s, 400 kg/m²s, 500 kg/m²s, 750 kg/m²s and 1000 kg/m²s were studied during experiments using the 75 mm long channel test section. On the other hand, the 100 kg/m²s and 1000 kg/m²s mass fluxes were not investigated in the 30 mm long channel test section for two different reasons. The lower pressure drop that occurred in the channel for the shorter test section resulted in the pump speed having to be significantly reduced. This resulted in the pump becoming unstable and a steady mass flux could not be achieved. Conversely, 1000 kg/m²s could not be investigated near CHF conditions once again due to the high pressure drop. As mentioned above, this meant that constant inlet conditions could not be achieved without the use of some form of active reservoir cooling.

Wall heat fluxes between 0 kW/m² and 435 kW/m² were investigated for both test sections. This correlated to either achieving a vapour quality of 1, an inability to maintain the experimental channel inlet pressure or CHF being achieved. The experimental operating conditions for both test sections are presented in Table 3.7.

Table 3.7. Experimental parameters.

Parameter	75 mm long channel test section	30 mm long channel test section	Overall
Channel inlet pressure (bara)	1 - 2	1.5 - 2	1 - 2
Inlet subcooling (K)	10	10	10
Mass flux (kg/m ² s)	100 - 1000	200 - 750	100 - 1000
Wall heat flux (kW/m ² s)	0 - 204.74	0 - 435	0 - 435
Exit vapour quality (-)	0-1	0 - 0.71	0 - 1

3.7 INSTRUMENTATION CALIBRATION PROCEDURES

Experimental data possesses an associated uncertainty that arises due to an accumulation of both bias and random errors present during data acquisition [386]. The bias errors, also designated the systematic errors, arise due to errors in measurement instrumentation and can be decreased through instrument calibration. This subsection outlines the procedures that were undertaken to calibrate the instrumentation that was used in the facility. Measurements taken during experimentation included the

reservoir pressure, reservoir temperature, mass flow rate, inlet plenum absolute pressure, plenum differential pressure, plenum temperatures and test section temperatures. These measurements required multiple thermocouples, two absolute pressure transducers, a differential pressure transducer and a Coriolis flow meter. The differential pressure transducer and Coriolis flow meters were calibrated by the manufacturers whilst all thermocouples and absolute pressure transducers were calibrated using the procedures in this section.

3.7.1 Thermocouple Calibration Procedure

Figure 3.34. shows the apparatus used to calibrate all thermocouples used for experimentation. A model T100-250 platinum resistance thermometer probe with a length of 250 mm and diameter of 6 mm was connected to an ASL F250 MK II precision thermometer. The probe was then placed into a stainless-steel calibration bath that was filled with sunflower oil. Sunflower oil was selected for this purpose to permit reference temperatures greater than 100 °C which would be the upper limit for target temperatures if water were selected. An Omega-HCTB-3030 thermo-regulator was attached to the oil bath. The regulator consisted of a circulation pump, immersion heater and PID temperature control system. This configuration allowed the oil temperature to be controlled between 25 °C and 165 °C. The system was able to reach an input target temperature in a quasi-steady state where, the system settled to the temperature within ± 5 mK. The reference probe resolution was 0.25 mK.

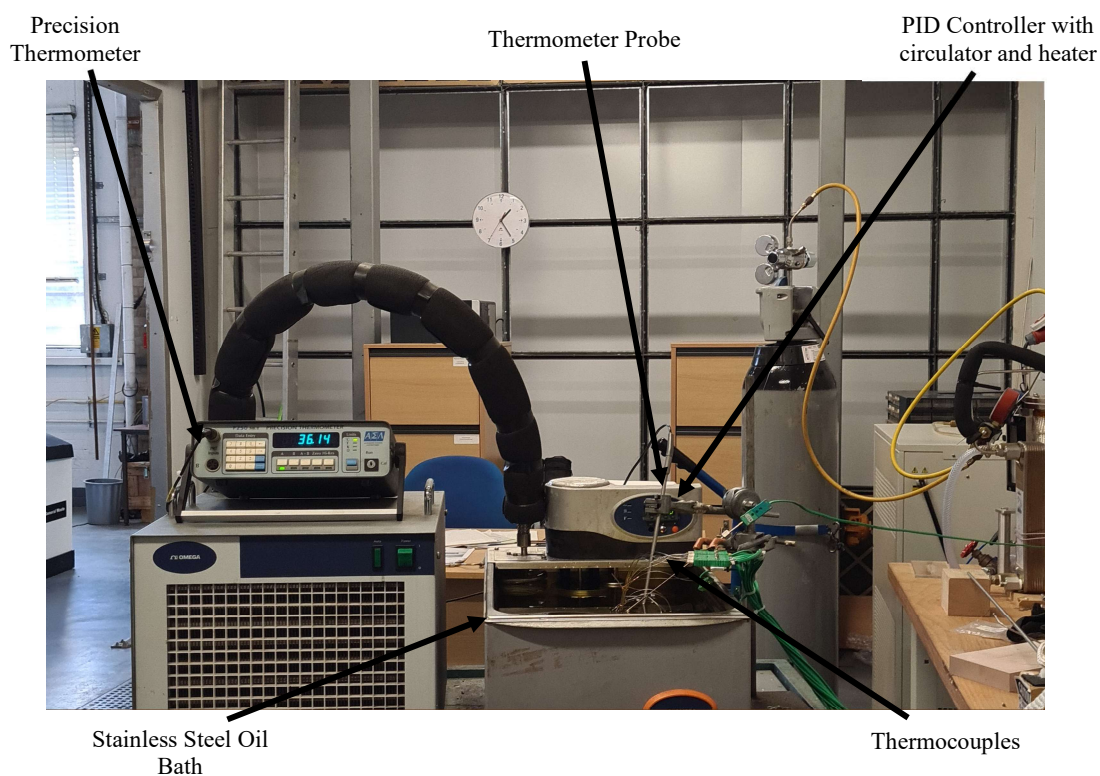


Figure 3.34. Thermocouple calibration procedure apparatus.

The thermocouple calibration process was as follows:

1. Immerse the top of the platinum probe in the circulating stainless-steel oil bath.
2. Immerse tips of thermocouples in circulating stainless-steel oil bath, ensuring the tips of the thermocouples are not in contact with each other or the steel bath itself. Locate thermocouples as close to the precision thermometer as possible.
3. Turn on precision thermometer.
4. Set initial target temperature of stainless-steel oil bath using the PID temperature controller.
5. Wait for oil bath to reach target temperature within quasi-steady state limits, i.e. settle to within ± 5 mK of the reference temperature.
6. Log precision thermometer reference values and record all thermocouple measurement values.
7. Repeat steps 5 and 6 for a range of target temperatures.
8. Repeat steps 5 to 7 a further two times for repeatability.
9. Turn off apparatus, then remove and clean thermocouples with a cloth prior to re-inserting them into the experimental facility.

The calibration lines between the precision probe reference temperature measurement and the thermocouple temperature measurements were calculated for each individual thermocouple using the Vandermonde method for polynomial interpolation. Figure 3.35 shows an example calibration curve for thermocouple T14, plotting the temperature recorded by the data-acquisition system against the reference probe temperature. The fitted line represents the calibration relationship obtained for that sensor. The resulting coefficients were implemented within the LabVIEW data-acquisition software, enabling automatic real-time correction of all thermocouple measurements. The uncertainty value of the measured temperatures were then found as standard deviations.

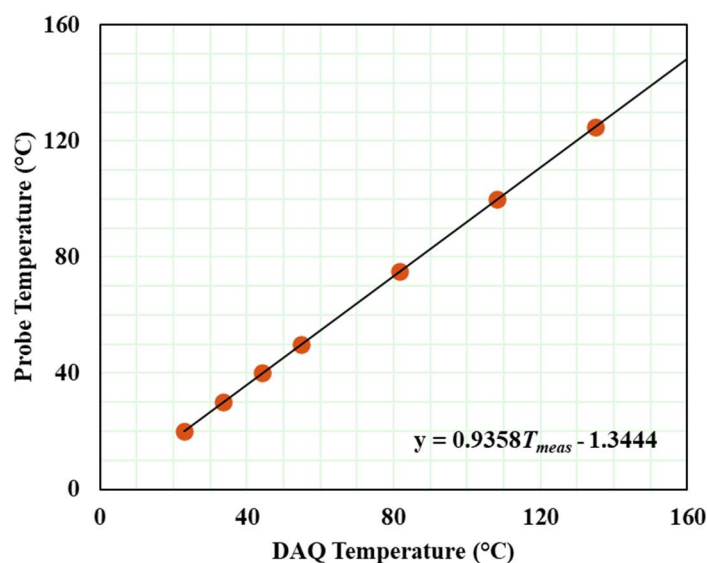


Figure 3.35. Calibration curve for thermocouple T14.

3.7.2 Absolute Pressure Transducer Calibration Procedure

The apparatus used to calibrate the absolute pressure transducers is presented in Figure 3.36. The pressure transducers were connected to a dead weight pressure gauge tester with an accuracy of $\pm 0.01\%$ (note that Figure 3.36 shows a differential pressure transducer as the apparatus was used to check the manufacturers calibration). Weights were placed on the left pin of the gauge tester that related to a given pressure acting on the pin. The pressure of the oil reservoir in the gauge tester was modified using a pressure adjustment wheel. The pin moved upwards when the pressure in the reservoir was greater than the pressure acting from the pin. This concept was used to adjust the oil pressure to match the pin pressure. The transducer then measured the pressure in the oil that corresponded to the dead weight pin pressure.

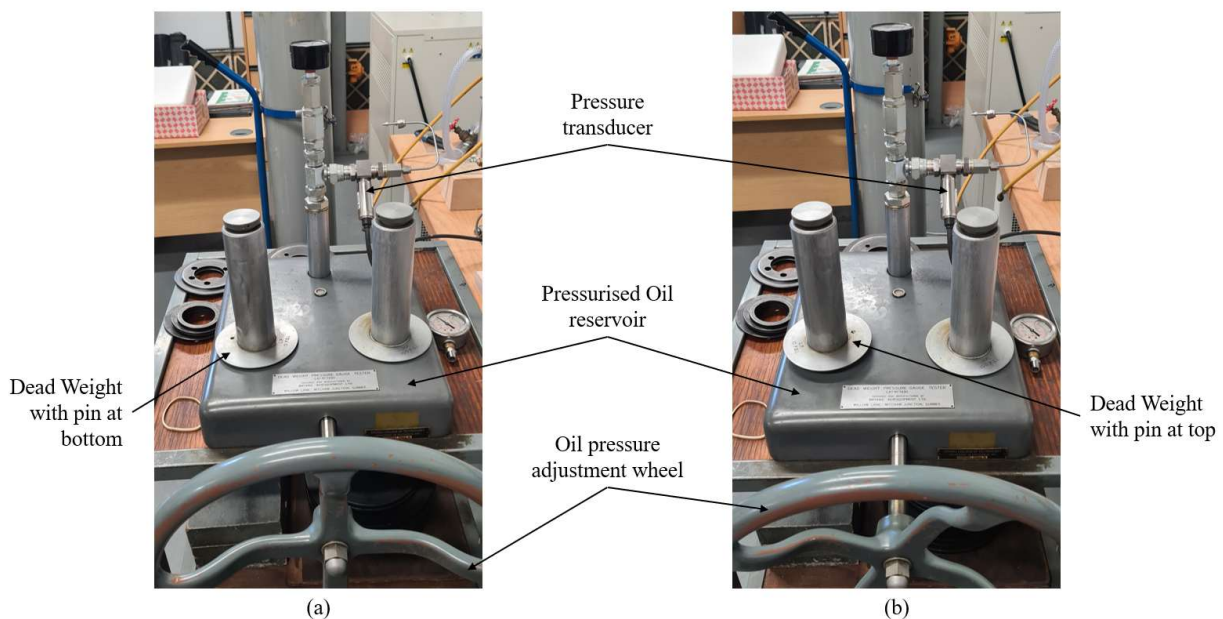


Figure 3.36. Dead weight gauge tester apparatus: (a) where oil reservoir pressure is lower than reference pressure; (b) where oil reservoir pressure is greater than reference pressure.

The process conducted to calibrate the absolute pressure transducers was as follows:

1. Connect the pressure transducer to the outlet port of the reservoir.
2. Place dead weight onto the pin of the oil reservoir.
3. Increase reservoir pressure using the oil pressure adjustment wheel, stopping when dead weight pin begins to float.
4. Measure the output voltage of the absolute pressure transducer and note corresponding oil pressure from dead weight.
5. Increased the weight and consequent pin pressure on the dead weight tester.
6. Repeat steps 3 to 5 for multiple pin pressures.
7. Repeat steps 2 to 6 a further two times for repeatability.

8. Decrease reservoir oil pressure to atmosphere and remove pressure transducer.
9. Clean pressure transducer with compressed nitrogen and a cloth before re-inserting into the experimental facility.

The calibration lines between the corresponding pin pressures from the dead weights and the absolute pressure transducer output voltages were calculated using the same method presented in Section 3.7.1

3.8 EXPERIMENTAL PROCEDURE

This subsection outlines the procedures undertaken to conduct the experiments necessary to study flow boiling in a single microchannel. The three main procedures include the degassing process, the single-phase validation experimental procedure and the flow boiling experimental procedure.

3.8.1 System Degassing Procedure

The degassing process is an essential pre-requisite to conducting both single-phase and flow boiling microchannel experiments. The presence of any dissolved gasses in the liquid phase may significantly affect heat transfer and pressure drop results [387]. Furthermore, the data reduction presented in Section 3.6 requires that the thermophysical properties of the working fluid are known. Dissolved gasses in the working fluid may alter overall fluid thermophysical properties and render the data reduction invalid. The degassing process attempts to purge the working fluid of all dissolved gasses.

The degassing process was as followed:

1. Open all isolation valves in the main closed flow loop and ensure that the globe valve under the reservoir tank is fully open.
2. Turn on the gear pump and set mass flow rate to $500 \text{ kg/m}^2\text{s}$ in the channel.
3. Turn on chiller system and preheater. Adjust the reservoir heat and preheat to achieve a channel inlet pressure of 1 bar and subcooling of 10 K.
4. Turn on the test section cartridge heaters to achieve an outlet thermodynamic vapour quality of approximately 0.8.
5. Run facility for one hour at steady state conditions.
6. Turn off test section heat and increase the mass flow rate to $5000 \text{ kg/m}^2\text{s}$ in the channel. Run for a further 10 minutes.
7. Close all valves in the main closed flow loop.
8. Attach vacuum pump to the valve on top of the reservoir.
9. Open the reservoir top valve and turn on the vacuum pump for 90 seconds.

10. Check the reservoir temperature and pressure measurements and compare to the saturation curve for the working fluid. Degassing process is complete if the values lie on the saturation curve. Repeat degassing process if the values do not lie on the saturation curve.

The procedure above was designed to degas the entire facility with a focus on both the reservoir and the test section. The copper test section was particularly prone to taking in trapped gasses that are released with changes in temperature. Conducting flow boiling in the test section ensured that trapped gasses were released and they would not be released during flow boiling experiments. The mass flow rate was subsequently increased to force any trapped gasses into the reservoir, where the working fluid would sit at the bottom of the tank and trapped gasses would rest at the top. Some trapped gasses would then re-dissolve into the liquid; however, the violent boiling that occurred during use of the vacuum pump ensured that the dissolved gasses were purged from the system.

3.8.2 Single-Phase Validation Experimental Procedure

Single-phase validation experiments were conducted to validate and commission the experimental facility. Single-phase validation was conducted on any test section prior to their use in the experimental facility. Single-phase liquid heat transfer and pressure drop results were compared to existing, well established literature correlations to ensure that the instrumentation was working correctly and the experimental methodology was theoretically sound. Typically, single-phase validation experiments could be performed in a single day. The process of conducting single-phase validation experiments differed from the flow boiling experimental procedure and was as follows:

1. Degas the experimental facility as per the procedure presented in Section 3.8.1.
2. Turn on the power supply for the pressure transducers, Coriolis flow meter signal converter, DAQ and cartridge heaters.
3. Turn on the water-glycol chiller system.
4. Turn on the gear pump to the maximum flow rate that will be observed.
5. Switch on the test section cartridge heater to heat required to create a temperature differential of 2 K between the inlet and outlet of the channel.
6. Adjust the reservoir tank heat using the variac to reach a steady state channel inlet pressure of 1 bar.
7. Allow the system to reach steady state conditions.
8. Log data for 60 seconds.
9. Lower flow rate to next desired flow rate and allow the system to reach steady state conditions, adjusting tank heat to maintain an inlet pressure of 1 bar if necessary.
10. Repeat steps 8 and 9 for all desired observable mass fluxes.

11. Turn off the test section heat supply and increase flow rate to maximum possible flow rate. This cools the test section and ensures the temperature will not rise when the pump is stopped.
12. Turn off all equipment and close all valves.

3.8.3 Flow Boiling Experimental Procedure

The flow boiling experiments were conducted in a similar manner to the single-phase experiments; however, the control of inlet subcooling was of more importance. Furthermore, it was required that a range of both heat and mass fluxes were investigated for each studied pressure. Experiments were conducted where an individual mass flux was investigated each day for the entire range of applied heat fluxes. Experiments were concluded when either of the following three scenarios were present:

- (1) An outlet vapour quality of one was reached and single-phase flow of vapour was present at the channel outlet.
- (2) Critical heat flux was reached and a sudden and dramatic increase in the test section temperature was observed. The experiment was terminated in this case to avoid any damage to the test section.
- (3) The desired inlet pressure could no longer be reached with the mass flux being studied. This phenomenon occurred when the pressure drop in the channel was significant and caused a rise in the inlet pressure, which could not be controlled by turning off the heat to the reservoir tank.

The flow boiling experimental procedure was as follows:

1. Degas the experimental facility as per the procedure presented in Section 3.8.1.
2. Turn on the power supply for the pressure transducers, Coriolis flow meter signal converter, DAQ and cartridge heaters.
3. Turn on the water-glycol chiller system.
4. Turn on the gear pump as set to the flow rate that observed that day.
5. Turn on the heat to test section to the lowest heat flux to be studied.
6. Adjust the preheater and reservoir tank heat to achieve desired inlet temperature and pressure conditions.
7. Allow the system to reach steady state conditions.
8. Log data for 60 seconds.
9. Record flow visualisation at inlet, middle and outlet of the channel.
10. Increase the test section heat supply to achieve the next desired wall heat flux and allow system to reach steady state conditions. Adjust reservoir tank heat supply if required to maintain constant inlet pressure.
11. Repeat steps 8 to 11 until one of the three experiment termination criteria are met.
12. Turn off the test section and preheater heat and increase flow rate to maximum possible flow rate.

13. Turn off all equipment and close all valves.

3.9 ESTIMATING HEAT LOSS IN THE TEST SECTION

Accurate estimation of the heat loss in the test sections was critical to the implementation of the methodology and data reduction presented in Section 3.5. It is therefore important to outline the full procedure and data reduction undertaken to determine the heat loss in the test sections, which are both presented in this subsection.

3.9.1 Heat Loss Estimation Experimental Procedure

The heat loss coefficient was calculated through the following experimental procedure.

1. The pipe-work connecting the sight glass to the valve prior to the test section was removed, leaving the valve and the inlet connecting joint exposed.
2. The valve downstream of the test section was fully sealed.
3. A vacuum pump was connected to the exposed inlet joint of the needle valve prior to the test section.
4. The vacuum pump was turned on and pressure measurements taken from the test section inlet pressure transducer.
5. A small amount of heat was added to the test section when a rough to medium vacuum (0.1 mbar) was achieved.
6. The heat supply was gradually increased until the temperature of the top surface of the copper test section was greater than the ambient temperature and steady state conditions were reached.
7. The ambient temperature and the thermocouple temperatures were logged.
8. Steps 6 and 7 were repeated until the top surface of the copper test section reached approximately 110°C.
9. The heating supply and vacuum pump were turned off.
10. The pipework was then reconnected to the valve to complete the main flow loop circuit.
11. The pump was run for an hour to circulate through the system.

The degassing procedure was conducted as air had entered the system during connecting of the vacuum pump to the flow loop system.

3.9.2 Heat Loss Estimation Data Reduction

The results from the procedure provided the thermocouple measurements from the copper test section and the ambient temperature, taken at a distance of 60 cm from the test section. The heat flux in the base of the test section was calculated using the same methodology presented in the data reduction section. It is presented again here in Equation (3.39).

$$q''_{base} = -k_{cu} \left. \frac{dT_{col}}{dy} \right|_{y=0} \quad (3.39)$$

Only datapoints in which the heat flux difference between the columns was within 10 % and the R^2 value of the linear fit was greater than 0.9 were used. This required that additional insulation was placed round the test section so that a large enough proportion of the heat was lost through the top section and not the surrounding sides.

It was then assumed that all of the vertical heat flux left through the top plate of the test section, such that:

$$q''_{base} A_{base} = Q_{top,loss} = h_{top} A_{top} (T_{ts} - T_{\infty}) \quad (3.40)$$

where h_{top} , A_{top} , T_{ts} and T_{∞} represent the heat transfer coefficient of heat loss through the top plate, the surface area of the top plate, the temperature of the top surface of the copper test section and the temperature of the ambient surroundings.

The temperature of the top surface was calculated as follows:

$$T_{ts} = \frac{y_{top} q''_{base}}{-k_{cu}} \quad (3.41)$$

where y_{top} is the vertical distance between the top of the copper test section and the top row of thermocouples, i.e. 2 mm.

The temperature distribution for the 75 mm long channel test section is shown in Figure 3.37 for a base heat flux of 9.65 kW/m².

A simple rearrangement of Equation (3.40) then provides the calculation for the product of the heat transfer coefficient for the heat lost through the top plate and the area of the top plate. Heat loss was estimated during the experiments by recording the ambient temperature and interpolating the product of the top plate's heat transfer coefficient, its area, and the temperature difference between the copper top surface and the surroundings. The corresponding results are shown in Figure 3.38.

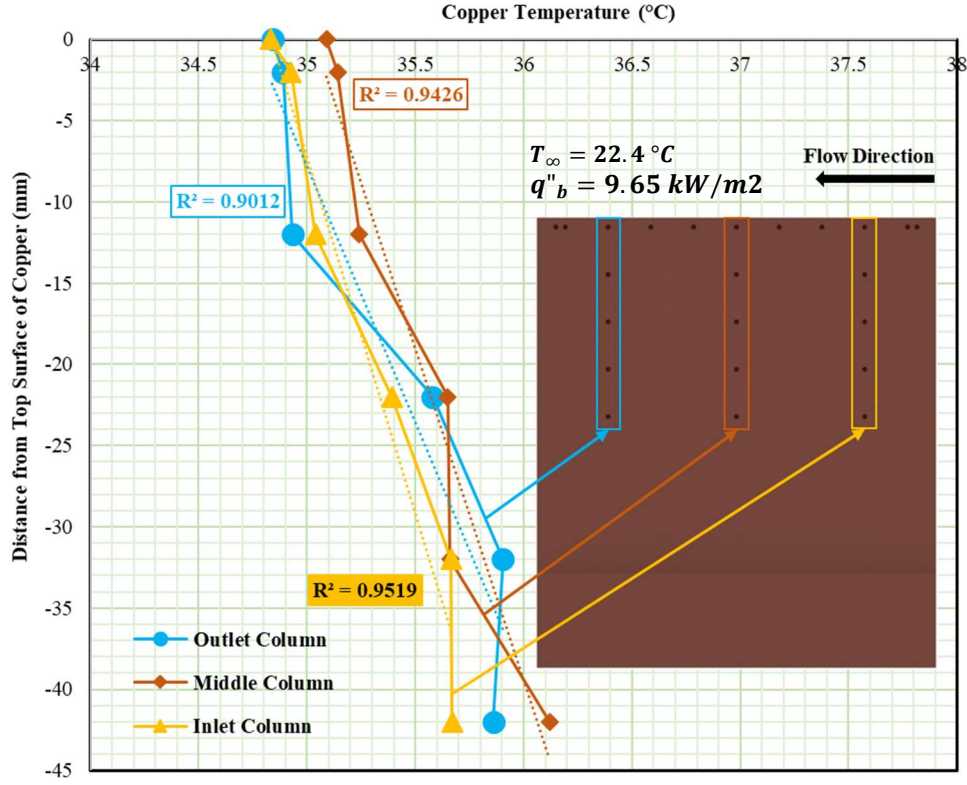


Figure 3.37. Temperature distribution in copper test section for heat loss experiment at a base heat flux of 9.65 kW/m^2 .

3.10 SINGLE-PHASE VALIDATION

Single-phase validation of the experimental facility was conducted through testing under both adiabatic and diabatic conditions. Experiments were carried out at a channel inlet pressure of 2 bar and varying mass flux levels to verify that all instruments operated correctly and maintained acceptable measurement accuracy. The experiments were conducted with an inlet fluid temperature of 20°C at input powers between 8 W and 15 W.

The experimental results were compared with existing established correlations for both the Fanning friction factor and the Nusselt number. The results were compared to the existing single-phase correlations using the mean absolute error percentage (MAE%) formula shown in Equation (3.42).

$$\text{MAE}\% = \frac{1}{n} \sum \left| \frac{X_{\text{exp}} - X_{\text{pred}}}{X_{\text{exp}}} \right| \times 100\% \quad (3.42)$$

Validation was carried out for both test sections to verify that the methodology appropriately accounts for the pressure drop external to the channel and accurately quantifies the heat losses through the top plates. The adiabatic experimental pressure drop for the 75 mm long channel test section are presented in Figure 3.39, which shows the Fanning friction factor plotted against Reynolds number.

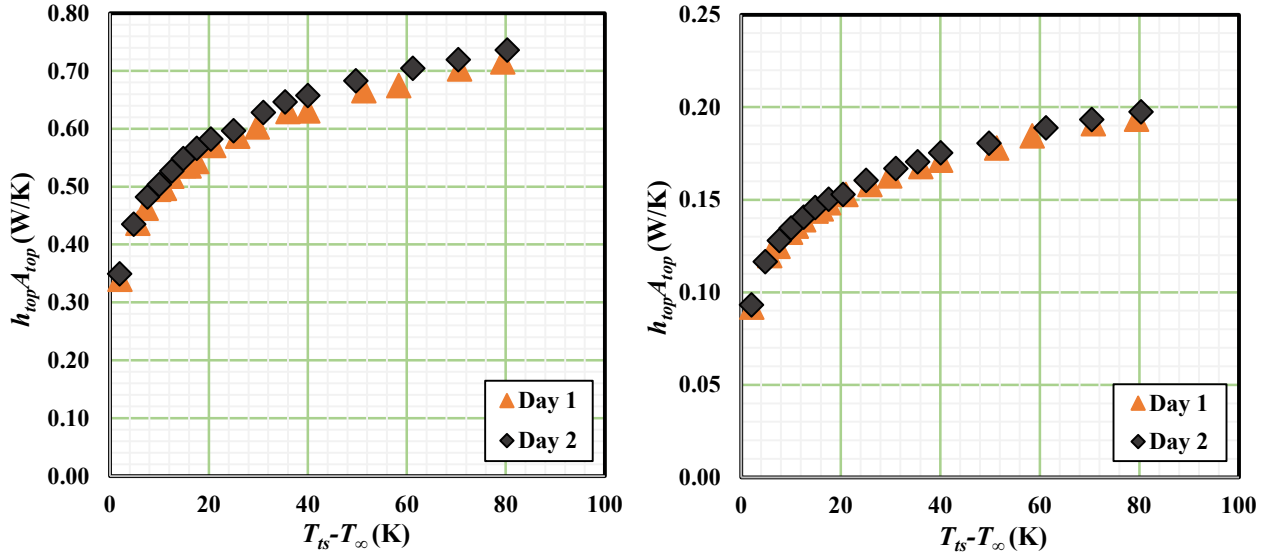


Figure 3.38. Top plate overall heat transfer coefficients for (left) the 75 mm long channel test section and (right) the 30 mm long test section.

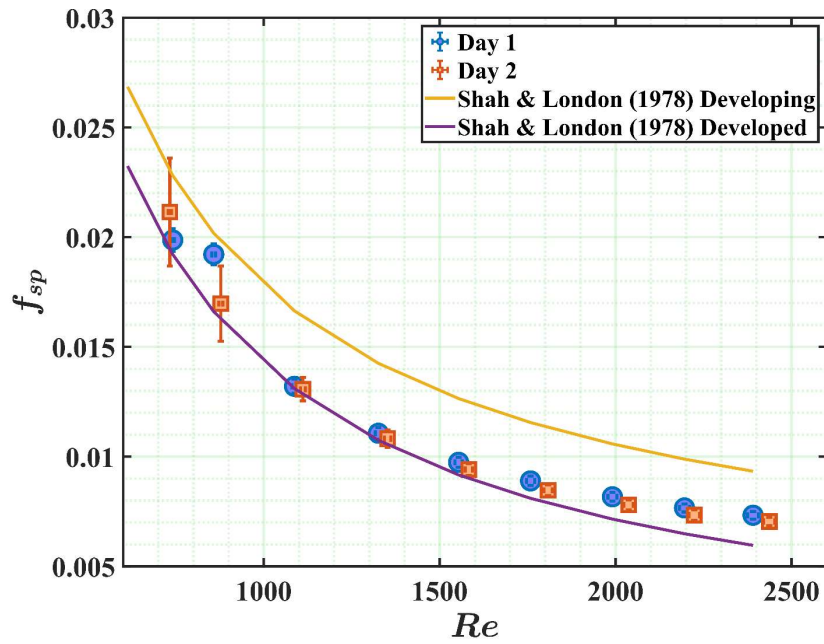


Figure 3.39. Single-phase Fanning friction factor vs Reynolds number for 75 mm long channel test section.

The adiabatic single-phase friction factor was evaluated against the correlations proposed by Shah and London [382] for both developing and fully developed laminar flow conditions in rectangular channels. The correlation for developed flow in the laminar region is presented in Equation (3.43) and the correlation for developing flows is shown in Equation (3.44) to Equation (3.45), which is dependent on the correlation for developed flows.

$$f_{sp} = \frac{3.44}{Re\sqrt{L^*}} + \frac{f_{FD}Re + \frac{K_{\infty}}{4L^*} - \frac{3.44}{\sqrt{L^*}}}{Re\left(1 + \frac{C}{L^{*-2}}\right)} \quad (3.43)$$

$$L^* = \frac{L_{sub}}{ReD_h} \quad (3.44)$$

$$f_{FD}Re = 24(1 - 1.355\beta^* + 1.946\beta^{*2} - 1.7012\beta^{*3} + 0.9564\beta^{*4} - 0.2537\beta^{*5}) \quad (3.45)$$

The experimental data showed better alignment with the correlation for developing laminar flow; however, the results generally fell between the predictions for fully developed and developing flow regimes. The hydrodynamic entry length in rectangular ducts for laminar flow can be estimated using Equation (3.46) [388]:

$$L_{ent} \approx 0.05 \cdot Re \cdot D_h \quad (3.46)$$

Based on this relation, approximately 63% of the channel length corresponds to fully developed flow at the lowest Reynolds number examined ($Re = 480$), while only 38% is developed at $Re = 1200$. At $Re = 1920$, the entire channel lies within the hydrodynamic development region. Consequently, it is reasonable to observe that the data increasingly aligns with the developing flow correlation as the Reynolds number rises, given that a larger portion of the channel is dominated by developing flow behaviour at higher Reynolds numbers. The MAE% between the experimental results and the correlations for developing and fully developed flow were 25.81 % and 8.36 %, respectively.

Figure 3.40 presents a comparison between the experimentally determined single-phase Nusselt number as a function of Reynolds number and the correlations proposed by Jiang et al. [389] and Lee and Garimella [390]. The correlation proposed by Jiang et al. [389] is presented in Equation (3.47) to Equation (3.49).

$$Nu = 0.52L^{*-0.62}, \text{ for } L^{*-0.62} < 0.05 \quad (3.47)$$

$$Nu = 2.02L^{*-0.32}, \text{ for } L^{*-0.32} > 0.05 \quad (3.48)$$

$$L^* = \frac{L_{ch}}{Re \cdot Pr \cdot D_h} \quad (3.49)$$

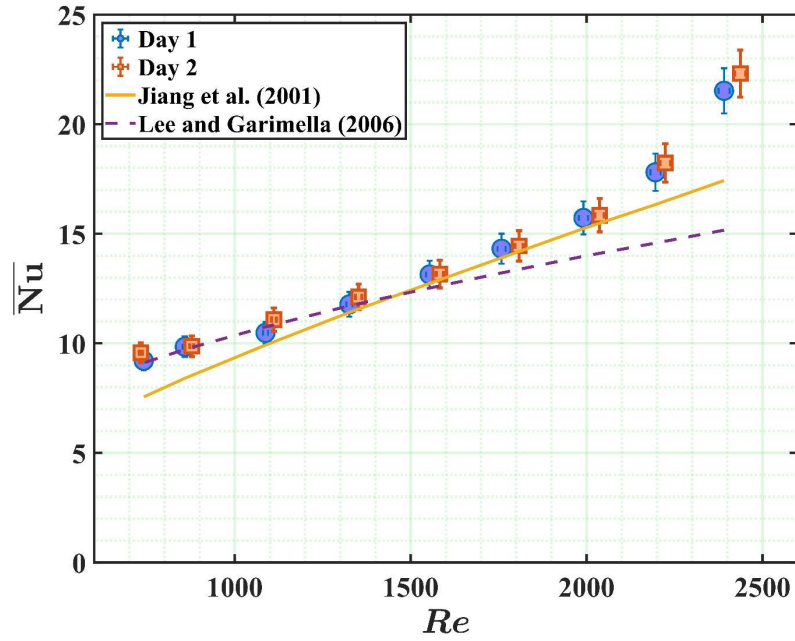


Figure 3.40. Nusselt number vs Reynolds number for 75 mm long channel test section.

The correlation for Nusselt number proposed by Lee and Garimella is shown in Equation (3.50). The experimental results showed strong agreement with the correlations proposed by Jiang et al. and Lee and Garimella, with MAE% of 9.6 % and 9.27%, respectively.

$$Nu = 1.766L^{*-0.378}\beta^{0.1224} \quad (3.50)$$

Figure 3.41 presents the measured friction factor for the 30 mm long channel test section. At lower Reynolds numbers, the friction factor is notably higher than both the developing and fully developed flow correlations. However, as the Reynolds number increases, the friction factor trends downward and eventually fell between the two correlations. This behaviour is somewhat unexpected, given that a greater portion of the channel remains in the developing flow regime at higher Reynolds numbers, with fully developed conditions estimated to occur around $Re \approx 765$. Despite this, the experimental data demonstrated reasonable agreement with the theoretical predictions, resulting in mean absolute errors (MAE) of 22.09 % and 36.41 % for the developing and fully developed flow correlations, respectively.

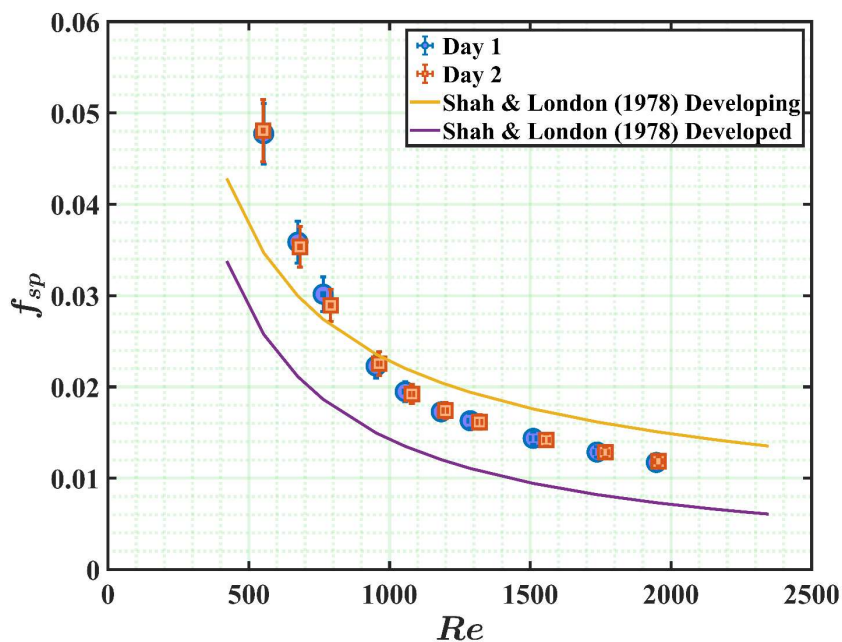


Figure 3.41. Single-phase Fanning friction factor vs Reynolds number for 30 mm long channel test section.

Figure 3.42 presents a comparison of the single-phase Nusselt number against Reynolds number for the 30 mm long channel test section, alongside the correlations proposed by Jiang et al. and Lee and Garimella. The experimental data showed good agreement with both correlations, with mean absolute errors (MAE) of 12.67% and 11.02%, respectively.

Overall, the validation confirmed that the experimental methodology was robust, the instrumentation performed as expected, and both the pressure drop external to the channel and heat losses were accurately accounted for. This provided the confidence that the two-phase flow boiling experimental results were accurate.

3.11 REPEATABILITY

The repeatability of the data was examined by conducting the same experiments over three separate days. Flow boiling experiments were conducted at 500 kg/m²s at an inlet pressure of 1 bar for the 75 mm long channel test section and at 1.5 bar for the 30 mm long test section. The repeatability results are displayed in Figure 3.43 for the 75 mm long channel test section and in Figure 3.44 for the 30 mm long test section.

Day 1 experiments for each test section were conducted prior to the flow boiling experiments used in this study. The experiments represented as Day 2 in Figures 3.43 and 3.44 correspond to the data used in this study and are presented in the subsequent chapters. Although this set of experiments

commenced on the day following Day 1, testing proceeded sequentially through the mass flux conditions for the specified channel inlet pressure. Consequently, the Day 2 run at 500 kg/m²s that is compared with Day 1 was performed between four and seven days after Day 1. Finally, the Day 3 repeatability experiment was conducted after completion of the experiments for the given channel inlet pressure and test section. This occurred between two and four days after the Day 2 experiment.

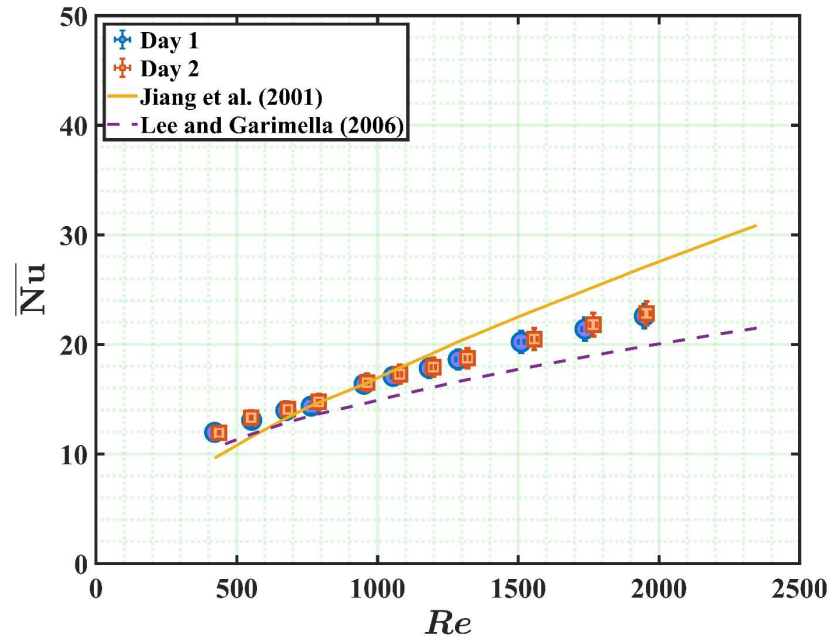


Figure 3.42. Nusselt number vs Reynolds number for 30 mm long channel test section.

The mean deviations between the Day 1 data and the utilised experimental data and between the Day 3 data and the utilised experimental data for the 75 mm long channel test section was 7.36% and 6.82%, respectively. The mean deviations between the Day 1 data and the utilised experimental data and between the Day 3 data and the utilised experimental data for the 30 mm long channel test section was 5.09% and 10.17%, respectively. This figure illustrates that the relationship between the wall heat flux and the temperature difference remained consistent across all three test days. The overall trend was reproducible, with only minor variations observed between Day 1, Day 2, and Day 3. These deviations were small and within the bounds of experimental uncertainty, indicating that the repeatability of the measurements was reliable and fell within acceptable limits for thermal performance testing.

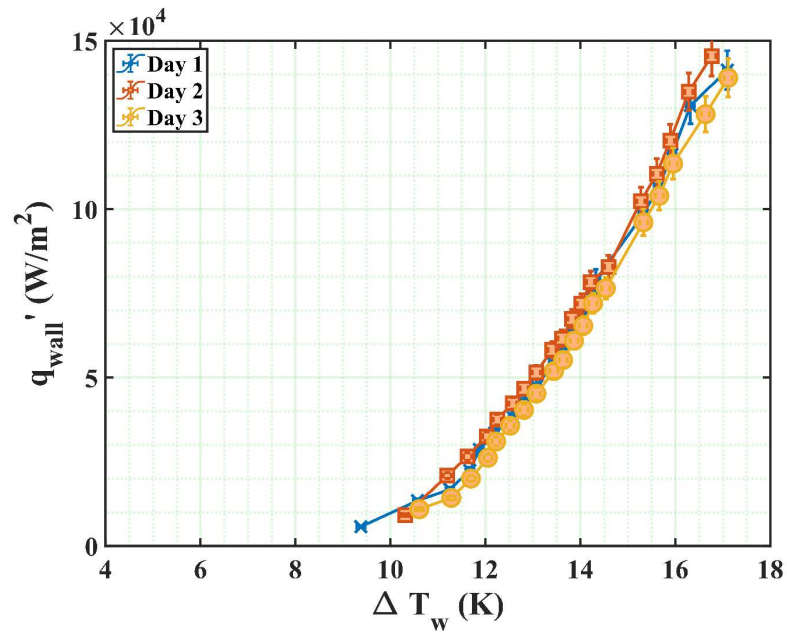


Figure 3.43. Repeatability results for 75 mm long channel test section for an inlet pressure of 1 bar and mass flux of 500 kg/m²s.

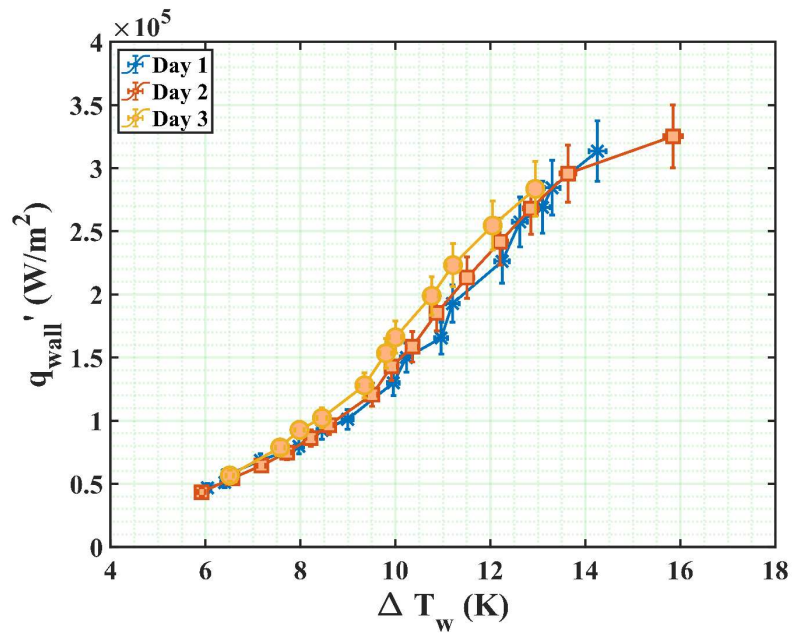


Figure 3.44. Repeatability results for 30 mm long channel test section for an inlet pressure of 1.5 bar and mass flux of 500 kg/m²s.

3.12 SUMMARY

Significant modifications were made to an existing experimental facility at Brunel University of London to enable the detailed investigation of flow boiling phenomena using HFE-7100 in a single

microchannel configuration. The development and integration of two newly designed test sections played a central role in this process. Both test sections shared the same hydraulic diameter to ensure consistent channel geometry, but differed in heated lengths (75 mm and 30 mm), to enable a systematic study of the effects of heated length on heat transfer characteristics and to investigate CHF. Further modifications were made by replacing the test section power supply, preheater and adding a gas loaded accumulator and flexi hose.

The facility instrumentation was thoroughly calibrated to ensure accurate measurements of key parameters, including pressures, temperatures and mass flow rates. Following calibration, the reliability and accuracy of the experimental setup were validated through a series of single-phase flow experiments. These included both adiabatic and diabatic tests to assess the pressure drop and heat transfer behaviour to well established correlations in the absence of phase change. The results from these validation experiments were consistent with theoretical expectations and existing literature, providing confidence in the measurement approach and system response.

In addition to validation, repeatability tests were conducted to assess the consistency of the experimental data. These tests confirmed that the system performance remained stable over multiple runs and that variations in measured quantities were within acceptable experimental error margins. This further reinforced the credibility of the experimental methodology and the robustness of the test facility.

Altogether, the successful design and implementation of the test sections, combined with rigorous calibration, validation, and repeatability testing, confirmed that the modified facility is both suitable and reliable for conducting high-fidelity flow boiling experiments with high-speed, high resolution flow visualisation. The methods established in this chapter provide a solid foundation for the subsequent experimental investigations and ensures that the research objectives, as outlined in Chapter 1, can be achieved with confidence.

CHAPTER 4

FLOW BOILING PATTERN RESULTS

4.1 INTRODUCTION

This chapter presents the flow visualisation during flow boiling experiments with increasing heat flux, from boiling incipience up to termination of the experiment, which occurred either at an outlet quality of unity, CHF or when the pressure drop became sufficient that a constant channel inlet pressure could not be achieved. All experiments were conducted with a subcooling of 10 K. The visualisation results are included in Section 4.2, where the generic flow pattern observations are presented. The flow pattern maps are exhibited in Section 4.3. with a discussion regarding the transition boundaries between flow patterns and the mechanisms affecting the transitions. The experimental flow pattern results are then compared to existing flow pattern boundaries in Section 4.4 before final conclusions are presented in Section 4.5

4.2 FLOW VISUALISATION

Flow visualisation was conducted at three locations for each test section using the Phantom Mirolab 110 high-speed camera and microscope. The visualisation locations for each test section were over the second thermocouple location, the fifth thermocouple location and the eighth thermocouple location (see Figure 3.21 and Figure 3.25), i.e. a visualisation of the channel inlet and outlet, as well as visualisation of the centre. These observation locations are presented in Figure 4.1.

Boiling incipience was initially found to occur at the corner of the channels with isolated bubbles in the subcooled region, as shown in Figure 4.2. The figure shows a single nucleation location where a fresh bubble is growing from the site at $t = 0$ ms. It should be noted that the first nucleation site appeared here by happenstance and regularly changed with each experiment. The bubble can be seen to have grown in size at $t = 0.161$ ms before departing in the image shown at $t = 0.333$ ms, where a fresh bubble is seen to occupy the nucleation site.

In the subcooled region, bubbles initially appeared along the channel corners and continued to expand while attached to the heated wall. Once detached, however, the absence of the superheated wall layer prevented further evaporation, and the bubbles began to condense as they entered the cooler bulk

liquid. This process is illustrated in the sequence of images in Figure 4.3. Bubble A, for instance, becomes anchored to the wall at $t = 0.81$ ms and grows steadily until the final recorded frame at $t = 9.516$ ms. By contrast, Bubble B emerges at $t = 0.81$ ms, detaches from the wall, and is carried downstream by the subcooled liquid flow. As it moves away, condensation dominates, causing the bubble to shrink progressively until the last observed frame at $t = 9.516$ ms.

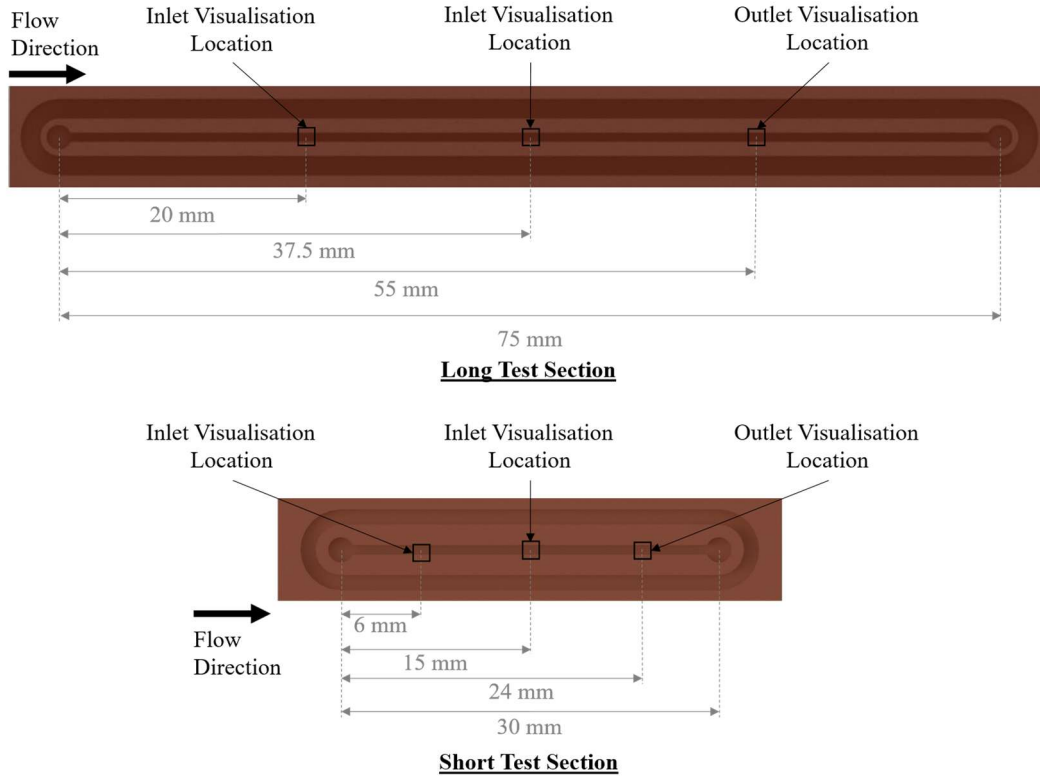


Figure 4.1. Flow visualisation locations.

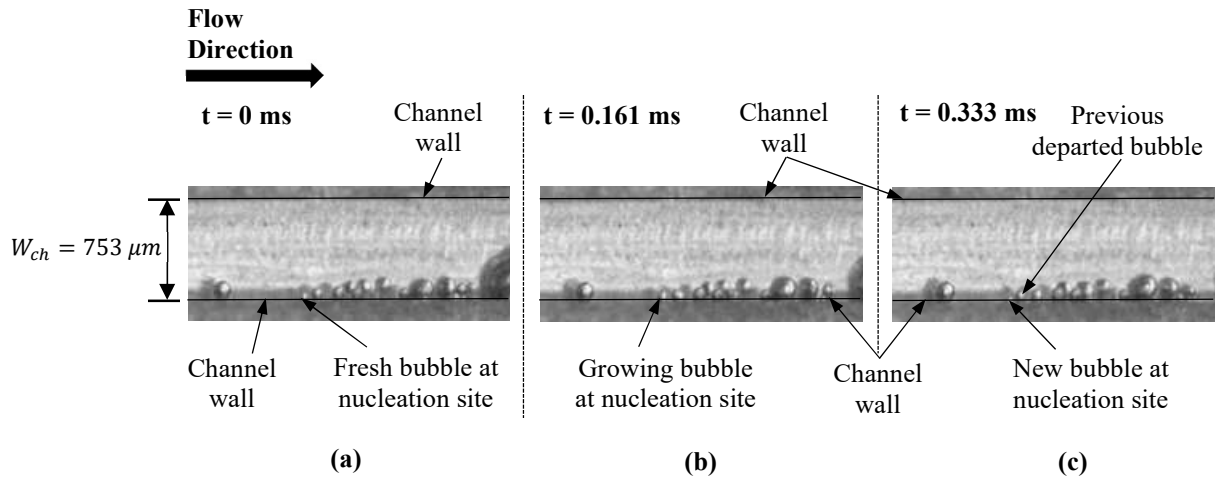


Figure 4.2. Onset of bubble nucleation at channel walls at the middle viewing location ($P_{in} = 1.5$ bar, $G = 750$ kg/m²s, $q''_w = 8.0593$ kW/m², $x_{out} = -0.0787$).

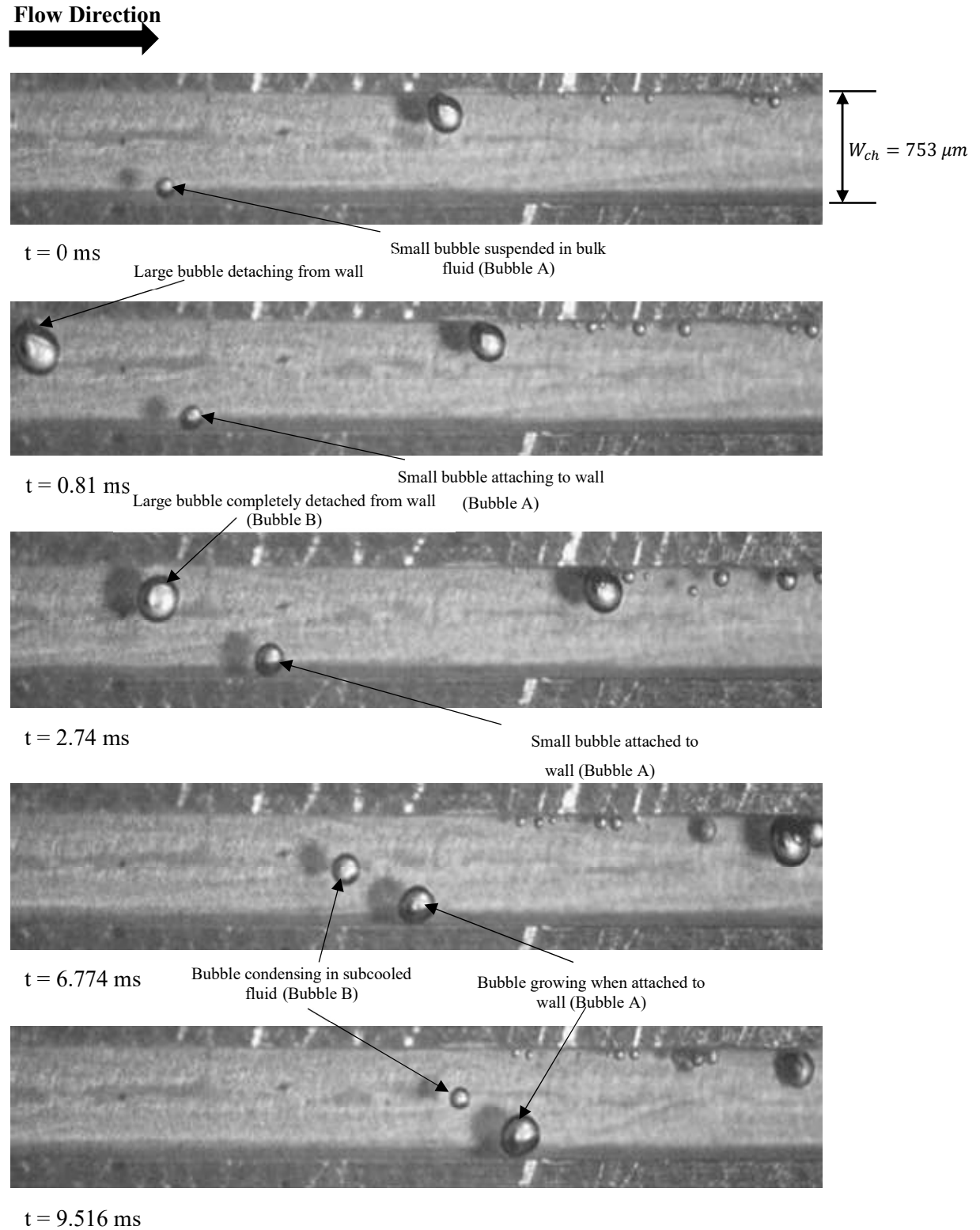


Figure 4.3. Bubble nucleation and departure at the channel outlet in the subcooled region ($P_{in} = 1.5$ bar, $G = 500 \text{ kg/m}^2\text{s}$, $q''_w = 19.79 \text{ kW/m}^2$, $x_{out} = -0.0072$).

Flow visualisation identified four fundamental flow regimes: bubbly, slug, churn, and annular, as per the review on existing literature presented in Chapter 2. These regimes are illustrated in Figure 4.4 for

a mass flux of $400 \text{ kg/m}^2\text{s}$. In the bubbly regime, observed in the saturated region, vapour bubbles continued to expand despite not being in direct contact with the channel walls. This growth was likely driven by interaction between the bubble interface and a superheated liquid film along the channel wall. As the bubbles travelled downstream, the film evaporated into the bubbles, promoting their enlargement. Since the bulk liquid was at saturation conditions, condensation did not occur, and the bubbles grew until they coalesced into larger, elongated bubbles characteristic of slug flow, shown in Figure 4.4(b).

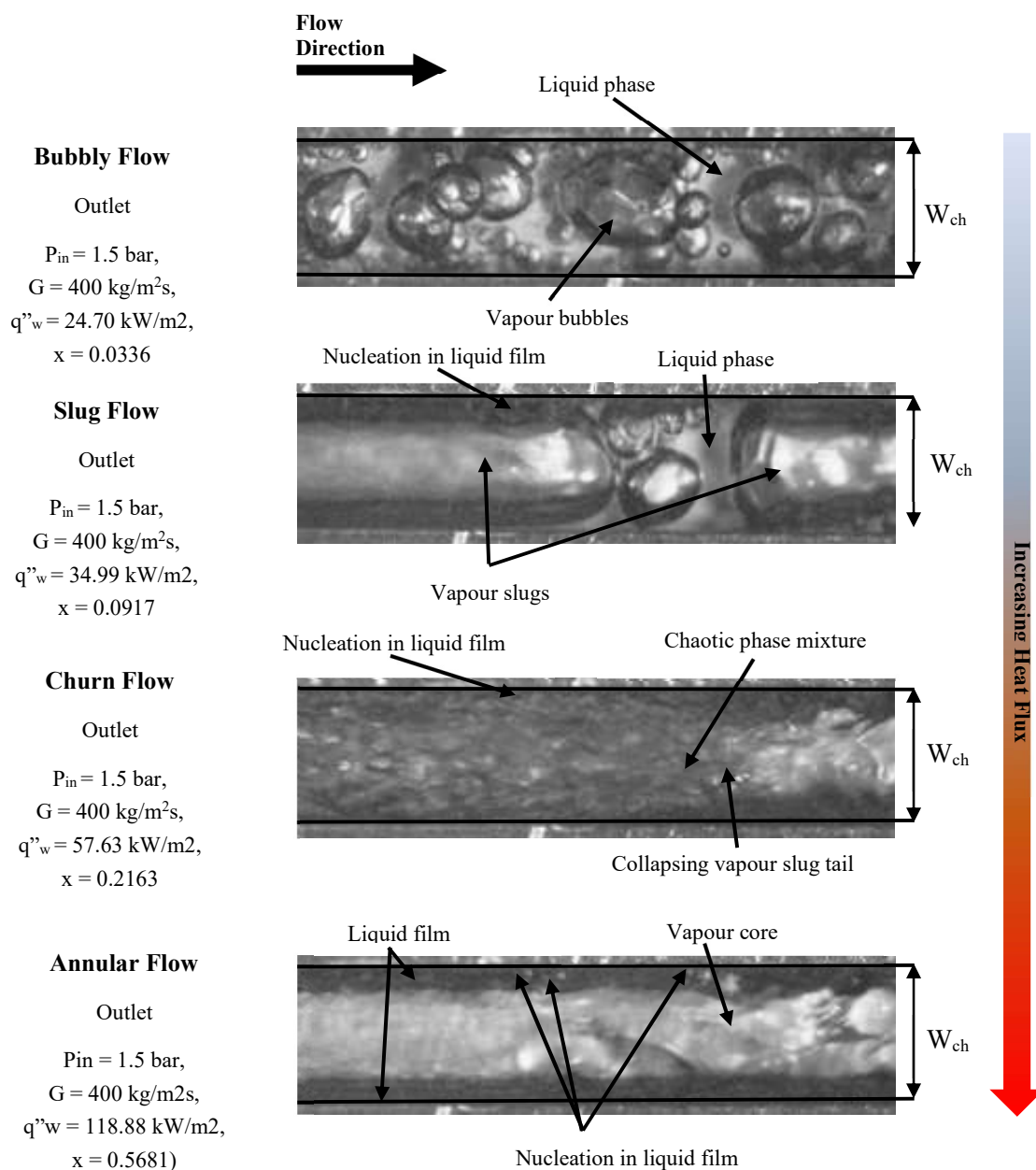


Figure 4.4. Observed flow patterns for the 75 mm long channel in the saturated region with generic classifications.

Further bubble nucleation still occurred within the liquid film during slug flow. This behaviour was linked to the rectangular channel geometry, which produced a thicker liquid film at the channel corners, as illustrated in Figure 4.5. Even in slug flow, occasional dry patches appeared on the heated surface. As the confined vapour bubbles enlarged, interfacial instability developed, giving rise to a chaotic churn flow regime depicted in Figure 4.4(c).

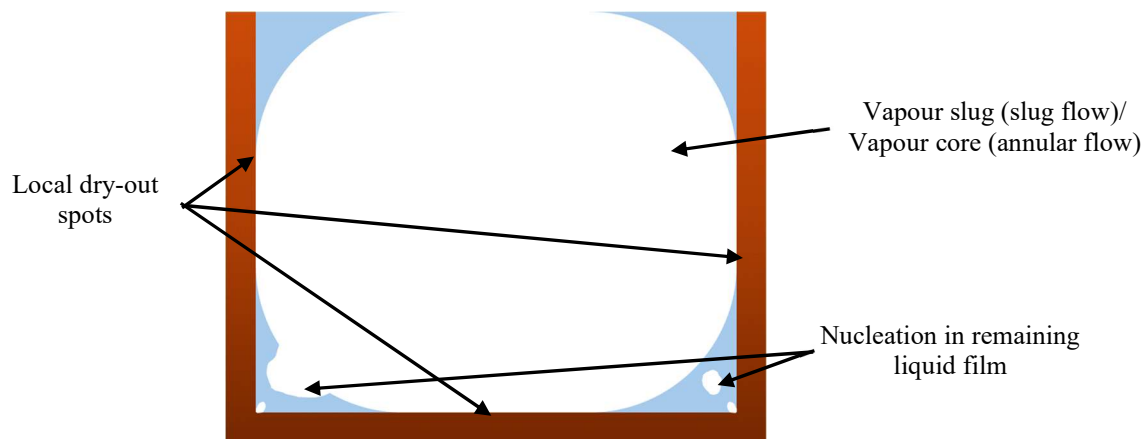


Figure 4.5. Cross section of channel with simultaneous dry spots and bubble nucleation.

The reorganisation of phases within churn flow led to an unsteady but recurring annular structure, shown in Figure 4.4(d). Here, a thin liquid film encased a central vapour core. Over time, thinning of the liquid film caused intermittent dry-out spots. At high vapour qualities near the channel exit, blockage of the outlet plenum occurred, producing compressibility effects that elevated the channel inlet pressure. Once this pressure was relieved, upstream churn flow rewetted the walls, restoring the annular configuration. Immediately after rewetting, the liquid film was thick enough to permit bubble nucleation in the channel corners, with microbubbles detaching into the vapour core. As the film thinned again and dry spots reappeared, nucleation diminished. Nevertheless, it is worth noting that some degree of bubble generation was observed across all four flow regimes.

4.3 EXPERIMENTAL FLOW PATTERN MAPS

As discussed in Section 2.3.3, flow patterns are typically presented in literature using flow pattern maps, which may use a range of coordinate systems. The reviewed literature includes coordinate systems that use the phase superficial velocities, mass flux and vapour quality, a combination of the boiling number, Bond number and liquid only Reynolds number, the liquid only Weber number and the confinement number. The experimental range for the parameters used in each coordinate system are presented in Table 4.1 for both test sections.

Table 4.1. Experimental range of flow pattern map coordinate parameters.

Parameter	75 mm Long Channel Test Section	30 mm Long Channel Test Section	Units
U_{gs}	0 - 40.13	0 - 26.52	m/s
U_{ls}	0 - 0.7358	0.0402 - 0.623	m/s
G	100 - 1000	200 - 750	kg/m ² s
x	0 - 1	0 - 0.71	-
Re_{lo}	198.4 - 2501.8	22.93 - 2763.4	-
Bo	1.4×10^{-5} - 0.0074	0.0011 - 0.0075	-
Bd	0.880 - 1.041	0.940 - 1.0556	-
We_{lo}	0.5419 - 76.67	2.452 - 44.15	-
We_{go}	57.20 - 8853.8	203.33 - 3925.4	-
Co	0.98 - 1.066	0.974 - 1.035	-

The experimental flow pattern maps for the 75 mm long channel test section are presented in Figure 4.6, where mass flux and vapour quality are used as mapping coordinates. It should be noted that there often existed multiple flow patterns at a large number of flow visualisation locations. For example, the collapsing of slugs that creates churn flow meant that slug flow was intermittently observed at a particular location before churn flow began to occur. Accordingly, each data point was classified according to the flow pattern that persisted for the longest period at its respective location during steady-state experimental conditions.

Figure 4.6 illustrates the progression of flow regimes across varying mass fluxes, beginning with bubbly flow at the lowest vapour qualities for each case. As vapour quality increases, the flow undergoes sequential transitions from bubbly to slug, followed by churn, and ultimately to annular flow. A clear trend emerges whereby the transition between these regimes occurs at progressively lower vapour qualities as the mass flux increases.

Whilst Figure 4.6 presents flow pattern maps in terms of mass flux and vapour quality, separating the individual influences of mass flux and wall heat flux is difficult because vapour quality is inherently dependent on both parameters. To better relate the imposed operating conditions to the observed transitions, maps were therefore first constructed using mass flux and wall heat flux. These representations indicate the heat flux associated with each transition at a given mass flux; however, they implicitly assume that the flow pattern map is effective over the entire channel length. Under subcooled inlet conditions, such as those used in this study, flow patterns outside of subcooled bubbly flow commenced downstream of the inlet once the liquid reached saturation. Consequently, for the same wall heat flux, different mass fluxes can produce substantially different lengths of two-phase flow, with higher mass fluxes generally resulting in a longer subcooled region and a shorter saturated region. It is therefore necessary to account for the location at which saturated conditions are established as flow regime transitions are governed by the behaviour within the saturated portion of the channel. To address

this, the dimensionless saturated length, defined as $(z - L_{sub})/D_h$, was introduced to scale the operating conditions according to the saturated length.

Additional maps were also constructed using the boiling number together with the dimensionless saturated length, since the wall heat flux alone does not fully represent the rate of vapour production when the effective boiling length varies. The resulting flow pattern maps for inlet pressures of 1 bar and 2 bar are shown in Figure 4.7. Two maps present mass flux versus the wall heat flux-dimensionless saturated length parameter, and two present mass flux versus the boiling number-dimensionless saturated length parameter.

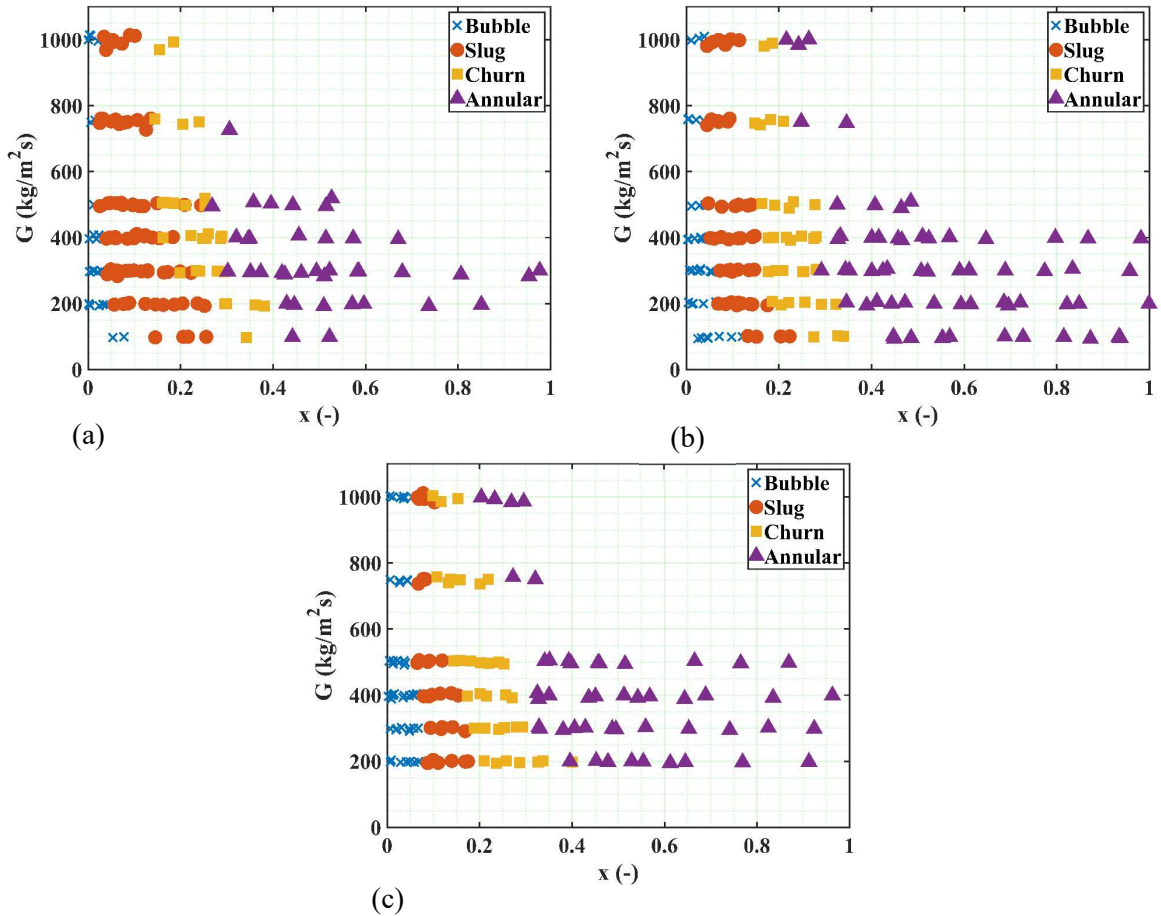


Figure 4.6. Flow pattern maps for 75 mm long test section using mass flux and vapour quality for: (a) 1 bar inlet pressure; (b) 1.5 bar inlet pressure; (c) 2 bar inlet pressure.

Figure 4.7(a) and Figure 4.7(c) demonstrate that the transition boundaries between successive flow regimes occur at progressively higher heat fluxes as the mass flux increases. This trend is expected, since at higher flow rates the liquid residence time within the heated channel is reduced, resulting in less vapour generation for a given heat input. In contrast, Figure 4.7(b) and Figure 4.7(d), which employ the boiling number as the normalising parameter, reveal that lower boiling numbers are associated with

regime transitions at higher flow rates. In other words, less vapour generation relative to the mass flow is needed to initiate these transitions under high-mass-flux conditions. Explaining this behaviour requires a closer examination of the underlying mechanisms that govern each transition

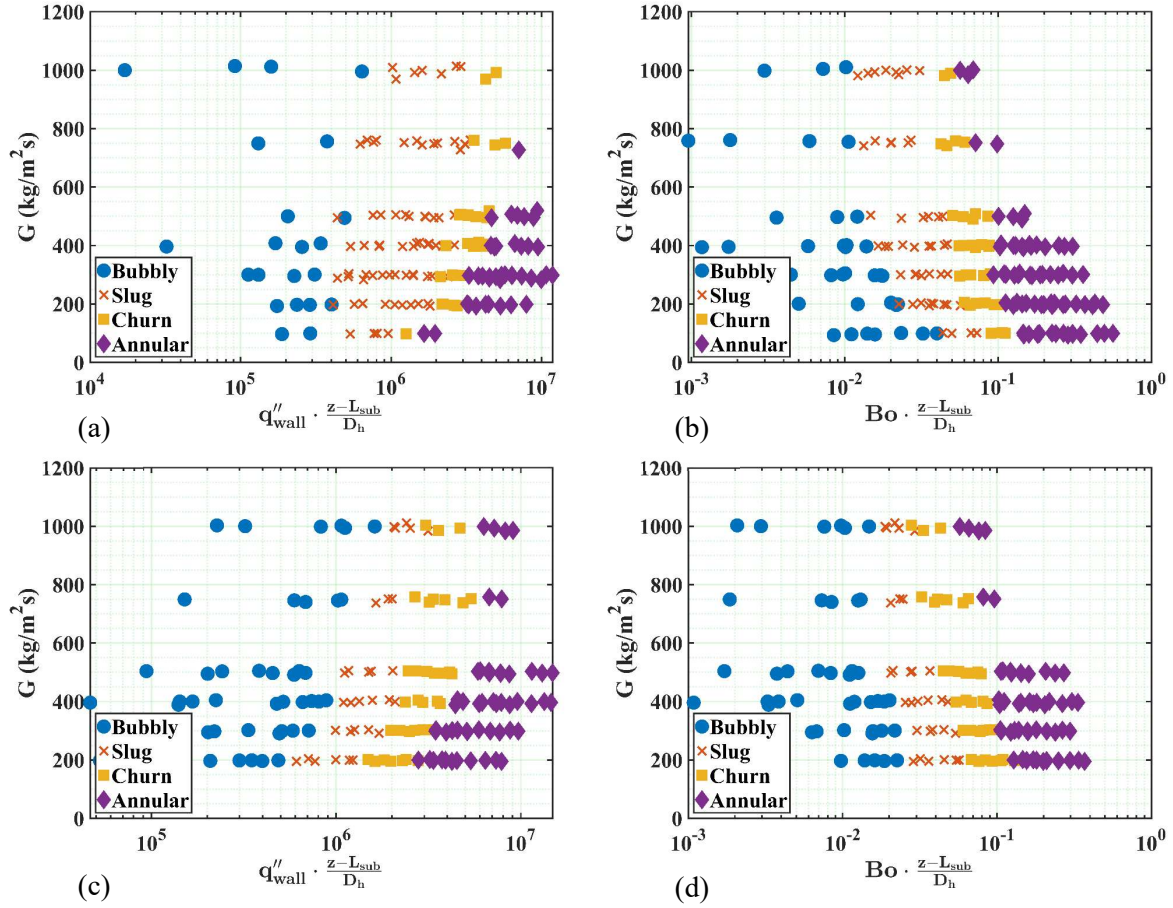


Figure 4.7. Flow pattern maps for 75 mm long test section using alternative coordinates: (a) 1 bar inlet pressure and heat flux; (b) 1 bar inlet pressure and boiling number; (c) 2 bar inlet pressure and heat flux; (d) 2 bar inlet pressure and boiling number flux.

4.3.1 Bubbly to Slug Transition

Firstly, the transition between bubble and slug flow is shown diagrammatically in Figure 4.8 for a mass flux of 400 kg/m²s. At $t = 0$ ms, the flow contains a collection of vapour bubbles of varying size. Several large bubbles, formed through earlier coalescence events, are visible alongside a dense cluster of smaller bubbles emerging from active nucleation sites on the heated wall. At this initial stage, the larger bubbles remain unconfined within the cross-section of the channel, and the flow is still categorised as bubbly. The small bubble cluster indicates continued nucleation activity from a nucleation site. At $t = 0.323$ ms, some coalescence has occurred. The previously distinct cluster of small bubbles has combined to form two larger bubbles. One of these begins to merge with a nearby large

bubble, distorting the vapour-liquid interface and altering the local flow structure. The second newly formed bubble merges with the trailing cluster, increasing its volume and approaching confinement within the channel dimensions.

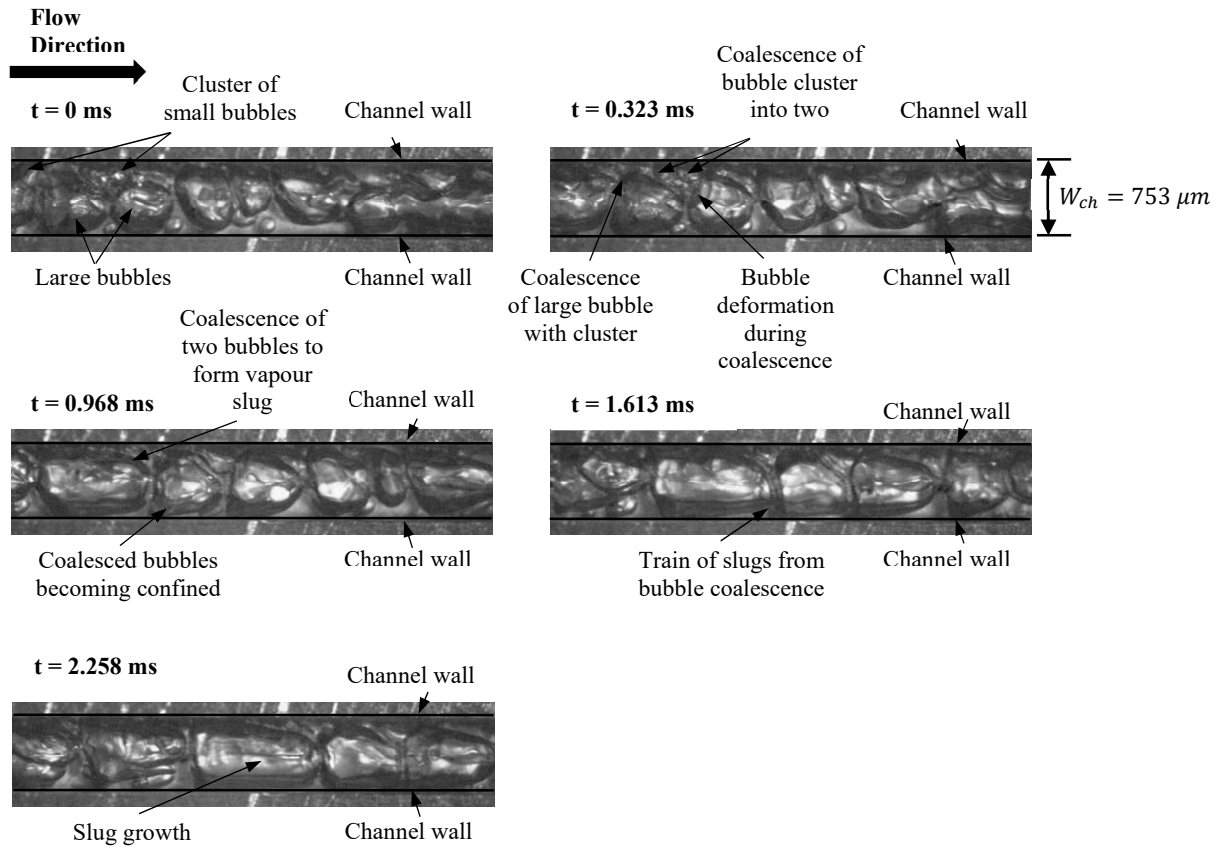


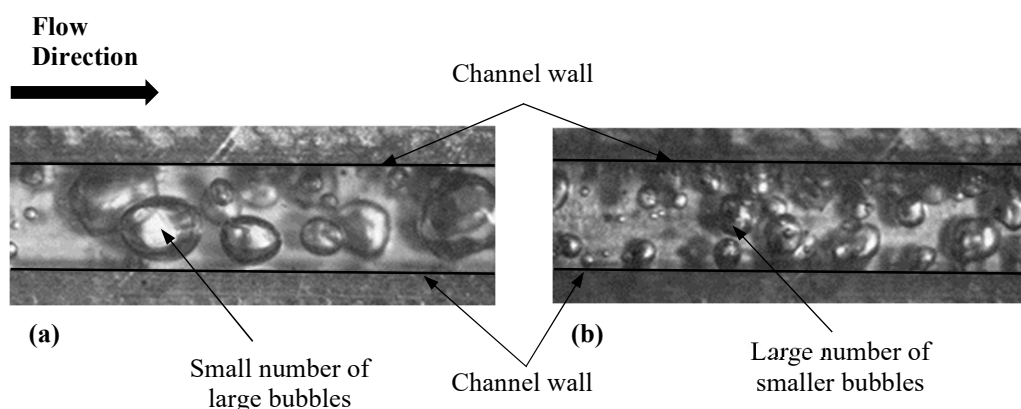
Figure 4.8. Flow visualisation of transition between bubbly flow and slug flow at outlet for $G = 400 \text{ kg/m}^2\text{s}$, $q''_{\text{wall}} = 24.79 \text{ kW/m}^2$ and $P_{\text{in}} = 1 \text{ bar}$.

At $t = 0.968 \text{ ms}$, two of the larger bubbles, already close to being confined by the channel cross-section, coalesce further to form an elongated vapour body. This marks the onset of slug formation, where the vapour begins to span a substantial portion of the channel diameter. By $t = 1.613 \text{ ms}$, the surrounding liquid film undergoes further evaporation, contributing to continued axial extension of the confined vapour slug. This development gives rise to a train of vapour slugs travelling along the channel, closely pressed together. Finally, at $t = 2.258 \text{ ms}$, the evolution of the slug train continues. The existing vapour slugs elongate further, while two smaller slugs begin to merge into a larger confined structure.

Bubble coalescence emerges as the primary mechanistic driver for overall bubble growth and the eventual confinement of vapour structures within the channel. This process governs the transition from bubbly flow to slug flow, as successive coalescence events reduce the number of discrete bubbles while increasing their size and axial dimensions. However, bubble coalescence is inherently complex and

strongly influenced by a range of interdependent factors. Such factors include bubble dimensions, surface tension [391], bubble velocity and bubble film effects [392]. The latter refer to the dynamics of the thin liquid layer that forms between approaching bubbles or between a bubble and the channel wall. The rate of drainage, rupture, and reformation of this intervening film determines whether bubbles rebound, remain temporarily separated, or merge.

Figure 4.10 presents bubbly flow visualisation for a boiling number of approximately 0.75 at the centre of the channel. Here, it can be seen that the bubbles are smaller for a mass flux of $750 \text{ kg/m}^2\text{s}$, in comparison to the bubbles at $300 \text{ kg/m}^2\text{s}$; however, there are significantly more bubbles. Typically, smaller bubbles are more stable and are less likely to coalesce, particularly under high shear or inertial forces, as they are more easily removed or suppressed before growth allows coalescence [393]. However, the greater number of bubbles and the increased velocity make bubble coalescence more likely due to the increased chance of random collision, despite the smaller bubble size. This results in the transition to slug flow to occur at higher heat fluxes for greater mass fluxes but occur at lower boiling numbers, due to the increased chance of bubble collisions at the higher phase velocities.



4.9. Bubbly flow visualisation at the middle of the 75 mm long channel test section for a boiling number of 0.75 for: (a) a mass flux of $300 \text{ kg/m}^2\text{s}$; (b) a mass flux of $750 \text{ kg/m}^2\text{s}$.

4.3.2 Slug to Churn Transition

Figure 4.9 provides a detailed sequence of images capturing the transition from slug flow to churn flow at a mass flux of $300 \text{ kg/m}^2\text{s}$ and an inlet pressure of 1 bar. The time-resolved frames allow the evolution of the flow structures to be examined closely, highlighting the processes of bubble coalescence, slug deformation, and eventual collapse into a churn regime. At the initial stage, corresponding to $t = 0 \text{ ms}$, there is an elongated vapour slug trailing behind both unconfined and confined bubbles in a train. At $t = 1.452 \text{ ms}$, the previously distinct bubble train begins to coalesce, merging into continuous vapour slug. The head of the elongated slug also becomes flattened against the merged preceding slug.

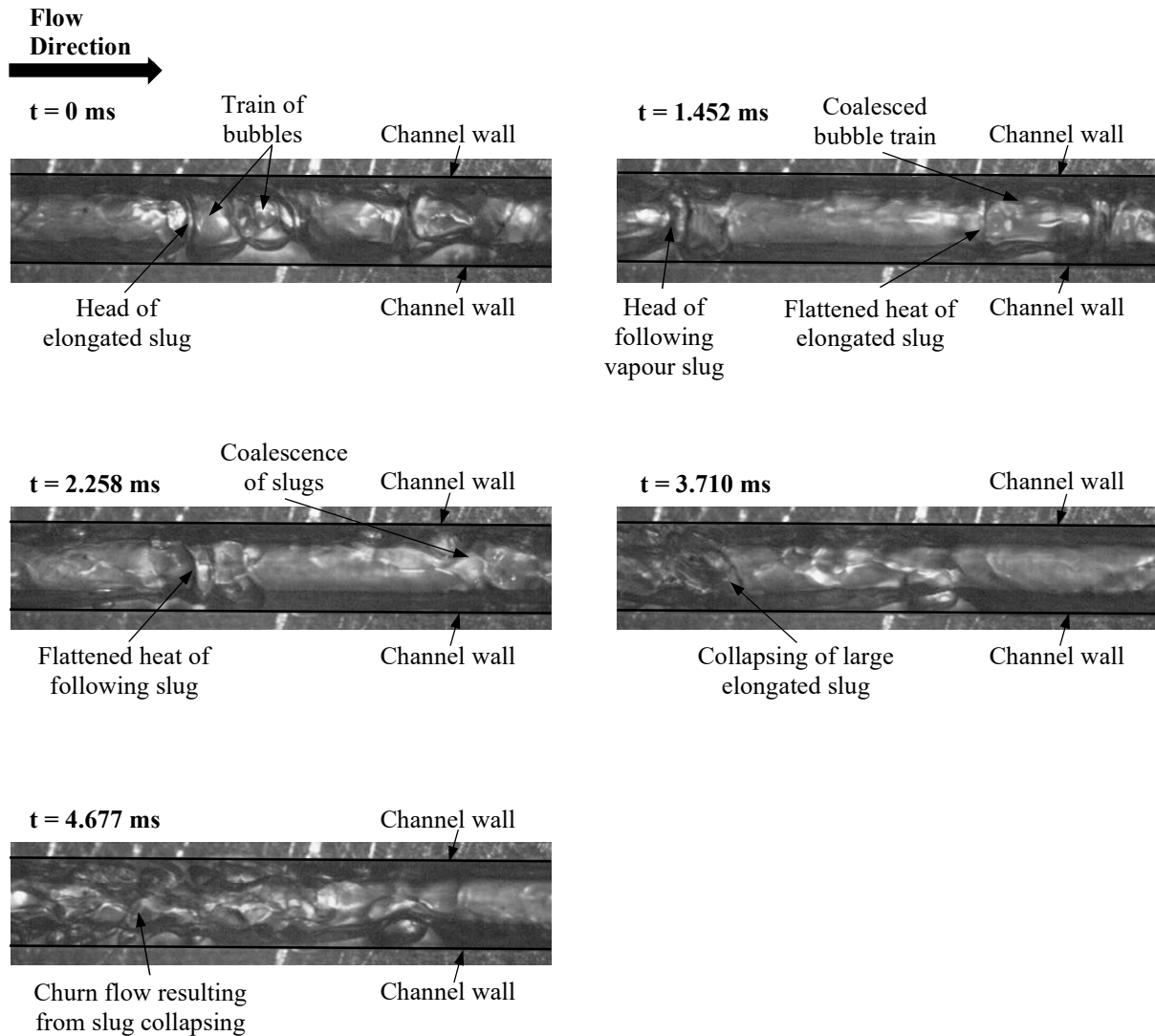


Figure 4.10. Flow visualisation of transition between slug flow and churn flow at outlet for $G = 300 \text{ kg/m}^2\text{s}$, $q''_{\text{wall}} = 40.43 \text{ kW/m}^2$ and $P_{\text{in}} = 1 \text{ bar}$.

At $t = 2.258 \text{ ms}$, the dynamics become more complex. The elongated slugs themselves begin to merge, with the tail of one slug being pressed against the flattened head of the trailing slug. This interaction between successive slugs creates a localised zone of high interfacial shear and distortion, with the phase boundary exhibiting visible disturbances that propagate along the length of the slug. Consequences of these disturbances are clearly observed by $t = 3.710 \text{ ms}$. The tail region of the slug begins to collapse under the influence of interfacial instabilities. Finally, at $t = 4.677 \text{ ms}$, the transition is complete. The collapsing slug tail generates a highly disturbed phase distribution, characterised by a chaotic mixing between the liquid and gas phases. The interface becomes indistinct, and instead of well-defined slugs or bubbles, the channel contains a disordered mixture. This state corresponds to churn flow, in which the phases are no longer separable into coherent structures.

The sequence in Figure 4.10 provides valuable insight into the physical mechanisms governing the transition from slug to churn flow. As described, the transformation is initiated by the coalescence of upstream bubble trains, followed by deformation of the slug head, slug to slug interactions, and the eventual collapse of the liquid-vapour interface. Therefore, the primary mechanism through which slug flow transitions to churn flow is through collapsing of the liquid-vapour interface. The onset of churn flow is shifted to higher values of the wall heat flux-dimensionless saturated length parameter at greater mass fluxes, as shown in Figure 4.7. This indicates that a larger effective vapour development within the saturated region is required to destabilise the slug structures when the mass flux is increased. At elevated mass flux, the fluid residence time within the channel is reduced and the subcooled region is extended, so a greater overall heat input is needed before sufficient vapour is generated within the available two-phase length to initiate interfacial collapse. However, when the boiling number is combined with the dimensionless saturated length, the transition boundaries shift to lower values of this parameter at higher mass fluxes. This suggests that, relative to the mass flow, a smaller amount of vapour generation within the saturated region is sufficient to trigger the transition under high mass flux conditions.

This behaviour can be interpreted using the scaled force analysis proposed by Kandlikar [79], presented in Section 2.2. Higher mass flux increases the inertial and shear forces acting on the vapour-liquid interface, which promote interfacial instability. At the same time, the evaporation momentum force, associated with phase change, drives axial growth of the vapour slugs and is not directly dependent on the mass flux when expressed per unit area. As a result, at higher mass fluxes, the increased destructive inertial and shear forces reduce the amount of vapour development within the saturated region required to destabilise the slug interface and produce churn flow.

4.3.3 Churn to Annular Transition

The transition between churn and annular flow at a mass flux of $500 \text{ kg/m}^2\text{s}$ is illustrated in Figure 4.11. At $t = 0 \text{ ms}$, churn flow is observed propagating through the channel, with remnants of collapsed vapour slugs visible upstream. Observed at $t = 0.161 \text{ ms}$, the churn front has almost exited the channel, and the initial formation of a vapour core is evident behind it, with the vapour phase beginning to push the liquid phase towards the channel walls. At $t = 0.645 \text{ ms}$, the churn region has fully left the channel, leaving a distinct vapour core. At this stage, the liquid film surrounding the core is highly disturbed, with pronounced interfacial waves and nucleation activity within the film. Between $t = 1.613 \text{ ms}$ and $t = 3.70 \text{ ms}$, the film progressively stabilises, becoming more uniform and representative of stabilised annular flow, although nucleation still persists within the developed liquid film. It should be noted that annular flow was frequently disrupted by the intermittent passage of churn flow through the channel.

Following these disturbances, annular flow would reform through the mechanism illustrated in Figure 4.11.

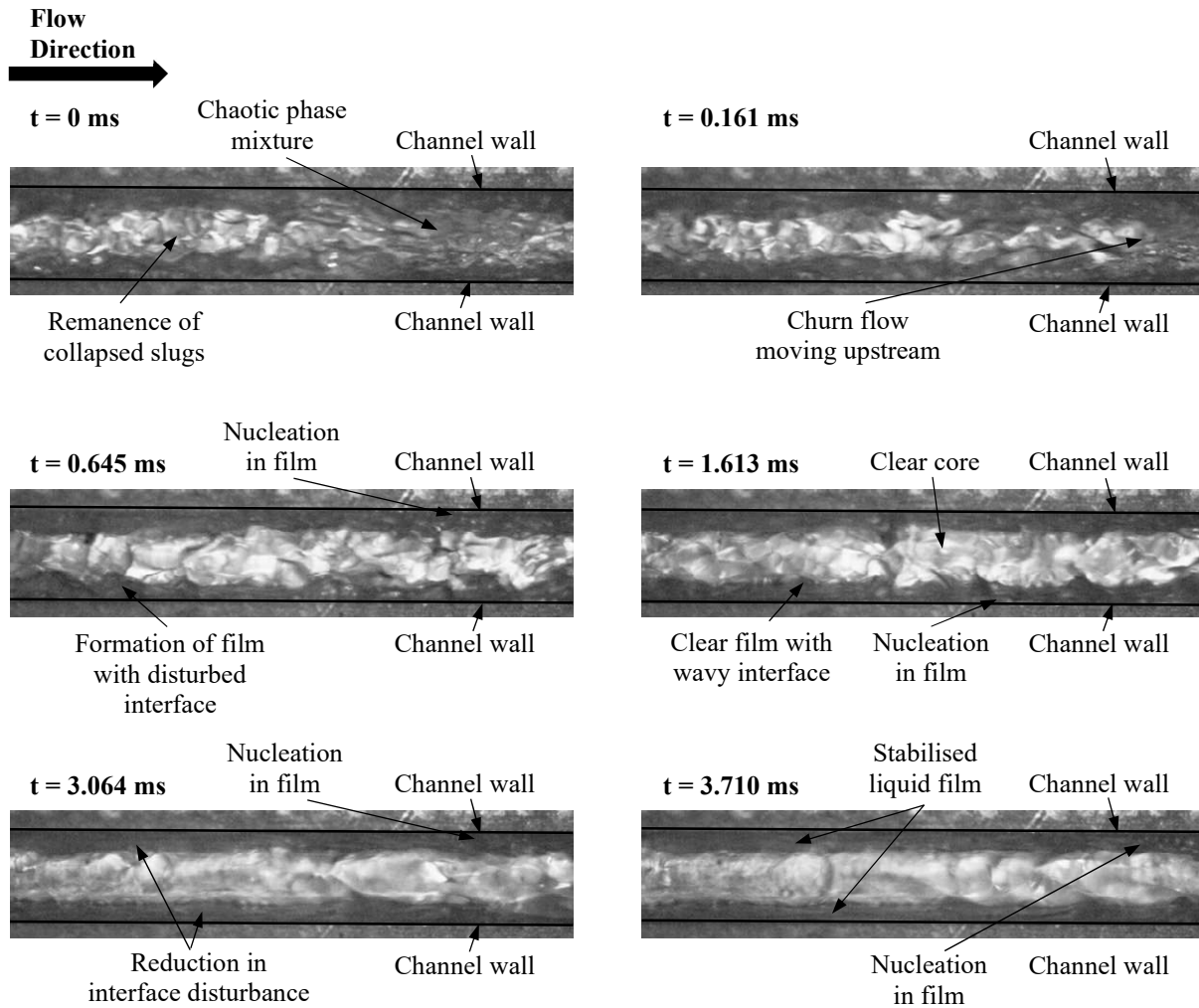


Figure 4.11. Flow visualisation of transition between churn flow and annular flow at outlet for $G = 500 \text{ kg/m}^2\text{s}$, $q''_{\text{wall}} = 78.98 \text{ kW/m}^2$ and $P_{\text{in}} = 1 \text{ bar}$.

Chen [198] reported two distinct mechanisms responsible for this churn-to-annular transition. The first mechanism occurs when the average void fraction becomes sufficiently high that elongated gas slugs can puncture the liquid bridges, eventually merging into a continuous vapour core. This mechanism dominates under low liquid velocity conditions, where the liquid phase offers limited resistance to vapour channel formation. The second mechanism is associated with higher liquid superficial velocities. In this regime, the liquid film along the tube wall becomes thicker, constricting the space available for the vapour core. As a result, the vapour is accelerated even at a fixed superficial velocity, producing a high-speed gas stream that shears against the liquid film and sustains its attachment to the wall. This interaction promotes the formation of a stable annular structure.

The flow behaviour recorded in Figure 4.11 corresponds to Chen's second mechanism, relating to the kinematic effect of superficial phase velocities. Figure 4.7 shows that, when expressed using the wall heat flux-dimensionless saturated length parameter, the churn to annular transition shifts to higher values as the mass flux increases. This indicates that a greater degree of effective vapour development within the available saturated length is required to establish and sustain a continuous vapour core at higher mass fluxes. At elevated mass flux, the liquid phase possesses greater inertia and offers increased resistance to displacement towards the channel wall, so a larger vapour volumetric flow within the two-phase region is necessary to form a stable annular structure. However, when the boiling number is combined with the dimensionless saturated length, the transition boundaries occur at lower values of this parameter as the mass flux increases. This suggests that, relative to the total mass flow, a smaller amount of vapour generation within the saturated region is sufficient to trigger the transition under high-mass-flux conditions.

The transition mechanism is fundamentally linked to the local phase velocities and the balance of inertial forces between the liquid and vapour phases. Consequently, a coordinate system based on superficial velocities provides a more direct representation of these effects. Figure 4.12 presents the flow pattern map in terms of superficial phase velocities and shows that higher liquid superficial velocities, associated with increased mass flux, require correspondingly higher vapour superficial velocities to produce the transition from churn to annular flow. This behaviour can be explained by the increased inertia of the liquid phase at higher flow rates. As the liquid superficial velocity increases, greater vapour momentum is required to displace the liquid towards the channel wall and establish a stable annular configuration.

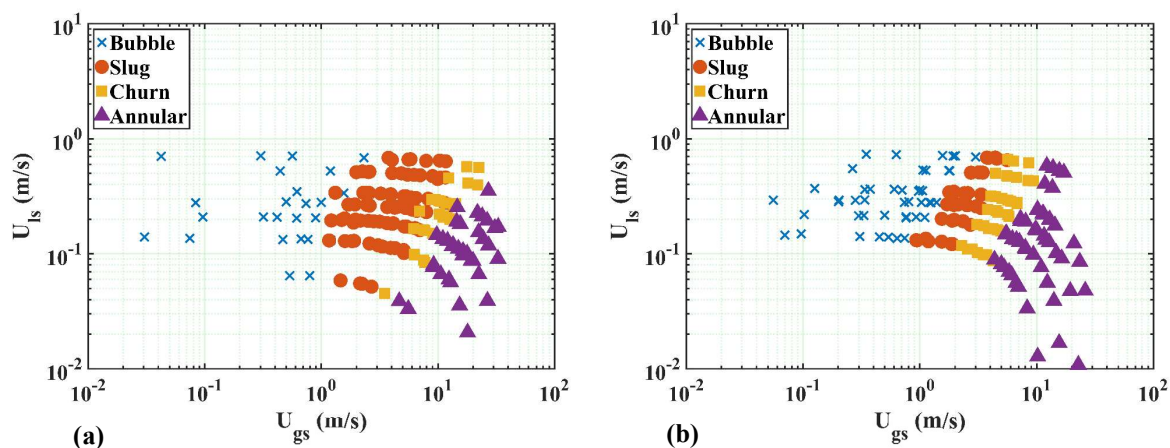


Figure 4.12. Flow pattern maps for 75 mm long test section using superficial velocity coordinates: (a) for 1 bar inlet pressure; (b) for 2 bar inlet pressure.

4.3.4 Effect of System Pressure

Figure 4.13 illustrates the effect of system pressure on the observed flow pattern transitions, presented in terms of mass flux vapour quality. The empirically determined transition boundaries for inlet pressures of 1 bar, 1.5 bar, and 2 bar are shown together for comparison. A horizontal green line is also included to indicate the experimental operating limit, beyond which no data were obtained. The figure therefore provides a direct visualisation of how the transition locations shift with channel inlet pressure across the investigated operating range. Comparison of the transition lines indicates that system pressure does not produce a consistent or monotonic shift in the vapour quality at which the slug to churn or churn to annular transitions occur. Although the transition boundaries differ slightly between pressures, no clear trend with increasing inlet pressure is evident for these two transitions. In contrast, the bubble to slug transition demonstrates a systematic dependence on pressure, with higher inlet pressures corresponding to higher vapour qualities at which confinement first occurs.

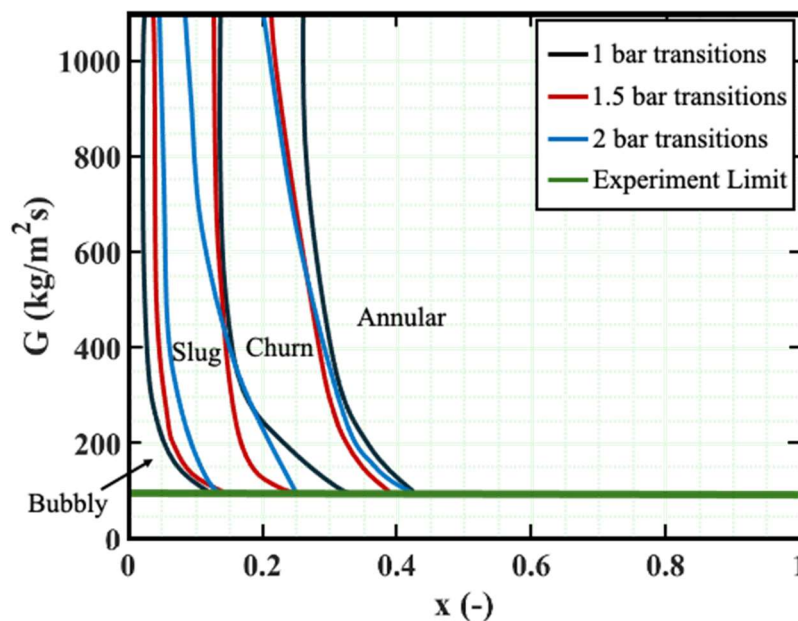
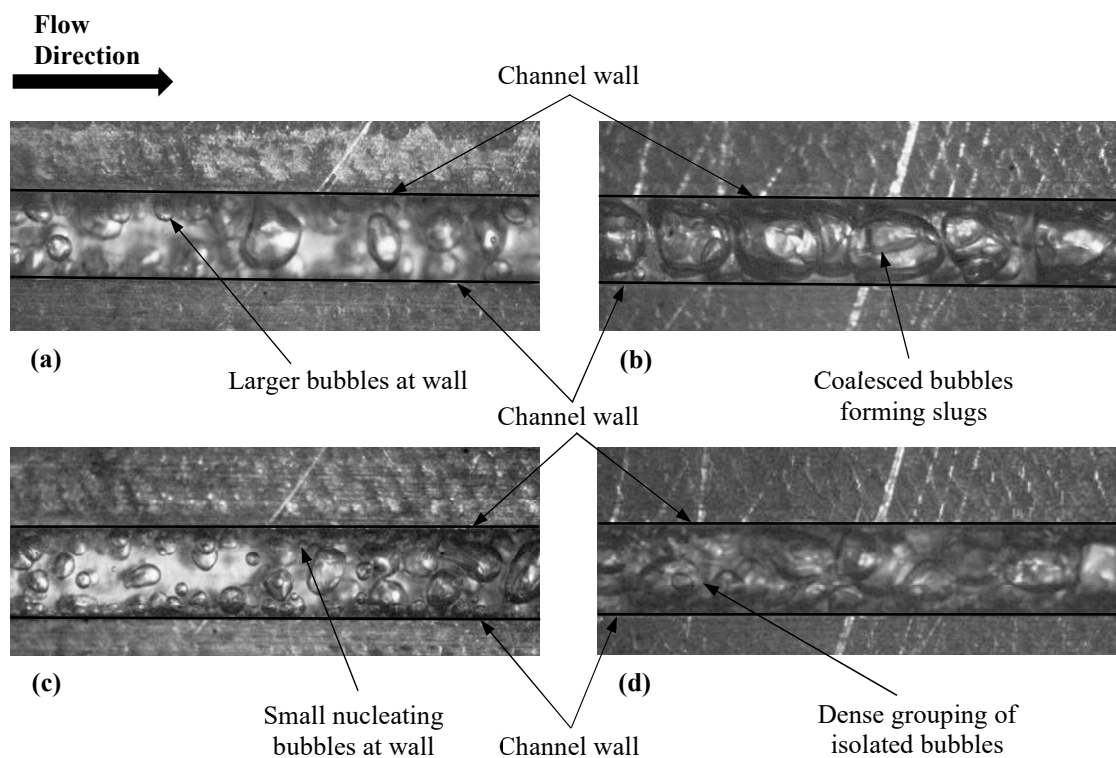


Figure 4.13. Effect of channel inlet pressure on flow pattern transition boundaries in the 75 mm long channel test section.

The absence of a clear influence of pressure on the slug to churn and churn to annular transitions may, at least in part, be due to the time-averaged and inherently subjective nature of the flow pattern classification. The collapse of slugs into churn flow, for instance, is highly dependent on the instantaneous structure of the vapour phase, which is governed by the random process of bubble coalescence. If numerous small slugs are present in close succession, additional coalescence events are required before collapse can occur, delaying the transition. Conversely, when longer vapour slugs are separated by thin liquid films, the probability of collapse is greater, and the transition may appear to

occur earlier. Similarly, distinguishing between churn and annular flow presents challenges, as it is often difficult to define the precise moment when the liquid film has stabilised sufficiently to constitute annular flow. As seen in Figure 4.11, annular structures often form in a transient manner following the passage of churn flow, and the decision to categorise the flow as annular may vary depending on the observation window.

However, the bubble to slug transition is less ambiguous and therefore more straightforward to classify, as it is clearly identified by the point at which a bubble spans the full channel width and subsequently begins to elongate axially. As already noted, Figure 4.13 demonstrates that higher vapour qualities are required for this transition at elevated inlet pressures. Closer inspection of this behaviour reveals that it is strongly linked to bubble departure size. Figure 4.14 presents flow visualisations at a mass flux of $300 \text{ kg/m}^2\text{s}$ and a heat flux of 19 kW/m^2 , taken at both the mid-channel and outlet locations.



4.14. Flow visualisation in the 75 mm long channel test section for a mass flux of $300 \text{ kg/m}^2\text{s}$ and heat flux of 19 kW/m^2 at: (a) 1 bar and middle location; (b) 2 bar and middle location; (c) 1 bar and outlet location; (d) 2 bar and outlet location.

Figure 4.14(a) and Figure 4.14(b) correspond to 1 bar inlet pressure, while Figure 4.14(c) and Figure 4.14(d) show the equivalent locations at 2 bar inlet pressure. At the channel middle for one bar inlet pressure, large bubbles are visible in the mid-channel, having grown and coalesced due to relatively large initial bubble departure diameters. In contrast, smaller bubbles dominate in Figure 4.14(c) and rather than coalescing, they were frequently observed to rebound from one another. The reduction in

bubble departure size at higher pressures is attributed to the decreased surface tension; while lower surface tension facilitates bubble detachment at smaller diameters, the resulting bubbles are more stable and less prone to coalescence. In Figure 4.14(c), large, coalesced bubbles are evident at the channel outlet, forming the onset of slug flow. Conversely, numerous smaller bubbles remain in close contact but do not merge, despite being pressed together in the outlet visualisations for an inlet pressure of 2 bar. These observations explain why the bubble to slug transition occurs at higher vapour qualities for higher pressures, as the smaller, more stable bubbles delay the coalescence events required for confinement.

4.3.5 Effect of Heated Length

The influence of heated length on flow boiling patterns was investigated by comparing the 30 mm and 75 mm long channels. The same generic flow pattern classifications were observed in both channels. The observed flow patterns at the channel outlet for the 30 mm heated length are shown in Figure 4.15. In the shorter channel, enhanced nucleation activity was evident within the liquid films of the slug, churn and annular regimes. Intermittent churn flow disrupting the annular flow was also observed in the 30 mm channel, as discussed in Section 4.3.3.

The experimental flow pattern map for the 30 mm channel is presented in Figure 4.16, where mass flux and vapour quality are represented on the vertical and horizontal axes, respectively. While this figure illustrates the flow regimes across the range of operating conditions, it does not capture the effect of heated length on flow patterns and their transitions in isolation.

Figure 4.17 presents a direct comparison of the empirically determined transition boundaries for the 30 mm and 75 mm heated channels, plotted in terms of mass flux and vapour quality. The transition lines for both heated lengths are shown together for each pressure condition, along with horizontal lines indicating the experimental operating limits for each test section. Figure 4.17(a) corresponds to an inlet pressure of 1.5 bar, while Figure 4.17(b) shows the comparison at 2 bar. The plots provide a direct visual comparison of how the transition boundaries shift with heated length across the accessible operating range.

From Figure 4.17, the effect of heated length on the churn to annular transition is not clearly defined. The transition boundaries for the two channel lengths do not exhibit a consistent or monotonic shift with heated length, and the observed differences fall within the scatter of the experimental data. This behaviour is consistent with the discussion in Section 4.3.4, where the classification of churn and annular flow was shown to be sensitive to transient behaviour and observation window.

In contrast, a clearer influence of heated length is observed for the bubbly to slug and slug to churn transitions. For both transitions, the boundaries for the shorter channel generally occur at lower vapour qualities compared with the 75 mm channel.

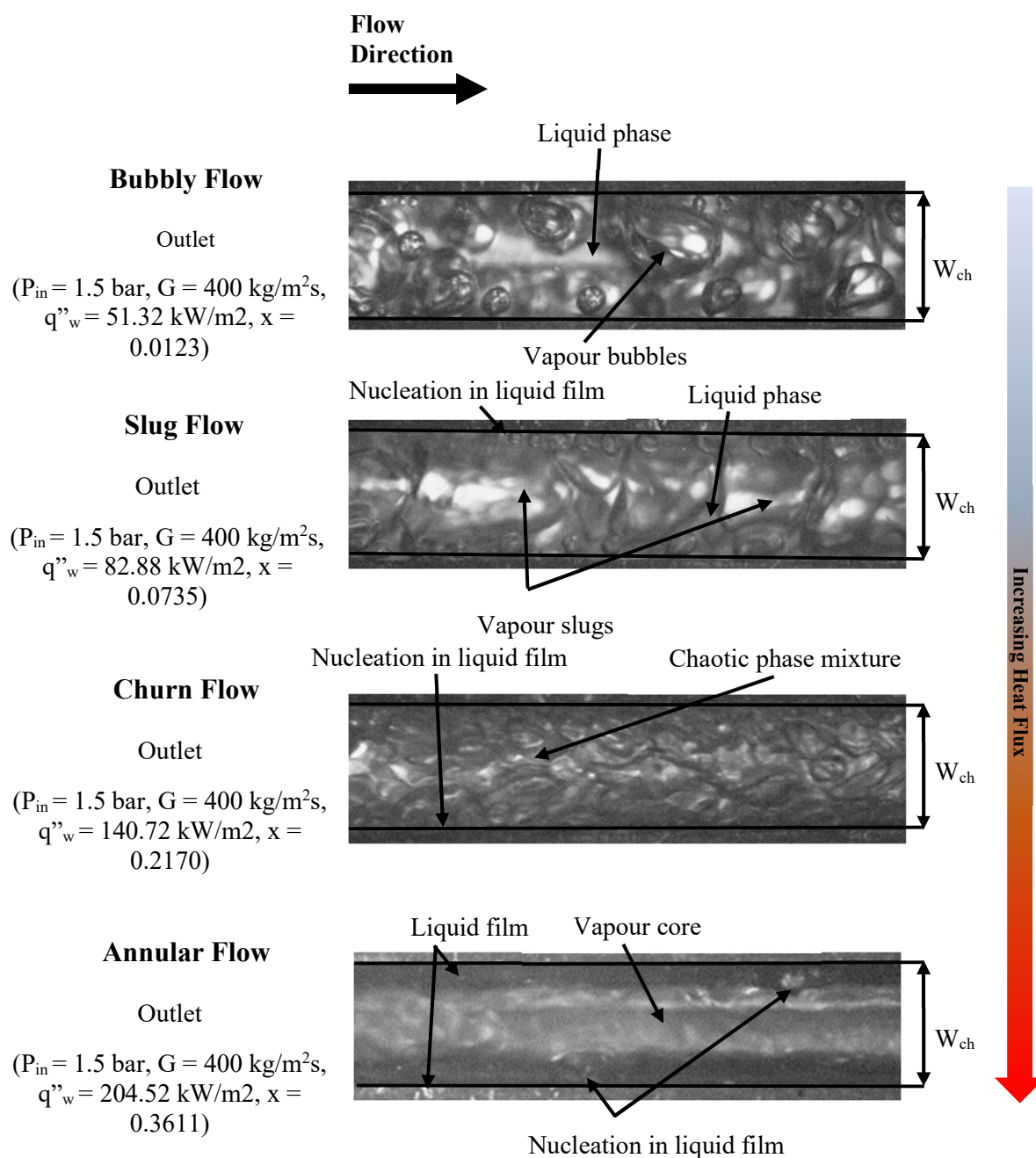


Figure 4.15. Observed flow patterns for the 30 mm long channel in the saturated region with generic classifications.

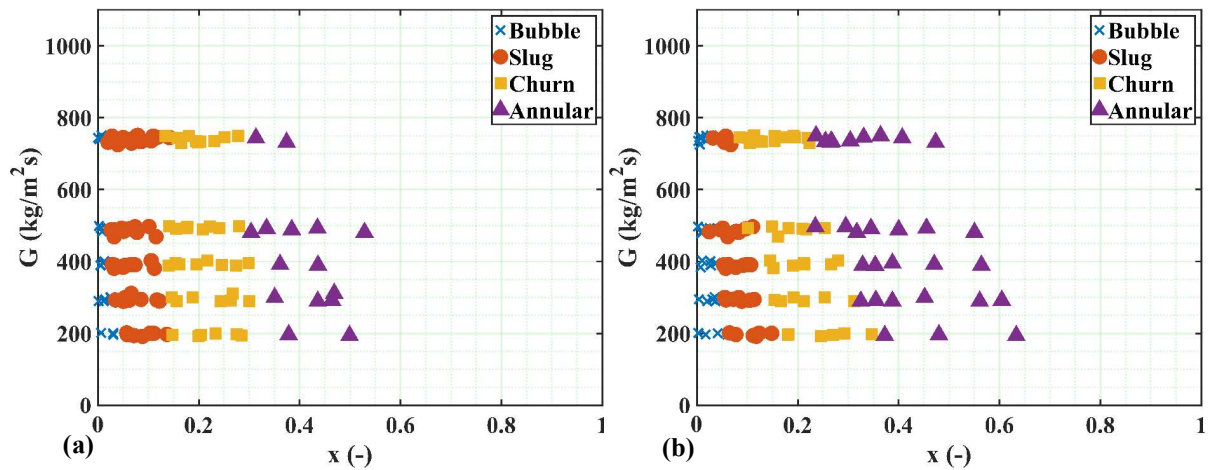


Figure 4.16. Flow pattern maps for 30 mm long test section using mass flux and vapour quality for: (a) 1.5 bar inlet pressure; (b) 2 bar inlet pressure.

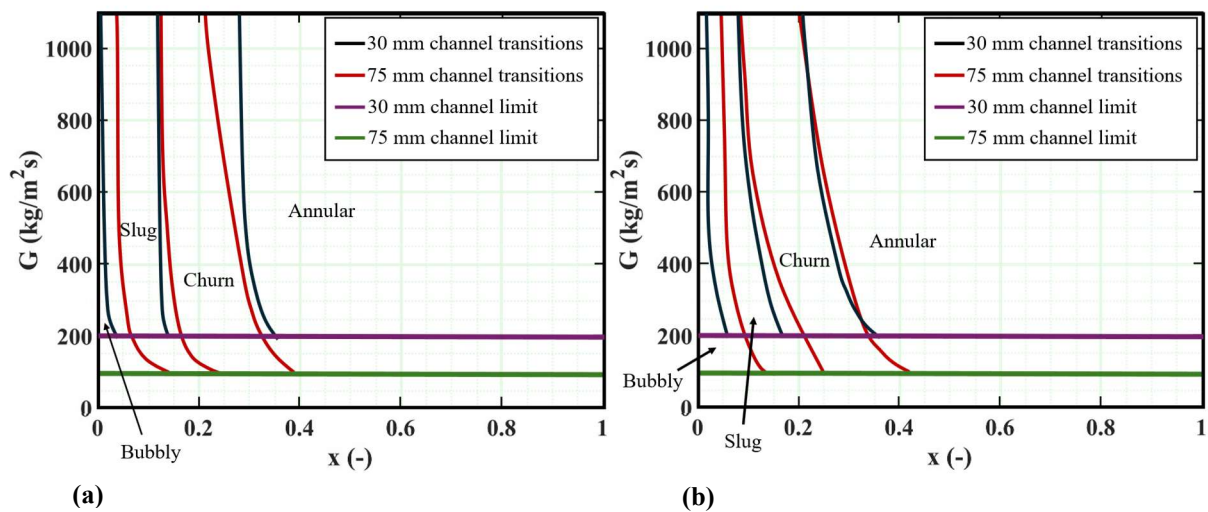


Figure 4.17. Effect of heated length on flow pattern transition boundaries for the 30 mm and 75 mm channels: (a) inlet pressure of 1.5 bar; (b) inlet pressure of 2 bar.

Beginning with the bubbly to slug transition, it is evident that the shorter heated length requires a higher heat flux to achieve the same vapour quality. Elevated heat fluxes are generally associated with the formation of larger bubbles, which in turn reduce the stability of the vapour- liquid interface. In addition, higher heat fluxes promote increased nucleation site density, leading to a greater number of bubbles being generated. The combination of larger bubble sizes and more active nucleation sites increases the likelihood of bubble interactions and coalescence. Consequently, the transition from bubbly to slug flow occurs at lower vapour qualities in shorter channels.

Turning to the slug to churn transition, an earlier onset of churn flow is also evident in the shorter channel. Confined vapour structures were observed to appear at lower vapour qualities, i.e. into slug

flow, as discussed above. As discussed in Section 4.3.2, the slug to churn transition was primarily driven by slugs growing and/or coalescing to lengths at which they are unstable. The growth of the confined vapour slugs is primarily driven by evaporation momentum at the surrounding liquid films. At the higher heat fluxes associated with the shorter heated length, bubble growth occurs more rapidly, producing elongated structures with unstable tails. These slugs would then be more prone to collapse into churn flow, even at the same vapour quality due to the higher heat fluxes. Taken together, these observations suggest that heated length influences not only the bubbly to slug transition, but also the boundary between slug and churn flow.

4.4 COMPARISON TO EXISTING FLOW BOILING PATTERN MAPS

This section provides an in-depth comparison between the flow pattern data obtained from the experimental facility and existing microscale flow pattern maps. A detailed presentation of existing flow pattern prediction tools is presented in Chapter 2; however, a brief overview is also presented in this section.

4.4.1 Comparison to Chen (2006)

The comparison between the experimental data from the current study and the flow pattern map proposed by Chen [198] is presented in Figure 4.18. Chen [198] developed detailed flow pattern maps based on indirect diabatic flow boiling experiments with R134a at 6 bar, 10 bar and 14 bar in upwards vertical tubes. Mass fluxes between 50-6200 kg/m²s and vapour qualities between 0.0005 and 0.9 were studied in tubes with internal diameters of 1.1 mm, 2.01 mm 2.88 mm and 4.26 mm.

The flow regime map provided only limited accuracy in identifying bubbly flow, with just a couple of points correctly predicted in this region, while the majority fell within the slug domain. Predictions for slug flow were partially successful, although a substantial proportion of slug data points were instead classified within the churn and annular regions. In particular, the map performed poorly in capturing churn flow behaviour, where significant discrepancies were observed. By contrast, the representation of annular flow was comparatively robust, with predictions showing a reasonable degree of agreement with the experimental data. However, this is purely because the transitions were predicted to occur at significantly lower superficial gas velocities, as opposed to accurate prediction of the churn to annular transition boundary itself. The map did perform slightly better for the 30 mm long channel test section due to the earlier transition between flow patterns.

4.4.2 Comparison to Mahmoud and Karayiannis (2016)

Mahmoud and Karayiannis [82] examined several established flow regime maps by benchmarking them against an experimental database generated using R245fa within a vertically upward flowing copper microtube of 1.1 mm internal diameter. The test section consisted of a 150 mm adiabatic calming section followed by a 300 mm electrically heated length, with flow visualisation conducted at the exit of the heated section. The operating conditions encompassed heat fluxes between 3 and 25 kW/m², mass fluxes from 100 to 400 kg/m²s, an inlet subcooling of 5 K, and an inlet pressure of 1.85 bar. Their comparison revealed that the majority of existing maps were unable to accurately capture the experimentally observed transition boundaries. Among the maps considered, Chen's [198] model showed the closest agreement with portions of the dataset; however, it failed to accurately represent the bubbly to slug flow transition. Therefore, Mahmoud and Karayiannis proposed an analytical revised transition criterion that incorporated the influence of inertial effects, tube diameter, and fluid thermophysical properties on bubble confinement.

The comparison between the current experimental data and the correlation provided by Mahmoud and Karayiannis [82] is presented in Figure 4.19. It can be seen that there is marginal improvement in the prediction between bubbly and slug flows; however, the improvement is minimal, and the trend appears to differ significantly from the experimental data. This can be seen where the gradient of the transition boundary appears to decrease with increasing superficial liquid velocities; however, the correlation proposed by Mahmoud and Karayiannis sees an increase with an increase in liquid superficial velocities.

A potential reason for the poor performance may be in the lack of consideration of the viscous force acting on the newly confined bubble. The original force balance is displayed and discussed in Chapter 2, Figure 2.26. A revised force balance is presented in Figure 4.20, which now includes the effect of viscous drag acting on the spherical bubble. Using Laplace's law, the capillary pressure jump term can be expressed as shown in Equation (4.1), as was also considered by Mahmoud and Karayiannis [82].

$$(P_g - P_l) = \frac{4\sigma}{D_b} \Rightarrow (P_g - P_l) \cdot \frac{\pi D_b^2}{4} = \pi D_b \sigma \quad (4.1)$$

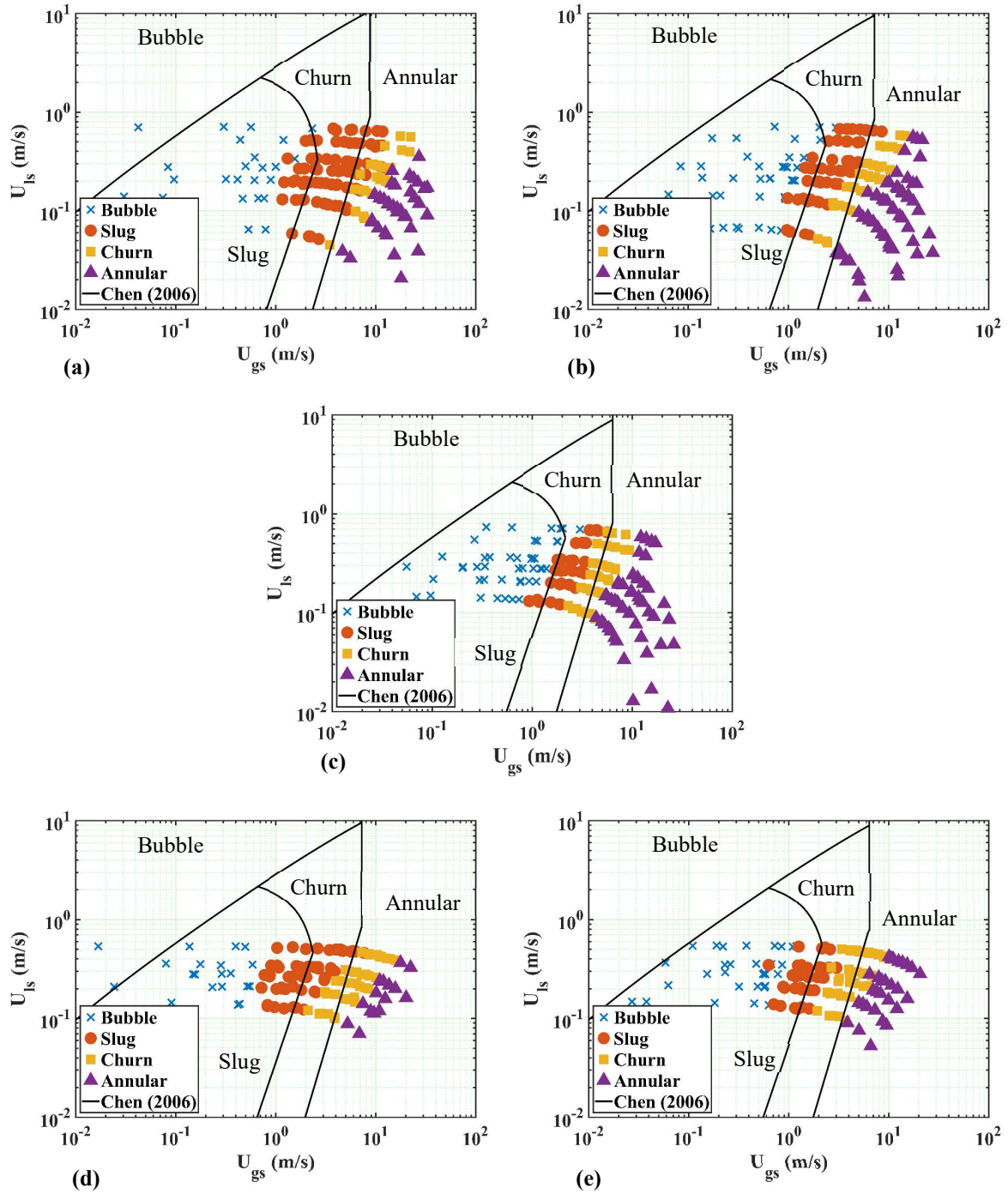


Figure 4.18. Comparison of experimental flow patterns with flow pattern map proposed by Chen [198]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.

The viscous drag term is more difficult to determine; however, an estimate can be derived. The formula for the drag force is as follows:

$$F_\mu = \frac{1}{2} C_d \rho_l U^2 A = \frac{1}{8} C_d \rho_l U^2 \pi D_b \quad (4.2)$$

where C_d represents the drag coefficient. A simplified value for the drag coefficient was proposed by Hadamard–Rybczynski for low Reynolds number viscous flow, in which $C_d = 16/Re$ [394]. Whilst the flow observed during experiments is laminar, it is not low Reynolds number viscous flow as this is flow in which $Re \approx 1$. Nevertheless, this method offers a straightforward means of estimating the drag force, which can then be integrated into a practical transition boundary prediction tool. The force balance using the Hadamard–Rybczynski drag coefficient on a spherical bubble is presented in Equation (4.3), where the final form produces a solvable quadratic.

$$\rho_l \frac{\pi D_b^2}{4} U_r^2 - 2\pi\mu_l D_b U_r - \pi D_b \sigma = 0 \quad (4.3)$$

The bubble can be assumed to become confined by the smallest channel dimension, in this case it was the channel width, as it was in the force balance proposed by Mahmoud and Karayiannis. Using this assumption, the final bubbly to slug flow pattern transition that includes consideration of the viscous drag force acting on the bubble is presented in Equation (4.4); however, the channel width can be replaced with the smallest channel dimension.

$$U_r = \frac{2(2\mu_l + \sqrt{\rho_l \cdot \min(W_{ch}, H_{ch})\sigma + 4\mu_l^2})}{\rho_l \cdot \min(W_{ch}, H_{ch})} \quad (4.4)$$

A comparison between the newly semi-analytically derived bubbly to slug flow pattern transition boundary is presented in Figure 4.21. Whilst it did not perform particularly well for the 75 mm long channel, the new bubbly to slug transition boundary did perform particularly well for the bubble to slug transition in the 30 mm long channel test section, especially for an inlet pressure of 1.5 bar. Nevertheless, the new transition boundary did overall better reflect the transition trend. The failure of the new transition boundary to predict the transition observed in the longer test section indicates that the influence of heated length and heat flux is not properly considered.

Furthermore, the new transition boundary indicates that the transition should occur at lower superficial gas velocities for lower system pressures, whereas the transition actually occurs at higher superficial gas velocities. This may be due to an over dependence on the change in fluid viscosity, which decreases with system pressure. Again, one way to improve the semi-analytical transition boundary may be to consider a more recent and accurate representation of the drag coefficient acting on the spherical bubble.

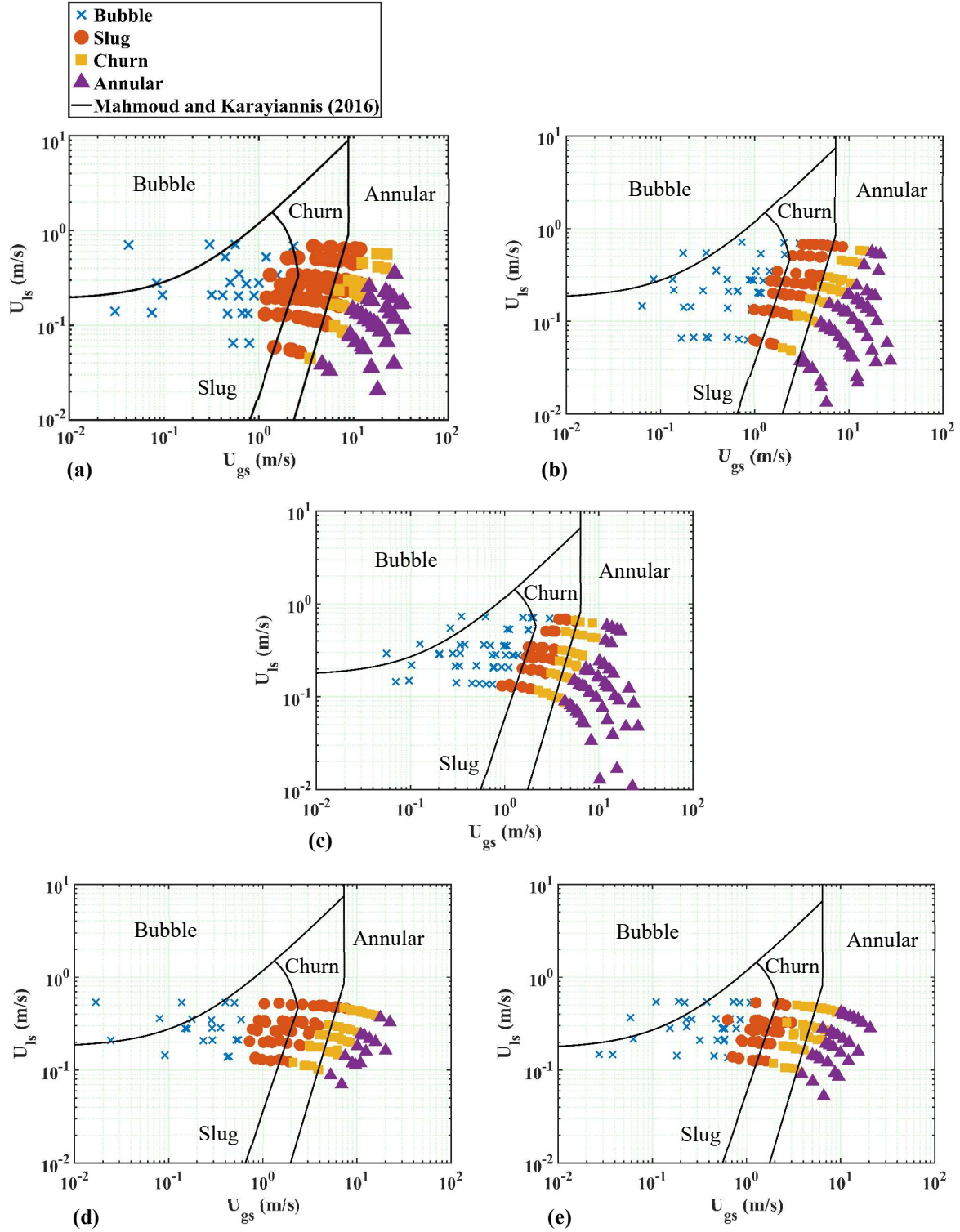


Figure 4.19. Comparison of experimental flow patterns with flow pattern map proposed by Mahmoud and Karayiannis [82]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.

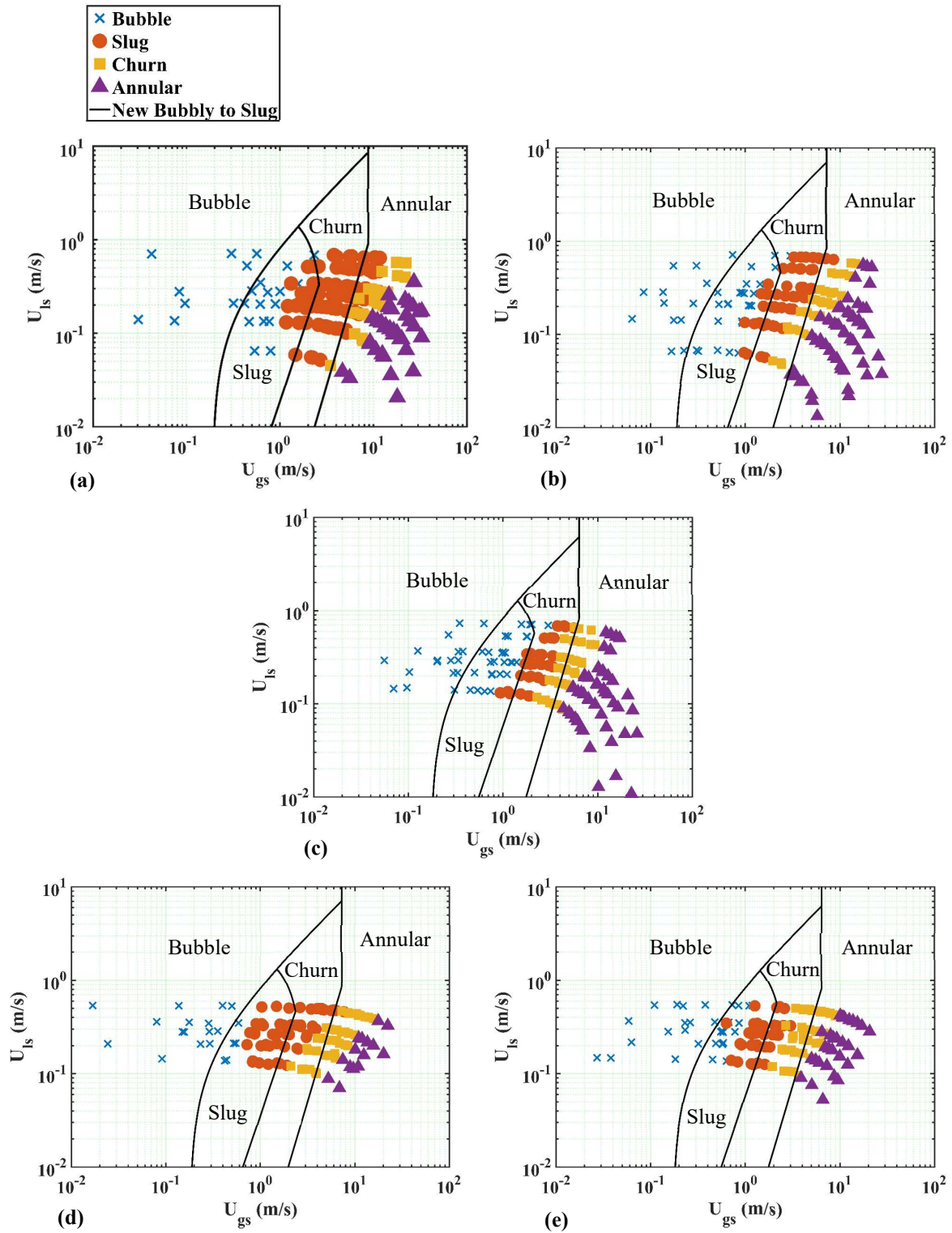


Figure 4.21. Comparison of experimental flow patterns with new semi-analytical bubbly to slug transition: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.

4.4.3 Comparison to Revellin and Thome (2007) and Ong and Thome (2011)

Figure 4.22 compares the experimentally observed flow regimes with the flow pattern transition boundaries proposed by Revellin and Thome [125], plotted in terms of mass flux and vapour quality. The reference map was constructed from experiments conducted with R-134a and R-245fa in horizontally oriented microtubes of 0.509 mm and 0.79 mm internal diameter. The test section consisted of an 80 mm preheater, followed by a 20 mm glass insulation section, a 110 mm stainless steel micro-evaporator, and a 100 mm glass section used for flow visualisation at the outlet. In developing the map, Revellin and Thome [125] assessed the influence of heat flux, mass flux, fluid properties, and channel size on flow behaviour. Three principal regimes were identified. The isolated bubble (IB) regime extends from the onset of vapour generation to the vapour quality corresponding to the maximum bubble count frequency, where discrete bubbles dominate the flow. The coalescing bubble (CB) regime occurs beyond this point, where increasing vapour production leads to frequent bubble interactions and the formation of elongated vapour structures. The annular regime is reached when the bubble count frequency falls to zero, indicating the presence of a continuous vapour core with a surrounding liquid film. The transitions between the IB–CB and CB–annular regimes form the boundaries of the proposed flow map.

The comparison shows that the isolated bubble to coalescing bubble transition performs well for the 75 mm long channel for all system pressures, assuming that the isolated bubble to coalescing bubble transition boundary is analogous to the bubbly to slug transition boundary. However, the boundary did not capture the transition for the 30 mm long channel as well, significantly overpredicting the vapour qualities required. This occurs despite the fact that the boiling number was considered in the correlation in an attempt to consider the heated length. The poor performance for the shorter channel is further surprising, considering the heated lengths used by Revellin and Thome [125] ranged from 20 mm to 70 mm.

Conversely, Figure 4.23 presents the comparison between the experimental data and the flow pattern map proposed by Ong and Thome [80]. This was developed using refrigerants R-134a, R-236fa and R-245fa in tubes with internal diameters of 1.03 mm, 2.2 mm and 3.04 mm. The same methodology of determining flow patterns used by Revellin and Thome [125] was also used, along with the data collected in the study. A revised empirical isolated bubble to coalescing bubble correlation was produced, with the new data being collected for heated lengths of 157.5 mm, 201.1 mm and 241.6 mm. They concluded that the confinement number was a relevant parameter to the transition, whilst also maintaining that the boiling number was a relevant parameter.

A comparison of the bubbly to slug transition with the revised isolated bubble to coalescing bubble boundary shows presents a high performance of the correlation, also appearing to better capture the effect of heated lengths. This may be true simply due to the larger range of fluids, heated lengths and

channel geometries through which the flow pattern transition boundary was developed. The empirical nature of the transition boundary means that certain parametric effects are more likely to be accurately captured if a wider range of data is utilised, as was true for the correlation developed by Ong and Thome [80] as opposed to Revellin and Thome [125].

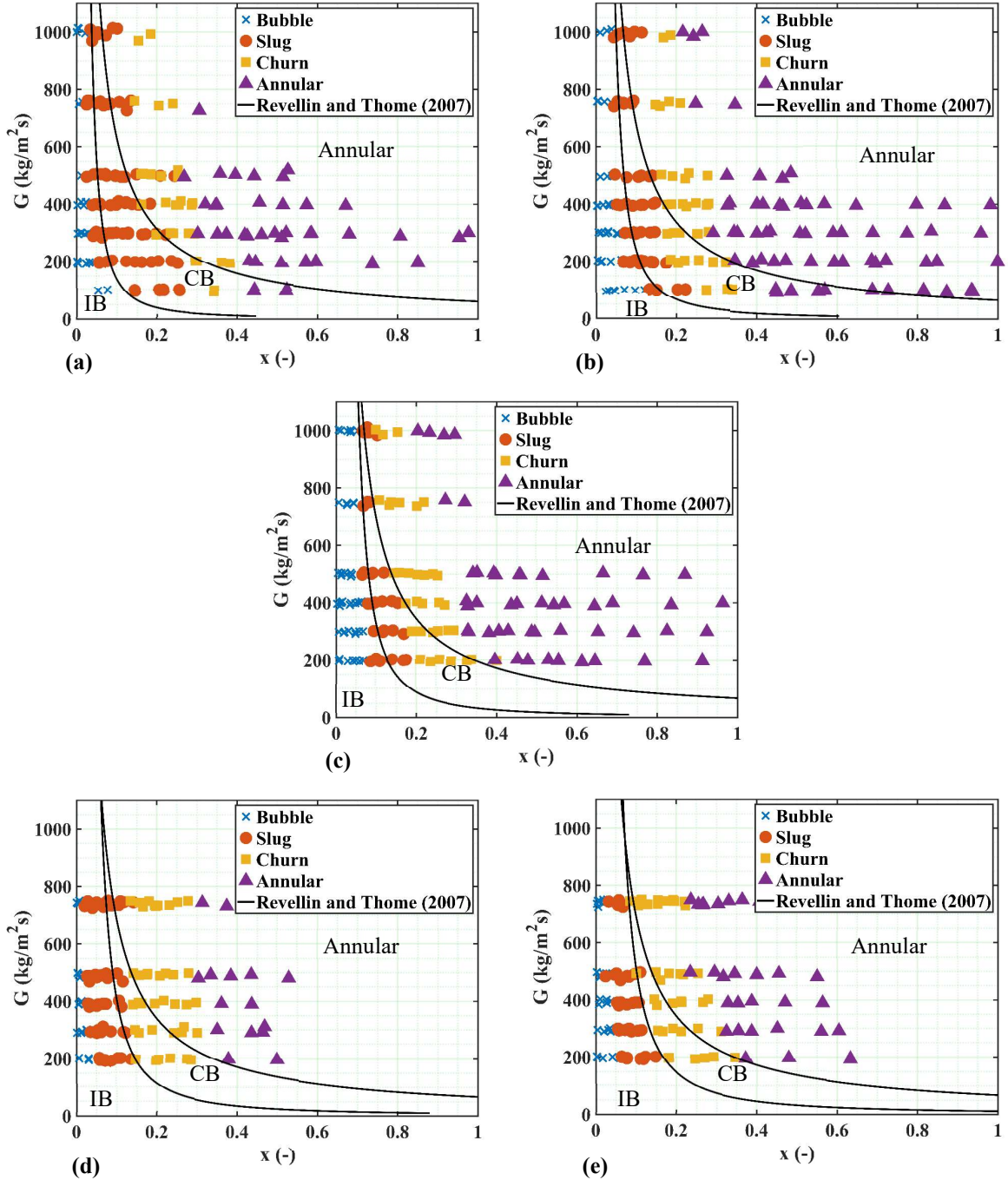


Figure 4.22. Comparison of experimental flow patterns with flow pattern map proposed by Revellin and Thome [125]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.

The coalescing bubble to annular regime did not appear to capture the data quite as well for both test sections. There is some difficulty in evaluating this transition boundary as it does not conform to the ‘classical’ flow pattern designations. However, it appears that for higher mass fluxes it is capturing the classical slug to churn transition for the experimental data and at lower mass fluxes it is capturing the classical churn to annular transition boundary.

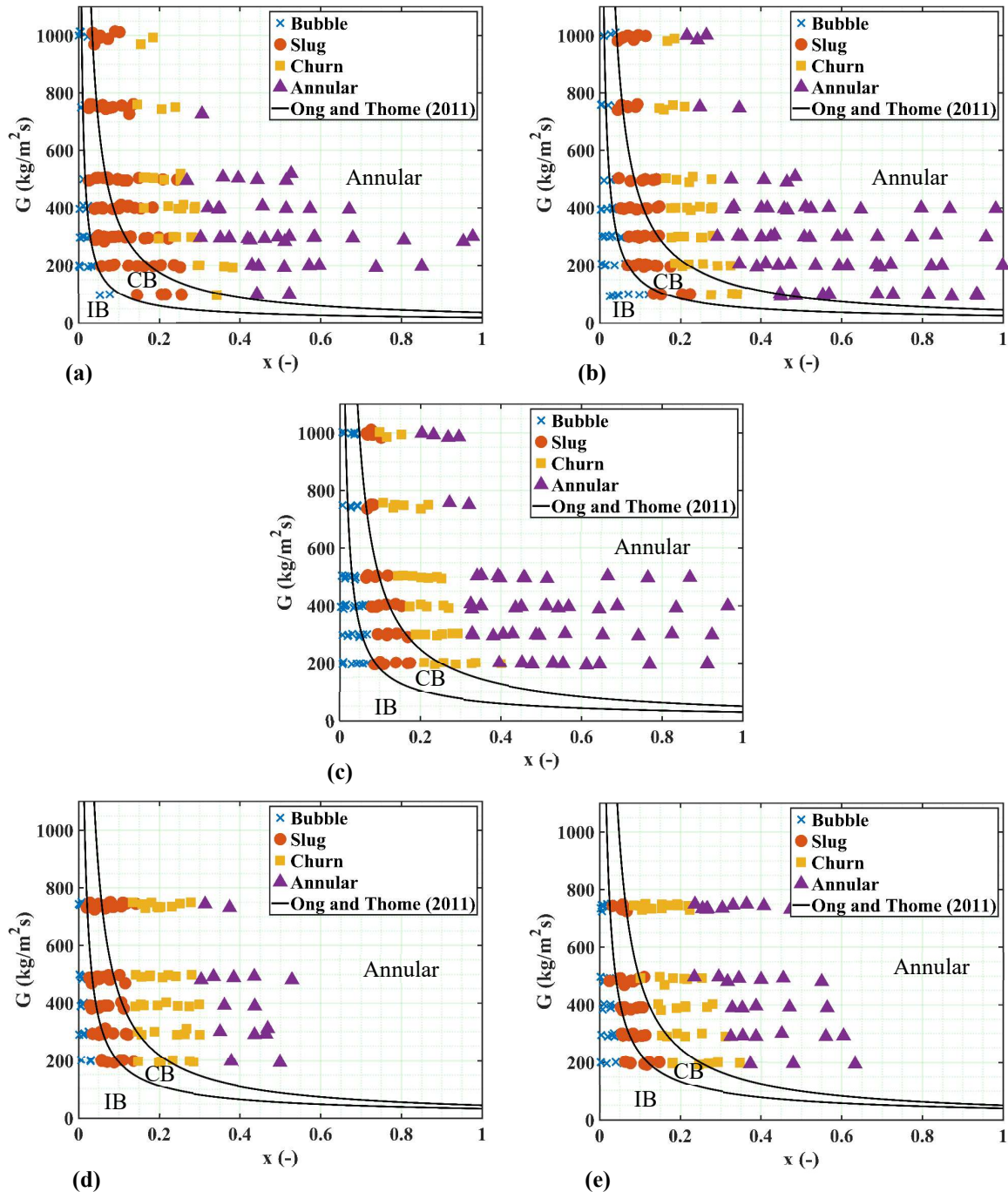


Figure 4.23. Comparison of experimental flow patterns with flow pattern map proposed by Ong and Thome [80]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.

4.4.4 Comparison to Harirchian and Garimella (2010) and Harirchian and Garimella (2012)

Figure 4.24 shows a comparison between the present experimental results and a flow pattern map developed by Harirchian and Garimella [202]. Their study investigated flow boiling of FC-77 in horizontal rectangular multi-microchannels, with hydraulic diameters ranging from 0.096 mm to 0.707 mm and channel lengths of 12.7 mm. Experiments were conducted over a heat flux range of 25–380 kW/m² and mass fluxes between 225 and 1420 kg/m²s. The authors proposed a flow pattern map that classifies regimes into two categories: confined flows (confined slug and confined annular) and unconfined flows (bubbly, churn, annular, and wispy-annular). The map was expressed using two dimensionless parameters: the convective-confinement number and the dimensionless heat flux. Figure 4.24 obviously shows that the flow pattern transitions did not perform particularly well and that, due to the flow patterns being recorded at several locations, the coordinates used did not actually fully separate the flow patterns into distinct regions.

The same authors later modified the flow pattern maps for the purpose of developing flow-regime based modelling of heat transfer and pressure drop [203]. The map was modified to use the phase change number ($\varepsilon\xi$), originally introduced by Saha et al. [204]. The phase change number is defined as the product of the frequency of vapour generation and the fluid particle residence time. However, the number can be reduced to Equation (4.5).

$$\varepsilon\xi = Bo \frac{L_{heat}}{D_h} \cdot \frac{\rho_l - \rho_g}{\rho_g} \quad (4.5)$$

Flow pattern measurements and categorisations were taken at various lengths along the channels and consequently, a new flow pattern map was developed using the new co-ordinates. The comparison between the experimental flow patterns and the later flow pattern map produced by Harirchian and Garimella [203] is presented in Figure 4.25. The figure shows a clear separation of the flow patterns, unlike the previous correlation. However, once again the flow pattern transition boundaries are not accurately predicted with the transition between bubbly and slug flow showing an entirely different trend. The transition into the annular region is fairly well predicted for the 75 mm long channel test section, particularly for an inlet pressure of 1 bar. However, it does not appear that the correlation accurately reflected the effect of heated length on the flow pattern transition boundaries.

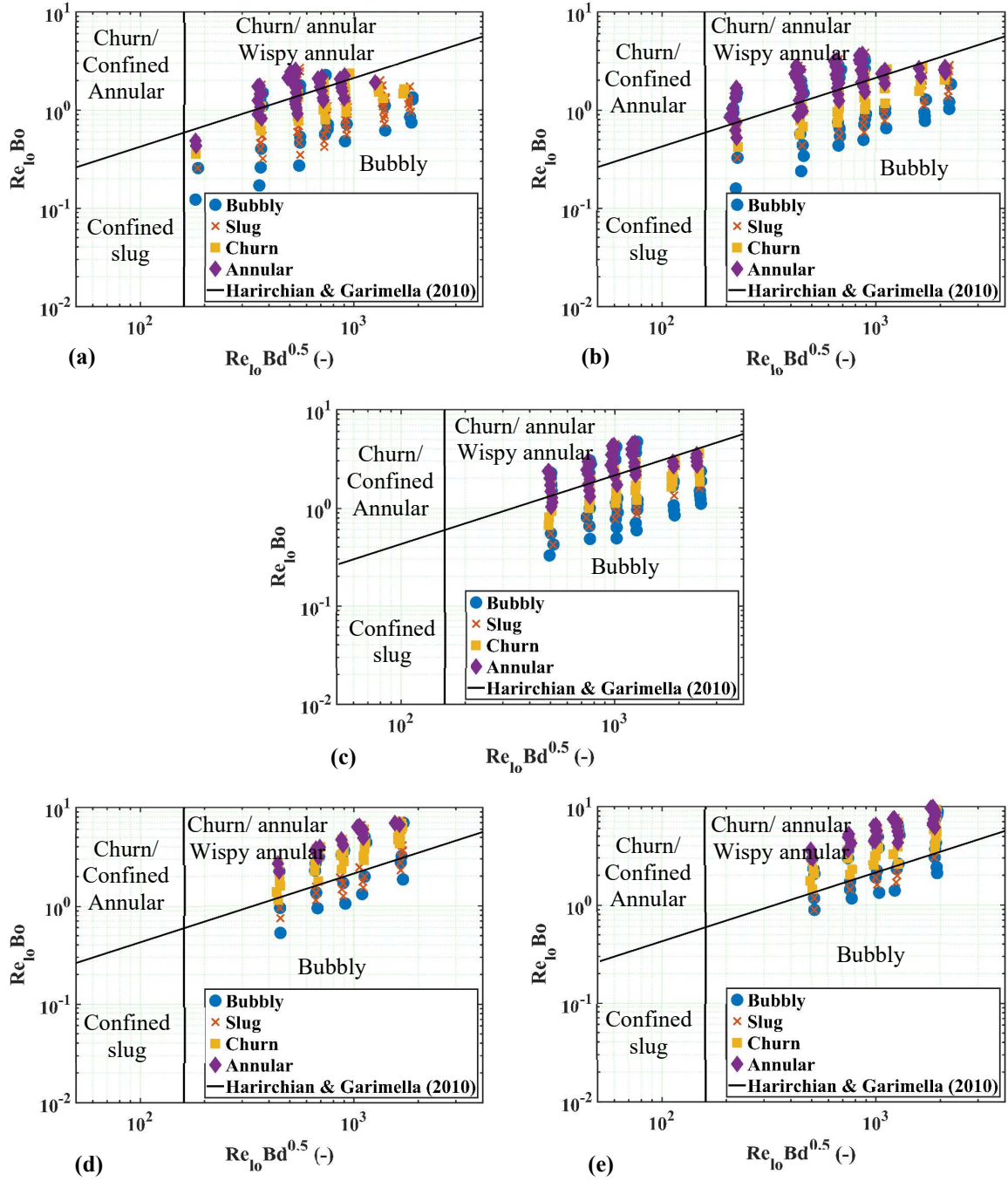


Figure 4.24. Comparison of experimental flow patterns with flow pattern map proposed by Harirchian and Garimella [202]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.

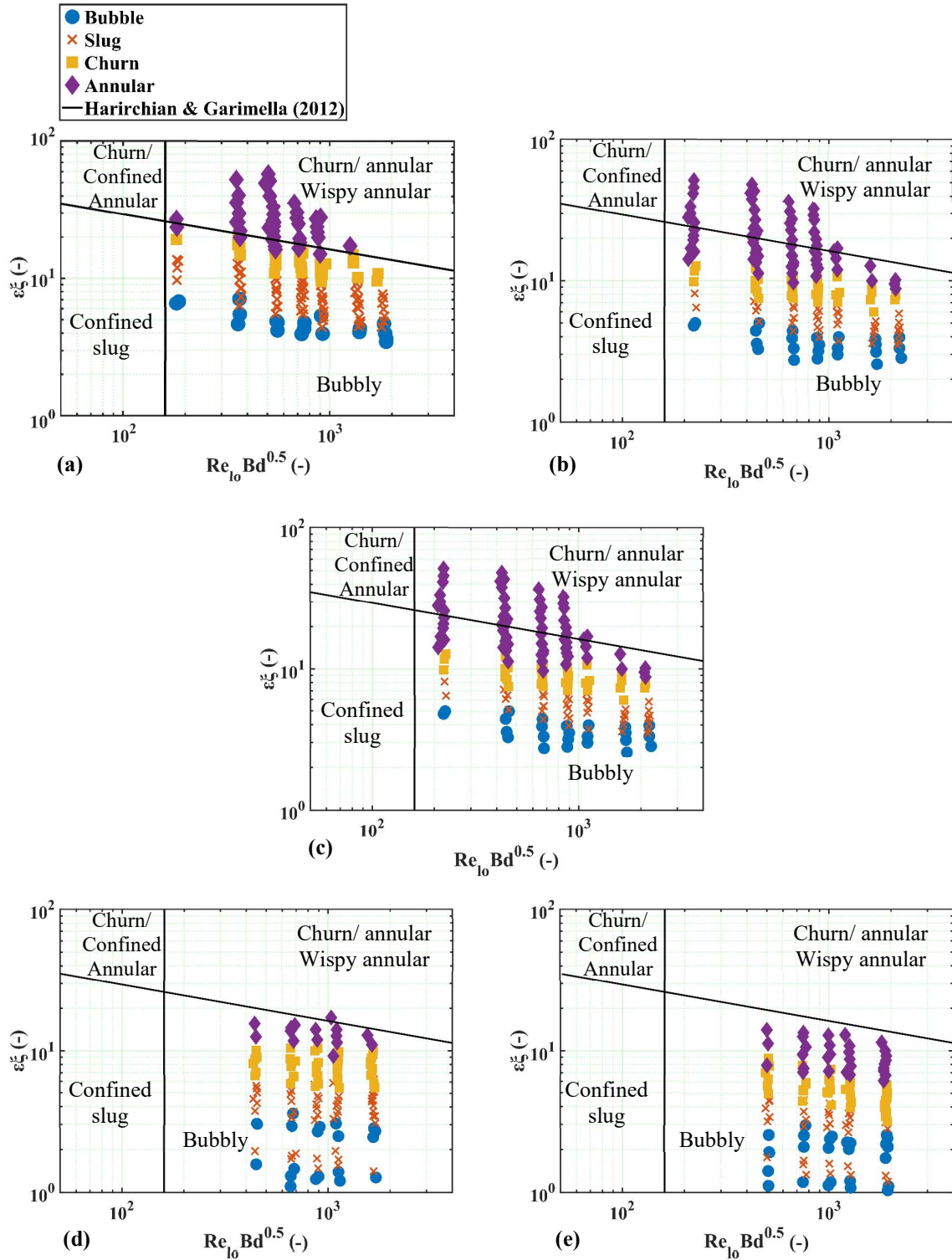


Figure 4.25. Comparison of experimental flow patterns with flow pattern map proposed by Harirchian and Garimella [203]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.

4.4.5 Comparison to Tibiri   et al. (2017)

Tibiri   et al. [186] developed a set of correlations intended to capture a range of microscale flow boiling parameters, including flow regime transitions. A key objective was to ensure that the correlations remained simple and practical to apply. To achieve this, the authors specified three requirements: each expression should take the form of a single continuous function with only a few dimensionless groups; it should be readily linearised; and it should be expressed in a way that allows the inverse function to be derived analytically. Four separate correlations were established for the bubble to slug, slug to churn, churn to annular, and annular to dry out transitions, each supported by an independent database. These databases included results for R-134a, R-236fa, R-245fa, R-1234ze(E), R-32, R-152a, and nitrogen, obtained using both direct and indirect diabatic experimental methods. A substantial proportion of the supporting data, however, was drawn from adiabatic experiments with air–water mixtures. The microchannels examined covered hydraulic diameters from 0.25 mm to 5.5 mm, while no ranges of heat flux or mass flux were reported.

A comparison between the proposed correlations and the current experimental data is presented in Figure 4.26. The flow pattern transition prediction models performed very poorly in comparison to the experimental data. The bubbly to slug transition remains close to 0 for the entire range of analysed mass fluxes. Overall, trends were fairly well captured but all flow pattern transition boundary vapour qualities were significantly underpredicted.

4.4.6 Comparison to Choi et al. (2017)

Choi et al. [151] investigated flow boiling of FC-72 in a horizontal multi-microchannel heat sink. The test section, fabricated from copper, contained fifteen parallel rectangular channels, each 60 mm long with a hydraulic diameter of 0.277 mm. Heating was applied from below, while the top was sealed with a polycarbonate cover. During the experiments, the inlet pressure was maintained between 1.05 and 1.1 bar, and the outlet vapour quality was controlled across a broad range from 0 to 0.96.

The authors proposed a transition criterion between bubbly/slug and annular flows, expressed as a function of the Weber number and vapour quality, which is discussed in detail in Chapter 2. Their resulting flow pattern map, plotted with Weber number against vapour quality, is reproduced in Figure 4.27. Only two regimes were distinguished in their map; namely, bubble/slug and annular. In contrast to this methodology, the present experiments revealed the presence of churn flow. For comparison, the departure into churn flow was considered as part of the annular region proposed by Choi et al. [151]. Overall, the correlation did not perform well in determining the transition from slug to churn flow or from churn to annular flow. This is likely a result of a different flow pattern framework being utilised to categorise flow patterns.

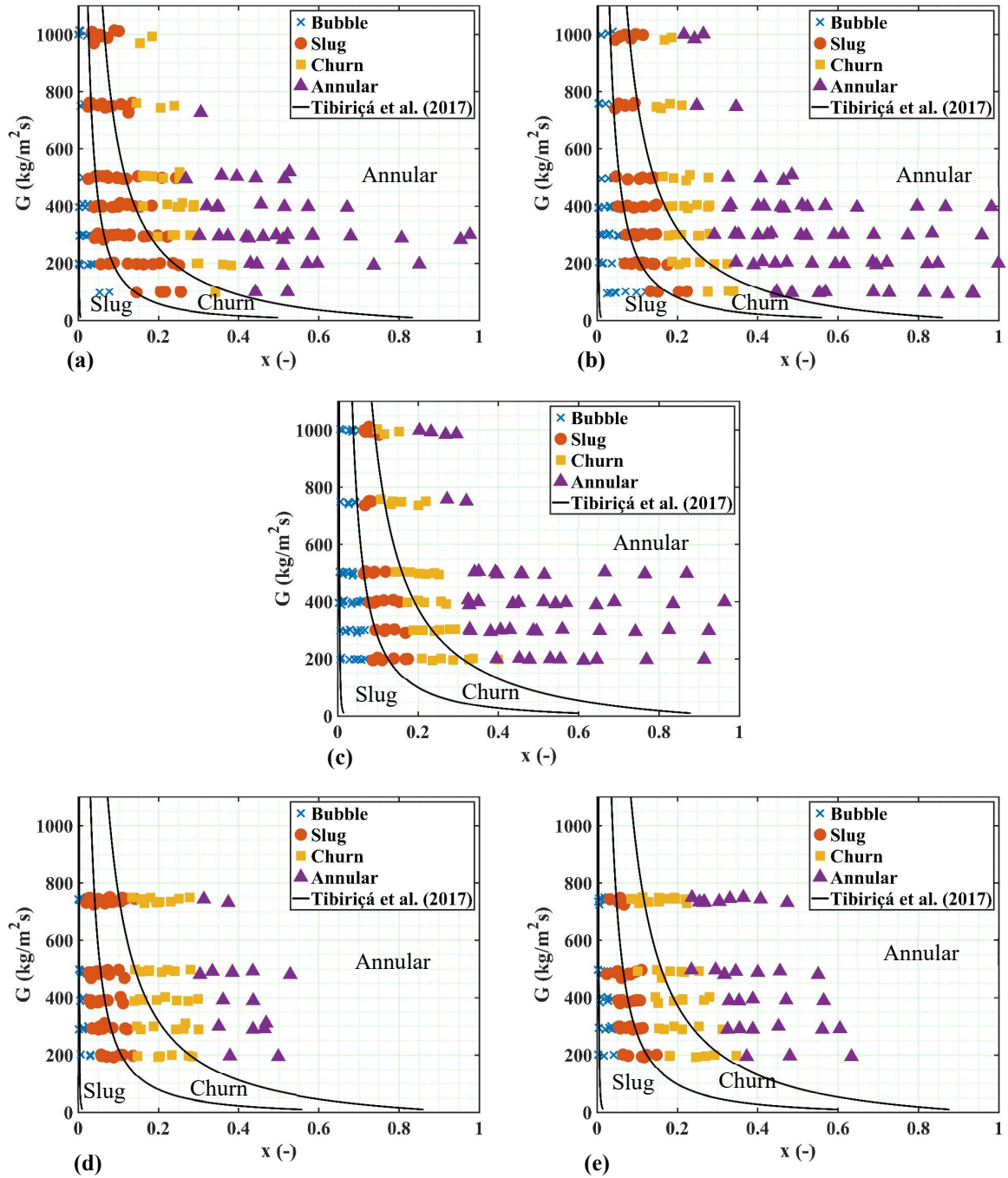


Figure 4.26. Comparison of experimental flow patterns with flow pattern map proposed by Tibirić et al. [186]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.

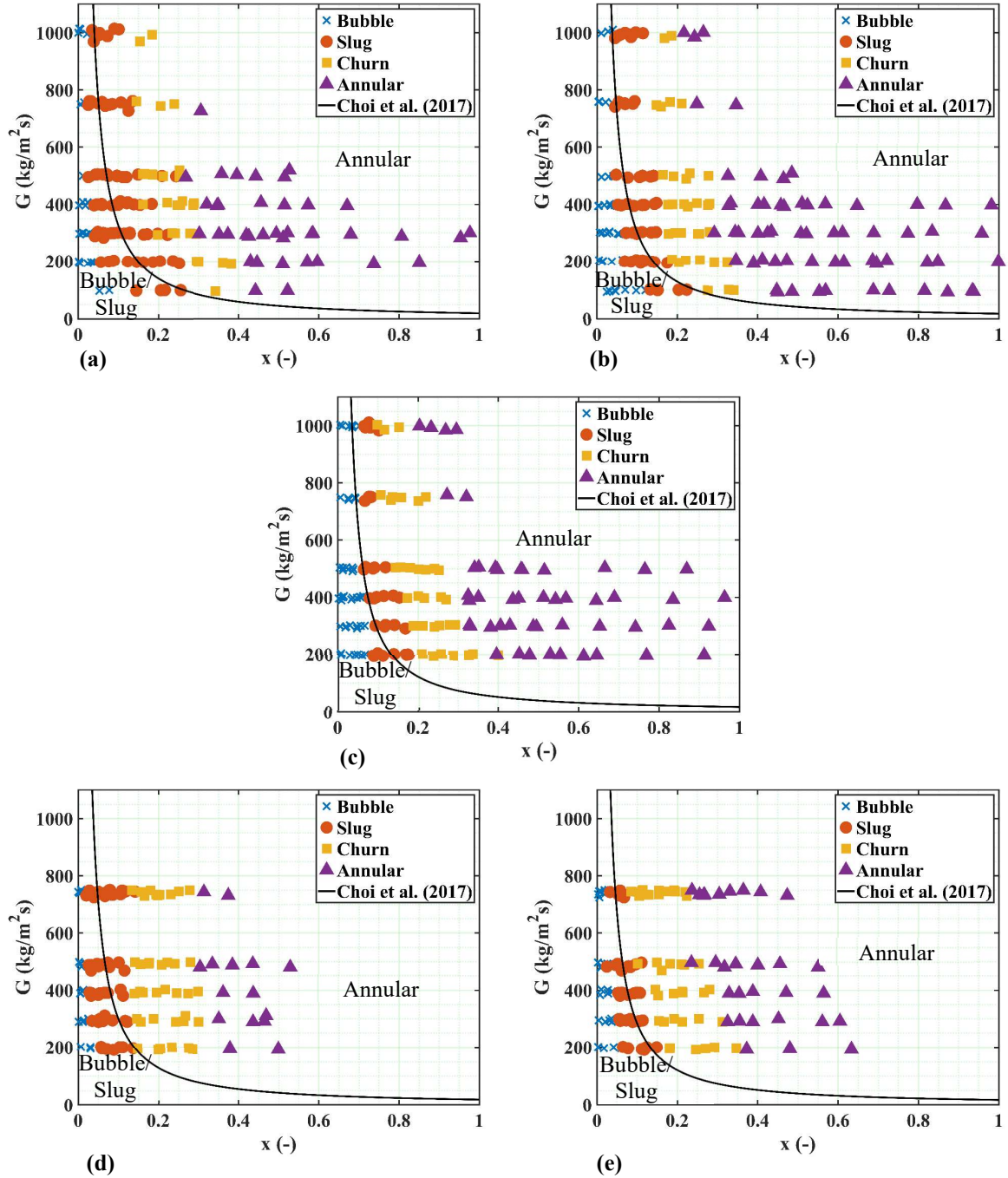


Figure 4.27. Comparison of experimental flow patterns with flow pattern map proposed by Choi et al. [151]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.

4.4.7 Comparison to Jige et al. (2018)

A comparison was made with the flow regime map of Jige et al. [205] in Figure 4.28, who investigated R-32 in horizontal multiport rectangular minichannels of 0.5 mm and 1.0 mm hydraulic diameter. Their experiments covered mass fluxes of 30 kg/m²s to 400 kg/m²s and vapour qualities between

0.1 and 0.9 at a saturation temperature of 15 °C. The observed flow was classified into only three regimes: intermittent, transition, and annular. Intermittent flow dominated at low qualities, while annular flow appeared only at high qualities and high mass fluxes. In contrast, the present work identifies four distinct regimes, with churn occupying a significant portion of the map. Jige et al. [205] effectively combined bubbly and slug into their intermittent regime and treated churn as part of a broader transition category. Their framework therefore cannot incorporate the churn regime observed during experiments, nor the clear sequence of regime development with increasing heat flux.

Differences in experimental configuration also contribute to the discrepancy. Jige et al [205] studied multiport channels, where flow maldistribution and cross-channel interactions allow elongated vapour plugs to persist across a larger portion of the parameter space. Hence, whilst the proposed map may provide a useful description for multiport geometries under the conditions investigated, it does not capture the detailed flow regime behaviour observed in the present single-channel experiments.

4.5 CONCLUSIONS

This chapter presented the experimental results for flow boiling patterns in a rectangular microchannel, with visualisation conducted at multiple channel locations using high-speed imaging. The experiments covered heat fluxes from boiling incipience through to vapour qualities approaching unity, at a controlled subcooling of 10 K. Observations confirmed the four generic flow patterns of bubbly, slug, churn and annular flows. Experimental flow pattern maps were presented using a variety of coordinate systems to capture transition trends. Results showed that regime boundaries shift systematically with mass flux, heat flux, system pressure, and heated length, with transitions occurring at lower vapour qualities under higher mass fluxes. Comparisons with existing predictive maps from the literature revealed partial agreement for some transitions but significant deviations were evident. This highlighted the limitations of established correlations in capturing microscale flow boiling pattern transition behaviour. The key findings were:

- Nucleation consistently initiated at the corners of the rectangular channels. In the subcooled region, bubbles grew while attached to the heated wall but condensed once detached, due to the surrounding subcooled liquid.
- The transition from bubbly to slug flow was primarily governed by bubble coalescence and consequent confinement. Initially, multiple discrete bubbles travelled downstream without filling the channel cross-section. As vapour production intensified, random collisions and coalescence resulted in confined vapour slugs. Higher mass fluxes delayed this transition to higher heat fluxes but lower boiling numbers, due to the increased frequency of bubble collisions at elevated phase velocities, which compensated for smaller individual bubble

sizes. Elevated inlet pressures reduced bubble departure diameters by lowering surface tension, producing more stable bubbles that coalesced less readily. This shifted the transition to higher vapour qualities at higher pressures.

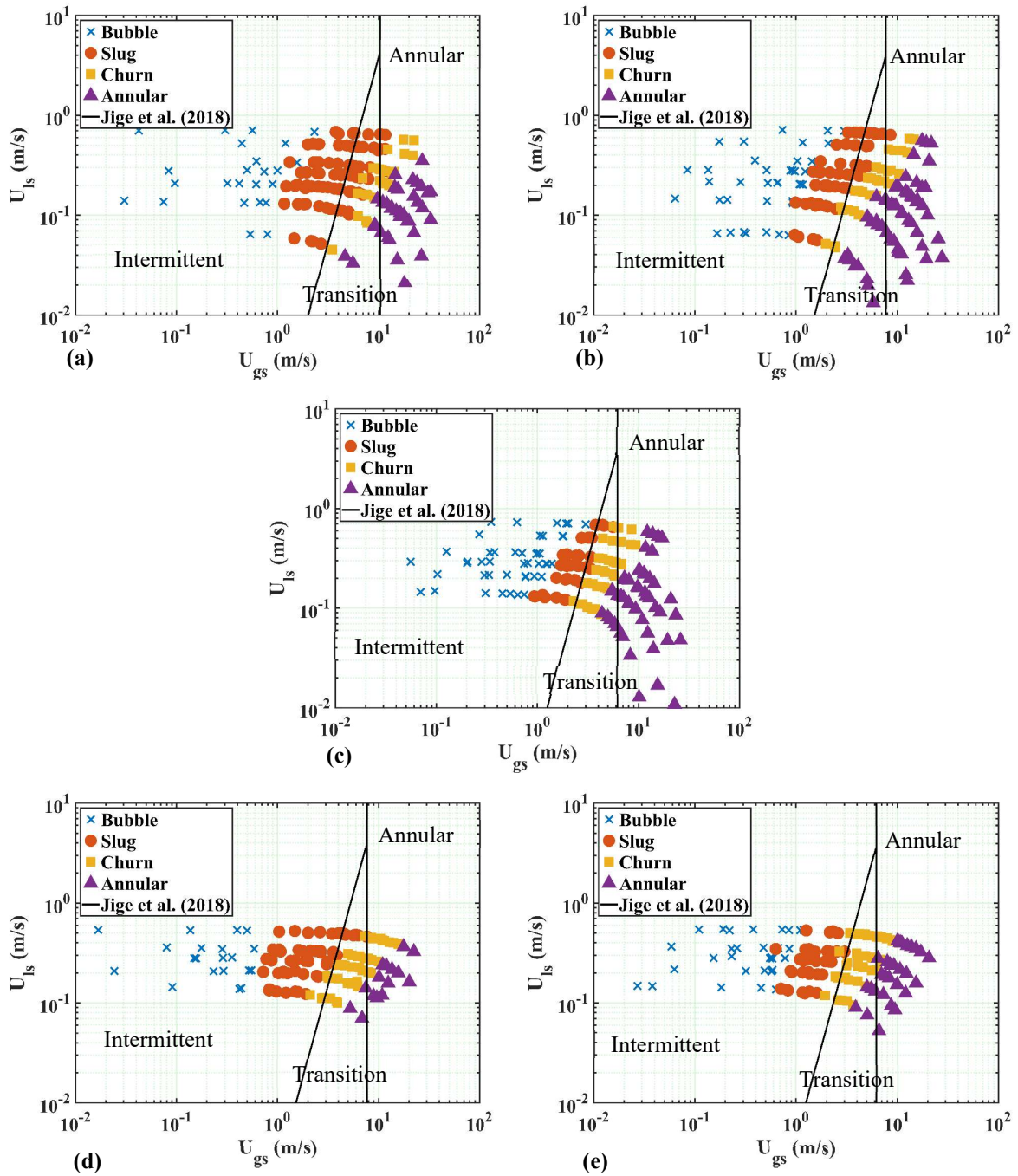


Figure 4.28. Comparison of experimental flow patterns with flow pattern map proposed by Jige et al. [205]: (a) for the 75 mm long test section at 1 bar; (b) for the 75 mm long test section at 1.5 bar; (c) for the 75 mm long test section at 2 bar; (d) for the 30 mm long test section at 1.5 bar; (e) for the 30 mm long test section at 2 bar.

- Shorter heated lengths promoted earlier onset of slug flow, because the same vapour qualities were achieved at higher local heat fluxes in the 30 mm channel compared to the 75 mm channel. Therefore, bubbles grew more rapidly and coalesced easier, reaching confinement at lower vapour qualities. Visualisations at equal vapour qualities revealed that bubbles were significantly larger in the shorter channel, confirming that the transition boundary is strongly affected by heated length and heat flux, factors not fully accounted for in most existing transition models.
- The transition from slug to churn flow was governed by the progressive elongation and destabilisation of confined vapour slugs. As the heat flux increased, individual slugs lengthened and successive structures merged, generating strong interfacial shear and deformation at the liquid–vapour boundary. These instabilities eventually caused the collapse of the coherent slug interface, producing the chaotic phase distribution characteristic of churn flow. The flow maps showed that, at higher mass fluxes, this transition occurred at lower values of the boiling number-dimensionless saturated length parameter, suggesting that reduced effective vapour development within the saturated region was sufficient to destabilise the slug structures under high-inertia conditions.
- The churn to annular transition was associated with the development of a continuous vapour core, which displaced the liquid to the channel walls and produced a thin liquid film. The formation of the liquid film was attributed to the formation of a stable annular structure through vapour acceleration at higher liquid velocities. Increasing mass flux shifted the transition to higher absolute heat fluxes but lower boiling numbers. However, the required vapour velocity was still increased for higher mass fluxes as sufficient phase momentum was required to push the liquid phase to the channel wall.
- Overall, existing flow pattern maps and correlations demonstrated limited predictive capability for microscale rectangular channels under the present experimental conditions. While some captured individual transition trends, none adequately accounted for the combined effects of mass flux, heat flux, heated length, pressure, and geometry. Mechanistic influences such as confinement, interfacial shear, evaporation momentum, and viscous drag were shown to be critical in determining regime transitions but were not consistently represented in current models.
- A modification of the analytical bubbly to slug transition boundary correlation by Mahmoud and Karayiannis [82] was proposed that included an attempt to consider the viscous drag acting on a confined bubble. The modification enhanced the performance of the correlation but did not accurately capture the effect of heated lengths or channel inlet pressure.

CHAPTER 5

HEAT TRANSFER RESULTS

5.1 INTRODUCTION

The heat transfer results for channel inlet pressures of 1 bar, 1.5 bar and 2 bar are presented in this chapter for the 75 mm long channel test section, where mass fluxes between 100 kg/m²s and 1000 kg/m²s and heat fluxes up to 204.75 kW/m² were investigated. The heat transfer results were correlated to the flow visualisation presented in Chapter 4 to explain the flow boiling heat transfer mechanisms. The flow boiling curves are presented and discussed and presented in Section 5.2. The parametric effects, such as mass flux, heat flux and channel inlet pressure, are then presented in Sections 5.3 through 5.6. The effect of heated length was then examined in Section 5.7 by comparing results to the 30 mm long channel test section, where mass fluxes between 200 kg/m²s and 750 kg/m²s and heat fluxes up to 435.00 kW/m² were studied for channel inlet pressures of 1.5 bar and 2 bar. Finally, the heat transfer results for both test sections were compared to existing flow boiling heat transfer prediction tools in Section 5.8 and a summary of the chapter conclusions presented in Section 5.9

5.2 BOILING INCIPIENCE

The resultant flow boiling curves at $z/L_{ch} = 0.5$ for the three studied channel inlet pressures in the 75 mm long test section are presented in Figure 5.1. The beginning of the boiling curves demonstrates the transition from single-phase flows to the onset of nucleate boiling (ONB). ONB occurs when the wall surface temperature is greater than the fluid saturation temperature by some degree, but does not necessarily correlate with the beginning of saturated flow boiling, i.e. subcooled flow boiling can occur. The datapoints at the ONB and the previous heat flux are displayed at each channel inlet pressure in Figure 5.1.

A key observation in most experiments is a sudden drop in wall superheat at ONB, a phenomenon known as temperature overshoot (TOS). TOS arises when the heat transfer rate associated with ONB is high enough to cause a significant reduction in wall superheat from the level required to initiate nucleate boiling. The presence or absence of TOS in the experiments can be attributed to the nature of boiling incipience, which involves the activation of vapour embryos in surface cavities leading to bubble nucleation [244]. TOS was observed in other flow boiling studies in which HFE-7100 was investigated [126, 395, 396].

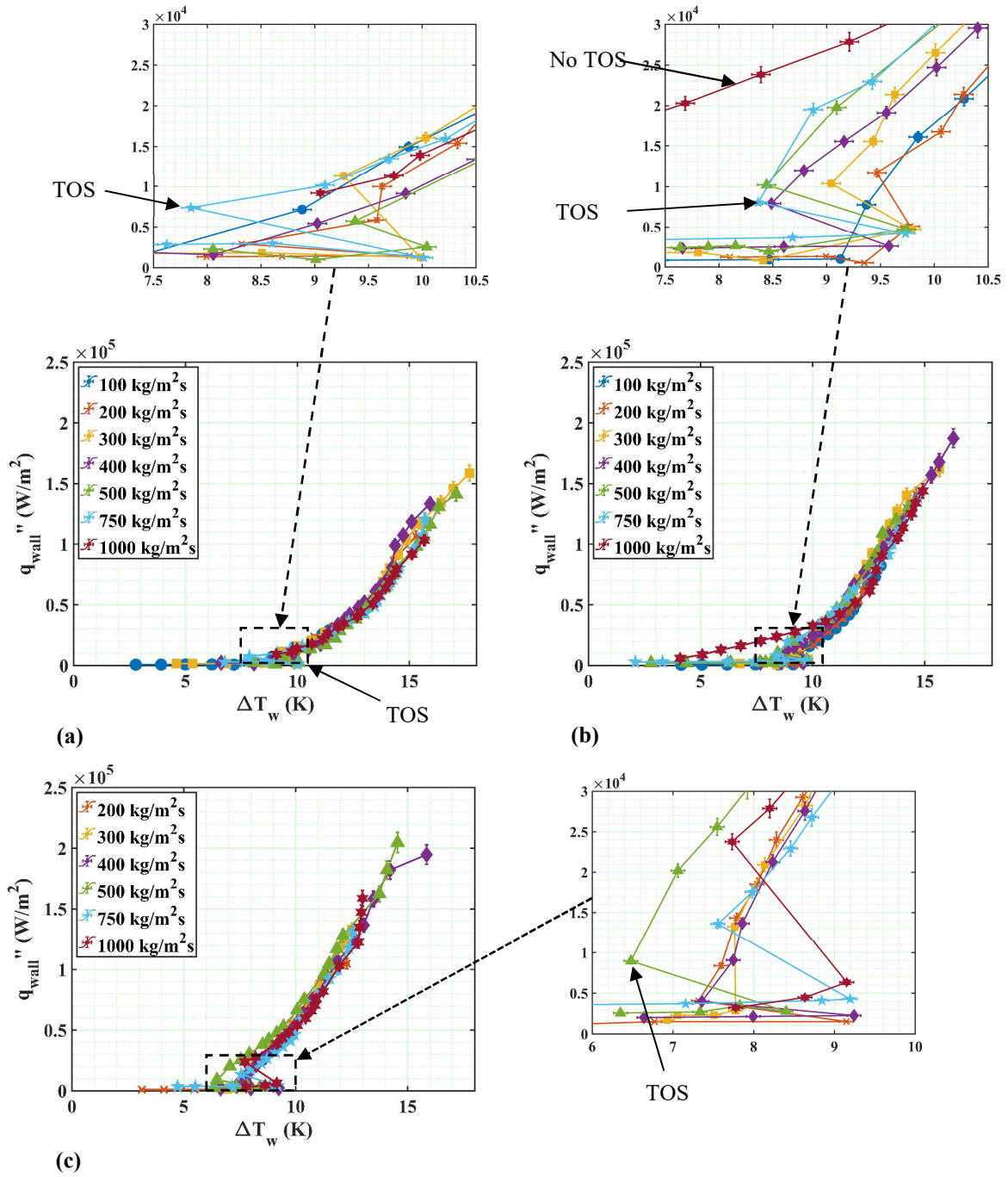


Figure 5.1. Flow boiling curves at $z/L_{ch} = 0.5$ for channel inlet pressures of: (a) 1 bar; (b) 1.5 bar; (c) 2 bar; in the 75 mm long channel test section.

The phenomenon of bubble generation, including bubble initiation, growth and departure from a heated surface is defined as heterogeneous nucleation [397]. As discussed in Chapter 2, it is widely agreed that there is a prerequisite for gas/vapour nuclei entrapment to occur in the surface cavities prior to nucleation. As presented in Chapter 2, Bankoff [245] and Lorenz [246] proposed a gas entrapment criterion for a conical cavity based on the displacement of the gas by the advancing liquid front towards

the cavity. Both agreed that the three-phase contact angle, θ , must be greater than the cavity cone angle, 2β , for gas entrapment, as was shown in Figure 2.34 in Chapter 2. This suggests that higher wetting fluids, i.e. fluids with low contact angles, will only be trapped for smaller cavities. HFE-7100 is a highly wetting fluid, such that accurate measurement of the contact angle becomes difficult; however, it has been assumed as 10° in the literature [397].

Boiling incipience is not only related to whether the cavities are capable of trapping the vapour, but also on the wall superheats at which the cavity containing the vapour embryo is active. Two main types of models have been proposed to predict boiling incipience superheat: namely, phase stability-based models and superheated boundary layer-based models. Phase stability models consider the conditions under which a liquid remains in a metastable state, where liquid is superheated beyond the thermodynamic equilibrium saturation temperature, but the mechanical equilibrium condition is still met [397]. It is proposed that boiling incipience occurs when the liquid then transitions to an unstable state.

Conversely, superheated boundary layer-based models consider the thermal boundary layer at the heated wall and suggest that nucleation occurs when the temperature of the liquid in the boundary layer is equal to or greater than that of the vapour trapped inside the bubble. Hsu [243] predicted the wall superheat required to activate a cavity containing trapped vapour using the superheated boundary layer-based incipience model, proposing the quadratic equation presented in Equation (5.1), below.

$$r_c(max, min) = \frac{\delta_{th}}{2} \cdot \frac{\sin\theta\Delta T_{sup}}{(1+\cos\theta)(\Delta T_{sup}+\Delta T_{sub})} \left[1 \pm \sqrt{1 - \frac{8(1+\cos\theta)\sigma T_{sat}(T_{sup}+T_{sub})}{\delta_{th}\rho_g i_{fg} \Delta T_{sup}^2}} \right] \quad (5.1)$$

In Equation (5.1), $r_c(max, min)$ denotes the upper and lower bounds of cavity mouth radii where a trapped vapour embryo can be activated. Cavities with mouth radii falling within this range will activate at a specific wall superheat. Figure 5.2 illustrates how the required wall superheat varies with cavity mouth radius for HFE-7100 at pressures of 1 bar, 1.5 bar, and 2 bar. θ in Equation (38) refers to the contact angle between the bubble embryo and the boiling surface. According to Hsu, this angle is 53.1° , based on the assumption that the liquid's temperature equals that at the top of the bubble. The thermal boundary layer thickness, δ_{th} , is approximated by dividing the liquid's thermal conductivity by the average single-phase heat transfer coefficient. A 5 K subcooling, representing the average between an inlet subcooling of 10 K and saturation, was used to estimate the liquid-phase thermophysical properties, while vapour properties were assessed at saturation temperature.

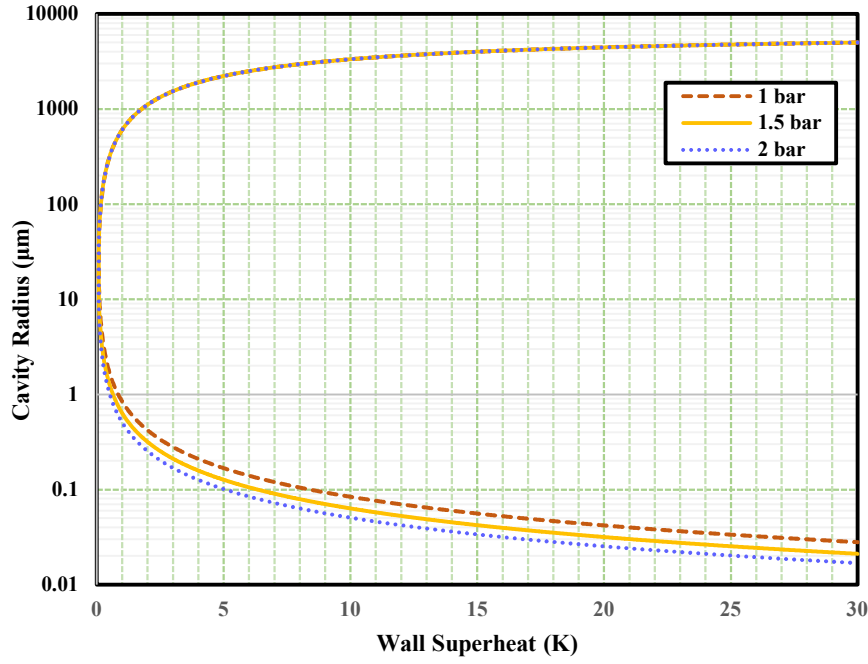


Figure 5.2. Effective nucleation cavity radii with respect to wall superheat according to Hsu [243].

According to the Hsu correlation for required wall superheats, cavities with mouth radii less than 10 μm require larger wall superheats to activate. However, according to the gas entrapment criteria provided by Bankoff [245] and Lorenz [246], it is the cavities with smaller mouth radii for a given cavity depth that will trap vapour. Therefore, greater wall superheats will be required for the small cavities that are prone to trapping vapour for highly wetting fluids such as HFE-7100 which may be the cause of the TOS observed for all pressures.

There may have been some air that entered the system prior to the experiment for the 1000 $\text{kg/m}^2\text{s}$ case at 1.5 bar, which could explain the absence of temperature overshoot (TOS) and the earlier onset of boiling at lower wall superheats. Considering the mechanical equilibrium of the vapour-liquid interface using the Young-Laplace equation, bubble growth of trapped vapour in a cavity occurs when:

$$P_{g,b} > \frac{2\sigma}{r_b} - P_l \quad (5.2)$$

In accordance with Dalton's law, the total pressure within the bubble in a cavity is the sum of the partial pressures of the vapour and non-condensable gas components, i.e. the HFE-7100 vapour and air. Here, the partial pressure is defined as the pressure that the gas would exert if it was alone in the occupied volume at the same pressure. Therefore, the bubble growth requirement, as described by the Young Laplace equation becomes:

$$P_{g,HFE-7100}(T_g) > \frac{2\sigma}{r_b} + P_l - P_{g,air} \quad (5.3)$$

The partial pressure of the HFE-7100 vapour inside the bubble, $P_{g,HFE-7100}$, matches the saturation pressure for the vapour temperature. As is evident from Equation (5.3), the presence of air reduces the required partial pressure of HFE-7100 vapour and, consequently, the required vapour temperature for bubble growth and nucleation. In such cases in which air was trapped, this may have resulted in early bubble nucleation and a lack of TOS.

Boiling incipience was observed to decrease with increasing system pressure, as shown in Figure 5.1. According to the Hsu correlation, while the system inlet pressure has minimal influence on the required wall superheat for activating larger cavity mouth radii, it significantly reduces the required superheat for smaller cavity mouth radii. This again is evident in Figure 5.2. This results in boiling incipience occurring at lower wall temperatures as pressure increases. Although temperature overshoot (TOS) is still present, it manifests at reduced wall superheats due to the earlier activation of smaller cavities.

5.3 EFFECT OF HEAT FLUX ON HEAT TRANSFER

The local heat transfer coefficients for a mass flux of 400 kg/m²s are plotted against vapour quality in Figure 5.3, showing results for heat fluxes of approximately 25 kW/m², 50 kW/m², 75 kW/m² and 100 kW/m². The figure also shows the flow patterns that are prevalent at each vapour quality. The figure clearly indicates that the heat transfer coefficient increases with an increase in heat flux. This has been seen for other studies in which flow boiling with HFE-7100 was investigated [40, 223].

In the subcooled region, it is noticeable that the wall heat flux still has an effect on the local heat transfer. Such a difference would be unexpected for single-phase subcooled flows; however, subcooled boiling also occurred in this region. The added agitation of the bulk fluid due to bubbles departing the surface will increase the local heat transfer coefficient in comparison to purely single-phase flows. Furthermore, the nucleation site density and bubble generation frequency rise as the heat flux is increased. The increased agitation of the increased number of bubbles departing the surface in the subcooled region therefore increased the local heat transfer coefficients, even in the subcooled region.

In the saturated region, the heat transfer coefficients were observed to decline with increasing local vapour quality, but they rose significantly at higher wall heat fluxes. This trend was also observed by [40, 223]. This trend can be related to the flow regimes discussed in Chapter 4. As the flow moves beyond the bubbly regime, there is less agitation due to bubbles departing the surface and heat transfer is primarily governed by the liquid film. Furthermore, slug flow was also prone to the appearance of dry spots which reduce the local heat transfer efficiency. However, the persistent dry nucleation in the liquid film meant that nucleation where spots did not appear was increased, thus increasing the overall HTC but reducing it in comparison to pure bubbly flow.

It should be noted that the experiments for the 75 mm long test section were not taken to CHF due to the physical limitations of the experimental facility, outlined in Section 3.8.3. Lee et al. [241] found that the local heat transfer coefficient began to increase again in the annular flow regime at high heat fluxes. This was attributed to a change in heat transfer mechanism from nucleate boiling to convective boiling heat transfer. They argued that thin-film evaporation was the main driver of heat transfer and the channel exit. Consequently, the thinner film thickness reduced the thermal resistance across the film and increased the heat transfer coefficient. This phenomenon was not observed in this study; however, the heat flux may not have been sufficiently high as bubble nucleation in the liquid film was observed for all flow patterns. This signifies that the liquid film thickness did not diminish enough to suppress bubble nucleation and was therefore not thin enough to enhance heat transfer through thin film evaporation.

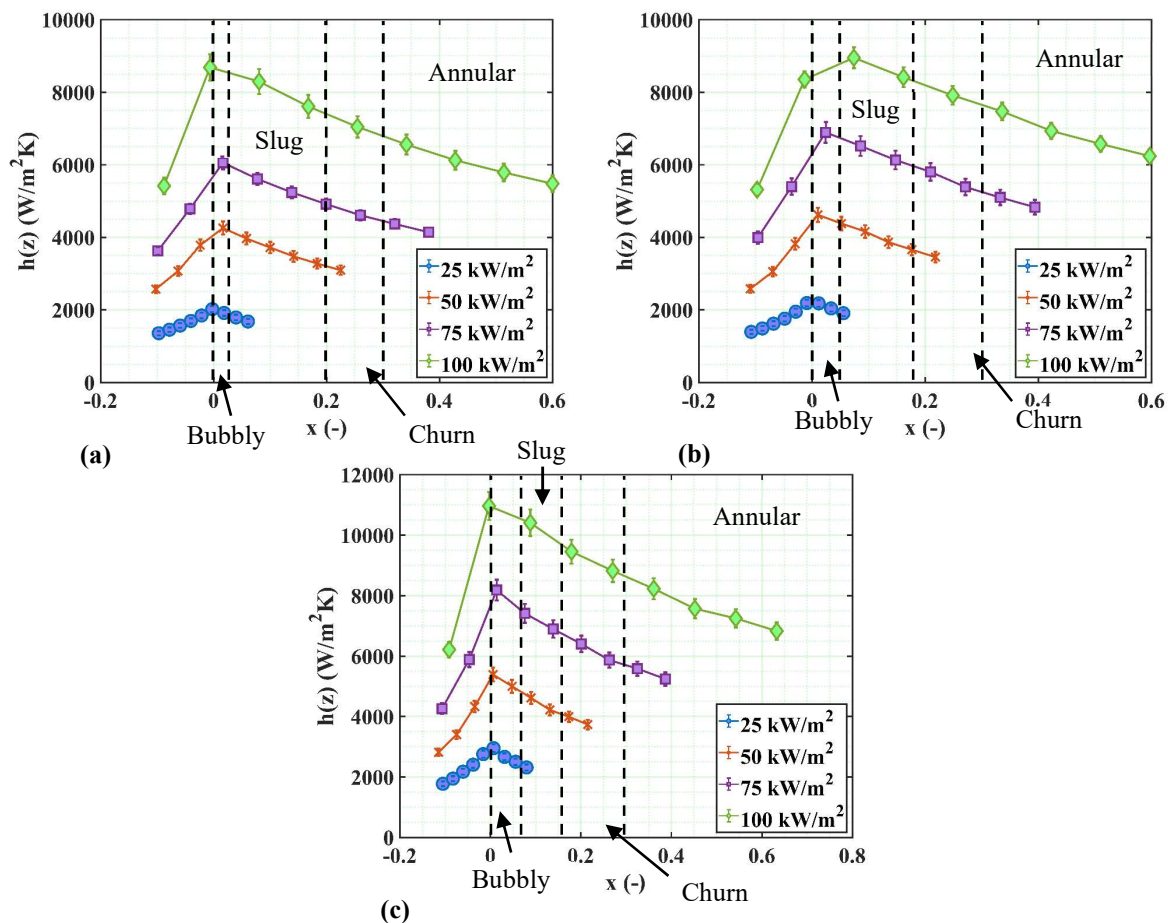


Figure 5.3. Local heat transfer coefficient and vapour quality for a mass flux of 400 kg/m²s: (a) at 1 bar; (b) at 1.5 bar; (c) at 2 bar; in the 75 mm long channel test section.

Figure 5.4 also illustrates the effect of wall heat flux on heat transfer through a plot of local heat transfer coefficient against dimensionless length. It shows that the local heat transfer coefficient was greater at all points along the channel as the wall heat flux increased. As expected, the local heat transfer

coefficient maximum, previously identified at the transition to saturated bubbly flow in Figure 5.3, occurs closer to the channel inlet at higher wall heat fluxes. This shift is associated with the earlier onset of saturated boiling, resulting from increased vapour generation and nucleation activity at elevated heat fluxes. In the subcooled region, the higher wall superheat promotes more intense subcooled boiling, with a greater number of active nucleation sites and greater bubble departure rates. This enhances local mixing and heat transfer, leading to increased local heat transfer coefficients upstream of the saturation point. As the heat flux increases, the subcooled region shortens and the intensity of subcooled boiling rises, producing the observed increase in local heat transfer in this part of the channel.

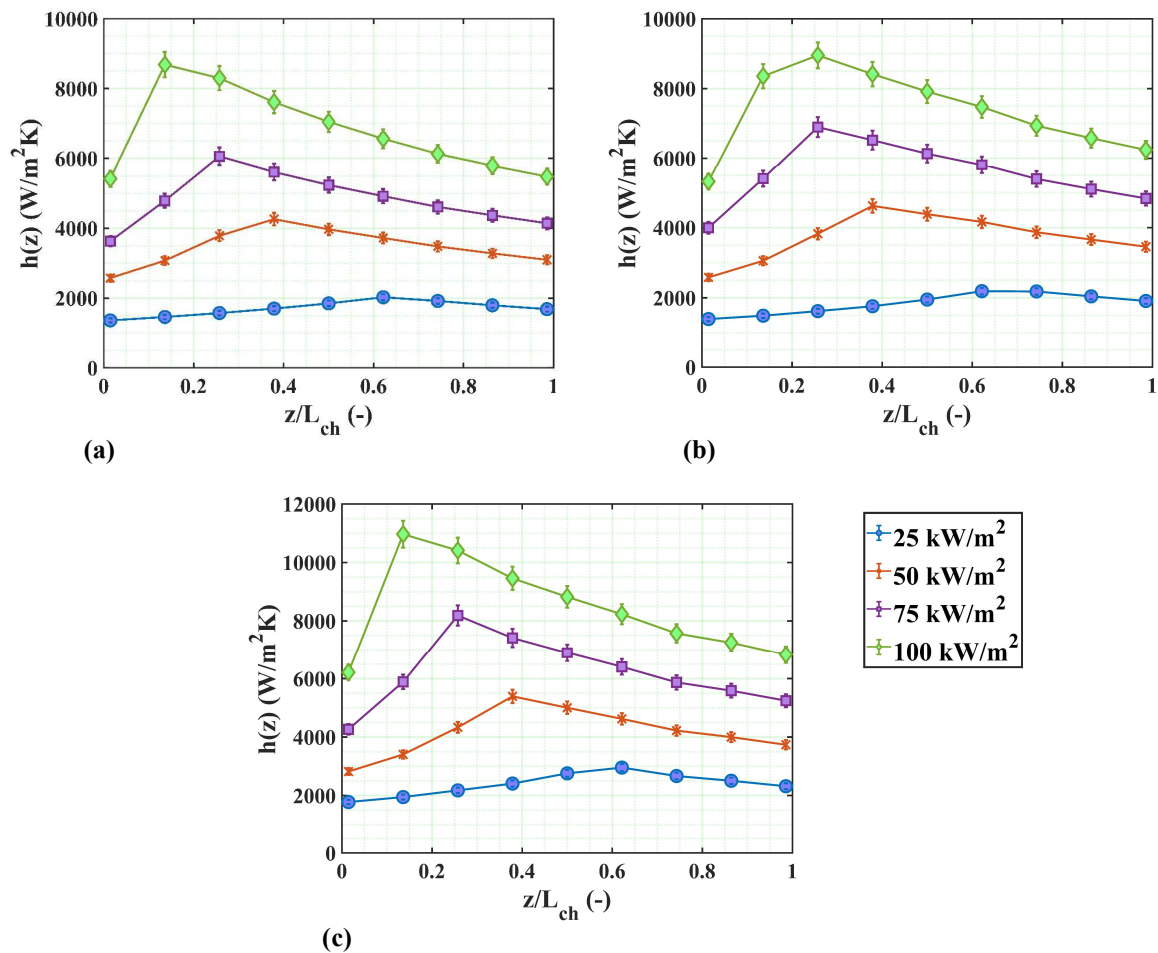


Figure 5.4. Local heat transfer coefficient and dimensionless length for a mass flux of 400 kg/m²s: (a) at 1 bar; (b) at 1.5 bar; (c) at 2 bar; in the 75 mm long channel test section.

5.4 EFFECT OF MASS FLUX ON HEAT TRANSFER

The average two-phase heat transfer coefficients are plotted against the wall heat flux in Figure 5.5 for the three channel inlet pressures. The figures show that there is no clear effect of mass flux on the average two-phase heat transfer coefficient. This phenomenon was also reported by other researchers

in the literature [127, 223, 398]. However, other researchers noticed a minimal impact of mass flux on heat transfer coefficients at lower heat fluxes in the nucleate boiling regime [129, 134, 217], but convective heat transfer dominates as the heat flux increases which causes an increase in heat transfer at higher mass fluxes. The limitations of the experimental facility, in which the experiments were concluded at the point in which a constant channel inlet pressure could no longer be maintained, may have meant that the scenario in which mass flux begins to increase heat transfer through convective effects was not observed. Therefore, it can only be concluded that the nucleate boiling mechanism appears to dominate for the range of heat and mass fluxes investigated in the 75 mm long channel test section.

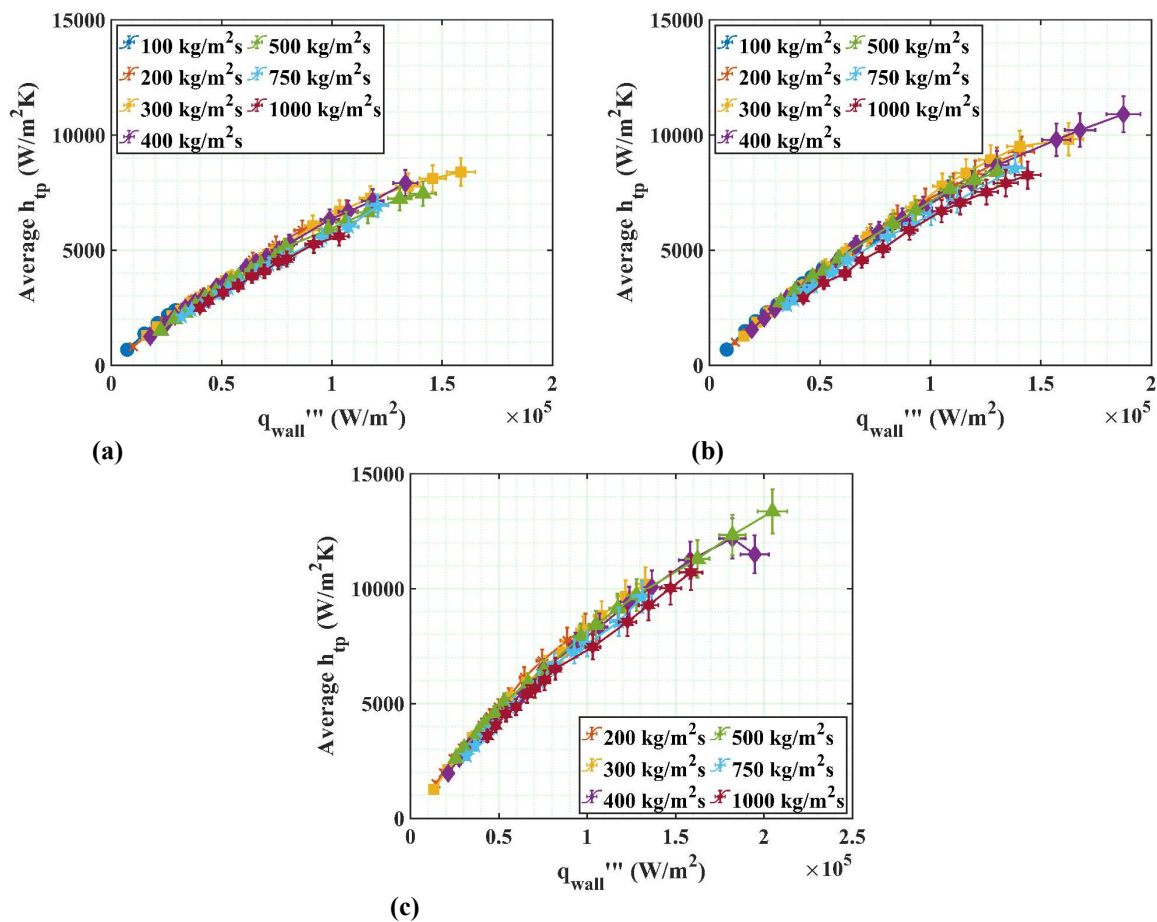


Figure 5.5. Average two-phase coefficient and wall heat flux for a heat flux of $50 kW/m^2$: (a) at 1 bar; (b) at 1.5 bar; (c) at 2 bar in the 75 mm long test section.

The negligible mass flux effect can also be observed in Figure 5.6, which plots the local heat transfer coefficient against dimensionless length. Figure 5.6 also shows that the increase in mass flux will result in a larger portion of the channel being in the subcooled region. This leads to a decrease in the local heat transfer coefficient, meaning it is lower than the values observed at lower mass fluxes where boiling occurred.

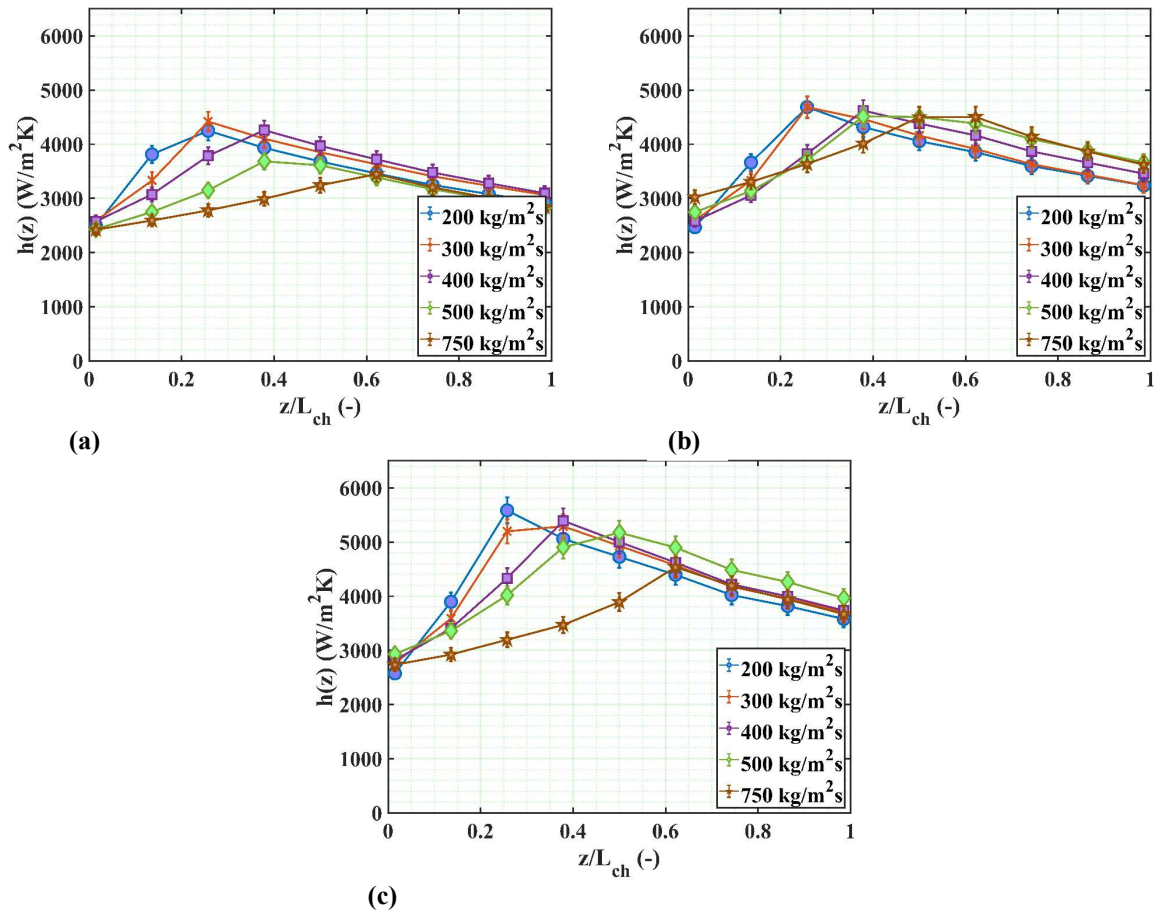


Figure 5.6. Local heat transfer coefficient and dimensionless length for a heat flux of 50 kW/m^2 : (a) at 1 bar; (b) at 1.5 bar; (c) at 2 bar in the 75 mm long test section.

If the average heat transfer coefficient is instead calculated over the entire heated length, including both the subcooled and saturated regions, a different trend is observed. When the local heat transfer coefficient is integrated along the channel length, the overall average heat transfer coefficient decreases with increasing mass flux. This behaviour is attributed to the axial redistribution of the boiling region. As shown in Figure 5.6, higher mass fluxes extend the subcooled region and reduce the proportion of the channel operating under saturated boiling conditions. Since the highest local heat transfer coefficients occur in the saturated region, the reduced two-phase length at higher mass flux leads to a lower overall average heat transfer coefficient, even though the local two-phase heat transfer coefficient remains largely unaffected by mass flux.

While mass flux had a minimal impact on heat transfer, it did slightly influence the wall temperature, especially at lower system pressures. This is illustrated in Figure 5.7, which shows both local fluid and wall temperatures. The two-phase pressure drop was observed to decrease with higher channel inlet pressure and increase with greater mass flux. As a result, the local fluid temperature dropped more significantly along the channel at higher mass fluxes. Consequently, the wall temperature was lower,

even though the local heat transfer coefficient remained mostly unchanged in the two-phase region. This effect diminished at higher channel inlet pressures due to the reduced two-phase pressure drop.

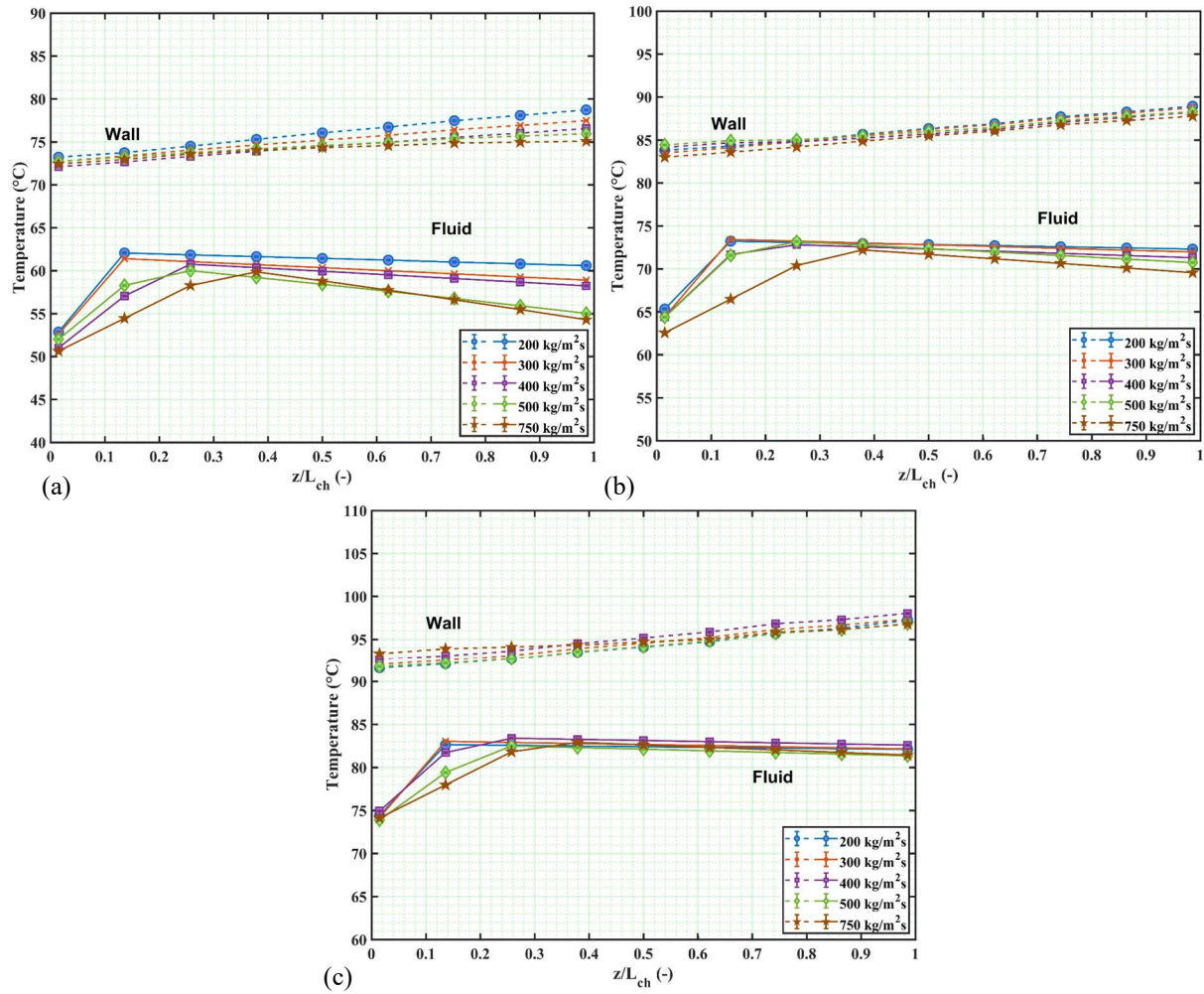


Figure 5.7. Effect of mass flux on local wall and fluid temperatures for a heat flux of 100 kW/m^2 : (a) at 1 bar; (b) at 1.5 bar; (c) at 2 bar in the 75 mm long test section.

5.5 EFFECT OF CHANNEL INLET PRESSURE ON HEAT TRANSFER

Figure 5.8 presents the average two-phase heat transfer coefficients against wall heat flux at the three channel inlet pressures for a mass flux of $500 \text{ kg/m}^2\text{s}$. It should be noted that whilst the channel inlet subcooling was kept at a constant 10 K, this required a change in channel inlet temperature to maintain this inlet subcooling for the three channel inlet pressures. The inlet temperatures were $50.67 \text{ }^{\circ}\text{C}$, $63.35 \text{ }^{\circ}\text{C}$ and $73.08 \text{ }^{\circ}\text{C}$ for channel inlet pressures of 1 bar, 1.5 bar and 2 bar, respectively. The figure shows that the average two-phase heat transfer coefficient was found to increase with system pressure. This increase in average two-phase heat transfer coefficient is small but apparent up to wall heat fluxes of $50 \text{ kW/m}^2\text{s}$ but becomes more significantly pronounced at higher heat fluxes.

The constant nature of the bubble nucleation that was observed across all heat fluxes and the absence of any clear mass flux effect indicates that the reason for the increase in heat transfer coefficient with system pressure is likely due to the mechanisms of bubble nucleation that occur. The graphical representation of the Hsu [243] correlation in Figure 5.2 showed that a greater range of cavities that have trapped vapour will be active at the same wall superheats. A comparison of the wall superheats and heat fluxes for the three channel inlet pressures is presented in Figure. 5.9.

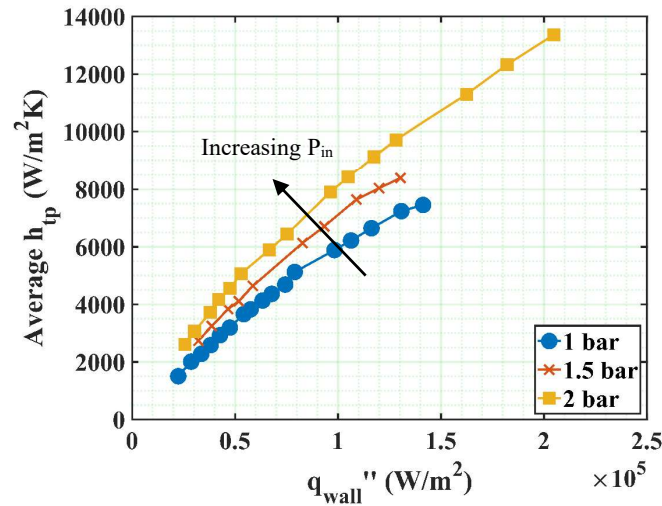


Figure 5.8. Comparison of average two-phase heat transfer coefficients for a mass flux of $500 \text{ kg/m}^2\text{s}$ at different channel inlet pressures in the 75 mm long test section.

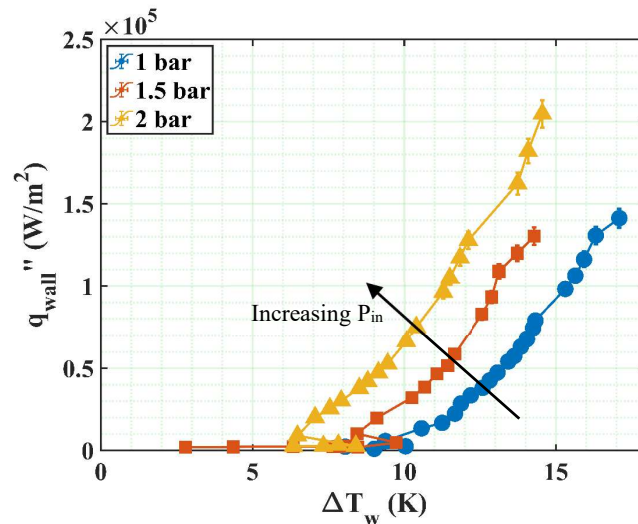


Figure 5.9. Comparison of flow boiling curves at $z/L_{ch} = 0.5$, for a mass flux of $500 \text{ kg/m}^2\text{s}$ at different channel inlet pressures in the 75 mm long test section.

The boiling curve presented in Figure 5.9, as previously discussed in Section 5.2, shows that boiling incipience occurs at lower wall superheats for higher system pressures. However, this does not help to

explain the significantly lower wall temperatures that occur at the onset of bubble nucleation and thus, greater heat transfer coefficients. Furthermore, the higher wall superheats at which boiling incipience occurs for the lower system pressures would result in a smaller difference in the range of cavities between the system pressures, according to Hsu. Whilst the Hsu correlation provides a description of whether the cavity can become active, the resultant lower heat transfer coefficients after boiling has begun will be dependent on whether the cavity can remain active.

Marto and Rohsenow [399] proposed a mechanism of nucleation site stability that was used to explain large fluctuations in wall temperature from experiments using the boiling of liquid metals in cylindrical cavities. They argued that vapour residuals are left at the cavity mouth when a bubble departs from a nucleation site. The departure of the bubble from the nucleation site has resulted in a lower wall surface temperature that may allow some of the vapour residuals to condense back into the cavity and penetrate the cavity. This condensation then reheats the cavity wall and the condensation effect ceases at a depth of x^* , referred to as the maximum penetration depth. This process is shown in Figure 5.10, where Figure 5.10a shows the vapour height at $t = 0$, soon after the bubble departs the cavity, and Figure 5.10b presents the vapour height at $t = t^*$, where the maximum penetration depth has been achieved prior to regrowth of the vapour embryo. Kosky [400] observed the same phenomena when studying the boiling of deionised water in a capillary tube.

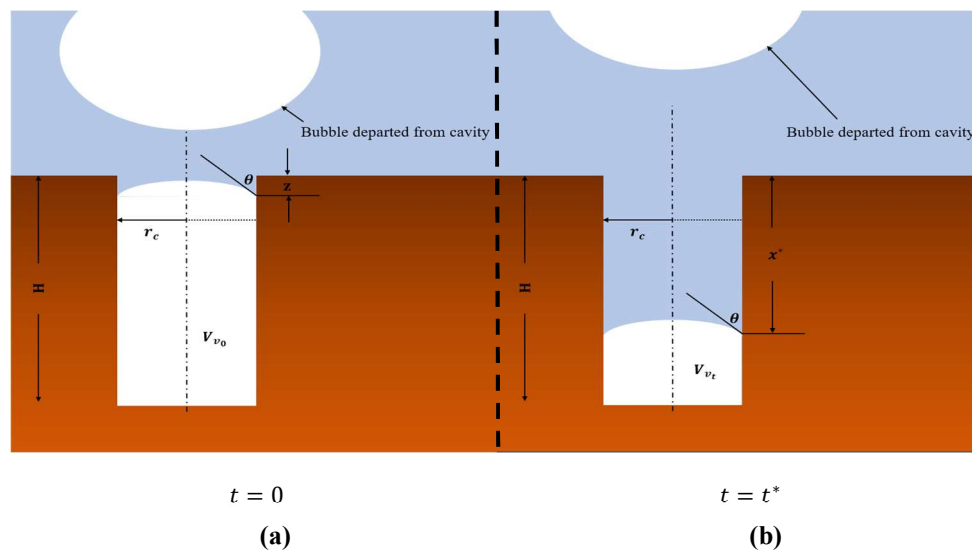


Figure 5.10. Vapour residual heights in cavities at: (a) time of bubble departure from cavity and (b) at time of maximum liquid penetration depth.

Marto and Rohsenow [399] therefore argued that a nucleation site would become unstable if the cavity depth, represented as H in Figure 5.10, was less than the maximum penetration depth, x^* . A theoretical criterion was provided by considering the transient heat conduction equation and the motion of the liquid vapour interface inside the cavity for contact angles less than $\pi/2$ (90°), which is likely

the case for a highly wetting such as HFE-7100. The criterion for a stable cavity as proposed by Marto and Rohsenow is presented in Equation (5.4),

$$x^* > \frac{4}{\pi} \left(\frac{T_{sat}^2}{\rho_v^3 i_{fg}^3} \right) \left(\frac{2\sigma \cos\theta}{r_c} \right) (1 + 0.5\varepsilon)^2 K^2 \left[\frac{\sqrt{\rho_w c_w k_w} \sqrt{\rho_l c_l k_l}}{20q''(1 + \sin\theta)} \right] \quad (5.4)$$

$$\text{where } K = \frac{aq''}{k_w \left(q'' \sqrt{\frac{f}{\alpha_w}} \right)^b}$$

where a , b , and ε were provided empirical parameters and f is the bubble generation frequency. The empirical parameters a and b were based on the wall surface material and the root mean square (RMS) surface roughness. Values of 41.2 for a and 0.706 for b were provided for aluminium with an RMS surface roughness of 13 μm . A value for a of 5.865 and a value for b of 0.63 were also provided for stainless steel with an RMS surface roughness of 19 μm . These values were taken from experiments investigating the transient local surface temperature during pool boiling of water by Hsu and Schmidt [401]. Finally, ε was an estimation of the wall temperature rise during the bubble waiting period, where Marto and Rohsenow recommended a value of 0.3.

The relationship between cavity mouth radius and the required cavity depth for sustained activation, based on Marto and Rohsenow [399] is shown in Figure 5.11 for HFE-7100. As is evident in the figure, the required cavity depth for a given cavity mouth radius becomes smaller as the channel inlet pressure increases. This indicates that a wider range of cavities will be stable for a given heat flux, likely leading to a greater nucleation site density and heightened agitation of the bulk fluids as bubbles depart the surface. As evident in the figure, the required cavity depth for a given cavity mouth radius decreases as the channel inlet pressure increases. This suggests that, at higher pressures, a wider range of surface cavities would remain stable and active under the same thermal conditions. However, it is difficult to relate this relationship directly to the present test section, since doing so would require precise measurements of the cavity mouth radii and depths on the heated surface. Such measurements were not available for the current experiments. Nevertheless, the analysis of Marto and Rohsenow indicates that higher pressures should promote a greater number of stable active cavities. This is consistent with the increased nucleation activity and higher heat transfer coefficients observed at elevated pressures in Figure 5.8.

The overall channel and two-phase pressure drop was found to decrease with an increase in channel inlet pressure, as is discussed in Chapter 6. Therefore, the local fluid pressure towards the end of the channel is not only greater because of the initial higher channel inlet pressure, but also due to the overall lower pressure drop. This is shown in Figure 5.12, which depicts the local fluid pressure along the channel for the three channel inlet pressures at a mass flux and heat flux of 500 $\text{kg/m}^2\text{s}$ and 130 kw/m^2 ,

respectively. An increase in wall heat flux also results in an increase in two-phase pressure drop, which again is presented in Chapter 6. Therefore, the results shown in Figure 5.11 would be further exacerbated at the end of the channel with increasing heat flux due to the greater differential in local pressure and the potential difference in range of active cavities even more significant. The heat transfer coefficient would therefore show a more significant difference between the channel inlet pressures at higher wall heat fluxes as observed in Figure 5.8. This would be true so long as bubble nucleation is not suppressed and still present, as was observed in the current study for the 75 mm long channel test section.

Furthermore, a detailed investigation into bubble nucleation site density, generation frequency and departure diameter was recently conducted by Al-Zaidi et al. [242] for HFE-7100 in a micro-gap heat exchanger. Not only did they find the nucleation site density increased at higher system pressures, as indicated from the above discussion, but they also found that bubble generation frequency also increased. They attributed this phenomenon to the reduced surface tension, which encourages bubble departure through the resultant force balance. The increased bubble generation frequency from the already greater number of active cavities further contributes to agitation of the bulk fluid and higher heat transfer for greater system pressures when nucleate boiling is present.

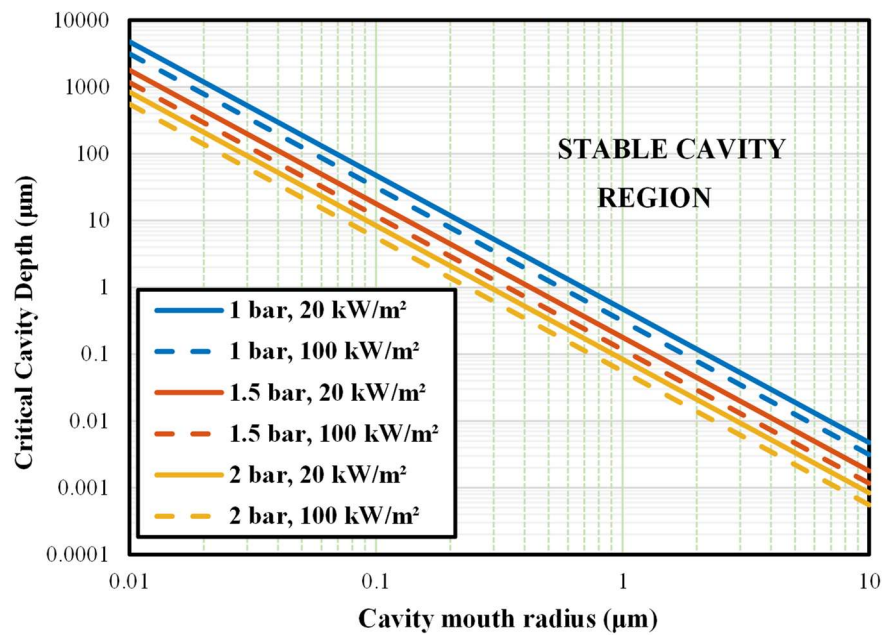


Figure 5.11. Required cavity depth at a given cavity mouth radius for stable cavity activation for HFE-1700 according to Marto and Rohsenow [399].

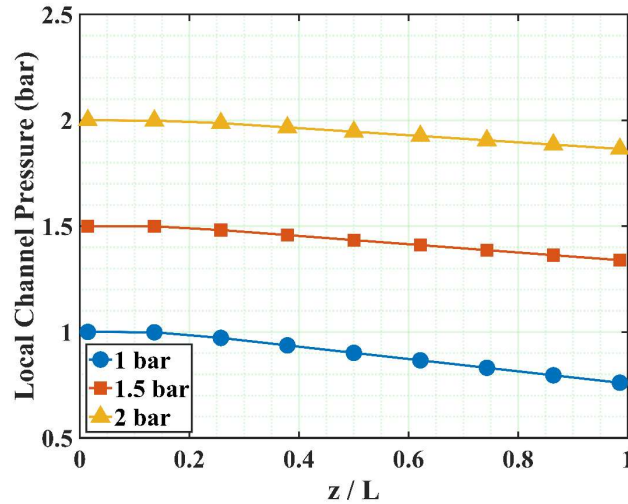


Figure 5.12. Local channel pressure at mass and heat fluxes of $500 \text{ kg/m}^2\text{s}$ and 130 kW/m^2 , respectively.

5.6 EFFECT OF HEATED LENGTH ON HEAT TRANSFER

The flow boiling curves for the 30 mm long channel test section at $z/L_{\text{ch}} = 0.5$ are displayed in Figure 5.13; however, only experiments for 1.5 bar and 2 bar inlet pressures were investigated in this channel due to the experimental limitations outlined in Chapter 3. The shorter test section also showed TOS prior to boiling incipience, the same as the 75 mm long channel. The shorter nature of the test section allowed the test section to be pushed to CHF for all mass fluxes, without any experimental issues regarding channel inlet pressure. This results in a greater range of observed wall heat fluxes for the 30 mm long test section.

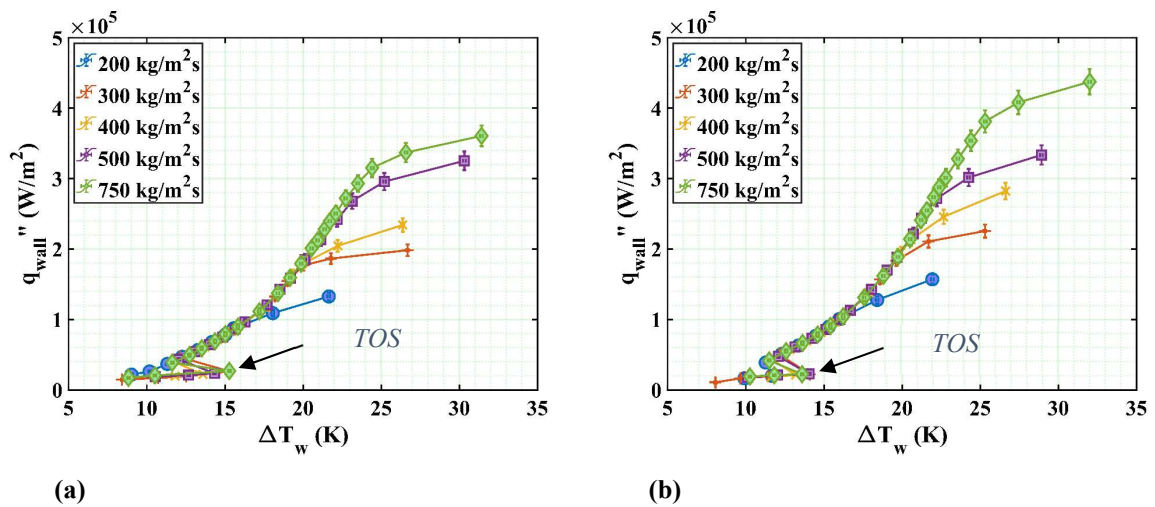


Figure 5.13. Flow boiling curves at $Z/L_{\text{ch}} = 0.5$ for channel inlet pressures of: (a) 1.5 bar; (b) 2 bar; in the 30 mm long channel test section.

It is noticeable that the wall superheat required for boiling incipience was higher for the 30 mm long channel test section than for the 75 mm long channel test section. The reason for this is unclear; however, this is likely due to any differences in surface structure and, in particular, cavity distribution. It may be that the distribution of cavity mouth radii and depth differ from the 75 mm long test section, where the 30 mm long channel test section possesses cavities with small cavity mouth radii. According to Hsu [243], this would require greater wall superheats to activate cavities in the 30 mm long test section. However, this phenomenon becomes difficult to prove as the cavity mouth radii range in discussion is in the region of $0.1\ \mu\text{m}$, i.e. microcavities. Cavities of this size are outside the range of the surface analysis conducted in Chapter 3 as typical surface metrology focuses on roughness for component finish rather than investigation of cavities for the purpose of boiling heat transfer.

A comparison between the average two-phase heat transfer coefficients for a mass flux of $500\ \text{kg/m}^2\text{s}$ and the range of heat fluxes investigated in the 75 mm long channel test section is presented in Figure 5.14. The results show that the average two-phase heat transfer coefficient decreases with channel length for a given applied wall heat flux. One may expect that the opposite would be true, as a greater percentage of the two-phase length in the 75 mm long channel is occupied by flow patterns other than bubbly flow for a given wall heat flux. A two-phase length occupied primarily by bubbly flow would be expected to provide a greater heat transfer coefficient, as it is the flow pattern from which the greatest local heat transfer coefficients are observed.

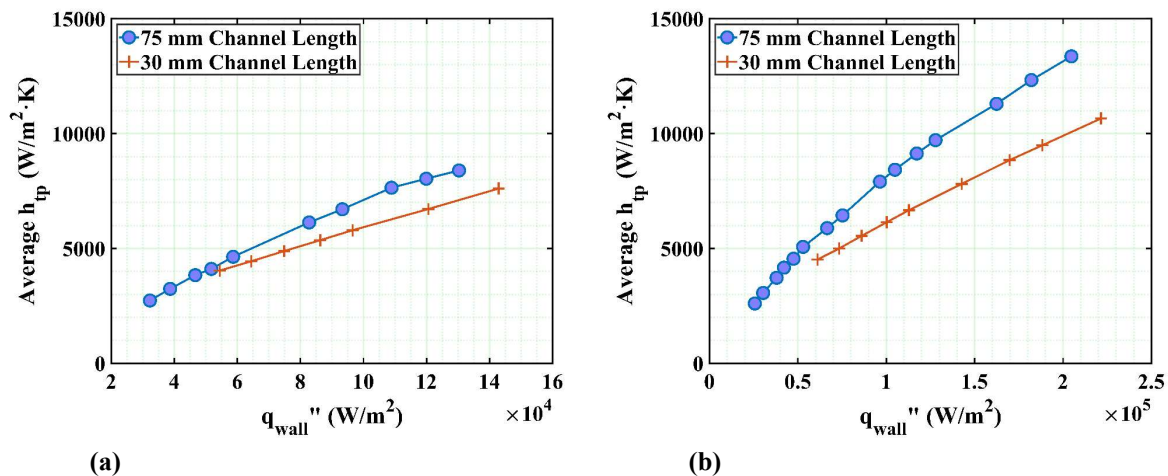


Figure 5.14. Effect of heated length on average two-phase HTC with wall heat flux at $500\ \text{kg/m}^2\text{s}$ for; (a) 1.5 bar inlet pressure; (b) 2 bar inlet pressure.

A constant inlet subcooling of 10 K was used for all experiments in both test sections. This results in a larger percentage of the channel length being occupied by the subcooled region in the shorter test section than in the longer test section. As a consequence, a greater portion of the applied heat in the

shorter channel is transferred under subcooled boiling conditions. The intensity of subcooled boiling increases with wall heat flux, as higher wall superheats activate more nucleation sites and promote more vigorous bubble growth and collapse within the liquid. This enhanced subcooled boiling can significantly influence the overall heat transfer behaviour, particularly in the shorter channel where it occupies a larger fraction of the heated length. Therefore, another useful comparison is the average two-phase heat transfer coefficient with outlet vapour quality, which is proportional to the heat into the channel.

The comparison of the average two-phase heat transfer coefficient for both test sections with outlet vapour quality is presented in Figure 5.15. The percentage of the channel length that is in the subcooled region is the same for both test sections, as is the heat applied to the channel. Figure 5.15 shows a significant increase in average two-phase heat transfer coefficients. An obvious explanation for this is that the wall heat flux required to achieve the same outlet vapour quality will be more than twice as high for the shorter channel test section. As per the definition of wall heat flux, the heat applied to the test section will be more concentrated and the length of the bubbly flow region will be lower, with the same amount of mass transfer. This results in more aggressive bubble nucleation and higher heat transfer coefficients. The phenomenon then holds true for the remainder of the flow patterns due to the appearance of bubble nucleation in the film for almost all datapoints, which is discussed in Chapter 7.

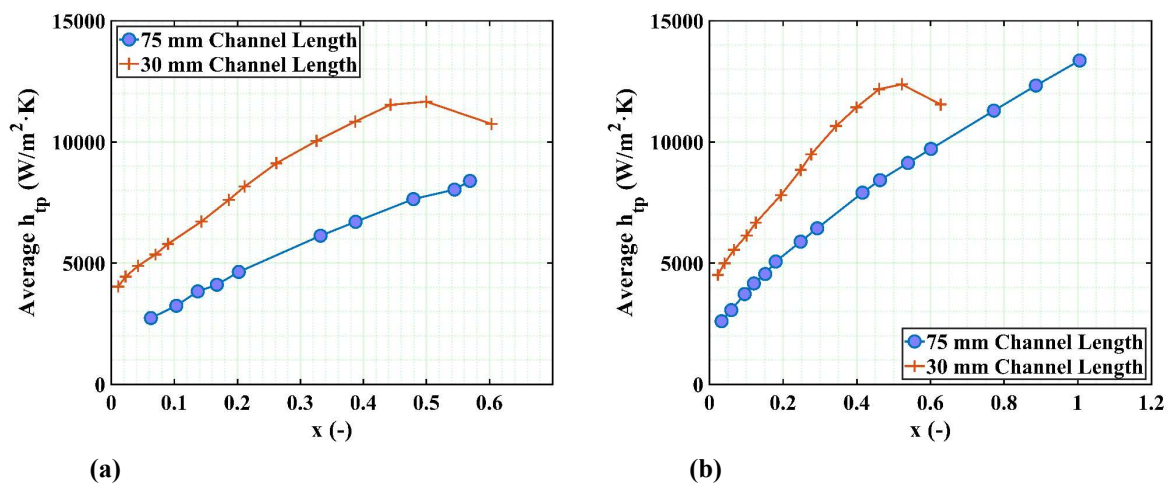


Figure 5.15. Effect of heated length on average two-phase HTC with outlet vapour quality at 500 kg/m^2s for; (a) 1.5 bar inlet pressure; (b) 2 bar inlet pressure.

5.7 COMPARISON TO EXISTING HEAT TRANSFER PREDICTION TOOLS

This section provides a comparison between the average two-phase heat transfer data obtained from the current flow boiling experiments and a selection of correlations developed for both conventional and microchannel applications. These selected correlations form the basis for evaluating how well the experimental findings align with previously established methods. An overview of each correlation,

including its origin and application context, is outlined in Chapter 2, whilst the correlations themselves are presented in Appendix A.

Some of the selected correlations were originally formulated for channels where all surfaces are subjected to uniform heating. Since the current experimental investigation was carried out using a single microchannel configuration in which the top wall remained unheated, a correction factor was necessary to account for this deviation in thermal boundary conditions. The adjustment defined by Equation (5.5) was applied to accommodate this, consistent with the methodology followed in earlier research efforts [2, 40, 402].

$$h_{tp,mod} = \left(\frac{Nu_3}{Nu_4} \right) h_{tp} \quad (5.5)$$

In Equation (5.5) $h_{tp,mod}$, is the modified two-phase heat transfer coefficient, Nu_3 and Nu_4 are the thermally developed and developing single-phase laminar flow Nusselt number for a channel with three and with four heated sides, respectively, and h_{tp} the predicted two-phase heat transfer coefficient.

Equations (5.6) and (5.7) present the Nusselt numbers corresponding to thermally developed flow in channels with three and four heated walls, respectively. These values were originally proposed by Shah and London [382].

$$Nu_3 = 8.235(1 - 1.833\alpha^* + 3.767\alpha^{*2} - 5.814\alpha^{*3} + 5.361\alpha^{*4} - 2\alpha^{*5}) \quad (5.6)$$

$$Nu_4 = 8.235(1 - 2.042\alpha^* + 3.085\alpha^{*2} - 2.477\alpha^{*3} + 1.058\alpha^{*4} - 0.186\alpha^{*5}) \quad (5.7)$$

Equation (5.8) was employed to calculate the MAE% for each correlation. Additionally, the proportion of data points falling within a $\pm 30\%$ deviation from the predicted values, denoted as θ_{30} , was determined.

$$\text{MAE}\% = \frac{1}{n} \sum \left| \frac{X_{exp} - X_{pred}}{X_{exp}} \right| \times 100\% \quad (5.8)$$

5.7.1 Comparison to Shah (1982)

The Shah [268] correlation, given in Appendix A, estimates the flow boiling heat transfer coefficient by selecting the larger of two components: nucleate boiling and convective boiling. The nucleate boiling contribution is expressed as a function of the Boiling number, representing the influence of wall heat flux, while the convective component depends on the Convection number. This correlation was originally developed using experimental data for conventional channels with diameters between 6 and

25.4 mm, and tested with various fluids including water, R-11, R-12, R-22 and R-113 in both vertical and horizontal orientations.

A comparison to the correlation and the experimental data is shown in Figure. 5.16. When applied to the 75 mm long channel test section, the correlation performed poorly. Only 35.2 % of the data points were predicted within $\pm 30\%$ of the measured values, and the mean absolute error was 58.47 %. The mean absolute error increased to 64.56 % for the 30 mm long channel test section, with only 30.84 % of data points within $\pm 30\%$ of predicted values. Since the flow boiling in the microscale channels was observed to be dominated by nucleate boiling mechanisms, one would expect the correlation to perform better by considering only the nucleate boiling component. However, it is clear that the correlation does not capture the relevant microscale effects adequately.

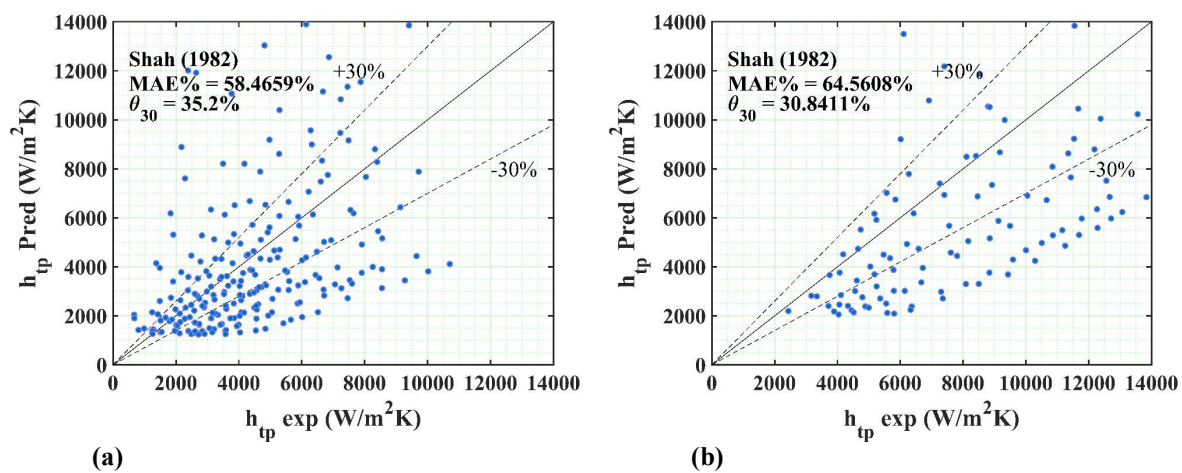


Figure 5.16. Comparison of experimental two-phase HTC to prediction by Shah [268] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

5.7.2 Comparison to Warriar et al. (2002)

The correlation proposed by Warriar et al. [269] and included in Appendix A was developed from experimental flow boiling data for FC-84 in horizontal rectangular multi-microchannels with a hydraulic diameter of 0.75 mm and a heated length of 306.75 mm. The model incorporates vapour quality and the Boiling number, along with a single-phase heat transfer coefficient based on fully developed laminar flow. It was validated within the ranges $0.03 \leq x \leq 0.55$ and $0.00027 \leq Bo \leq 0.00089$.

In the 75 mm long channel test section, the Boiling number ranged from 0.0000136 to 0.0071, with only 55.14 % of the data points falling within the correlation's valid range. The model performed poorly in this case, with only 38 % of the data points predicted within $\pm 30\%$ of the measured values and a MAE% of 48.20 %. This is consistent with the narrow development range of the correlation, which was derived using a single working fluid, fixed geometry and a limited boiling number window.

In contrast, the 30 mm long channel test section had a Boiling number range of 0.00020744 to 0.0075, with just 18.5 % of the data points lying within the valid bounds of the correlation. Surprisingly, despite the lower proportion of valid data, the correlation performed significantly better in this case. As shown in Figure 5.17, it predicted 100 % of the data points within ± 30 % error, with a MAE% of only 13.51 %. This counterintuitive result may be partially explained by the fact that the 30 mm test section operated at lower outlet vapour qualities, which aligns more closely with the validation range of the original correlation. Nevertheless, it remains surprising given the much shorter heated length and differences in fluid properties and geometry.

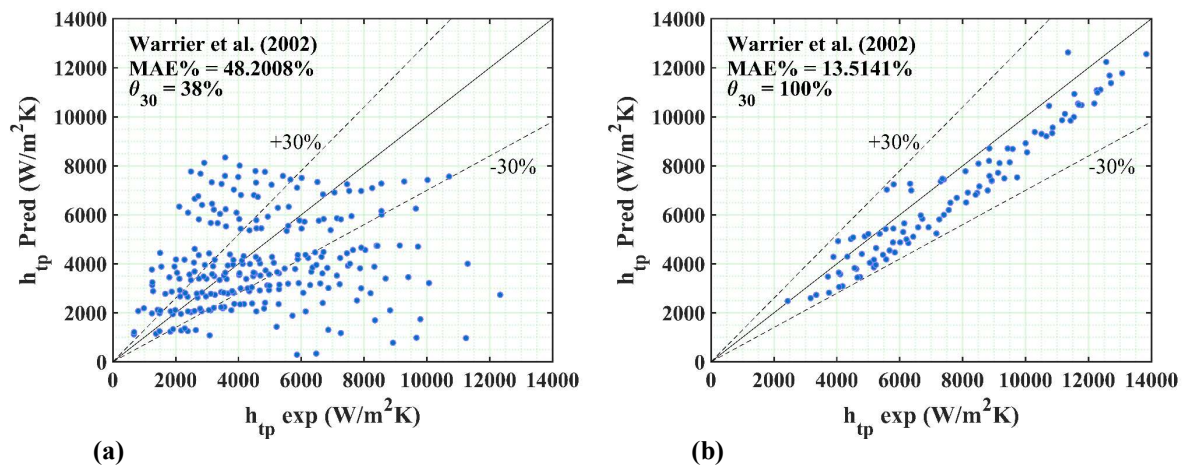


Figure 5.17. Comparison of experimental two-phase HTC to prediction by Warrier et al. [269].

5.7.3 Comparison to Lee and Mudawar (2005)

Lee and Mudawar [266] developed a set of flow boiling heat transfer correlations, which are summarised in Appendix A, based on experimental data obtained using water and R-134a in horizontal parallel rectangular microchannels with a hydraulic diameter of 0.348 mm. The microchannels were fabricated from oxygen-free copper and tested across a heat flux range of 159 kW/m² to 938 kW/m² and mass fluxes between 127 kg/m²s and 654 kg/m²s. Three separate correlations were proposed to account for different dominant boiling mechanisms across vapour quality ranges: $x < 0.05$, $0.05 < x < 0.55$, and $x > 0.55$. Nucleate boiling was reported to dominate at low vapour qualities, while annular film evaporation (i.e. convective boiling) was dominant at moderate and high vapour qualities. The heat transfer coefficient was expressed as a function of several dimensionless parameters, including the liquid only Weber number, boiling number, Martinelli parameter and single-phase heat transfer coefficient. Despite the comprehensive approach, the correlation did not perform well when applied to the present data, as shown in Figure 5.18.

In the 75 mm long channel test section, it predicted only 18 % of data points within ± 30 % of the experimental values, with a MAE% of 77.24 %. The segmentation of boiling mechanisms by vapour

quality likely contributed to the poor agreement. In contrast to the assumptions made by Lee and Mudawar, the current experiments exhibited nucleate boiling characteristics across the full range of vapour qualities, with no clear transition to convective boiling dominance at higher qualities. This fundamental difference in boiling regime behaviour appears to limit the applicability of the correlation.

The performance remained poor in the 30 mm long channel test section, where the correlation predicted only 9.35 % of the data within ± 30 % error and yielded a MAE% of 66.80 %. As shown in Figure 5.18, the prediction error remains high and the scatter significant. The continued presence of nucleate boiling throughout the vapour quality range is inconsistent with the segmented structure of the model, which presumes a mechanistic shift in dominant heat transfer mechanism with increasing vapour quality.

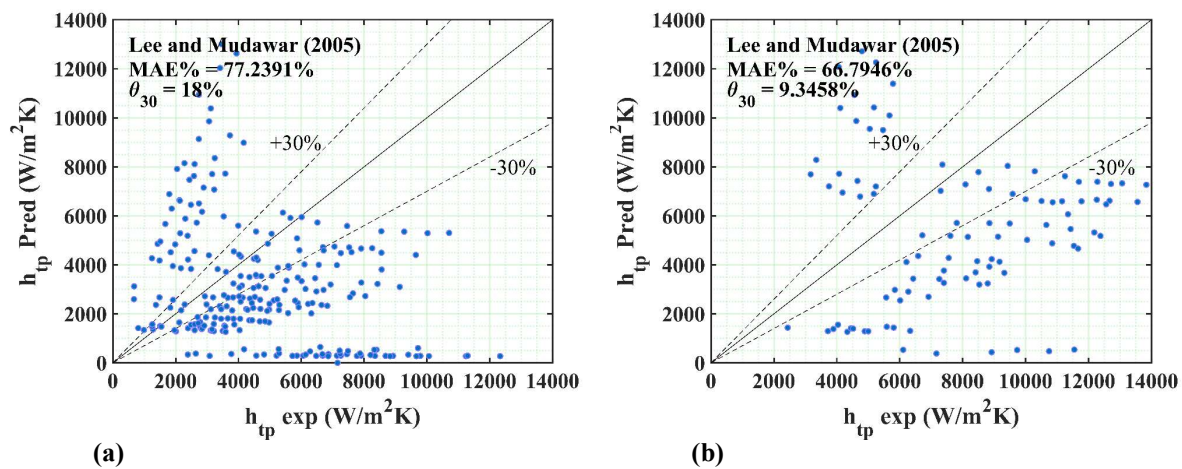


Figure 5.18. Comparison of experimental two-phase HTC to prediction by Lee and Mudawar [266] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

5.7.4 Comparison to Mahmoud and Karayiannis (2013)

Mahmoud and Karayiannis [270] investigated flow boiling heat transfer of R-134a in vertical tubes with internal diameters ranging from 0.52 mm to 4.26 mm. Their empirical correlation was developed using a modified version of the Chen [271] approach, incorporating an enhancement factor (F) and a nucleate boiling contribution modelled using Cooper's [272] pool boiling correlation. This formulation allowed the model to account for both nucleate and convective boiling contributions, even in microscale systems.

Figure 5.19 shows the comparison between the new experimental data and the correlation proposed by Mahmoud and Karayiannis [270], given in Appendix A. In the 75 mm long channel test section, this correlation demonstrated strong predictive capability, with a MAE% of 16.8 % and 87.6 % of the data falling within ± 30 % of the predicted values. The good agreement is likely due to the model's effective treatment of nucleate boiling through the Cooper pool boiling component, where the comparison is

presented in Figure 5.20. As noted in the previous analysis, nucleate boiling remains significant throughout the flow regimes in this study, contrasting with models that rely on distinct transitions to convective boiling at higher vapour qualities.

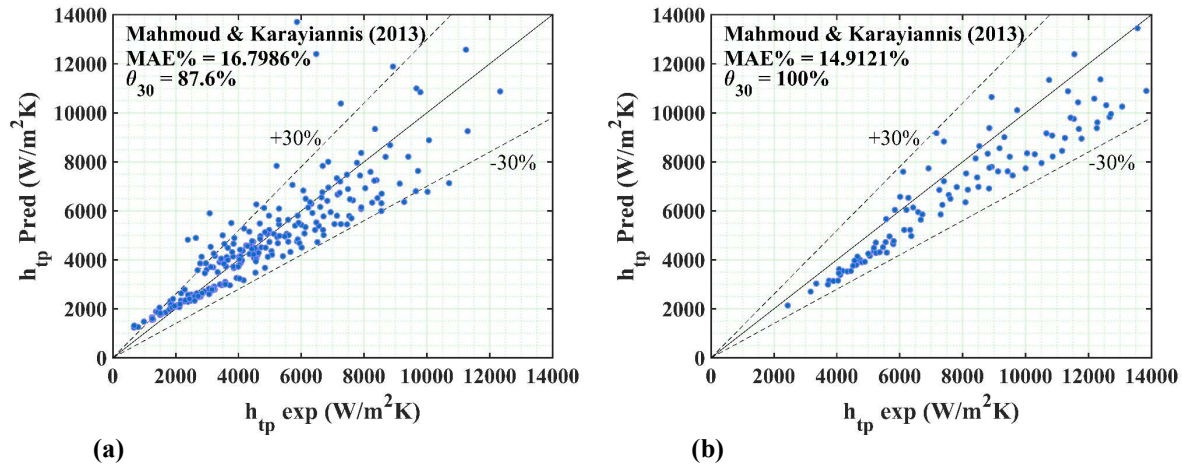


Figure 5.19. Comparison of experimental two-phase HTC to prediction by Mahmoud and Karayiannis [270] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

In the 30 mm long channel test section, the performance of the correlation further improved. As shown in Figure 5.20, the model predicted 100 % of the data within $\pm 30\%$ error, with a reduced MAE of 14.91 %. This improvement corresponds closely with the heat transfer behaviour revealed in Figure 5.20, where the nucleate boiling contribution becomes even more dominant compared to the longer test section. The increased nucleate influence in the 30 mm channel, relative to the 75 mm configuration, aligns with the core structure of the correlation and explains the enhanced predictive accuracy. The experimental findings for multi-channels reported by Al-Zaidi et al. [40], Fayyadh et al. [23], and Criscuolo et al. [403], as well as those for vertical minichannels by Anwar et al. [404], and Asim et al. [405], also showed good agreement with this correlation across a broad spectrum of refrigerants.

5.7.5 Comparison to Li and Jia (2015)

Li and Jia [232] performed flow boiling experiments using R-134a in horizontal rectangular multi-microchannels with a hydraulic diameter of 0.5 mm and three-sided heating. Their results were classified into three flow boiling regimes. In Regime I, corresponding to nucleate boiling, the heat transfer coefficient increased with heat flux across all mass fluxes, although the influence of mass flux was minimal. A slight decline in heat transfer coefficient at higher heat fluxes was also reported. Regime II described a convective boiling regime, in which mass flux became the dominant factor, while heat

flux still contributed. Regime III, associated with dry-out, exhibited a decline in heat transfer coefficient with increasing heat flux.

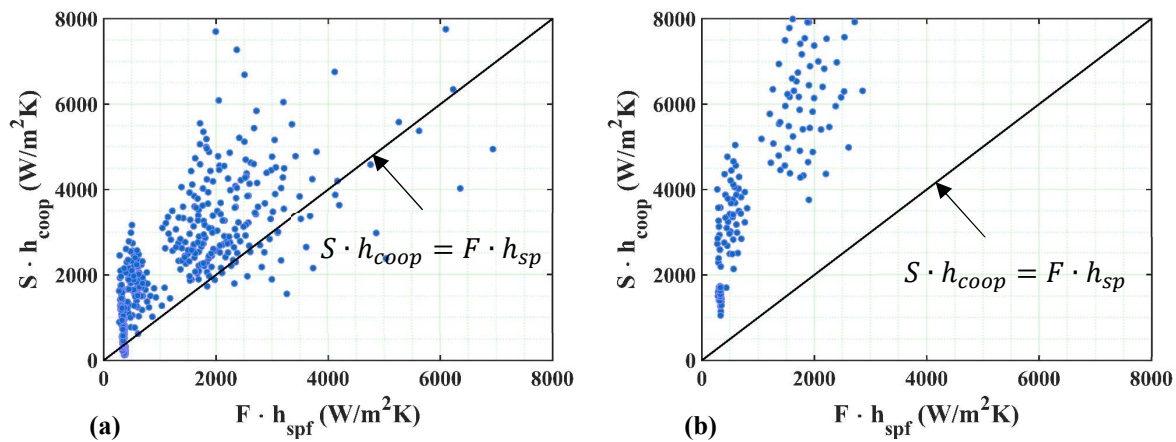


Figure 5.20. Comparison of predicted heat transfer mechanism components from the correlation developed by Mahmoud and Karayiannis [270] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

However, both correlations, included in Appendix A, performed poorly when applied to the present data, as shown in Figure 5.21. For the 75 mm long channel test section, the nucleate boiling correlation showed a very large MAE of 152.29 % with only 13.6 % of the data predicted within ± 30 % error. Although the model's performance improved in the 30 mm long channel test section, it remained unsatisfactory, with an MAE of 59.83 % and only 38.32 % of data points falling within the ± 30 % error margin, as shown in Figure 5.21. The poor predictive capability is likely due to the segmented regime-based approach adopted by Lin and Jia, which assumes clear transitions between nucleate and convective boiling regimes. This segmentation contrasts with the present experimental observations, where nucleate boiling effects persist across the entire range of vapour qualities, limiting the model's applicability and accuracy in describing the heat transfer characteristics under these conditions.

5.7.6 Comparison to Thiangtham et al. (2016)

Thiangtham et al. [149] conducted an experimental investigation of flow boiling heat transfer of R-134a in horizontal rectangular multi-microchannels with a hydraulic diameter of 0.421 mm. Their results demonstrated a clear distinction between two boiling regimes: at low heat fluxes, the heat transfer coefficient increased with heat flux, indicating the dominance of nucleate boiling; conversely, at higher heat fluxes, the heat transfer coefficient increased with mass flux, suggesting the onset of convective boiling as the dominant mechanism. Based on these observations, the authors developed a flow boiling heat transfer correlation, given in Appendix A, expressed in terms of dimensionless groups.

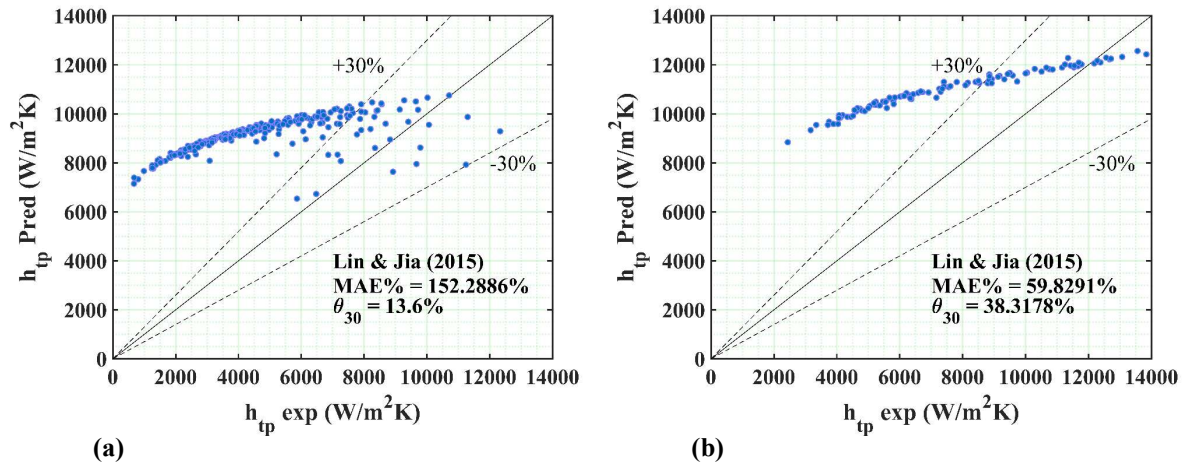


Figure 5.21. Comparison of experimental two-phase HTC to prediction by Li and Jia [232] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

When evaluated against the current experimental data, this correlation exhibited poor predictive capability, as shown in Figure 5.22. The comparison showed that no data points fell within $\pm 30\%$ of the predicted values and poor prediction was observed with MAE% of 181.53 % and 104.65% for the 75 mm and 30 mm long channel test sections, respectively. The large deviation can be attributed primarily to the inclusion of convective boiling effects, particularly the strong influence of mass flux at high heat fluxes, which was not supported by the present experimental findings. In the current study, mass flux showed an insignificant effect on heat transfer coefficient across the tested heat flux range, indicating that nucleate boiling remained the dominant mechanism even at elevated heat flux levels.

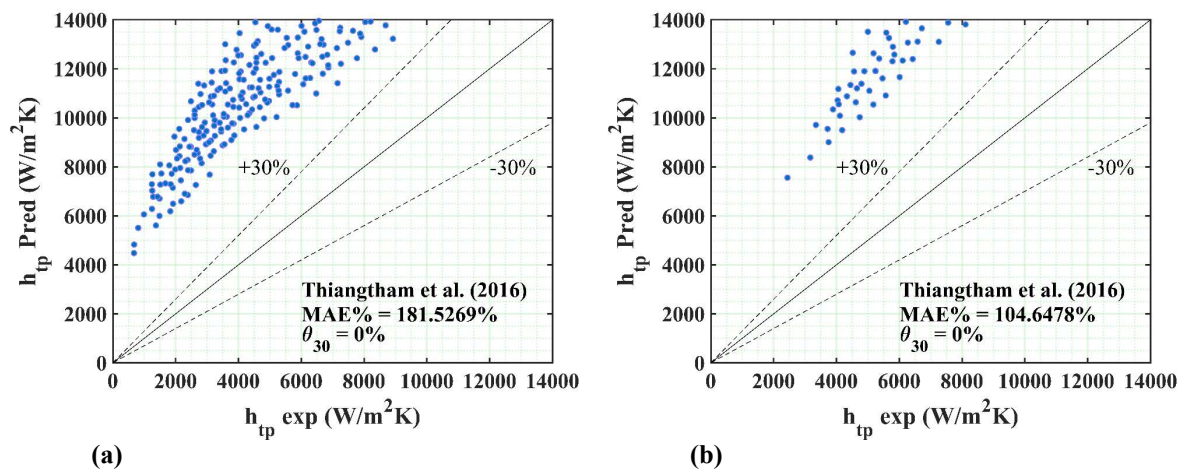


Figure 5.22. Comparison of experimental two-phase HTC to prediction by Thiangtham et al. [149] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

5.7.7 Comparison to Shah (2017)

Shah [273] re-evaluated their earlier 1982 correlation using a large experimental database and identified underprediction of heat transfer coefficients at low mass fluxes, which they attributed to the influence of surface tension. To address this, they introduced a new correction factor (F), incorporating the effects of the Weber number and Boiling number, see Appendix A. The updated correlation was validated across thirty-one fluids, a wide range of flow orientations (horizontal and vertical), various channel shapes, and diameters ranging from 0.38 mm to 27.1 mm.

When applied to the 75 mm long test section, the correlation performed poorly, as depicted in Figure 5.23. It resulted in an MAE of 244.32 %, with only 35.2 % of the data falling within ± 30 % of the predicted values. This significant deviation is likely due to the correlation's heavy reliance on modelling of convective boiling, which does not align well with the persistent nucleate boiling behaviour observed in the present study across all vapour qualities and mass fluxes. While the correction factor attempts to account for surface tension effects, it does not adequately reflect the complex role of nucleation and confinement-dominated boiling in microchannels.

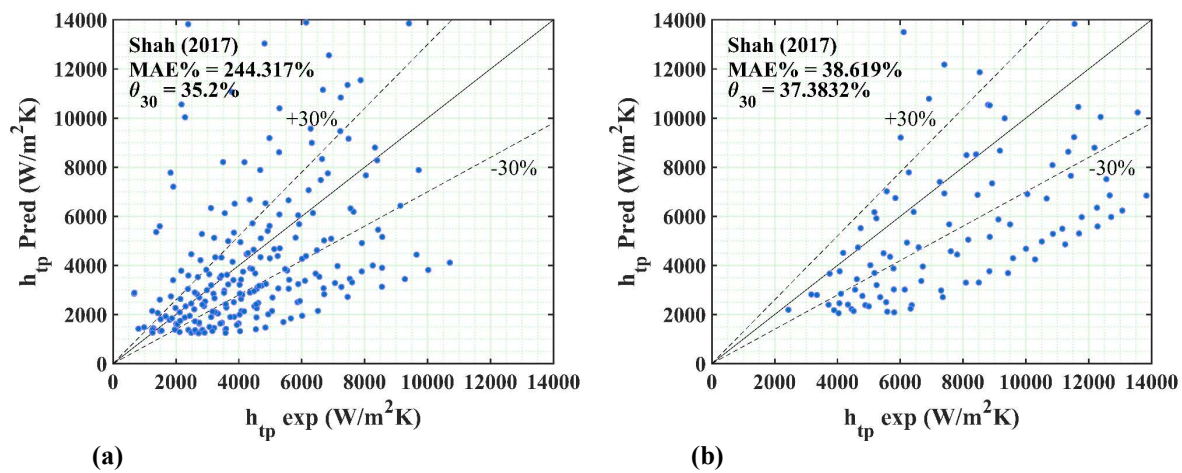


Figure 5.23. Comparison of experimental two-phase HTC to prediction by Shah [273] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

In contrast, for the 30 mm long test section, the performance improved markedly. The MAE dropped to 38.62 %, and θ_{30} rose to 37.38 %. However, the lack of an explicit nucleate boiling model still limits the predictive capacity of this correlation, even in the shorter channel.

5.7.8 Comparison to Wen et al. (2019)

Wen et al. [260] conducted an experimental investigation into the flow boiling heat transfer of R-134a using aluminium minichannels enhanced with serrated fins. Two rectangular channels were

fabricated, with hydraulic diameters of 1.28 mm and 1.59 mm, and the serrated fin geometry was specifically designed to promote surface area and turbulence (see Chapter 2 for more detail). They reported that at low vapour qualities, the local heat transfer coefficient increased with heat flux and saturation pressure, while the effect of mass flux was negligible. Conversely, at high vapour qualities, the influence of heat flux diminished, and the heat transfer coefficient increased with mass flux, indicating a transition toward convective boiling. Since existing models failed to predict their data accurately, they developed an empirical correlation using 1,429 data points, incorporating the Boiling number, Weber number, Froude number, and Cooper's [272] pool boiling correlation.

However, this correlation, provided in Appendix A, performed poorly when applied to the present study's data, as presented in Figure 5.24. For the 75 mm test section, it yielded an MAE of 44.00%, with only 38.47% of the data falling within $\pm 30\%$ of the predicted values. For the 30 mm test section, the predictive accuracy improved marginally, with an MAE of 39.76% and $\theta_{30} = 28.97\%$.

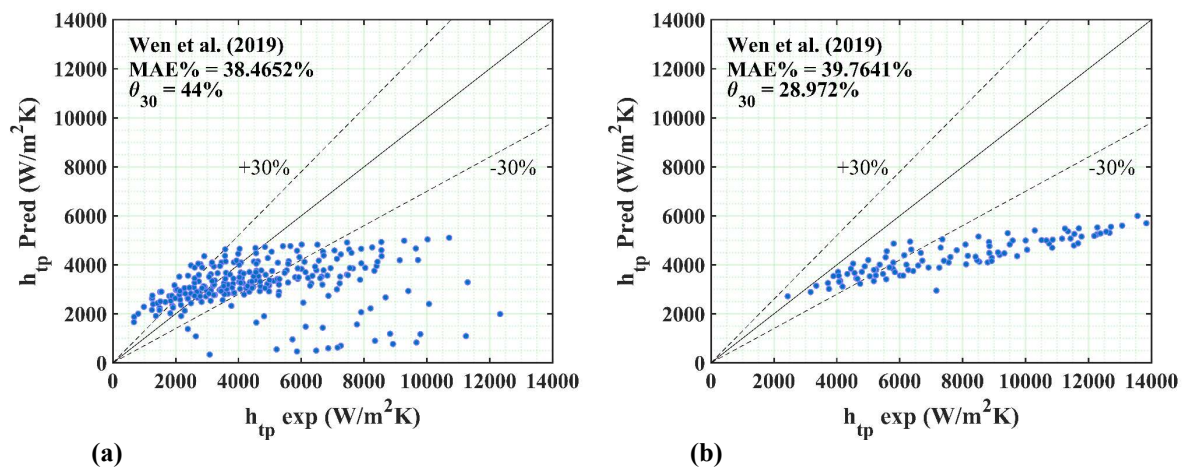


Figure 5.24. Comparison of experimental two-phase HTC to prediction by Wen et al. [260] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

The poor predictive performance can be primarily attributed to the substantial difference in geometry. Unlike the plain-wall, smooth rectangular channels used in the current study, Wen et al.'s use of serrated fins fundamentally alters the local heat transfer mechanisms by enhancing turbulence, increasing surface area, and possibly disrupting the vapour-liquid interface. These factors directly influence bubble nucleation, growth, and departure dynamics, mechanisms which are highly sensitive to surface structure. As a result, correlations derived from such geometrically enhanced channels are unlikely to transfer reliably to smooth-walled microchannels. Moreover, Wen et al. observed a regime-dependent behaviour with transitions from nucleate to convective boiling, whereas the present study consistently observed nucleate boiling dominance across all flow regimes, with minimal effect from mass flux even at elevated qualities and heat fluxes.

5.7.9 Comparison to Zhang et al. (2025)

Zhang et al. [210] conducted a visualisation-based experimental study to investigate the influence of hydraulic diameter on microchannel flow boiling heat transfer characteristics. Their experiments used DI water as the working fluid in three parallel microchannels with hydraulic diameters ranging from 0.25 mm to 1.5 mm. Heat fluxes were varied between 203 kW/m² and 880 kW/m², with an outlet pressure of 101.325 kPa and pressure drops ranging from 0.71 to 18.02 kPa. The resulting vapour quality ranged from 0.00183 to 0.37536. Owing to poor agreement between their data and existing predictive models, Zhang et al. proposed a new empirical correlation incorporating the liquid-only Reynolds number, Boiling number and liquid-only Weber number, see Appendix A.

The comparison between the results from the present study and the proposed correlation by Zhang et al. is presented in Figure 5.25. Zhang et al.'s correlation exhibited limited predictive accuracy, when applied to the present data. For the 30 mm test section, the correlation yielded an MAE of 53.92%, with only 14.02% of data points predicted within $\pm 30\%$ of the experimental values. Similarly, for the 75 mm test section, the MAE was 51.5%, with 40.0% of the data captured within $\pm 30\%$ agreement. These results demonstrate a consistent underperformance across both channel lengths.

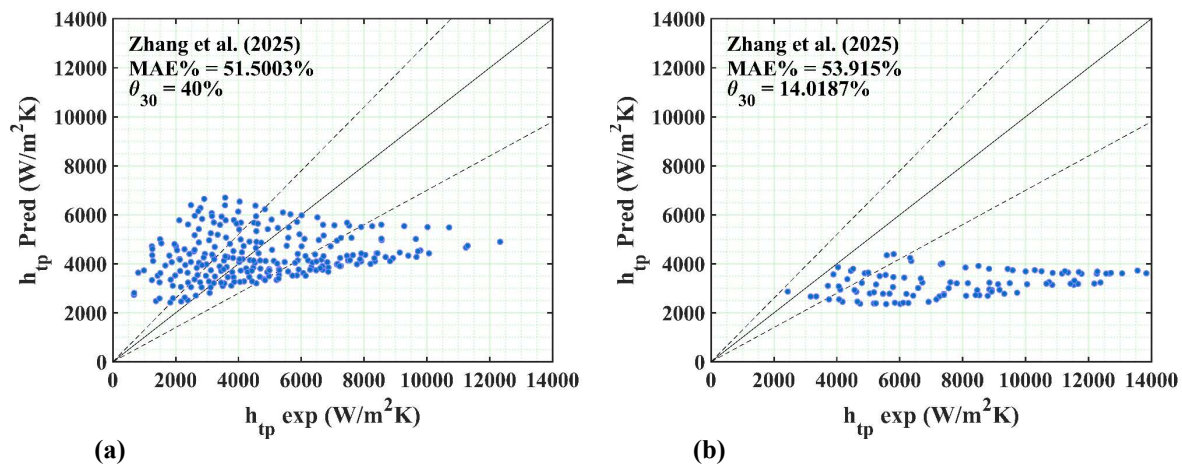


Figure 5.25. Comparison of experimental two-phase HTC to prediction by Zhang et al. [210] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

Several factors likely contribute to this discrepancy. Firstly, the correlation was developed specifically for DI water, whose thermophysical properties, including surface tension, viscosity and latent heat, differ substantially from those of HFE-7100. These differences affect nucleation, bubble dynamics and the transition between nucleate and convective boiling regimes, resulting in markedly different heat transfer behaviour. Secondly, the correlation is explicitly applicable only to vapour qualities below 0.4, whereas the present data, particularly from the 75 mm test section, extend well

beyond this range. It is therefore unlikely that the model captures key aspects of the dominant heat transfer mechanisms under the current experimental conditions.

5.7.10 Comparison to Patra et al. (2025)

Patra et al. [225] proposed an empirical correlation, which is included in Appendix A, for flow boiling heat transfer in parallel aluminium microchannels using acetone and methanol as working fluids. Their experiments were carried out in square microchannels with a hydraulic diameter of 0.5 mm and a heated length of 45 mm. The study identified three distinct boiling regimes based on visual observations. In Boiling Zone I, the heat transfer coefficient increased with heat flux, indicating nucleate boiling dominance. In Boiling Zones II and III, a stronger dependence on vapour quality and mass flux was observed, indicating a shift toward convective boiling. The correlation incorporated the Boiling number, Weber number, and Froude number and was shown to perform well with their dataset, achieving an MAE of 14% and predicting 92% of data points within $\pm 30\%$ of the measured values.

The correlation proposed by Patra et al. resulted in an MAE of 140.84% when compared to the current study, with only 22.73% of the data points predicted within $\pm 30\%$ error for the 75 mm long channel test section, as shown in Figure 5.26. In comparison, the MAE reduced to 50.80%, and the percentage of data captured within $\pm 30\%$ improved to 35.51%, for the 30 mm long channel test section. Although the correlation showed better agreement in the shorter channel, its overall predictive accuracy remained limited.

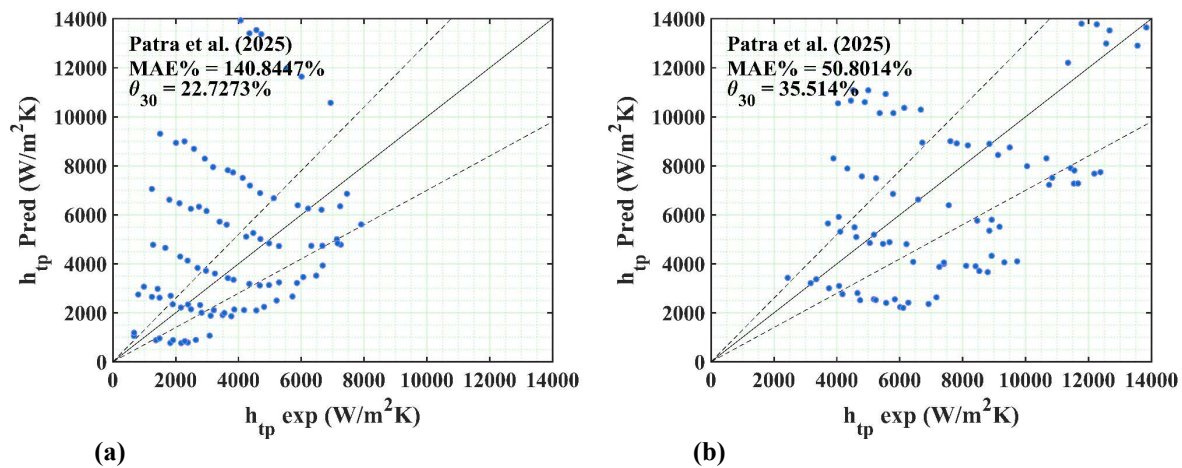


Figure 5.26. Comparison of experimental two-phase HTC to prediction by Patra et al [225] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

In applying the model, the version tuned for acetone was used, as this provided the best alignment with the present data. The discrepancy in predictive accuracy can be attributed to several key differences between the two studies. Patra et al. developed their correlation using acetone and methanol as working

fluids, while the current study uses HFE-7100, which has substantially different thermophysical properties. These include lower surface tension, higher density, and different latent heat of vaporisation, all of which influence boiling initiation and heat transfer behaviour. Furthermore, the channel geometry used by Patra et al. consisted of square aluminium microchannels with a heated length of 45 mm, whereas the present experiments were conducted in horizontal rectangular multi-microchannels made from oxygen-free copper, with lengths of 30 mm and 75 mm. These differences in material, geometry and orientation have a significant impact on confinement, nucleation site density and vapour–liquid distribution.

Most notably, the present experiments consistently demonstrated nucleate boiling existence across the entire vapour quality range, without a clear transition to convective boiling at higher qualities. This contrasts with Patra et al.'s findings, which indicated a regime shift towards convective boiling as vapour quality increased. The structure of their correlation, which was built around this regime transition, does not reflect the boiling behaviour observed in the present study. The relatively better performance in the 30 mm test section is likely due to its similar heated length to Patra et al.'s 45 mm channels, but this similarity is not sufficient to overcome the broader mechanistic differences.

Among the twelve evaluated heat transfer correlations, the model proposed by Mahmoud and Karayiannis [270] demonstrated the most accurate and consistent predictive performance across the experimental data, particularly for the 30 mm long channel test section. This correlation, which is based on a modified Chen-type model and incorporates the Cooper pool boiling correlation for the nucleate boiling contribution, effectively captured the dominant boiling mechanism observed throughout the study.

This strong performance is attributed to the model's ability to account for persistent nucleate boiling, which was consistently observed in both test sections across the full range of vapour qualities and heat fluxes. Unlike correlations that assume a mechanistic transition to convective boiling at higher vapour qualities, the Mahmoud and Karayiannis [270] correlation aligns well with the confinement-dominated boiling characteristics and the minimal mass flux dependence documented in the current experiments.

In contrast, correlations developed based on macro-scale flow conditions or those that emphasized convective boiling contributions, such as those by Shah [268, 273], Lee and Mudawar [266] and Thiangtham et al. [149], exhibited significant predictive discrepancies, particularly for the 75 mm long test section. These models tended to overemphasize mass flux effects and underrepresent the role of nucleate boiling, leading to high MAEs often exceeding 50%, and in some cases, approaching or exceeding 100%.

Additionally, many of the less accurate correlations were developed using fluids, surface structures, or geometric configurations that differ considerably from the smooth-walled copper channels and HFE-

7100 used in this study. These disparities further limited their applicability, particularly in scenarios where surface microstructure and channel confinement effects dominate heat transfer behaviour.

Overall, the comparison highlights the critical importance of selecting correlations that accurately reflect the underlying heat transfer mechanisms and flow regime characteristics of the system under study. In the case of microscale rectangular channels with dominant nucleate boiling and weak convective effects, as observed in this study, the Mahmoud and Karayiannis [270] correlation currently provides the most robust and applicable predictive tool.

5.8 CONCLUSIONS

The following conclusions have been made based on the presented flow boiling heat transfer results:

- Boiling incipience occurred at progressively lower wall superheats with increasing channel inlet pressure. This trend was consistent with the Hsu correlation [243], indicating more active cavities at higher pressures. TOS was observed in almost all cases, where trapped air was cited as a potential reason for earlier initiated nucleation in which it did not occur.
- Wall heat flux significantly influenced the heat transfer coefficient. Higher heat fluxes resulted in increased bubble nucleation frequency and site density, which enhanced local and average heat transfer coefficients. Even in the subcooled region, subcooled boiling increased heat transfer beyond what would be expected in purely single-phase flow.
- Mass flux had an insignificant effect on both local and average heat transfer coefficients across the entire heat flux range. However, higher mass fluxes extended the subcooled region and influenced wall temperatures due to associated pressure drops, especially at lower system pressures.
- Channel inlet pressure positively impacted heat transfer. At higher pressures, increased nucleation site density and bubble frequency enhanced heat transfer, particularly at elevated heat fluxes. This was linked to reduced required superheat for nucleation and more stable cavity activation.
- Heated length significantly influenced thermal performance. The shorter channel showed higher average heat transfer coefficients for the same outlet vapour quality compared to the longer channel due to concentrated heat input and enhanced nucleation intensity over a shorter distance.
- Across both channel lengths, nucleate boiling was the dominant mechanism and continued to persist throughout the vapour quality range. This consistent presence of nucleate boiling limited the applicability of many existing correlations that rely on transitions between boiling regimes.

- Among twelve evaluated heat transfer correlations, the model by Mahmoud and Karayiannis [270] provided the best agreement with experimental data, particularly for the 30 mm long channel. It achieved a MAE of 14.91% and captured nucleate boiling dominance effectively. In contrast, other models that relied heavily on convective boiling assumptions exhibited poor predictive performance, especially in the 75 mm long test section.
- The results emphasise the importance of tailoring heat transfer correlations to account for persistent nucleate boiling and microscale effects in short, rectangular microchannels with high wetting fluids like HFE-7100.

CHAPTER 6

PRESSURE DROP RESULTS

6.1 INTRODUCTION

In this chapter, the results for the flow boiling pressure drop are presented and discussed. The discussion focusses solely on the pressure drop in the saturated region, using the data reduction analysis discussed in Chapter 3. Firstly, the effect of wall heat flux and mass flux on the two-phase pressure drop in the 75 mm long channel test section are presented and discussed in regards to the components of two-phase heat transfer coefficient, namely the acceleration and frictional components. The effect of channel inlet pressure on the two-phase pressure drop in the 75 mm long channel is then discussed with reference to the resultant change in fluid thermophysical properties that arise due to the increased channel inlet pressure. A comparison between the two-phase pressure drop results in the 30 mm long channel test section and 75 mm long test section is made to determine any potential effects of heated length. Finally, the results are compared to existing microscale flow boiling pressure drop correlations.

6.2 EFFECT OF HEAT FLUX AND MASS FLUX ON PRESSURE DROP

The two-phase pressure drop is presented as a function of heat flux in Figure 6.1 for each of the channel inlet pressures. The figure shows a clear increase in two-phase pressure drop with increasing heat flux. As reported in Chapter 4, bubble nucleation was observed to occur for all flow patterns in the 75 mm long channel. The number of bubble nucleation sites and bubble departure frequency therefore increased with the rise in heat flux. The increase in bubble generation provides additional surface area and consequent frictional pressure drop between the two-phases. Furthermore, there is an increased difference in superficial phase velocities with an increased heat flux due to the elevated vapour generation. This results in a greater shear across the liquid-vapour interface and also results in an additional greater frictional pressure drop during the flow regime. Finally, the increased vapour generation also increases acceleration pressure simply as a result of the occurring mass transfer and momentum change between phases.

At lower heat fluxes, the two-phase pressure drop is lower for higher mass fluxes, as highlighted by the magnified images shown in Figure 6.1. This behaviour is partly due to the shorter effective two-phase development length at higher mass fluxes. For a fixed inlet subcooling, increasing the mass flux increases the thermal capacity of the liquid, delaying the onset of boiling and extending the subcooled

region. As a result, a smaller portion of the channel experiences two-phase flow, including bubbly flow, which reduces the overall two-phase pressure drop.

In addition, under these conditions the flow remains primarily in the bubbly regime, where vapour quality is relatively low and interfacial interactions are limited. The absence of larger flow structures, such as slugs or annular films, reduces flow obstruction and frictional losses. In contrast, at lower mass fluxes and equivalent heat fluxes, the vapour quality is higher and the flow transitions earlier from bubbly to slug or annular regimes. These regimes are associated with increased interfacial contact and partial flow blockage, leading to higher frictional pressure drops. However, at higher vapour qualities, the two-phase pressure drop was found to increase with mass flux due to the increasing phase velocities and associated shear.

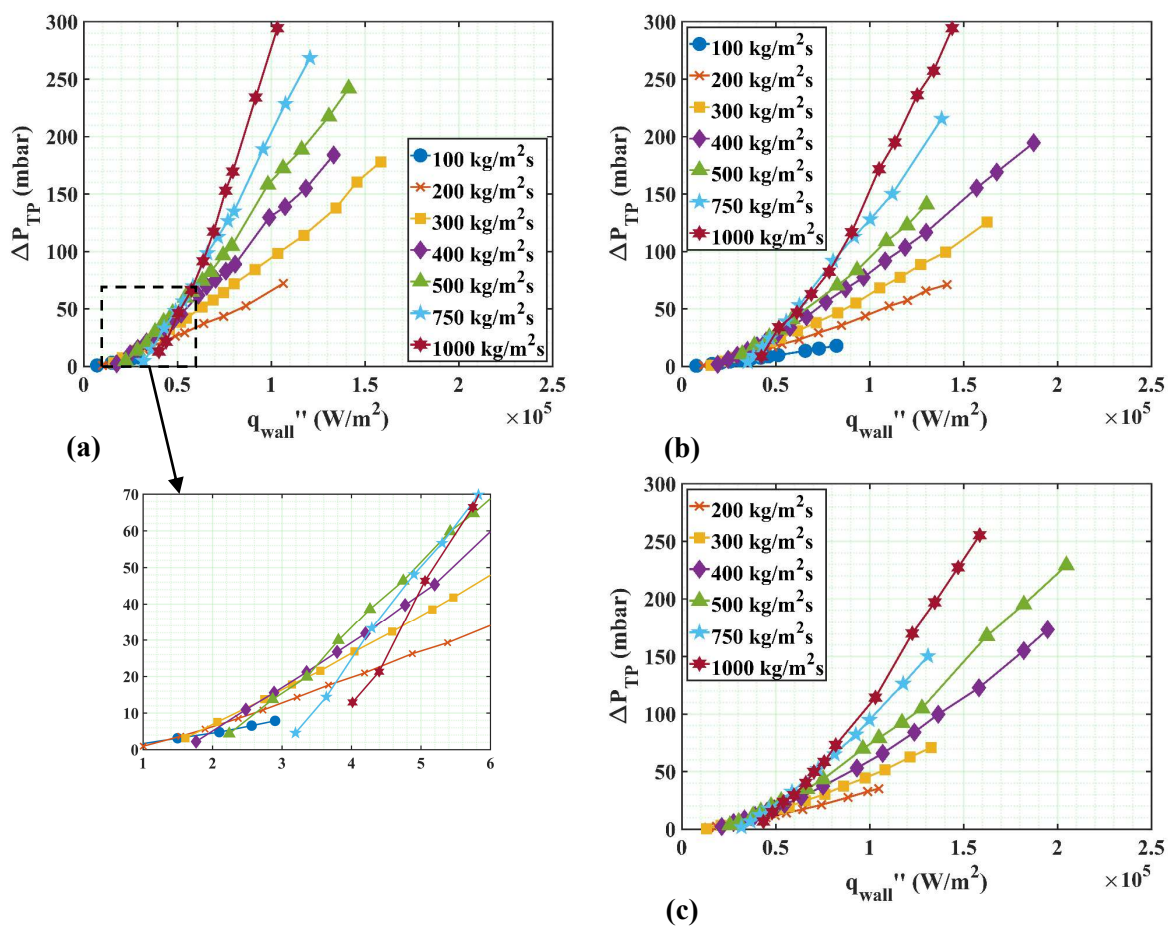


Figure 6.1. Two-phase pressure drop vs wall heat flux for channel inlet pressures of; (a) 1 bar; (b) 1.5 bar; (c) 2 bar; in the 75 mm long channel test section.

Figure 6.2 presents the flow boiling pressure drop as a function of exit vapour quality for the 75 mm long test section across all channel inlet pressures. The data are shown for exit vapour qualities up to one, corresponding to increasing heat flux. It is evident that, for a given exit vapour quality, the flow boiling pressure drop increases with increasing mass flux. This is because, as reported in Chapter 4, the

transition from bubbly flow occurs at lower vapour qualities for higher mass fluxes. This behaviour is consistent across all inlet pressure conditions. The increase in mass flux results in higher flow velocities and greater shear forces between the liquid and vapour phases, which in turn contribute to an elevated two-phase pressure drop.

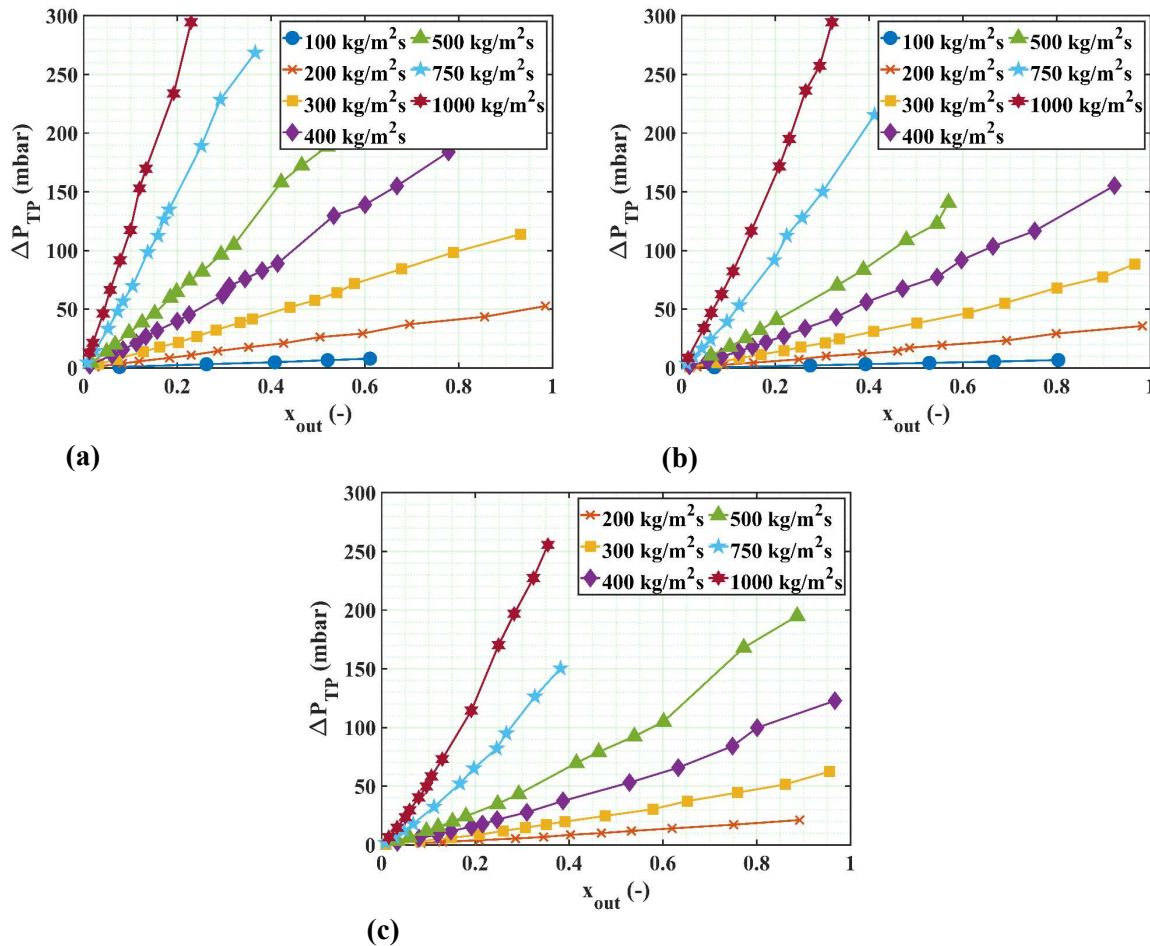


Figure 6.2. Two-phase pressure drop vs channel outlet quality for channel inlet pressures of; (a) 1 bar; (b) 1.5 bar; (c) 2 bar; in the 75 mm long channel test section.

6.3 EFFECT OF SYSTEM PRESSURE ON PRESSURE DROP

The two-phase pressure drop is plotted with increasing wall heat flux for all channel inlet pressures at mass fluxes of $200 \text{ kg/m}^2\text{s}$ and $500 \text{ kg/m}^2\text{s}$ in Figure 6.3. As is evident in both Figure 6.3(a) and Figure 6.3(b), the two-phase pressure drop was found to decrease with increasing system pressure, a phenomenon observed by other researchers [213, 227, 241, 406]. One possible explanation for this phenomenon is the delayed transition from bubbly flow that occurs with increasing channel inlet pressure, as presented in Chapter 4. As mentioned previously, the lack of larger flow structures like slugs or annular films minimises flow resistance and decreases frictional losses. Therefore, the occurrence of bubbly flow in a larger portion of the channel reduces the overall two-phase pressure

drop. This explanation may help to explain why the pressure drop results are fairly similar at low heat fluxes, where bubbly flow tends to dominate for the entirety of the channel saturated region.

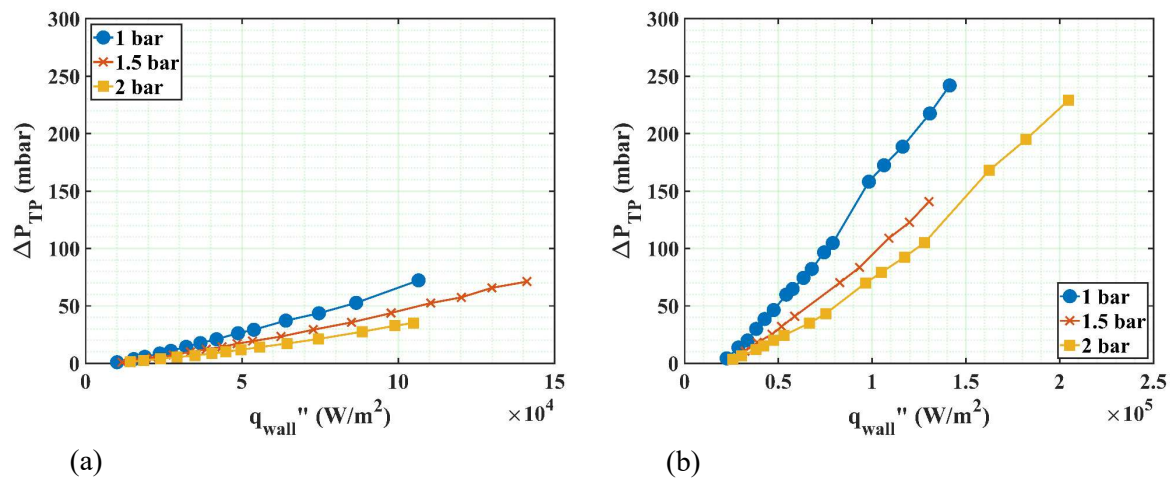


Figure 6.3. Comparison of two-phase pressure drop for varying channel inlet pressures in the 75 mm long channel test section for: (a) $200 \text{ kg/m}^2\text{s}$; (b) $500 \text{ kg/m}^2\text{s}$; in the 75 mm long channel test section.

The difference between the two-phase pressure drop values observed for various channel inlet pressures can be seen to increase at lower inlet pressures and higher wall heat fluxes, i.e. where larger proportions of the channel tend to be in annular flow. It is in the annular regime where significant slip occurs between the phases and the frictional component becomes more dominant. As will be shown in Section 6.5, the correlation proposed by Gao et al. [299] provided the closest overall prediction of the two-phase pressure drop when separated-flow effects were considered.

It should be noted, however, that the experimental data obtained by Gao et al. [299] consisted of adiabatic two-phase flows, in which phase change occurred mainly due to flashing rather than continuous wall heating. In their formulation, the acceleration pressure drop was calculated using a drift-flux-based void fraction model proposed by [407]. In contrast, the present study considers flow boiling with continuous phase change along the heated channel. Therefore, the acceleration pressure drop used in the present analysis was calculated using the method proposed by Lockhart and Martinelli [294], which are shown in Equations (6.1) and (6.2), with Equation (6.1) giving the acceleration pressure drop and Equation (6.2) providing the corresponding void fraction.

Figure 6.4 presents the effect of channel inlet pressure on the predicted acceleration and frictional pressure drops. The acceleration pressure drop was calculated using the saturated fluid properties for HFE-7100 at various pressures shown in Table 6.1. These properties were also presented in Chapter 3, with the addition of the liquid-to-vapour density and viscosity ratios. The effect of the fluid thermophysical properties on the predicted acceleration pressure drop is difficult to observe analytically. Therefore, the acceleration pressure drop calculated was plotted in Figure 6.5 for a mass

flux of 500 kg/m²s and an outlet vapour quality of 0.4 with varying fluid properties. Figure 6.5(a) presents the effect of changing phase density ratios on the predicted acceleration pressure drop with the viscosity ratio fixed at 19.84, corresponding to the saturation viscosity ratio at 1 bar. Conversely, Figure 6.5(b) presents the effect of changing phase viscosity ratios with the phase density ratio fixed at 148 and the liquid density also fixed at 1420 kg/m³.

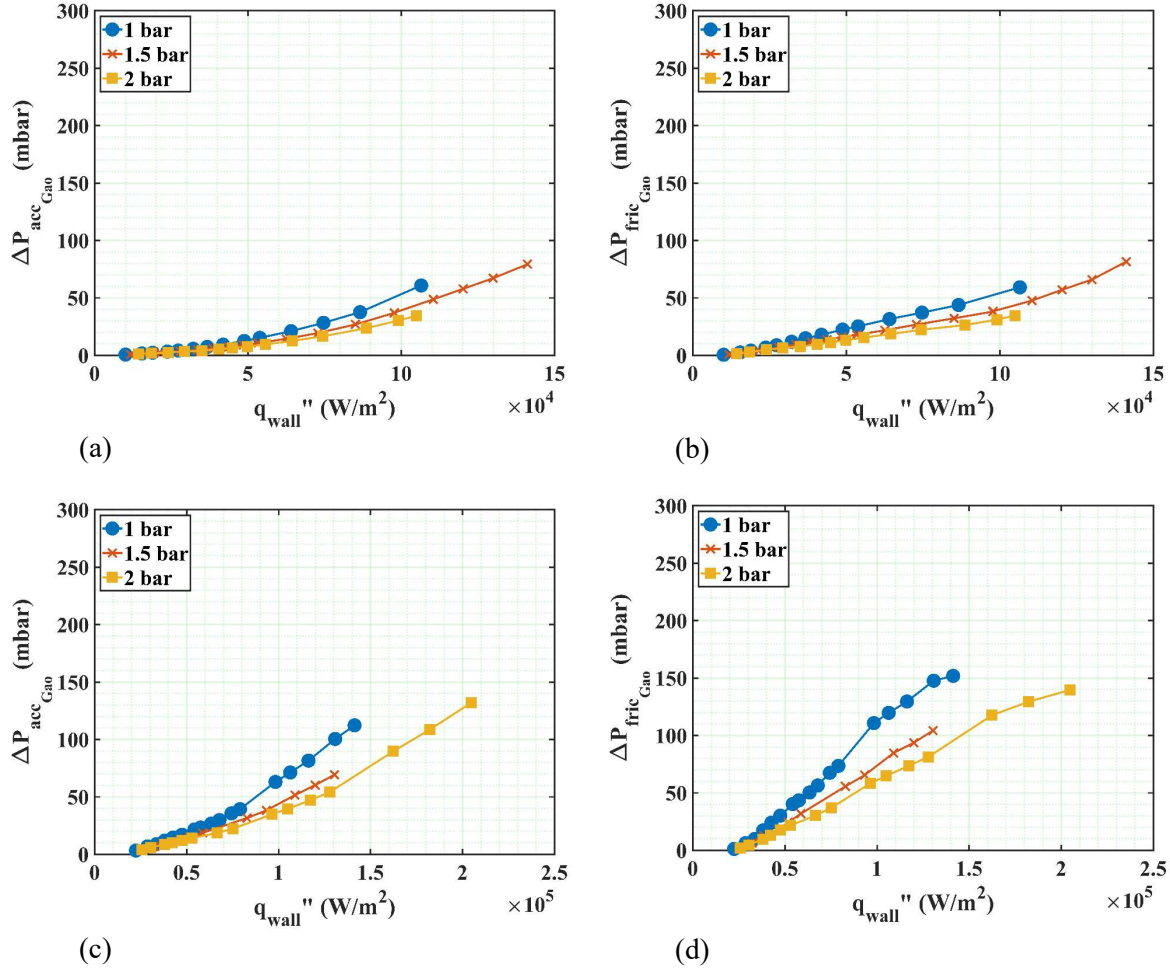


Figure 6.4. Predicted pressure drop components by Gao et al. [299]: (a) acceleration for 200 kg/m²s; (b) friction for 200 kg/m²s; (c) acceleration for 500 kg/m²s; (d) friction for 500 kg/m²s; in the 75 mm long channel test section.

$$\Delta P_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{out}^2}{\alpha_{out}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{out})}{(1 - \alpha_{out})} - 1 \right] \quad (6.1)$$

$$\alpha_{out} = \left[1 + 0.28 \left(\frac{1 - x_{out}}{x_{out}} \right)^{0.64} \left(\frac{\rho_g}{\rho_l} \right)^{0.36} \left(\frac{\mu_l}{\mu_g} \right)^{0.07} \right]^{-1} \quad (6.2)$$

Table 6.1. Thermophysical properties of HFE-7100 at saturation.

P_{in} [bar]	T_{sat} [°C]	ρ_l [kg/m ³]	ρ_g [kg/m ³]	ρ_l/ρ_g [-]	i_{fg} [kJ/kg]	μ_l [μPa·s]	μ_g [μPa·s]	μ_l/μ_g [-]	σ [N/m]
1	60.67	1420	9.58	148	115.7	393.7	19.84	19.84	0.0096
1.5	73.35	1383	14.13	98	111.3	341.6	20.62	16.54	0.0085
2	83.08	1353	18.66	73	107.8	312.2	21.22	14.71	0.0077

Table 6.1. shows that both the liquid to vapour density and viscosity ratios decrease with increased system pressure. Figure 6.5(a) shows that the acceleration pressure drop component is predicted to decrease with a decrease in liquid to vapour phase density ratio for HFE-7100. However, the acceleration pressure drop component is expected to increase with decreased liquid to vapour phase viscosity ratios. The result of an increase in two-phase pressure drop due to an increase in liquid to vapour phase density ratios is expected, as the acceleration pressure drop is a result of change in fluid velocity in the channel. The two-phase density can be calculated using Equation (6.3).

$$\rho_{tp} = \left(\frac{x}{\rho_g} + \frac{1-x}{\rho_l} \right)^{-1} \quad (6.3)$$

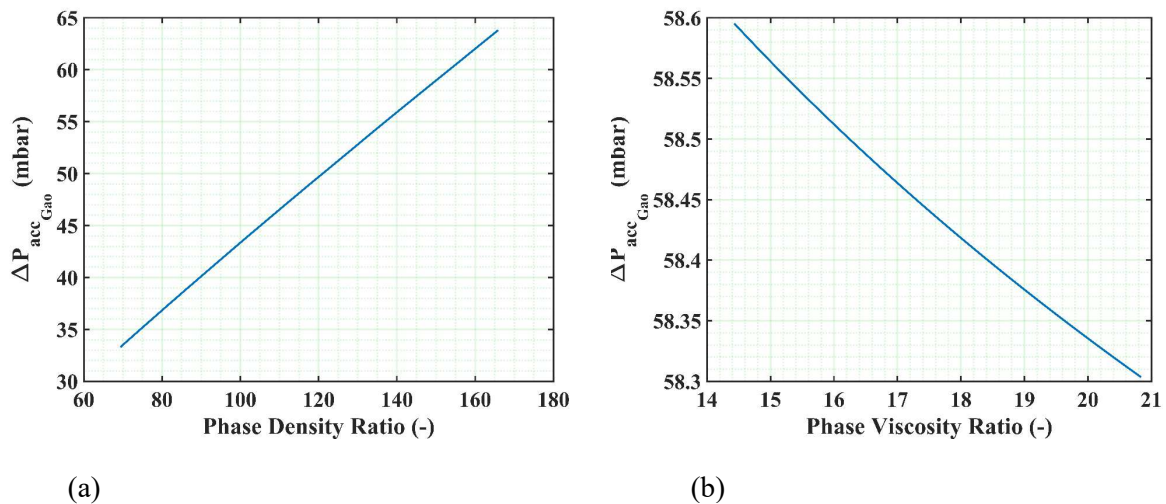


Figure 6.5. Predicted acceleration pressure drop component at 500 kg/m²s and an outlet vapour quality of 0.4: (a) for constant viscosity; (b) for constant densities; in the 75 mm long channel test section.

The vapour quality is zero at the beginning of the saturated region, meaning that the density is equal to that of the liquid phase. Again, assuming an outlet vapour quality of 0.4, the two-phase density at the outlet for 1 bar is 23.81 kg/m³, whilst the two-phase density at the outlet for 2 bar is 45.66 kg/m³. Compare this to the fact that the initial liquid density is lower at 2 bar, there is a comparatively smaller

density change and thus a smaller impact on change in flow velocity, thus reducing the acceleration component of the pressure drop.

The increase in acceleration pressure drop with a decrease in liquid to vapour phase viscosity ratios can be attributed to the viscosity ratio's role in determining the void fraction, as expressed in Equation (6.2). Specifically, a lower viscosity ratio reduces the flow asymmetry between phases and decreases the slip, resulting in a higher void fraction for a given vapour quality. As the void fraction increases, the mixture density decreases, and the corresponding increase in flow velocity causes a greater momentum change and thus a higher acceleration pressure drop. However, as shown in Table 6.1, the relative change in viscosity ratio across the range of system pressures is significantly smaller than the corresponding change in density ratio. This can also be seen in Figure 6.5, where the change in predicted acceleration drop is significantly greater for the change in densities. Therefore, while the viscosity ratio does influence the predicted acceleration pressure drop, its effect is comparatively minor due to the limited variation in viscosity relative to density over the same pressure range.

The frictional pressure drop prediction proposed by Gao et al. [299] is shown in Appendix B. Assessing the parametric effects on the frictional pressure drop is more difficult because of the dependence on the Bond number, the liquid only Reynolds number and the vapour only Reynolds number. The Bond number, which describes the ratio of buoyancy to surface tension forces is a function of the surface tension. A similar analysis on the effects of changing thermophysical properties on the predicted frictional pressure drop is displayed in Figure 6.6. This analysis now includes the effect of the changing surface tension; however, not all correlations included this parameter in the consideration of the frictional pressure drop. Other correlations in which this was considered as a relevant thermophysical property were provided by Keepaiboon et al. [296], Markal et al. [298] and Moradkhani et al. [301]. A constant surface tension value of 0.0096 N/m was used where the surface tension value was kept constant.

Figure 6.6(a) shows that the predicted frictional pressure drop decreases with increasing liquid to vapour phase density ratio. This behaviour can be explained by the influence of the density ratio on the void fraction and subsequently the two-phase mixture velocity. A higher density ratio leads to a smaller void fraction for a given vapour quality, meaning a greater proportion of the flow volume is occupied by the denser liquid phase. This increases the mixture density and reduces the average mixture velocity, thereby decreasing wall shear and frictional pressure losses. Additionally, the lower void fraction corresponds to reduced interfacial area and phase slippage, both of which contribute to lower frictional resistance in the flow.

Figure 6.6(b) illustrates that the frictional pressure drop increases with an increasing liquid to vapour viscosity ratio. This occurs because a higher viscosity ratio implies a relatively more viscous liquid phase compared to the vapour phase, which increases the overall flow resistance. The liquid's higher

viscosity directly contributes to greater shear stresses at the channel walls, elevating frictional losses. Moreover, an increased viscosity ratio tends to reduce slip between phases, promoting a more uniform velocity profile but also increasing the effective viscosity of the mixture. Consequently, the mixture experiences greater viscous resistance, resulting in a higher frictional pressure drop.

Figure 6.6(c) shows that the frictional pressure drop decreases with increasing surface tension. This appears counter intuitive as higher surface tension suppresses the formation of small bubbles and fine dispersed phases, promoting the presence of larger bubbles. A potential reason may be that the overall interfacial area is reduced with larger bubbles. The resultant smaller interfacial area and lower bulk fluid agitation may result in lower frictional pressure drops. Conversely, at lower surface tension, numerous small bubbles and dispersed phases are generated, increasing interfacial area and promoting turbulence and viscous dissipation, thereby increasing frictional losses. Regardless, Figure 6.6(c) shows that the effect of surface tension is minimal.

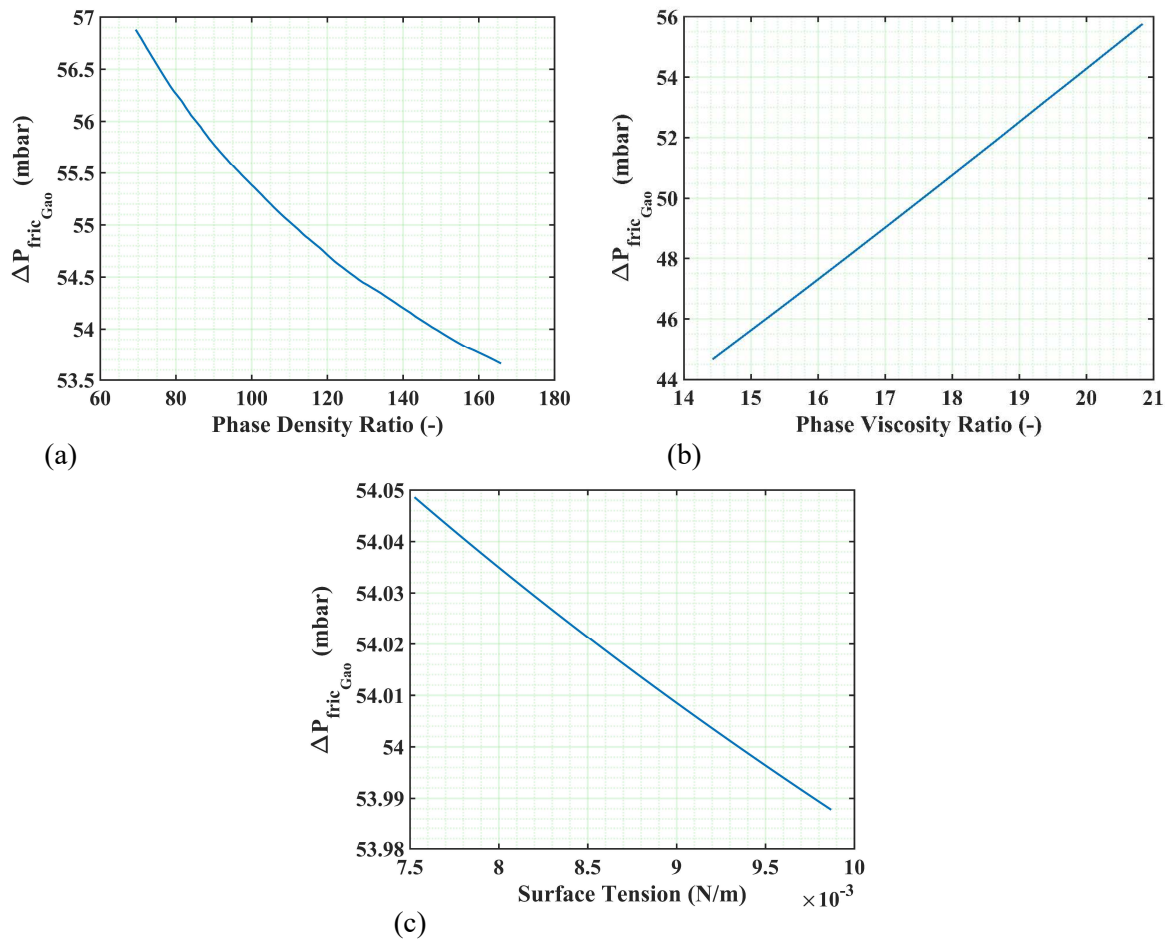


Figure 6.6. Predicted acceleration pressure drop component at 500 kg/m²s and an outlet vapour quality of 0.4: (a) for constant viscosity and surface tension; (b) for constant densities and surface tension; (c) for constant viscosity and densities; in the 75 mm long channel test section.

Figure 6.6 clearly shows that changes in viscosity have the largest influence on the frictional pressure drop. While increases in phase density and surface tension tend to increase frictional losses, the reduction in the liquid to vapour viscosity ratio with rising system pressure dominates these effects. As a result, despite the opposing influences of density and surface tension, the overall frictional pressure drop decreases with increasing pressure due to the significant impact of viscosity changes.

6.4 EFFECT OF HEATED LENGTH ON PRESSURE DROP

Figure 6.7 displays the effect of heated length on the two-phase pressure drop per unit length at 500 kg/m²s with outlet vapour quality. Figure 6.7 reveals that, for a given exit vapour quality, the pressure drop per unit length is substantially greater in the shorter channel. This indicates a steeper pressure gradient in the shorter test sections compared to longer ones under identical outlet conditions. The underlying reason is likely the higher heat flux imposed on the shorter channel to reach the same exit quality. This would promote significantly more bubble nucleation in the shorter channel to achieve the same vapour quality, a phenomenon observed across all flow patterns due to the nucleation in the liquid film. The intensified nucleation near the heated wall leads to the formation of numerous small vapour bubbles, which increase interfacial interactions within the liquid film. This interaction likely contributes to a higher frictional pressure drop.

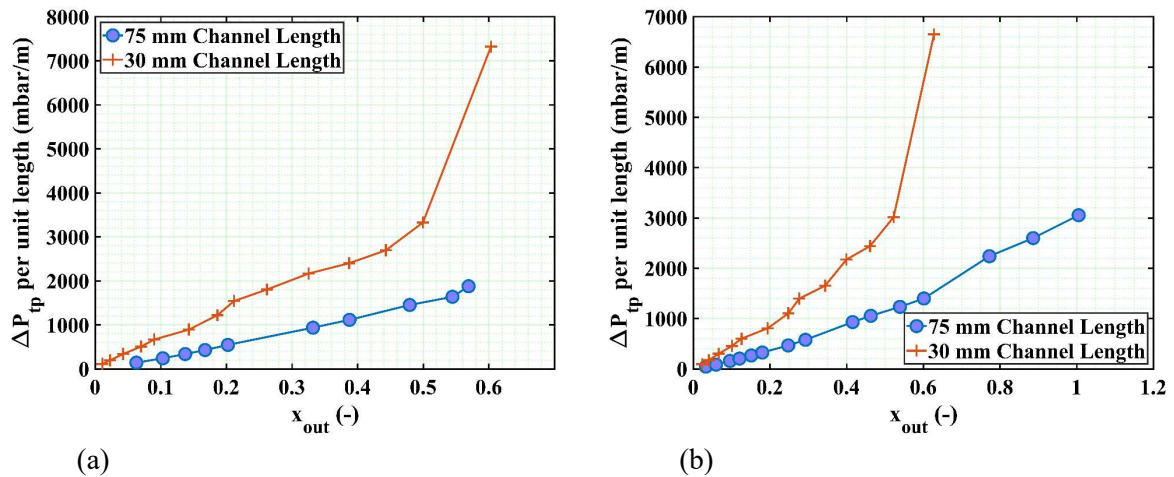


Figure 6.7. Effect of heated length on two-phase pressure drop with outlet vapour quality at 500 kg/m²s for; (a) 1.5 bar inlet pressure; (b) 2 bar inlet pressure.

6.5 COMPARISON TO EXISTING FLOW BOILING PRESSURE DROP CORRELATIONS

This section presents a comparison between the current flow boiling pressure drop measurements and several existing correlations developed for both conventional and microchannels. The chosen

correlations serve as a reference for assessing the agreement between the experimental results and established predictive methods. A summary of each correlation, including its background and intended application, is provided in Chapter 2, with the full formulations listed in Appendix B.

As previously outlined in Chapter 5, the MAE% was used to evaluate the accuracy of each correlation. For the two-phase pressure drop analysis, the same approach is adopted here, and the MAE% is recalculated using Equation (6.4), which is repeated from Equation (5.6) for clarity. In addition, the proportion of data points falling within a $\pm 30\%$ deviation from the predicted values, denoted as θ_{30} , is also determined.

$$\text{MAE}\% = \frac{1}{n} \sum \left| \frac{X_{exp} - X_{pred}}{X_{exp}} \right| \times 100\% \quad (6.4)$$

6.5.1 Comparison to Lockhart and Martinelli (1949)

Figure 6.8 presents a comparison between the current experimental results and the correlation proposed by Lockhart and Martinelli [294] that estimates frictional pressure drop using a two-phase multiplier approach. This method was developed from a broad dataset involving several fluids tested in horizontal circular channels with diameters spanning 1.5 to 25.8 mm. To account for differences in flow behaviour, the model assigns one of four constants depending on the specific combination of laminar or turbulent conditions in each phase. For the acceleration component, a void fraction relationship derived from Martinelli's approach was applied. The proposed prediction method was found to significantly overpredict the two-phase pressure drop, achieving MAE%s of 194.6 % and 86.0 % for the 75 mm long channel test section and the 30 mm long channel test section, respectively. Furthermore, only 16 % of data points fell within the 30 % limits for the longer channel, whilst an even smaller 11.2 % fell within the 30% limits for the shorter test section.

The overprediction of the Lockhart-Martinelli correlation for microscale flows has been observed extensively in literature [209, 371, 408]. This overprediction was attributed by Al-Zaidi [127] to the difference between the thermophysical properties of fluids used in the study, (air/water, air/benzene, air/kerosene and air/oils), and HFE-7100. Furthermore, they added that the void fraction model used in this correlation was originally formulated under the assumption of annular flow. Since each regime exhibits distinct phase distribution and flow characteristics, the resulting pressure drops can differ considerably. As a result, the accuracy and applicability of the correlation may be limited outside the specific conditions under which it was developed.

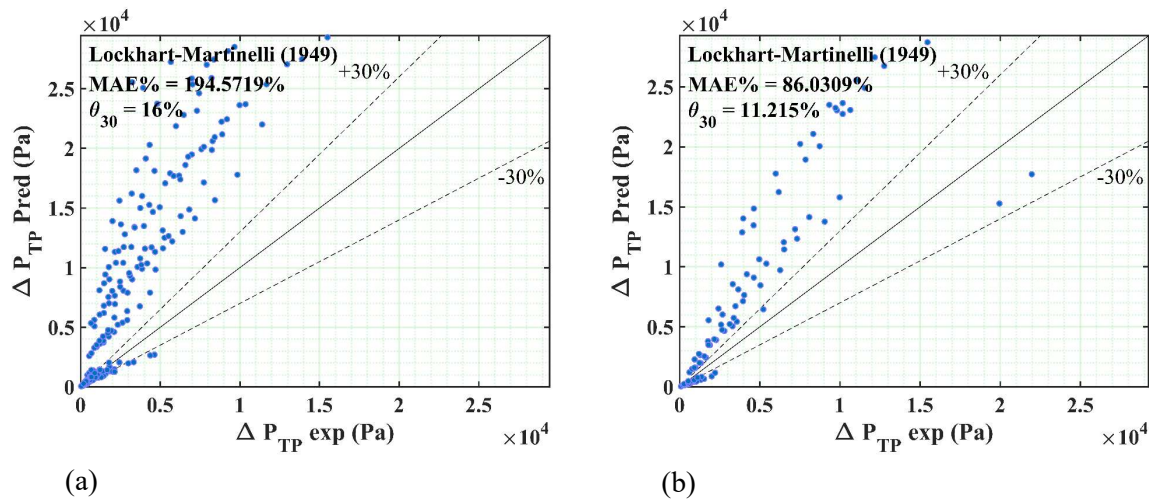


Figure 6.8. Comparison of experimental two-phase pressure drop to prediction by Lockhart and Martinelli [294] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

6.5.2 Comparison to Homogeneous Flow Model

The homogeneous flow model treats the liquid and vapour as a single, combined fluid with no relative motion between the phases, effectively assuming negligible slip [89]. This therefore makes the model particularly useful for dispersed bubbly flows, where the vapour phase moves with the same mean velocity as the liquid phase. The dispersed bubbly flow pattern was not particularly observed in this study. Therefore, it would be expected that the model would not perform particularly well when compared to the present data. However, the homogeneous flow model performed the best out of all investigated prediction tools. The comparison shown in Figure 6.9 shows that the MAE% of the homogeneous flow model was only 10.7 % for the 75 mm long channel test section, whilst 92.8 % of the datapoints fell within the 30 % range. The performance improved further when compared to the 30 mm long channel test section, achieving an MAE of 8.4 % and 93.5 % of the data points falling within the 30 % range.

Other researchers have reported a strong performance of the homogeneous flow model in predicting microscale flow boiling pressure drop. Shiferaw et al. [409] found that their results for R-134a in 0.52 mm to 4.26 mm vertical tubes agreed well with the homogeneous flow model. Al-Zaidi [127] found particularly good agreement for aluminium microchannels with an aspect ratio of 1 using HFE-7100. Mirmanto [410] found that the homogeneous flow model presented the best performance for water in a single microchannel for hydraulic diameters greater than 0.438 mm.

The strong performance of the homogeneous flow model is clearly not limited to the present operating conditions; rather, it appears to perform unexpectedly well across a range of microscale flow boiling applications. One possible explanation for this success lies in the selection of the homogeneous two-phase friction factor. In this study, a constant value of $f_{tp} = 0.003$, as proposed by Qu and

Mudawar [275], was employed, despite many different values and correlations being proposed in literature for conventionally sized tubes [89]. Notably, this value is consistent with the range originally recommended by several authors between the 1930s and 1950s for low-pressure flashing steam-water flows in conventional sized tubes, where suggested two-phase friction factors ranged from 0.0029 to 0.0033 [411–413]. These early recommendations were derived for conditions in which rapid phase change induces large momentum variations, resulting in the acceleration component dominating the total pressure drop.

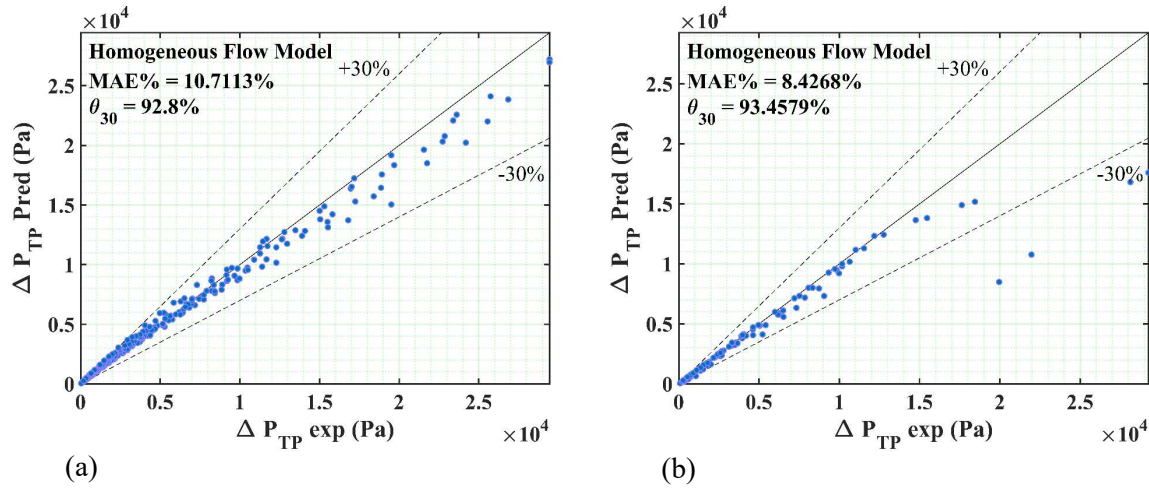


Figure 6.9. Comparison of experimental two-phase pressure drop to homogeneous flow model for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

A similar mechanism may explain the effectiveness of the homogeneous flow model in microscale channels. Due to the high surface area to flow area ratio, vapour generated during boiling becomes confined, leading to rapid acceleration of the vapour phase. As a result, the acceleration component of the pressure drop becomes the primary contributor, overshadowing the frictional component arising from slip between phases. The only regime in which the vapour phase is not confined is bubbly flow; however, in this regime, slip between the liquid and vapour phases is minimal, making the homogeneous flow assumption valid anyway. This behaviour is illustrated in Figure 6.10, which presents the experimentally derived acceleration pressure drop component, given in Equation (6.5) and the frictional pressure drop component, provided in Equation (6.6), using the homogeneous flow model.

$$\Delta P_{acc} = G^2(v_g - v_l)x_{out} \quad (6.5)$$

$$\Delta P_{fric} = \frac{0.006G^2v_lL_{tp}}{D_h} \left[1 + \frac{x_{out}}{2} \left(\frac{(v_g - v_l)}{v_l} \right) \right] \quad (6.6)$$

Figure 6.10 illustrates that the acceleration pressure drop component is significantly larger than the frictional component. The model performs better for the shorter test section, where the predicted acceleration pressure drop is higher, indicating that acceleration is likely the dominant mechanism.

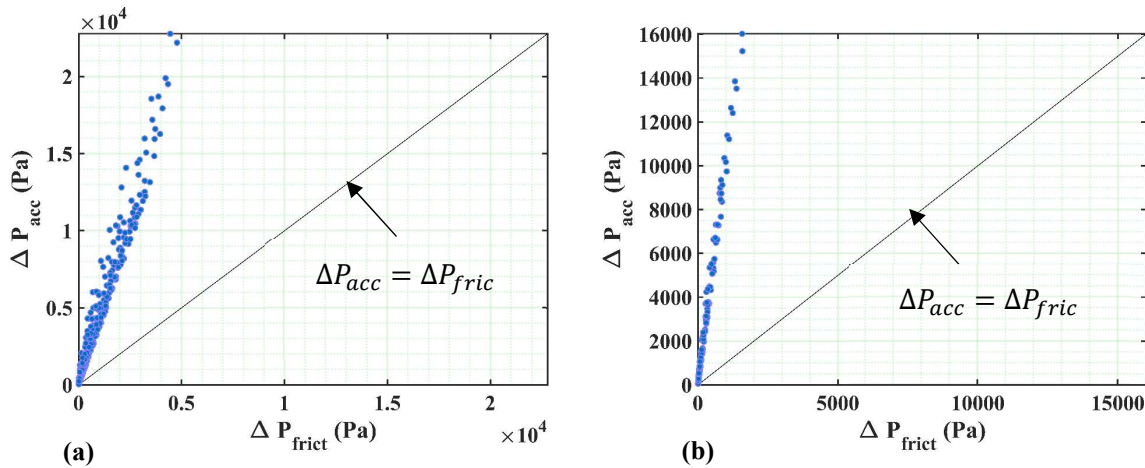


Figure 6.10. Comparison of acceleration and frictional pressure drop components using the homogeneous flow model for: (a) the 75 mm long channel test section; (b) the 30 mm long channel test section.

6.5.3 Comparison to Qu and Mudawar (2003)

Qu and Mudawar [275] conducted an experimental study on flow boiling of water within a horizontal heat sink composed of 21 rectangular microchannels, each with a hydraulic diameter of 0.35 mm. In their analysis, the Chisholm constant was adapted to account for variations in mass flux, allowing their correlation to better reflect the influence of flow rate on pressure drop. The correlation was found to overpredict the data obtained in the presented study, with MAE% of 134.1 % and 56.1 % for the long and short test sections, respectively.

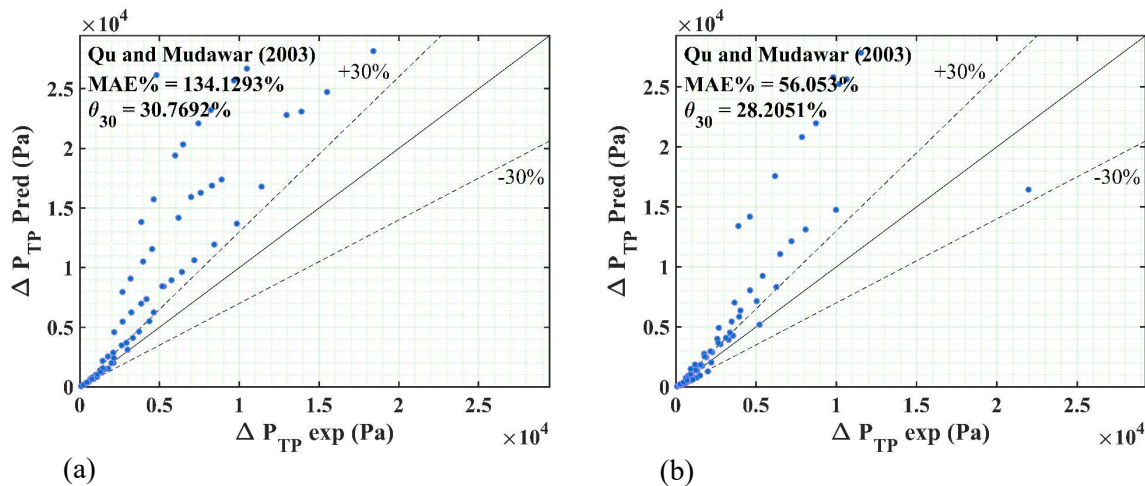


Figure 6.11. Comparison of experimental two-phase pressure drop to prediction by Qu and Mudawar [275] for: (a) 75 mm long channel test section; (b) 30 mm long channel test section.

While the inclusion of mass flux in the Chisholm constant may contribute to improved prediction accuracy under certain conditions, this effect was not consistently observed in the present study. Unlike the lower mass flux range used in Qu and Mudawar's original work (135–400 kg/m²s), the experimental

conditions in this study extended to significantly higher mass fluxes, up to 1000 kg/m²s. At these elevated flow rates, the modified Chisholm constant tended to overestimate the frictional pressure drop, resulting in substantial deviation from the experimental data. This suggests that while the correlation may capture trends at moderate mass fluxes, it does not scale effectively with higher flow rates, limiting its applicability across the full range of operating conditions tested in this study.

6.5.4 Comparison to Lee and Garimella (2008)

Lee and Garimella [217] explored the characteristics of flow boiling in water through horizontally oriented microchannels with rectangular cross-sections. Their experiments spanned a range of channel sizes, with hydraulic diameters from 0.162 mm to 0.571 mm. To better capture the influence of operating conditions, their pressure drop model introduced a modified form of the Chisholm constant that responds to changes in both mass flux and channel geometry. Figure 6.12 presents how this correlation compares to the experimental data obtained in the present work. Again, the model was found to significantly overpredict the data in the current study with large MAE% of 490.5 % and 12.1 % observed.

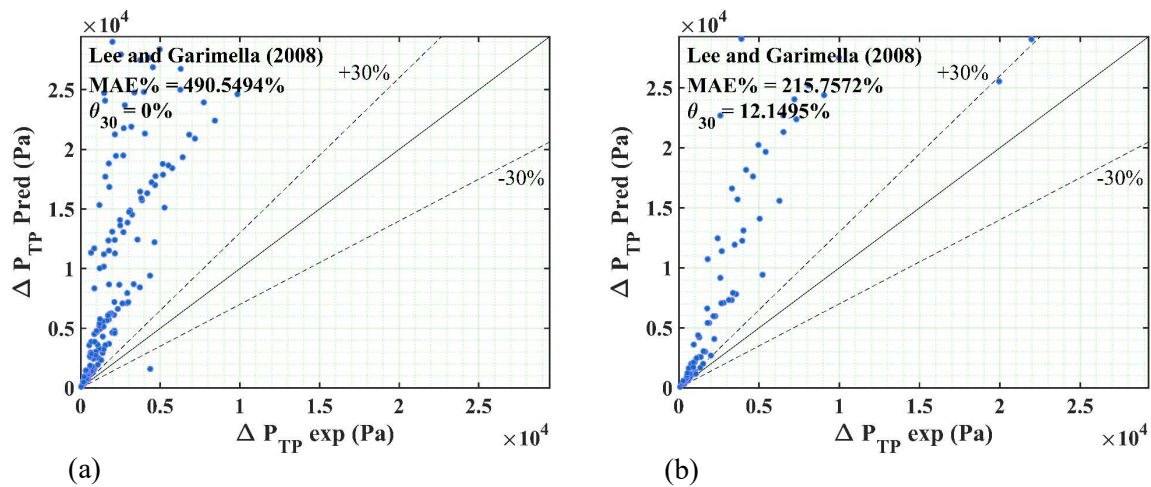


Figure 6.12. Comparison of experimental two-phase pressure drop to prediction by Lee and Garimella [217] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

The limited accuracy of their correlation may stem from the specific flow regime assumptions used in its development. Their formulation of the Chisholm constant within the two-phase frictional multiplier was tailored for conditions where both the liquid and vapour phases are laminar. In contrast, the present study includes not only laminar liquid–laminar vapour flow but also cases where the vapour phase becomes turbulent while the liquid remains laminar. This mismatch in assumed flow regimes likely contributes to the observed discrepancies between the predicted and experimental pressure drops.

6.5.5 Comparison to Keepaiboon et al. (2016)

Keepaiboon et al. [296] carried out an experimental study on flow boiling pressure drop using R-134a as the working fluid. Their experiments were conducted in a horizontal microchannel with a rectangular cross section and a hydraulic diameter of 0.68 mm. Based on their findings, they developed a modified Chisholm constant to be used within the two-phase frictional multiplier. This revised coefficient was expressed in terms of several dimensionless parameters, including the liquid only Reynolds number, the confinement number, and the Martinelli parameter, allowing the model to better reflect microscale flow dynamics. The model found mixed agreement, with an MAE% of 34.8 % for the 75 mm long channel test section; however, only 44.8 % of the data points in the current study fell within the 30 % error margins of the correlation. The performance improved in the 30 mm long channel test section with a lower MAE% of 29.1% and 64.5% of the data points within 30% of the predicted value.

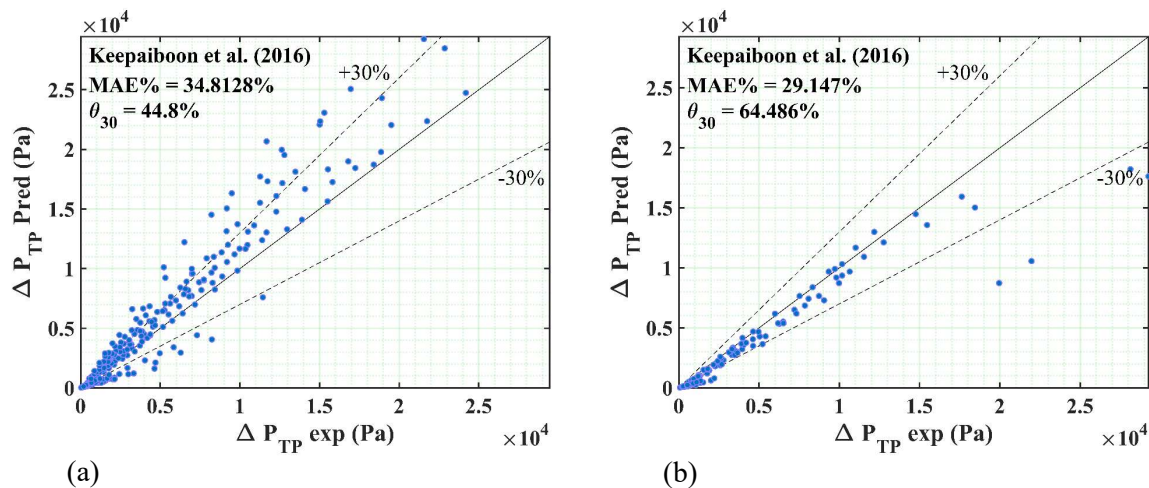


Figure 6.13. Comparison of experimental two-phase pressure drop to prediction by Keepaiboon et al. [296] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

The correlation performed reasonably well, likely due in part to the hydraulic diameter used in the present study, 0.75 mm, being relatively close to the 0.68 mm channel employed in their experiments. This dimensional similarity may have preserved comparable confinement effects. Although their correlation was developed for a single channel and applied here to a similar configuration, its predictive accuracy may still be influenced by differences in fluid properties, working fluid, and flow regimes. Interestingly, the agreement tended to improve at higher mass fluxes, suggesting that the inclusion of the Reynolds number in the Chisholm constant formulation helped capture the pressure drop behaviour more effectively under those conditions. Additionally, the confinement number used in their model accounts for the influence of surface tension and hydraulic diameter, which may have improved its applicability in bubbly and slug flow regimes present in the current experiments. However, some

deviation is still expected due to the inherent sensitivity of two-phase pressure drop to local flow structure and fluid distribution, especially in microchannel flows.

6.5.6 Comparison to Huang and Thome (2017)

Huang and Thome [297] conducted an experimental investigation into flow boiling pressure drop using three refrigerants, namely R-1233zd(E), R-245fa and R-236fa. The study was carried out in a heat sink composed of 67 horizontal square microchannels, each with a hydraulic diameter of 0.1 mm. When comparing their results with existing correlations, they found that models developed for mini and microchannels provided more accurate predictions than those intended for macrochannels. However, these microchannel correlations significantly underpredicted their data when the frictional pressure drop exceeded 15 kPa. To improve predictive accuracy, the authors introduced a revised Chisholm constant that accounted for mass flux and vapour quality, reflecting the high flow rates used in their experiments. This updated coefficient was formulated using the superficial vapour Reynolds number and the liquid-only Reynolds number. The resulting correlation is compared with the present data in Figure 6.14.

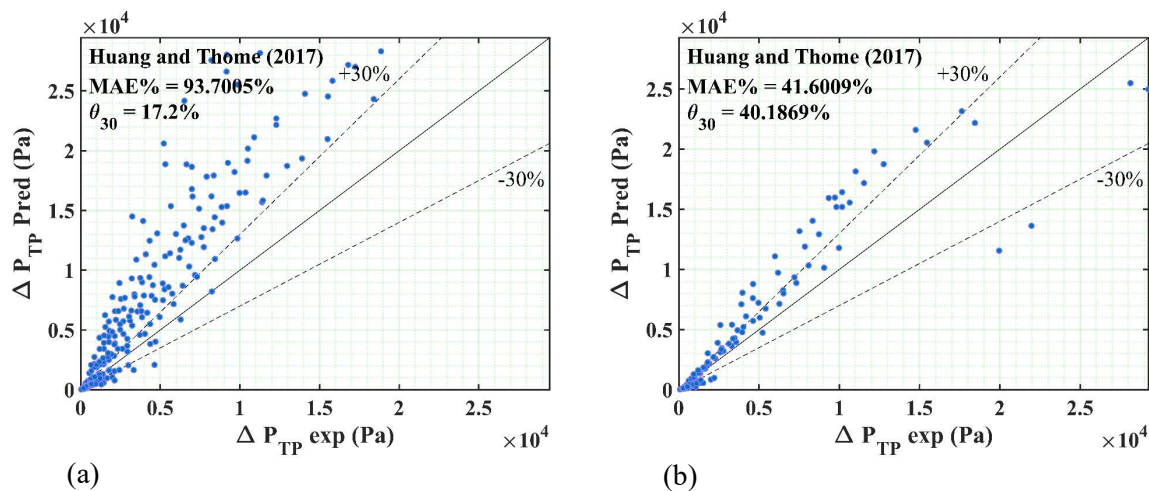


Figure 6.14. Comparison of experimental two-phase pressure drop to prediction by Huang and Thome [297] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

Once again, this model showed a clear tendency to substantially overestimate the pressure drop observed in the present experiments. The resulting mean absolute error percentages were notably high, reaching 93.7 % for the 75 mm test section and 41.6 % for the 30 mm test section. Furthermore, only 17.2 % of data points for the 75 mm test section and 40.2 % for the 30 mm test section fell within a $\pm 30\%$ deviation from the model predictions. The mass fluxes studied during the development of the correlation were between 1250 kg/m²s and 2750 kg/m²s, i.e. outside the range of the mass fluxes observed in this study. Much like the correlations by Qu and Mudawar [275] and Lee and Garimella

[217], it may be that the proposed Chisholm constant did not extend well beyond the parameters used to develop the two-phase pressure drop prediction tool.

6.5.7 Comparison to Markal et al. (2019)

Markal et al. [298] developed two empirical correlations to predict flow boiling pressure drop, based on experiments using DI water in horizontal rectangular multi-microchannels. Their study explored a range of aspect ratios between 0.37 and 5, with hydraulic diameters varying from 0.1 mm to 0.25 mm. The first of these correlations focused specifically on estimating the frictional component of pressure drop, while the second aimed to predict the total two-phase pressure drop under bubble confinement flow conditions, although the precise criteria for such flows were not clearly defined. The first model introduced a modified Chisholm constant expressed as a function of multiple parameters, including aspect ratio, liquid-only Reynolds number, Weber number, Boiling number, two-phase flow length, hydraulic diameter, vapour quality, and the liquid-to-vapour density ratio. A comparison between the predicted two-phase pressure drop correlation and the present experimental results is shown in Figure 6.15.

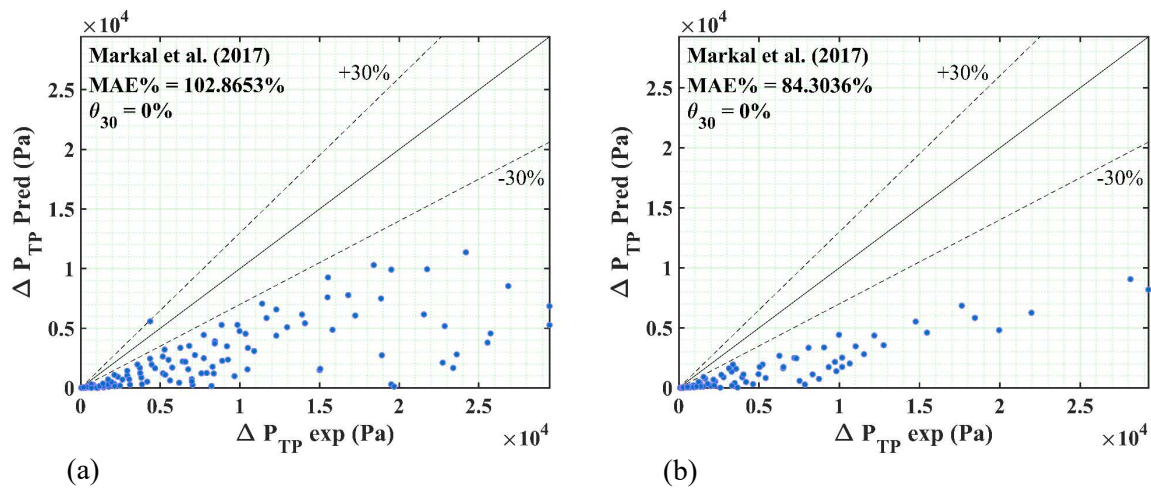


Figure 6.15. Comparison of experimental two-phase pressure drop to prediction by Markal et al. [298] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

The correlation was found to significantly underpredict the pressure drop across both test sections in the present study, yielding high MAE%s of 102.9 % for the 75 mm long channel and 84.3 % for the 30 mm long channel. Despite accounting for channel aspect ratio in their model, the predictive performance remained poor. One possible explanation is the difference in thermophysical properties between the working fluid used in their study, water, and HFE-7100, which was used in the current experiments. This contrast in fluid characteristics likely contributed to the limited applicability of their correlation under the conditions tested here.

6.5.8 Comparison to Gao et al. (2019)

Gao et al. [299] attempted to improve predictions of frictional pressure drop in two-phase flow by developing a new correlation based on experimental observations using ammonia as the working fluid. Their tests were conducted in horizontal stainless-steel tubes with an internal diameter of 8 mm and mass fluxes ranging from 20 kg/m²s to 100 kg/m²s, across a saturation temperature range of −15.8 °C to 4.6 °C. In addition, they incorporated earlier data from experiments using 4 mm tubes. Initial comparisons with established models revealed that the Müller-Steinhagen and Heck [300] correlation offered the closest match. However, recognising limitations in existing methods, particularly under conditions where forces beyond viscosity dominate, they proposed a revised correlation. This new model was developed by extending the Muller-Steinhagen and Heck [300] approach to include key dimensionless parameters: the Reynolds number, representing the balance between inertial and viscous forces, and the Bond number, which reflects the interplay between gravitational and surface tension effects. Their revised model was benchmarked against a comprehensive database that included eleven independent studies using fluids such as R-22, R-134a, CO₂, and ammonia, spanning a wide range of tube diameters (0.81 mm–14 mm) and flow conditions (50 kg/m²s – 800 kg/m²s). Figure 6.16 compares the predictive performance of their modified correlation with the current study's results.

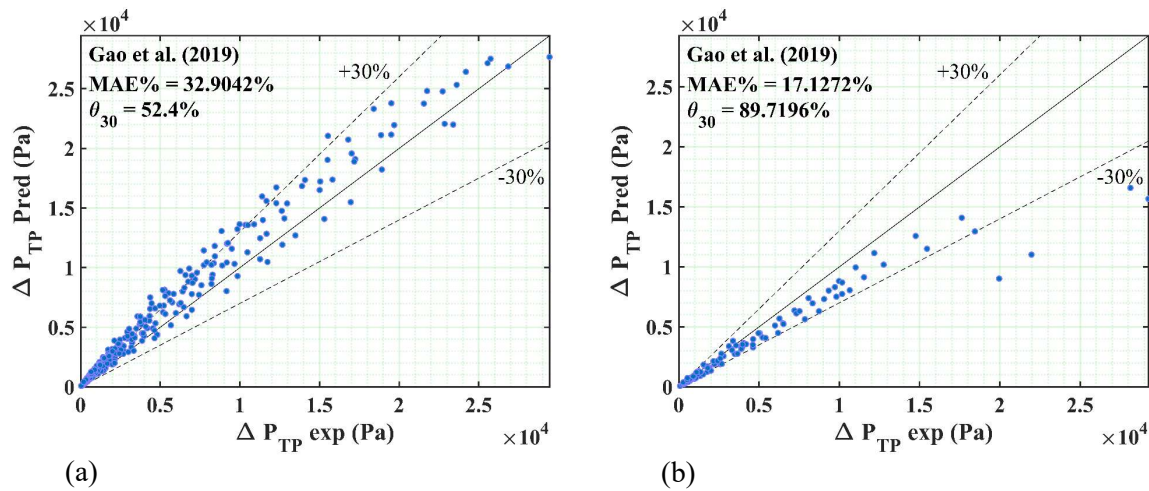


Figure 6.16. Comparison of experimental two-phase pressure drop to prediction by Gao et al. [299] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

The proposed correlation delivered mixed results overall but was the most accurate among the microscale models evaluated. It yielded a mean absolute error of 32.9 % for the 75 mm long channel and improved performance in the 30 mm test section with an MAE% of just 17.1 %. Interestingly, this stronger agreement in the shorter channel is somewhat unexpected, given that the correlation was developed using data from a much longer 1000 mm test section. This relatively strong agreement is likely influenced by the similarity between the present operating conditions and those included in the database used to develop the correlation. The dataset considered by Gao et al. [299] spans tube

diameters from 0.81 mm to 14 mm and mass fluxes from 50 kg/m²s to 800 kg/m²s, which closely overlap with the present study's hydraulic diameters of approximately 0.78 mm and mass flux range of 100 kg/m²s to 1000 kg/m²s. In addition, several refrigerants with thermophysical properties comparable to the present working fluid were included in their database.

The inclusion of the Bond number may also contribute to the improved performance of the correlation. The Bond number represents the relative importance of gravitational and surface tension forces and therefore helps account for the increased dominance of surface tension effects in microscale channels. In small-diameter tubes, where capillary forces strongly influence flow structure and pressure drop, the use of this parameter can improve the scaling of the model across different channel sizes and fluid properties. However, it should be noted that several other correlations also incorporate the Bond number or similar dimensionless groups, yet do not demonstrate the same level of agreement with the present data. This suggests that the improved predictive capability of the Gao et al. [299] correlation is not solely due to the inclusion of the Bond number, but more likely results from the combination of appropriate scaling parameters and the strong overlap between the present operating conditions and those represented in the correlation's development database.

6.5.9 Comparison to Moradkhani et al. (2021)

A comprehensive empirical model was developed by Moradkhani et al. [301] to estimate saturated flow boiling frictional pressure drop across both macro- and mini/micro-scale channels. Their approach employed genetic programming to derive an explicit two-phase multiplier based on a modified form of the classical Lockhart-Martinelli framework. Further details regarding genetic programming and this correlation have been presented in Chapter 2. The model was trained and validated using an extensive dataset of 6021 experimental points compiled from 42 different sources, encompassing a wide range of channel sizes (0.0695 mm to 14 mm), working fluids, and mass fluxes (50 kg/m²s to 1504 kg/m²s). Inputs of the Reynolds number ratio (Re_{go}/Re_{lo}), Bond number, exit vapour quality, pressure drop ratio and reduced pressure were used in the genetic programming. Figure 6.17 presents the comparison between the correlation and the current study.

Again, the performance of the correlation was significantly mixed, presenting MAE values of 95.4 % and 34.1 % for the 75 mm and 30 mm channel test sections, respectively. The correlation was found to overpredict the pressure drop in the 75 mm channel, while underpredicting it in the 30 mm channel.

A plausible explanation for the poor performance can be found in the composition of the database used to develop the correlation. Moradkhani et al. [301] constructed their model using 39 datasets, the majority of which were obtained in circular channels. Only three datasets were based exclusively on rectangular channels [171, 406, 414], with one additional dataset including a mixture of circular, rectangular, and triangular geometries [415]. Even within the training data, the correlation performed

significantly worse for the rectangular-channel datasets, with reported MAE values of 49.59 %, 44.11 %, and 34 %, compared with an overall MAE of 21.34 % for the full database.

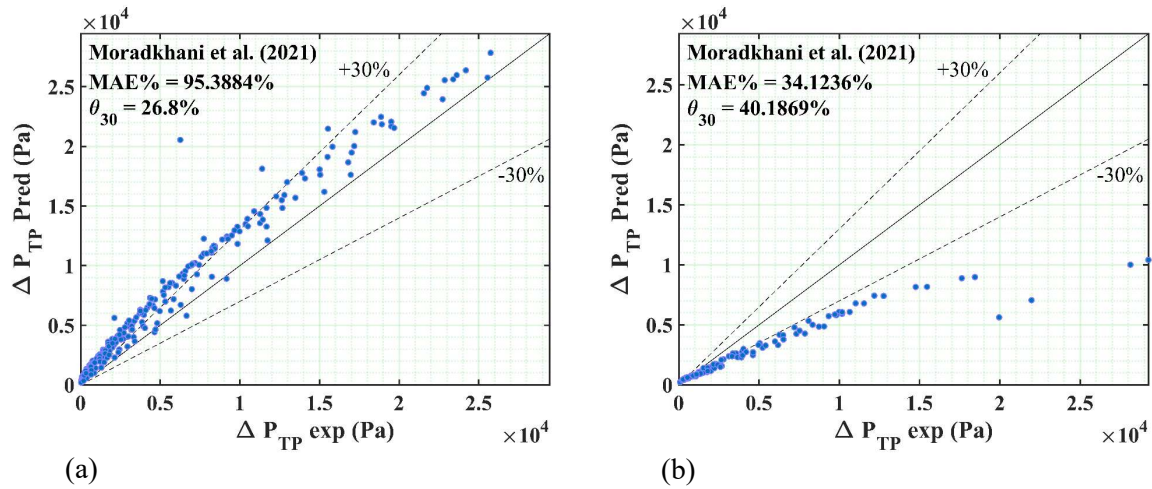


Figure 6.17. Comparison of experimental two-phase pressure drop to prediction by Moradkhani et al. [301] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

Furthermore, the independent validation performed by Moradkhani et al. [301] used four additional datasets not used to develop the model, all obtained in circular channels, for which the correlation achieved an MAE% of 21.32 %. No independent validation was conducted using rectangular channel data. This suggests that the model was primarily tuned and verified against circular channel data and may not adequately capture the differences that arise in rectangular microchannels. The relatively small proportion of rectangular-channel data in the training set, combined with their higher associated prediction errors, indicates that the model does not generalise as effectively to non-circular geometries. This likely contributes to the poor agreement observed for the present rectangular microchannel experiments.

However, the good performance reported for circular channels indicates that the inclusion of the Bond number in the correlation may still provide a useful scaling of surface tension and gravitational effects across different channel sizes. This suggests that the Bond number may be a contributing parameter in capturing the pressure drop behaviour, particularly in conventional and circular microchannels, although it is not sufficient on its own to ensure accurate predictions for non-circular geometries.

6.5.10 Comparison to Al-Zaidi et al. (2024)

Al-Zaidi et al. [303] developed two correlations for microscale pressure drop using a large microscale pressure drop databank for vertical tubes and horizontal heat sinks. Fluids R-134a, R-245fa, HFE-7100 and HFE-7200 were included in the databank, with hydraulic diameters 0.46 mm to 4.26 mm, heated lengths of 20 mm to 500 mm, system pressures of 1 bar to 14 bar, mass fluxes of 50 kg/m²s to 700 kg/m²s and wall heat fluxes up to 234 kW/m². The correlations attempted to predict the two-phase multipliers, which were found to be strongly dependent on the Boiling number, although the Martinelli parameter, the superficial Reynolds number and the reduced pressure were also used in both correlations. A comparison to the multichannel heat sink correlation is presented in Figure 6.18, whilst the comparison to the vertical tube correlation is displayed in Figure 6.19.

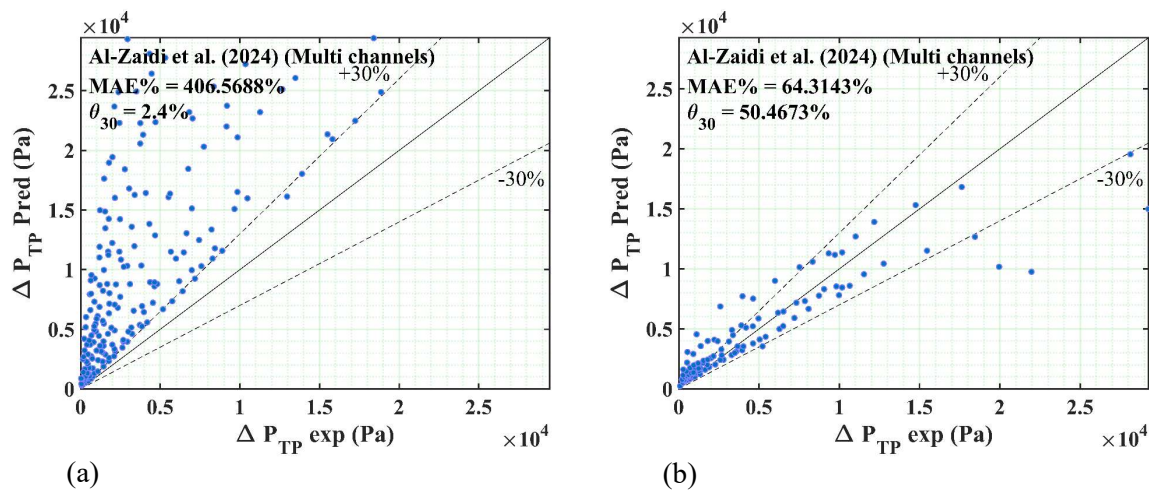


Figure 6.18. Comparison of experimental two-phase pressure drop to multi-channel heat sink prediction by Al-Zaidi et al. [303] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

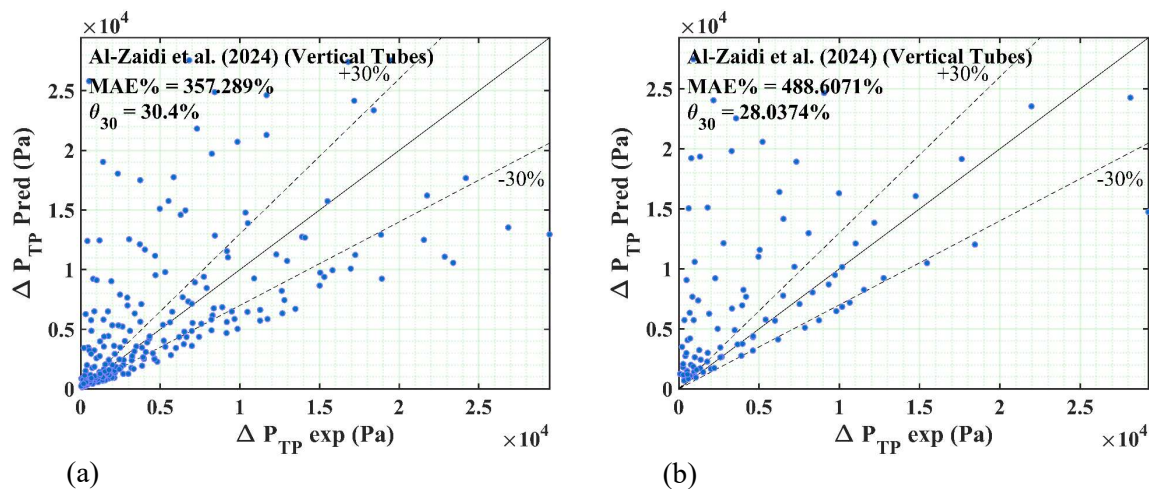


Figure 6.19. Comparison of experimental two-phase pressure drop to vertical tube prediction by Al-Zaidi et al. [303] for; (a) 75 mm long channel test section; (b) 30 mm long channel test section.

The correlation for multichannels did not perform well for either of the test sections; however, the correlation performed significantly worse for the longer test section with a derived MAE% of 406.6 %. Assessing the experiments used for the development, it can be seen that the heated length of the test sections used to develop the correlation for horizontal microchannels were only between 20 mm and 25 mm. Thus, the effect of heated length may not have been accurately represented in the correlation. The MAE% in comparison to the results for the shorter, more comparable, 30 mm long test section improved significantly, although the MAE% was still rather high at 64.3 %. Referring to the mass fluxes used in the experiments, it can be seen that the correlation was only written using mass fluxes between 50 and 400 kg/m²s, which is at the lower range of what was investigated in the current study. Therefore, the correlation may also not capture the effect of frictional pressure drop at higher mass fluxes. Finally, only two hydraulic diameters were used in this study, namely 0.46 mm and 0.48 mm. These two values are very close and therefore the effect of hydraulic diameter may not have been fully captured in this study.

The empirical correlation for vertical tubes generally performed poorly across both test sections, achieving an MAE% of 357.3 % for the 75 mm long channel test section and 488 % for the 30 mm long channel test section. As discussed in Chapters 4 and 5, nucleation was constantly observed in the film across all flow patterns, which may have only been permissible due to the geometry of the rectangular test section. This is discussed further in Chapter 7. Hence, the overall mechanism of frictional pressure drop between phases in slug, churn and annular flow regimes may be significantly different due to the slippage between bubbles in the liquid film. This change in mechanism between cylindrical tubes and rectangular microchannels may be a potential reason why the correlation did not accurately predict the pressure drop in a single rectangular microchannel.

The Lockhart and Martinelli [294] correlation, although widely used, consistently overpredicted the pressure drop for both channel lengths. This poor agreement was attributed to the correlation's original development for macroscale flows and its assumption of annular flow conditions, which were not consistently present in the current study. Similarly, other traditional two-phase multipliers, including those by Qu and Mudawar [275] and Lee and Garimella [217], produced large errors, especially at higher mass fluxes or in longer channels, suggesting a limited applicability outside their original development parameters.

In contrast, the homogeneous flow model performed unexpectedly well, providing the closest match to the experimental data across both channel lengths. This was surprising, but agrees with other studies reporting similar findings in microscale geometries. Potential reasoning was proposed using the two-phase friction factor used which was originally proposed for flashing of steam-water mixtures. The rapid vapour acceleration that relates to flashing steam-water mixtures may be similar to the confined

vapour generation occurring in microchannels, actually making the acceleration component dominate. The simplicity of the homogeneous model and its consistent performance make it a useful benchmark.

Among the microscale-specific correlations, Gao et al.'s [299] model delivered the most accurate predictions, particularly for the 30 mm test section. This improved agreement is likely influenced by the strong overlap between the operating conditions used in the present study and those included in the database from which the correlation was developed, including similar tube diameters, mass flux ranges, and refrigerant-type working fluids. The incorporation of the Bond number and Reynolds number into the model may also have contributed to the improved performance by helping to capture microscale effects such as surface tension dominance and phase confinement. Other microscale models, such as those by Keepaiboon et al. [296] and Moradkhani [301], showed improved performance at shorter lengths and higher mass fluxes but were less accurate overall. This may reflect their reliance on datasets not fully representative of the current flow conditions, particularly in terms of working fluid, geometry and flow regime.

6.6 CONCLUSIONS

This chapter presented a detailed investigation into two-phase flow boiling pressure drop across microchannels of varying lengths, mass fluxes, heat fluxes and system pressures using HFE-7100 as the working fluid. The effects of key operating parameters were systematically explored, and the experimental results were compared to a range of existing pressure drop correlations. The following bullet points summarise the principal findings and observations drawn from the analyses conducted throughout this chapter:

- The two-phase pressure drop in the 75 mm test section increased with wall heat flux due to increased vapour generation, elevated interfacial shear, and enhanced momentum exchange between the liquid and vapour phases. At lower heat fluxes, higher mass fluxes produced lower pressure drops, consistent with bubbly flow and limited interfacial disruption. However, as the heat flux increased and the flow transitioned to slug and annular regimes, the pressure drop increased with mass flux due to higher vapour velocities and intensified interfacial shear. For a given outlet vapour quality, pressure drop consistently increased with mass flux across all inlet pressures, confirming the combined influence of flow velocity and regime transitions on both frictional and acceleration components.
- Increasing system pressure resulted in an overall reduction in pressure drop. This behaviour was linked to both delayed flow regime transitions and changes in thermophysical properties. As pressure increased, the liquid to vapour density and viscosity ratios decreased, influencing both acceleration and frictional components of the pressure drop. The density ratio was found to have a stronger effect on the acceleration component due to its influence on void fraction and phase

velocity differences. Although surface tension, density ratio, and viscosity ratio affected the frictional pressure drop in competing ways, the overall trend at higher pressures was a reduction in pressure drop, primarily driven by changes in viscosity.

- The shorter, 30 mm heated test section exhibited significantly higher pressure drop per unit length than the 75 mm channel at the same outlet vapour quality. This was attributed to the higher wall heat fluxes required in the shorter channel to achieve the same vapour quality, resulting in more intense bubble nucleation, increased interfacial interactions, and steeper pressure gradients. The more aggressive boiling conditions in the shorter channel therefore led to greater frictional losses.
- Comparison with existing pressure drop correlations revealed large discrepancies in most cases, particularly for models developed for macroscale channels or different fluids. The homogeneous flow model provided the best overall agreement with the experimental data, especially for the 30 mm channel, suggesting that the acceleration component may dominate the total pressure drop under the present conditions.
- Among the microscale-specific correlations, the model of Gao et al. [299] produced the closest agreement with the present data. This improved performance is likely related to the strong overlap between the experimental conditions used in this study and those included in the correlation's development database, as well as the inclusion of parameters such as the Bond and Reynolds numbers to account for microscale effects.

CHAPTER 7

CRITICAL HEAT FLUX RESULTS

7.1 INTRODUCTION

This chapter presents the experimental results from the investigation of critical heat flux in the 30 mm long channel microchannel test section. As seen in Chapters 4, 5 and 6, critical heat flux was not reached in the 75 mm long channel as either vapour qualities of unity were achieved, or the pressure drop across the channel did not permit a constant channel inlet pressure to be obtained. Section 7.2 focuses on the experimental results and discusses the observed CHF mechanisms through both quantitative data and high-speed flow visualisation. The progression of interfacial flow phenomena leading up to CHF is described in detail, with particular emphasis on the relationship between local flow structure, dryout behaviour, and the ultimate heat flux limit. In Section 7.3, the measured CHF data are systematically compared with a range of existing microscale CHF prediction tools. Finally, Section 7.4 summarises the key findings and highlights the implications of the results for understanding and modelling CHF in microscale boiling systems.

7.2 CRITICAL HEAT FLUX MECHANISMS

The boiling curves for the 30 mm long channel test section are presented in Figure 7.1, despite already being presented in Chapter 5. However, noted in Figure 7.1 is the dryout region and the CHF. As was also discussed in Chapter 5, evidence of TOS was recorded at each inlet pressure tested, but the phenomenon occurred inconsistently, appearing only intermittently and not at all mass fluxes. Figure 7.2 presents the two-phase heat transfer coefficients in the 30 mm long channel test section, showing no effect of mass flux until the dry-out region. The influence of mass flux in the dry-out region is illustrated by the average two-phase HTC's presented in Figure 7.2, which indicate that increasing mass flux delays the onset of dry-out and consequently results in higher heat transfer coefficients.

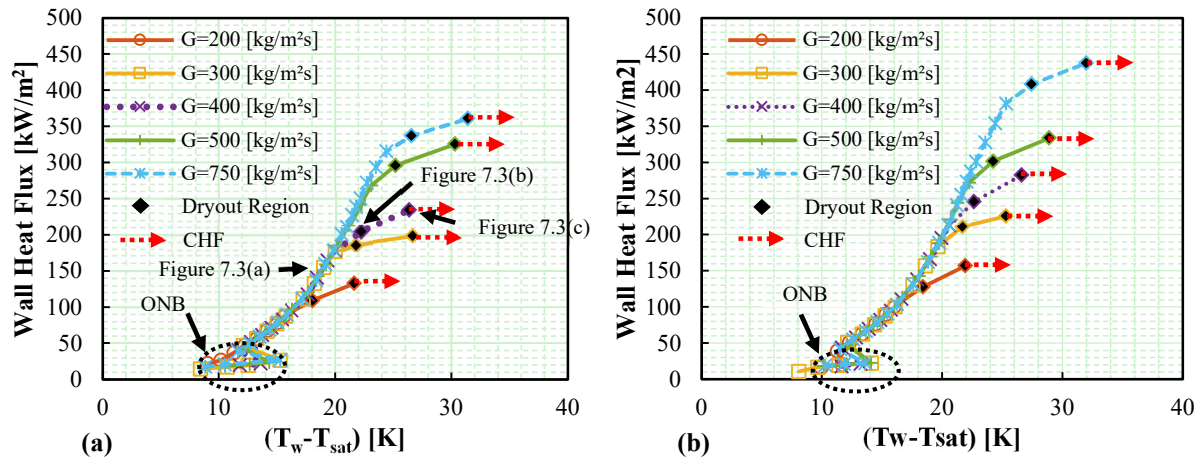


Figure 7.1. Boiling curves at $z/L = 0.5$ for the 30 mm long channel test section with CHF indicated and inlet pressure of: (a) 1.5 bar (b) 2 bar.

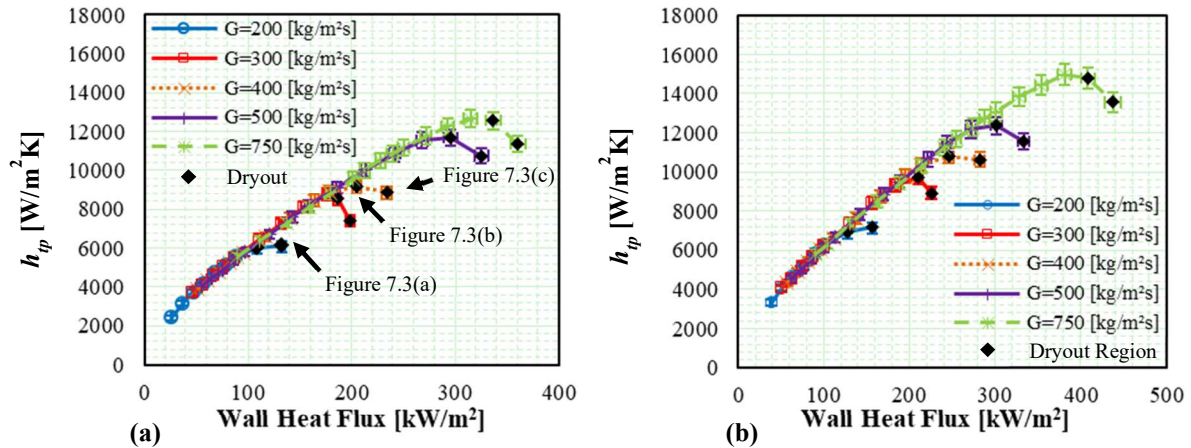


Figure 7.2. Average two-phase heat transfer coefficient for the 30 mm long channel test section and inlet pressure of: (a) 1.5 bar (b) 2 bar. Points indicated refer to flow patterns seen in Figure 7.3.

It should be emphasised that the CHF was identified as the last recorded data point prior to the onset of the expected rapid rise in wall temperature, since stable operation at CHF cannot be maintained because of the very high wall temperatures that develop. Moreover, the wall heat flux was increased in discrete steps, which restricted the ability to approach the exact CHF value with high accuracy. Despite this limitation, a distinct reduction in the two-phase heat transfer coefficient was observed as CHF was approached for almost all tested mass fluxes at both inlet pressures. To help interpret this behaviour, flow visualisation was carried out at the channel outlet for three heat flux conditions close to CHF. The corresponding images, linked to the heat flux levels indicated in Figures 7.1 and 7.2, are shown in Figure 7.3.

Figure 7.3(a) illustrates annular flow with intense bubble activity occurring within a comparatively thick liquid film at a mass flux of $400 \text{ kg/m}^2\text{s}$ and a wall heat flux of 140.72 kW/m^2 , for an inlet pressure

of 1.5 bar. Under these conditions, the influence of mass flux on the heat transfer behaviour is not yet apparent, which is consistent with the trends presented in Figure 7.2. It should nevertheless be recognised that the CHF increases with increasing mass flux. In Figure 7.3(b), corresponding to a wall heat flux of 204.52 kW/m^2 , the liquid film has become substantially thinner. Bubble nucleation is no longer visible at the outlet, and multiple dry patches have started to form along the heated surface. Even with the onset of local dryout and the absence of nucleation in this region, the average two-phase heat transfer coefficient continues to rise with increasing wall heat flux. This behaviour is most likely associated with enhanced nucleate boiling in the upstream section of the channel that remains fully wetted, where the higher applied heat flux promotes more vigorous nucleation and contributes significantly to the overall heat transfer.

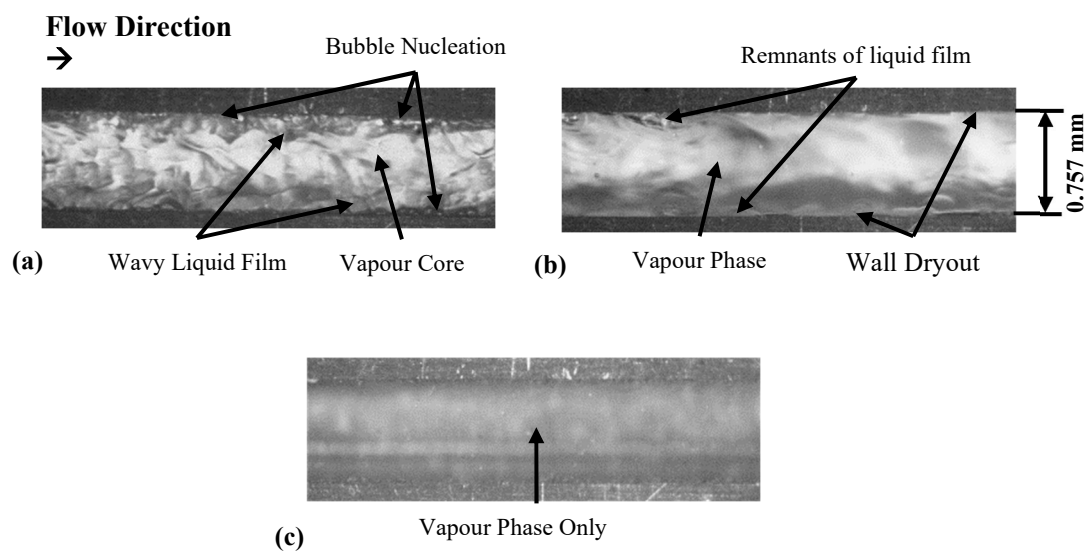


Figure 7.3. Flow visualisation at the 30 mm long channel outlet for $400 \text{ kg/m}^2 \text{ s}$ mass flux, inlet pressure of 1.5 bar and wall heat fluxes of: (a) 140.72 kW/m^2 (b) 204.52 kW/m^2 (c) 232.13 kW/m^2 .

Figure 7.3(c) illustrates the channel under fully dried-out conditions at a wall heat flux of 232.13 kW/m^2 , although this state was not sustained. Because steady operation at CHF is not possible, flow visualisation could only be conducted for a short period, approximately one minute, after the onset of CHF. Beyond this point, the wall temperature increased to a level that required the electrical input to the test section to be shut down in order to protect the experimental apparatus. This brief observation period was possible due to the relatively low latent heat of vaporisation of the working fluid compared with water, combined with the comparatively large thermal mass of the copper test block relative to the mass flow rate. These factors produced a slower, though still exponential, increase in wall temperature. The sequence of flow visualisation images captured during this temperature rise at CHF is presented in Figure 7.4.

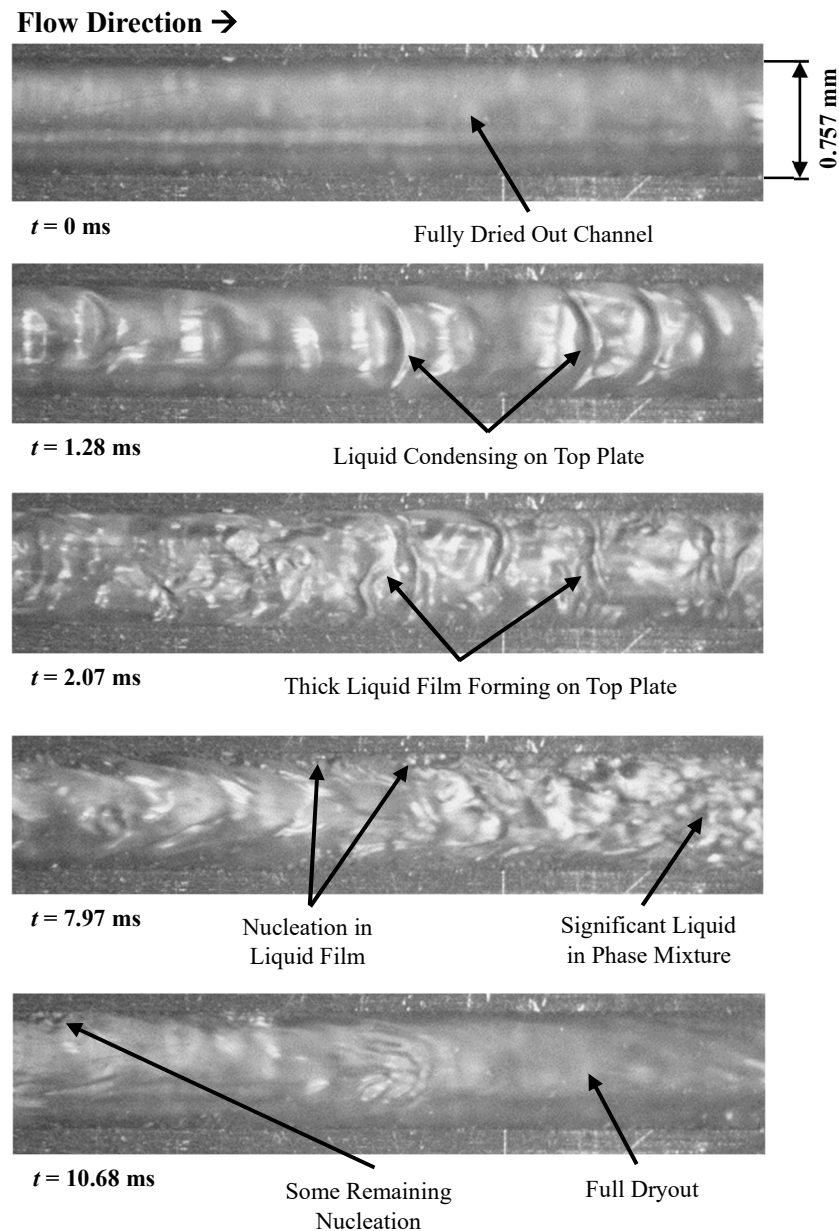


Figure 7.4. Outlet flow visualisation during rise of wall temperature at CHF (after 234.13 kW/m^2) for the 30 mm long channel at mass flux of $400 \text{ kg/m}^2 \text{ s}$ and an inlet pressure of 1.5 bar.

The first frame in Figure 7.4, captured at $t = 0$ ms, shows the outlet region of the channel in a fully dried-out condition, with vapour occupying the entire cross-section. At this point, the upstream flow appeared to stop, and the inlet pressure increased rapidly. This behaviour suggests that vapour accumulation in the outlet plenum obstructed the flow, producing a backpressure and raising the overall channel pressure.

At $t = 1.28$ ms, condensation became visible on the polycarbonate viewing plate. This may have occurred either because the stationary vapour created favourable conditions for condensation on the cooler top surface, or because the pressure rise increased the saturation temperature of the fluid,

producing a sufficient temperature difference between the vapour and the viewing plate. The mass of condensed vapour was likely enough to partially relieve the vapour blockage, allowing a surge of mixed liquid and vapour into the outlet region, as seen at $t = 7.97$ ms. This inflow of liquid temporarily rewetted the channel walls and re-established nucleation, causing the flow to briefly resemble annular conditions. As the liquid film subsequently thinned, dry patches reappeared and gradually expanded until the outlet returned to a fully dried-out state at $t = 10.68$ ms. This cyclic dryout behaviour was observed at a wall heat flux of 232.13 kW/m^2 . However, the duration of complete outlet dryout at this condition was significantly shorter than that observed during the approach to CHF. At 232.13 kW/m^2 , the fully dry period lasted approximately 0.823 ms, whereas a dryout duration of about 5.12 ms was recorded during the wall-temperature rise leading to CHF. The latter value may have increased further as the wall temperature continued to rise, although only a short visualisation interval was captured.

The periodic flow cycle at the outlet, shown in Figure 7.4, helps explain why steady wall temperatures were achieved at 232.13 kW/m^2 but not at higher heat fluxes. The inflow of liquid and the associated nucleation between approximately 7.97 ms and 10.98 ms likely produced high local heat transfer coefficients, whereas the dry periods reduced the overall heat removal capability. At this heat flux, the time-averaged heat transfer over successive wetting and drying cycles was likely sufficient to maintain stable conditions. However, once a critical wall temperature was reached, the duration of the dry periods increased rapidly, leading to thermal instability and the onset of CHF.

The blockage of the outlet by vapour in the fully dry state is linked to the dryout of the annular film. Figure 7.1 shows that higher CHF values were associated with increased inlet pressures and mass fluxes, while Figure 7.2 indicates that the average two-phase heat transfer coefficient also rose with system pressure. Figure 7.5 compares outlet annular flow visualisations at different inlet pressures for a mass flux of $400 \text{ kg/m}^2\text{s}$ and a wall heat flux of approximately 200 kW/m^2 . At an inlet pressure of 1.5 bar, dry patches formed on the surface without clear nucleation. In contrast, at 2 bar, the liquid film appeared thicker and active nucleation sites were visible, suggesting that the film thickness was sufficient to sustain bubble formation.

These observations suggest that variations in annular film thickness with system pressure may play an important role in the heat transfer behaviour and the onset of dryout. It is generally recognised that pressure can influence the liquid film thickness during annular flow. However, the literature presents some disagreement regarding this effect for refrigerants in metallic channels. Some studies have reported an increase in film thickness with increasing pressure [416, 417], while others have observed a reduction under similar conditions [262].

A range of techniques has been used to measure liquid film thickness, including optical reflection methods, high-speed image processing, and laser confocal displacement meters. These approaches are typically applied in transparent tubes or in metallic channels incorporating a transparent section. In the

present experimental configuration, which employed a metallic channel with a transparent top cover, accurate measurement of the liquid film thickness was not feasible. Therefore, an idealised annular flow model, assuming a circular vapour core, was adopted to estimate the liquid film thickness at the channel corners, as illustrated in Figure 7.6. In addition, the interfacial shear stress between the vapour core and the liquid film was evaluated, as this parameter can influence the film thickness and help clarify the effect of system pressure.

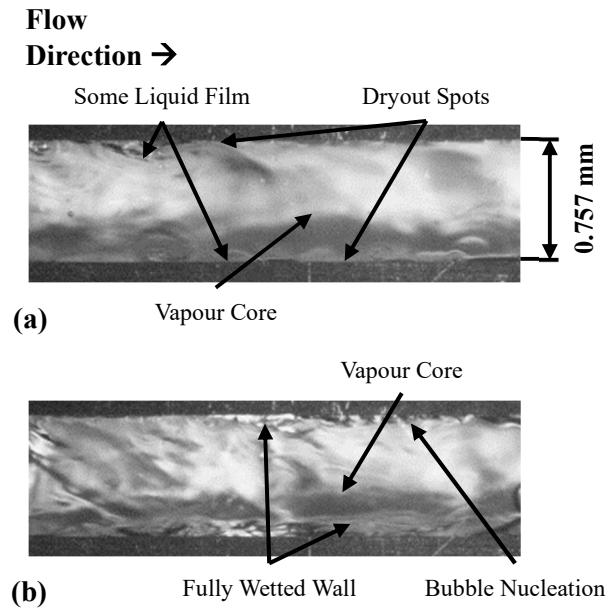


Figure 7.5. Outlet annular flow visualisation for the single channel, 400 kg/m²s mass flux and approximately 200 kW/m² for inlet pressures of: (a) 1.5 bar (b) 2 bar.

The liquid film thickness at the channel corners was determined using Equation (7.1):

$$\delta = \sqrt{\left(\frac{H_{ch}}{2}\right)^2 + \left(\frac{W_{ch}}{2}\right)^2} - r \quad (7.1)$$

The radius of the vapour core, was obtained from the void fraction, since the void fraction represents the ratio of the vapour cross-sectional area to the total channel cross-sectional area [418]:

$$\alpha = \frac{A_g}{A_{sec}} \quad (7.2)$$

$$A_g = \pi r^2 \quad (7.3)$$

The void fraction was estimated using the correlation proposed by Zivi [419], given by Equation (7.4):

$$\alpha = \left[1 + \frac{1-x}{x} \left(\frac{\rho_g}{\rho_l} \right)^{0.67} \right]^{-1} \quad (7.4)$$

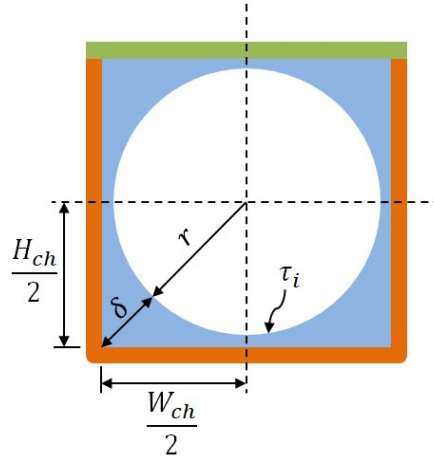


Figure 7.6. Idealised scenario of annular flow with perfectly circular vapour core cross-section.

The interfacial shear stress between the vapour core and the liquid film was calculated following the approach described by Patel et al. [195]:

$$\tau_i = \frac{1}{2} \rho_g f_i u_g^2 \quad (7.5)$$

where the vapour velocity is defined as:

$$u_g = \frac{\dot{m}_g}{\rho_g A_g} \quad (7.6)$$

and the vapour mass flow rate is given by:

$$\dot{m}_g = x G A_{sec} \quad (7.7)$$

The interfacial friction factor, f_i , was evaluated using the annular flow correlation proposed by Moeck [420]:

$$f_i = 0.005 \left[1 + 1458 \left(\frac{\delta}{D_h} \right)^{1.42} \right] \quad (7.8)$$

Figure 7.7 presents the calculated liquid film thickness and interfacial shear stress at two different inlet pressures. The results correspond to a single channel with dimensions of 0.75×0.75 mm, a length of 30 mm, a mass flux of $400 \text{ kg/m}^2\text{s}$, an inlet subcooling of 10 K, and applied heat fluxes up to 300 kW/m^2 .

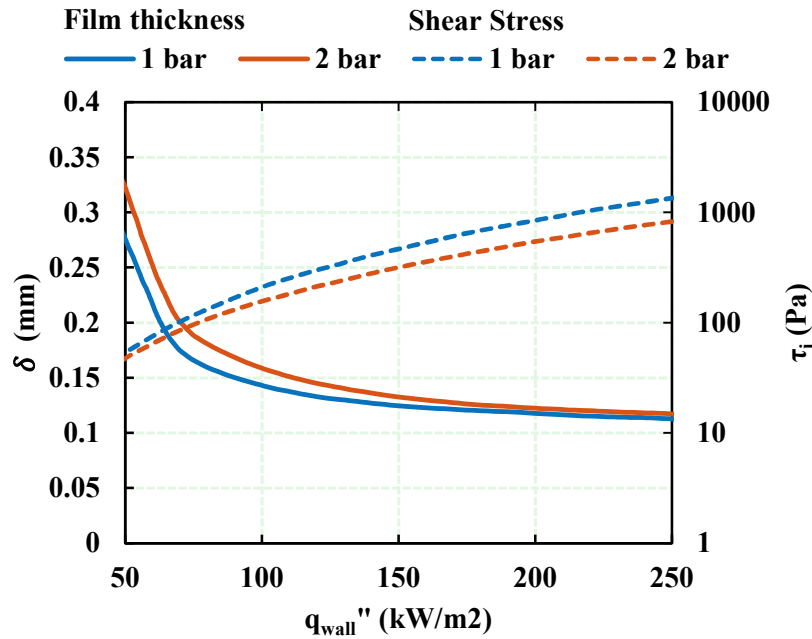


Figure 7.7. Liquid film thickness and interfacial shear stress of idealised scenario of annular flow.

Figure 7.7 indicates that the liquid film thickness decreases with increasing heat flux, or equivalently with increasing vapour quality, which is consistent with expected annular flow behaviour. At the same time, higher heat flux conditions produce greater interfacial shear stress due to the increased vapour superficial velocity. The figure also shows that raising the inlet pressure from 1 to 2 bar results in a reduction in interfacial shear stress and a corresponding increase in liquid film thickness. This behaviour is linked to the variation of fluid thermophysical properties with pressure. Increasing the inlet pressure from 1 to 2 bar reduces the liquid-to-vapour density ratio by approximately 51%, while the latent heat of vaporisation decreases by only about 7%, as summarised in Table 6.1. The reduced density ratio lowers the vapour superficial velocity, which in turn decreases the interfacial shear stress acting on the liquid film. As a result, less liquid is entrained from the wall, promoting the formation of a thicker liquid film at higher pressures.

The presence of a thicker film can delay the development of dryout regions and, consequently, the onset of CHF. It may also support enhanced nucleation activity within the annular flow regime, as suggested by the present flow visualisation results. In addition, higher pressures are known to produce smaller nucleating bubbles [242], which further increases the likelihood of nucleation within a thicker liquid film.

7.3 COMPARISON TO EXISTING MICROSCALE CRITICAL HEAT FLUX PREDICTION TOOLS

Figure 7.8 presents a comparison between the experimentally determined CHF values for the single microchannel and eight existing CHF prediction correlations developed for small to microscale passages. The correlations are presented in full in Appendix C. The comparison is shown in the form of a thermal map. Black symbols indicate the CHF points determined experimentally. Orange symbols mark the onset of dry-out, and green symbols represent heat flux values below these conditions, corresponding to the safe operational range of the heat sink in a thermal management system. Operating within this green region avoids CHF and the associated risk of catastrophic system failure.

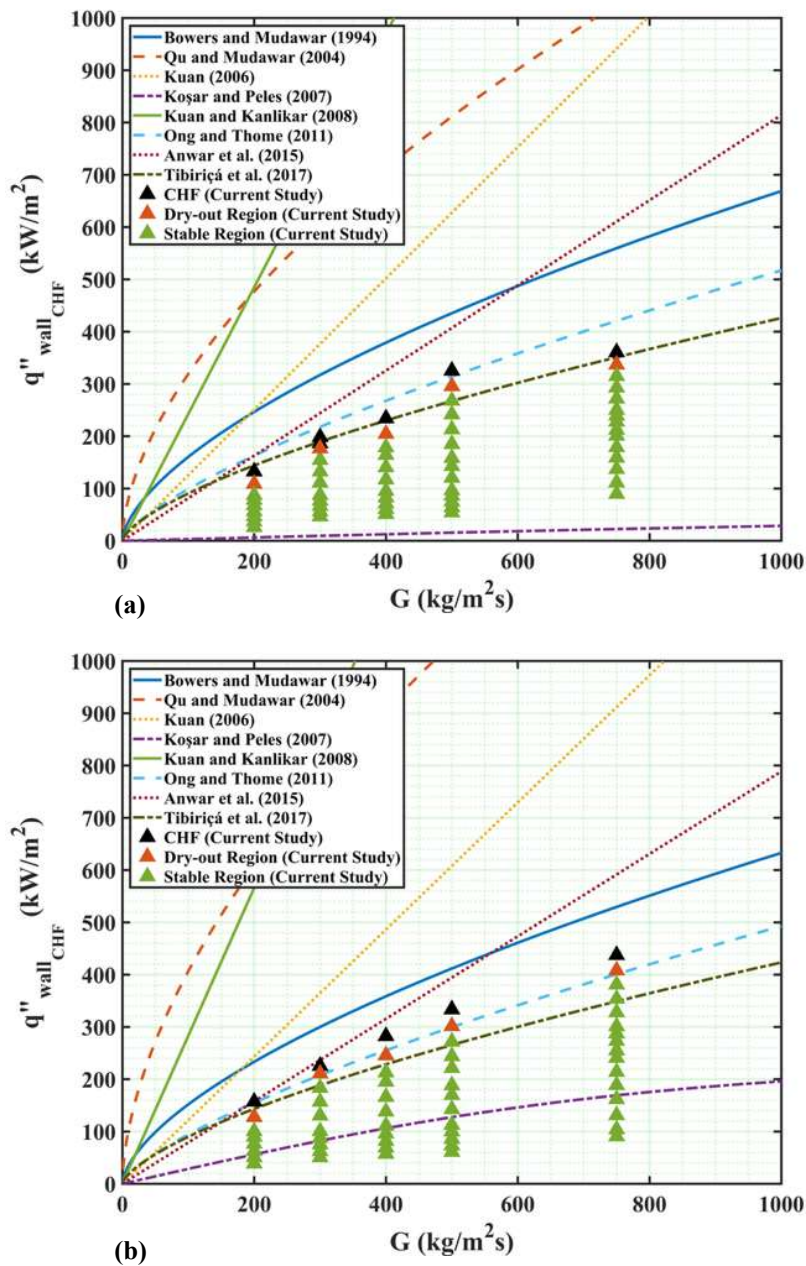


Figure 7.8. Comparison between CHF prediction tools and the CHF results for inlet pressure of: (a) 1.5 bar (b) 2 bar.

The correlation proposed by Bowers and Mudawar [325] was applied to the present dataset, and its predictive accuracy was assessed. This correlation over predicted the CHF values, resulting in an MAE of 45%. The correlation developed by Qu and Mudawar [309] performed poorly, substantially over-predicting CHF across all conditions, with an MAE of 216%. The correlation proposed by Kuan [326] also showed limited accuracy, with an MAE of 93%. Similarly, the correlation developed by Kuan and Kandlikar [327] produced poor predictions, over-predicting CHF with an MAE of 305%. The correlation by Koşar and Peles [31] under-predicted the experimental data, with an MAE of 79%.

Several correlations predicted the experimental CHF values with better accuracy. The correlations by Ong and Thome [208] and Anwar et al. [330] gave MAEs of 10% and 25%, respectively. The correlation developed by Tibiriçá et al. [186] showed strong predictive performance when applied to the present dataset, with an MAE of 12%.

The comparison highlights several key factors affecting the predictive performance of existing CHF correlations. Many of these correlations were developed for operating conditions that differ significantly from those used in the present study. For example, the correlations by Qu and Mudawar [309] and Bowers and Mudawar [325] were formulated for cases with much higher inlet subcooling levels (10 K to 70 K), which may contribute to their poor agreement with the current data. The Qu and Mudawar [309] correlation was also based on a CHF mechanism dominated by flow reversal, which contrasts with the liquid film dry-out mechanism identified here. This difference in dominant CHF mechanism likely explains the substantial over-prediction observed for that correlation.

The use of the receding contact angle in the correlation proposed by Kuan [326] significantly affects its predictive capability. Accurate measurement of this angle is challenging, especially for super-hydrophilic fluids with high evaporation rates, and this may have contributed to the discrepancies observed. Heated length is another key parameter influencing CHF. The correlations by Kuan [326] and Anwar et al. [330] were developed for channels with heated lengths of 63.5 mm and 213-245 mm, respectively, which are considerably longer than the 30 mm length used in the present experiments. This difference in length may partly explain their predictive performance.

The Koşar and Peles [31] correlation captures the correct qualitative pressure trend but shows significant quantitative disagreement with the present data. Their correlation was derived for a single working fluid (R-123), and the pressure property relationship obtained for that fluid may not generalise well to HFE-7100. The short heated length of 10 mm used in their study is another likely factor contributing to the discrepancy.

The Ong and Thome [208] and Tibiriçá et al. [186] correlations showed the best overall predictive performance. Both were developed using datasets that included a variety of working fluids, a wide range of operating conditions, and different channel geometries, which may have improved their generality. Nevertheless, neither correlation fully captured the weak pressure dependence observed in the present

results. Both predicted a decreasing CHF trend with increasing pressure, which is consistent with the current data but does not reproduce the measured values precisely.

The evaluation of existing CHF correlations confirms the significant influence of mass flux on CHF in single microchannels. The effect of inlet pressure is more complex and leads to noticeable differences among the correlations. Within the range examined in this study, the liquid to vapour density ratio appears to be the dominant pressure dependent parameter, exerting a stronger influence than latent heat.

7.4 CONCLUSIONS

The experimental investigation of critical heat flux in the 30 mm long channel test section was presented in this chapter. The results clarified the relationship between flow structure, local heat transfer behaviour, and operating conditions, and enabled a quantitative comparison between experimental data and established microscale CHF prediction tools. The conclusions from the chapter include:

- CHF was governed primarily by liquid film dry out in the annular flow regime. Visualisation near the outlet confirmed the transition from annular flow with some nucleation sites to a fully vapour filled channel, accompanied by cyclic wetting and drying of the heated surface. These oscillations in wall temperature and local wetting conditions were shown to drive the rapid wall temperature rise associated with CHF onset.
- Mass flux significantly affected CHF, with higher values extending the CHF limit to higher wall heat fluxes. The mass flux influence was dominant, attributed to lower vapour qualities for a given heat flux, enhanced inertia and more efficient bubble removal. Increased pressure produced thicker liquid films and delayed dry out, though its effect was comparatively weak. The observed weak pressure dependence aligns with trends reported for flow boiling, contrasting the stronger effects seen in pool boiling.
- The two-phase HTC remained nearly constant before dry out and declined sharply as dry out progressed. Flow visualisation demonstrated that local nucleation activity persisted at higher mass fluxes due to thicker liquid films, delaying CHF. The onset of dry out was identified as a critical transitional region where wall temperature stability depended on the time averaged balance between wetting and vapour coverage.
- Eight existing microscale CHF correlations were evaluated against the experimental data. The Ong and Thome [208] and Tibiriçá et al. [186] models provided the most accurate predictions, with mean absolute errors of approximately 10–12%. Correlations developed for longer channels or different working fluids, such as those by Qu and Mudawar [309], significantly over-predicted CHF, underscoring the importance of heated length, fluid properties, degree of subcooling and dominant CHF mechanism in correlation development.

- The findings confirm that CHF in microchannels is dictated by a variety of fundamental dry out mechanisms. This suggests that further work in the area should be conducted on finding under which conditions each mechanism occurs, such that the onset of dry out can be modelled for each mechanism and degradation of system performance accurately predicted and avoided.

CHAPTER 8

NEW FEED-FORWARD NEURAL NETWORK FLOW PATTERN MODEL

8.1 INTRODUCTION

This chapter presents the development of a new novel feed-forward neural network for predicting flow pattern transitions in the microscale. Section 8.2 discusses the overall machine learning methodology, including the obtained data used to develop the model. Section 8.3. then discusses the results of the machine learning model and further development through data processing techniques and network hyperparameter tuning. The final model's success is then compared to the flow patterns observed in the current study in Section 8.4 with final conclusions made in Section 8.5.

8.2 MACHINE LEARNING METHODOLOGY

As discussed in Chapter 4, none of the existing empirical or semi-empirical flow pattern maps were able to satisfactorily predict the microscale flow regimes observed in this study. This limitation reflects the broader challenges reported in the literature, where conventional maps struggle to account for the multidimensional and highly non-linear interactions between fluid properties, channel geometry and operating conditions at the microscale. In contrast, recent advances in artificial intelligence, particularly feed-forward and convolutional neural networks, have demonstrated strong capabilities in classifying flow patterns and predicting heat transfer characteristics across diverse data sets (see Section 2.7). These methods are especially suited to problems where governing mechanisms are complex or only partially understood, making them ideal for addressing the shortcomings of current predictive tools.

The Brunel Two-Phase Flow database of high-fidelity experimental data contains 11,349 bubbly, slug, churn and annular flow data points for vertical and horizontal channels, single circular channels, singular rectangular channels and microchannel heat sinks taken from three separate experimental facilities. Fluids included R-134a, R-245fa, R-1234yf, HFE-7100, HFE-7200 and DI water. The sources of the used experimental data are displayed in Table 8.1, where data points were both extracted and extrapolated. This databank was used to develop and train a neural network model capable of predicting

prevailing microscale flow patterns and transition boundaries using data based on the working fluid, channel geometries and operating conditions.

Based on the literature, FNNs have repeatedly shown strong performance in two-phase flow classification and heat transfer prediction problems, often outperforming traditional correlations in accuracy and generalisability. FNNs are particularly well suited to handling multi-parameter inputs without requiring explicit programming of physical relationships, allowing the model to identify complex, non-linear patterns in the data. Their use in microscale flow boiling applications, while relatively new in microscale boiling, builds upon promising results in macro and mini scale applications, e.g. Scalabrin et al. [347] and Guo et al. [351]. For these reasons, an FNN architecture was selected as the foundation of the new model.

The task of predicting flow boiling patterns can be considered a classification problem and therefore, required a classification algorithm. In this scenario, the set of relevant features (fluid properties, channel geometry and operating conditions) in the data set were related to known prevailing flow patterns, which were used as the data set labels. The prior knowledge of the prevailing flow patterns therefore results in the requirement for a supervised machine learning algorithm. In this work an FNN algorithm was developed to predict flow boiling patterns based on inlet fluid properties, channel geometry and operating conditions.

Table 8.1. Brunel Two-Phase Flow database of high-fidelity experimental data sources.

Experimental Facility	Test Section Configuration	Hydraulic Diameter	Fluids
1	Single Vertical Tube	0.5 - 4.26	R-134a, R-245fa [198, 421–424]
2	Single Horizontal Microchannel	0.438 - 0.561	DI Water [263, 410]
3	Microchannel Heat Sinks	0.45 - 0.48	R-134a, R-1234yf, HFE-7100, HFE-7200 [127, 408, 425]

A typical FNN architecture was presented in Chapter 2; however, it is presented again in Figure 8.1 to aid in the understanding of how the new novel architecture differs. The modified FNN architecture was developed for the purpose of classifying flow boiling patterns in the microscale in which four individual but connected FNNs formed the model. The FNN architecture is shown diagrammatically in Figure 2.2. Three individual networks were used to predict the transition from bubbly to slug flows, slug to churn flows and then churn to annular flows to allow feature selection of each individual network and obtain knowledge of the most relevant features for each transition boundary. Each individual transition boundary was trained using only flow patterns relevant to its own transition boundary, i.e. the bubbly to slug transition boundary was only trained using bubbly and slug data. This meant that each network may present unexpected results when faced with data points that would not be classified by the

transition boundary, such as presenting the bubbly to slug transition boundary network with an annular flow data point. To overcome this obstacle, a final “classification” FNN was placed after each transition boundary. This was trained by obtaining predictions for the data set from each transition boundary, regardless of flow pattern, and making a final 4 class classification based on the three inputs from the transition boundary FNNs.

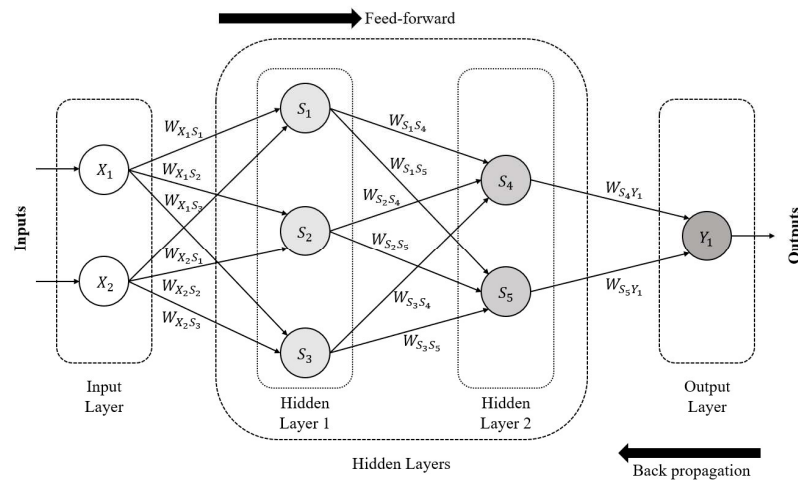


Figure 8.1. Typical FNN architecture.

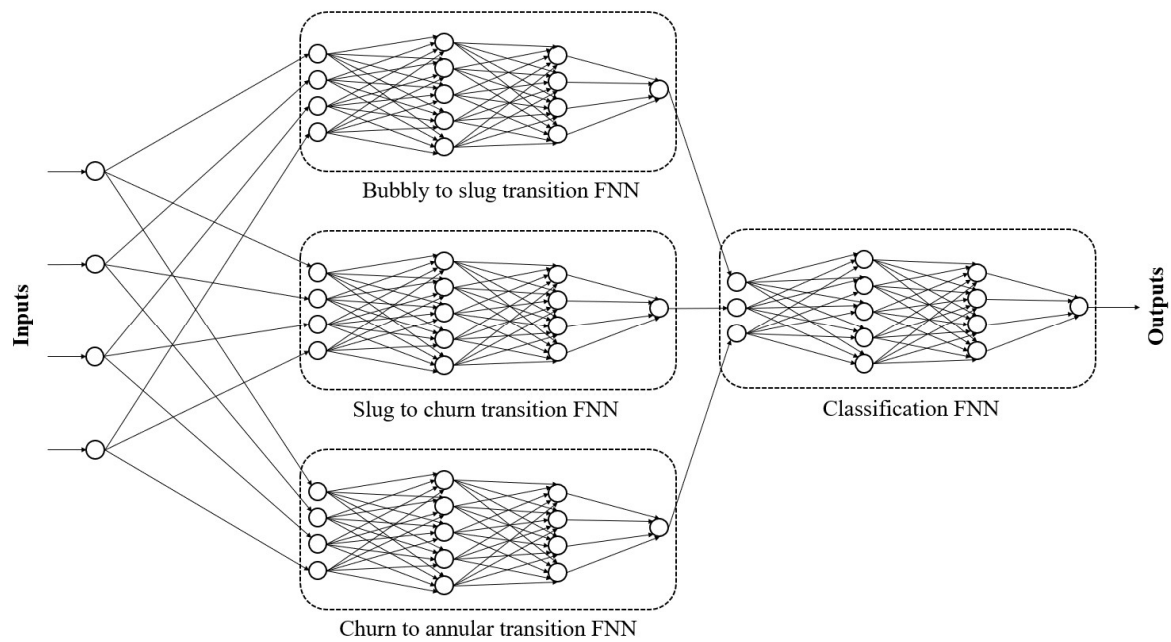


Figure 8.2. Multiple FNN model architecture schematic.

8.3 FEED-FORWARD NEURAL NETWORK RESULTS

8.3.1 Data Splitting

It is essential that a data splitting process is implemented at the beginning of training the model to assess the generalisation performance and to prevent overfitting. Overfitting refers to the phenomenon in which a machine learning model learns the training data too well and captures specific details that are not general to the problem, resulting in an inability to generalise new, unseen data, and hence decreases the performance when applied to the training data set. Hence, the opposing term would to overfitting would be generalisation [426]. Separating the data permits an evaluation of the model's ability to generalise to new examples and applicability in real scenarios. There is a strong possibility that a model will perform exceptionally well when examined against the training data, but will perform very poorly when implemented in real world examples.

Using a training split in which too much training data is used may also lead to overfitting and may capture noise and data outliers rather than understanding the underlying patterns. Consequently, the model may not be able to adapt to the new data. Conversely, using too little testing data may cause the model to overlook some of the required patterns to adequately predict problems. Hence, it is vital to maintain a sensible ratio between training and testing data, such that a reasonable evaluation of the model's capability can be conducted. Attaining an optimal test/train data split contributes to building an FNN that is well suited to the training data and also competent at making accurate predictions for unseen examples. A typical split using 80 % training data and 20 % testing data was utilised for the development of the FNN models.

The architecture of the neural network resulted into further data splitting as well as a conventional 80/20 training and testing data split. The splits are explained diagrammatically in Figure 8.3. Firstly, the entire data set was split into the flow patterns relevant to each transition boundary. This left three data sets, one that contained bubbly and slug flow data, one that contained slug and churn flow data and the final set contained churn and annular flow data points. The individual transition boundary FNNs were then trained using the data sets for each transition boundary. Predictions were then made for the entire data set using the trained transition boundary algorithms. The predictions from each algorithm were then used to train the classification FNN. Whilst the individual transition boundary FNNs should be capable of predicting the flow patterns from the relevant data set, the networks may produce unexpected results when provided with data that is dissimilar to that it was trained with. The addition of a classification FNN allows this unexpected behaviour to be accounted for in the full multiple FNN model.

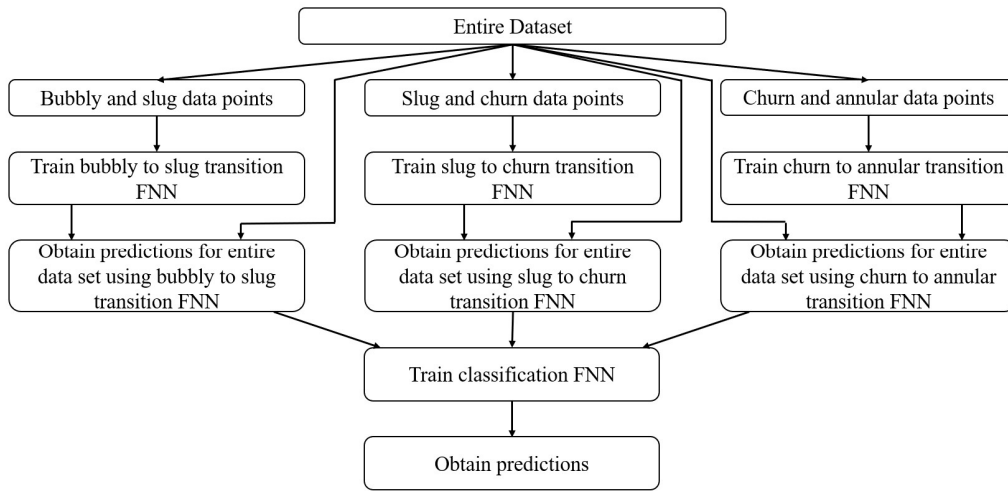


Figure. 8.3. Use of data set to train multiple FNN and obtain predictions.

8.3.2 Data Scaling

Scaling the data is an integral pre-processing step in training an FNN and neural networks in general. It plays a crucial role in enhancing convergence of the model and the overall performance. Typically, neural networks rely on optimisation algorithms such as gradient descent to minimise the loss function during training. The loss function in a neural network quantifies the difference between the predicted output and the actual target values and hence, the purpose of training the algorithm is to minimise this function. Hence, the performance of an FNN can be significantly limited by the optimisation performance [427]. Algorithms such as gradient descent are highly sensitive to the scale of input features and uneven updates of weights in the neural network can occur when features are considerably different in magnitude. Therefore, scaling or normalising the data can significantly aid in improving algorithm performance [428].

Neural networks with multiple layers can experience gradients of the loss function becoming excessively small or large and contributes to vanishing or exploding gradient problems. This may make it difficult for the network to train. The process of scaling or normalising parameters ensures that the gradients are kept within a functional range and allows smoother weight updates and overall stability in training. Furthermore, scaling or normalising features helps to prevent any single input feature from dominating the learning process as a result of its larger magnitude. For example, the scales of the values of fluid enthalpy are considerably greater than that of liquid or vapour phase viscosity in SI units. The effect of fluid enthalpy would significantly dominate the network in comparison to any phase viscosity if a scaling process is not undertaken. As such, a normalisation formula was applied to the training data set as shown in Equation (8.1), where $\hat{X}_{i,n_{train}}$ is the normalised value of the n^{th} data point in feature i in the training data set, $X_{i,n_{train}}$ is the original data value, $\mu_{i_{train}}$ is the mean of the values of the i^{th}

feature in the training data set and $\sigma_{i_{train}}$ is the standard deviation of the values of the i^{th} feature in the training data set.

$$\hat{X}_{i,n_{train}} = \frac{X_{i,n_{train}} - \mu_{i_{train}}}{\sigma_{i_{train}}} \quad (8.1)$$

It is also required that a normalisation process is also undertaken on the testing data and any future data that the network wants to predict. However, as the network was trained using the normalised training data, the testing and prediction data must be normalised in accordance with the mean and standard deviation values of the training data. The testing data is normalised separately to the training data to avoid some leakage of information between the testing data and training data and reduce the risk of overfitting. The normalisation process of the training data is displayed in Equation (8.2), where $\hat{X}_{i,n_{train}}$ is the normalised value of the n^{th} data point in feature i in the testing data set and $X_{i,n_{train}}$ is the original pre-normalised value.

$$\hat{X}_{i,m_{test}} = \frac{X_{i,m_{test}} - \mu_{i_{train}}}{\sigma_{i_{train}}} \quad (8.2)$$

8.3.3 Model Cross-Validation

A K-fold cross validation method was built in to both models to obtain a more robust and significant evaluation of the model than only using a test-train split. The general K-fold methodology is represented in the schematic depicted in Figure 8.4. The entire data set is first split into data used for training and data used for testing. Then the training data is split into K subsets and one subset is set aside for testing, i.e. it is removed from the training data set. The model is then trained using the remaining folds and the fold that was set aside is used to test the algorithm performance. The process is then repeated K times with a different fold being set aside for testing each time. The final algorithm performance is then calculated as the mean of the K number of performance values. This methodology helps to reduce any potential bias that may be caused by a particular test-train split and is more representative of how the model may perform in real scenarios.

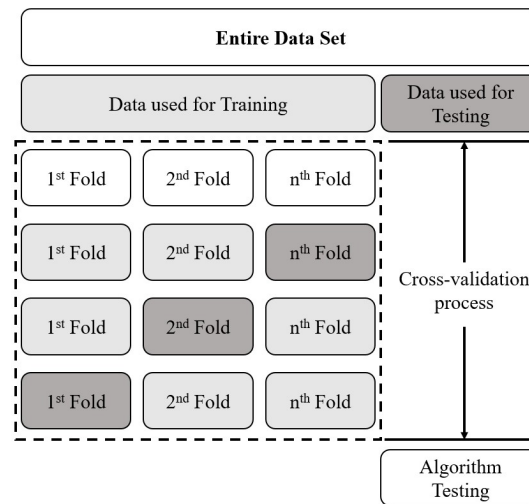


Figure 8.4. K-Fold methodology schematic in which n folds are used (modified from [429]).

The ability to make efficient use of the available data and to permit the high 80/20 training split are further advantages of the K-fold cross validation method [429]. The distribution of flow pattern data is displayed in Figure 8.5, where the number of data points for bubbly, slug, churn and annular flows are 2417, 2782, 2976 and 3174, respectively. There is some disparity between the number of bubbly and annular data points; however, this disparity is not significantly high. The K-fold cross method allows the data set to more efficiently use any evaluation of the model that may be difficult with this number of data points. The systematic cycling through the folds in the training data set allows for a more comprehensive understanding of how well a model is able to generalise certain patterns in the data and improves the FNNs applicability to new data sets. As such, a $K=10$ cross validation was performed for each training and assessment of any FNN created, both for the single classifier and the network of FNNs classifier, due to the fairly low bias and variance [430].

8.3.4 Feature Selection and Hyperparameter Tuning

Feature selection attempts to reduce the input dimension of the model and can play a pivotal role in improving both the performance and interpretability of machine learning models [431]. Features would be classified as any fluid thermophysical property, channel geometric parameter or operational parameter in the case of classifying flow boiling patterns. The improvement in interpretability of the models is particularly true in cases where a large number of features are present. A successful feature selection process will identify the most informative variables to the problem and retain them in the model. The consequent reduction in dimensionality often results in an alleviation of any potential overfitting risks and eliminates noise. Models become more focussed on the essential patterns that lie within the data when a suitable subset of features is selected which allows for better algorithm generalisation to unseen data and also improves computational efficiency.

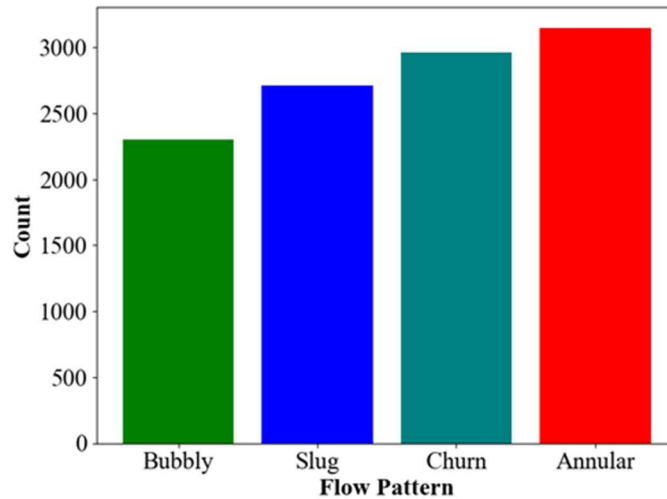


Figure 8.5. Distribution of flow pattern data points.

Feature selection can offer worthwhile insights into the most important factors that influence the system, especially when applied to physical parameters for a given problem. On the topic of flow patterns, feature selection can identify the key variables that dominate the observed phenomena and may provide valuable insight into the forces upon which flow pattern transitions are dependent. This makes feature selection a particularly useful tool for engineers and scientists that wish to gain insight into physical phenomena from deep learning algorithms without simply treating them as a “black box”. Thus, feature selection is not only capable of improving model performance but can also present a method to detect and highlight vital physical parameters that shape the dynamics of highly complex systems.

The implementation of an analysis of variance (ANOVA) F-test and SelectKBest feature selection process was utilised to form a supervised feature selection in the development of network algorithms. The ANOVA F-test is a statistical methodology that is used to evaluate if the mean values of two groups are significantly different from each other. Overall, the concept is to divide the total variability observed in a dataset into two distinct components; namely, the variability between the group means and the variability within the group itself [432]. The F-test from the variance results were then evaluated to determine if the variance between the groups is statistically significant. The SelectKBest method then takes the most relevant features from the ANOVA F-test and trains the model using these features. The model was then trained using these features and the performance of the algorithm examined using the testing data set was analysed. The feature importance rankings from the ANOVA F-test for each flow pattern transition boundary data set, i.e. bubbly & slug, slug & churn and churn & annular, are displayed in Table 8.2. These rankings were then incorporated with the SelectKBest methodology for development of the single FNN model to find the optimal feature subset that maximises the classification accuracy [433].

Table 8.2. ANOVA F-test rankings for transition boundary FNNs.

Feature Rank	Bubbly to slug feature rankings	Slug to churn feature rankings	Churn to annular feature rankings
1	Mass Flux	Superficial gas velocity	Mass Flux
2	Superficial liquid velocity	Thermodynamic vapour quality	Superficial liquid velocity
3	Thermodynamic vapour quality	Width of channel	Liquid viscosity at saturation
4	Vapour density at saturation	Vapour viscosity at saturation	Vapour density at saturation
5	Superficial gas velocity	Degree of subcooling at inlet	Superficial gas velocity
6	Hydraulic diameter of channel	Vapour conductivity at saturation	Thermodynamic vapour quality
7	Liquid Viscosity	Surface tension at saturation	Hydraulic diameter of channel
8	Height of channel	Hydraulic diameter of channel	Surface tension at saturation
9	Distance along channel	Latent heat of vapourisation	Vapour viscosity at saturation
10	Width of channel	Liquid conductivity at saturation	Degree of subcooling at inlet
11	Wall heat flux	Vapour density at saturation	Liquid specific heat at saturation
12	Flow area of channel	Liquid density at saturation	Latent heat of vapourisation
13	Vapour viscosity at saturation	Superficial liquid velocity	Liquid conductivity at saturation
14	Liquid density at saturation	Height of channel	Wall heat flux
15	Degree of subcooling at inlet	Mass Flux	Inlet pressure
16	Surface tension at saturation	Flow orientation	Distance along channel
17	Liquid specific heat at saturation	Flow area of channel	Enthalpy of fluid at inlet
18	Heated perimeter of channel	Number of heated corners	Enthalpy of liquid at saturation
19	Flow orientation	Wall heat flux	Number of channels
20	Number of heated corners	Enthalpy of vapour at saturation	Enthalpy of vapour at saturation
21	Number of channels	Liquid specific heat at saturation	Liquid density at saturation
22	Enthalpy of fluid at inlet	Enthalpy of liquid at saturation	Vapour conductivity at saturation
23	Enthalpy of liquid at saturation	Number of channels	Number of heated corners
24	Inlet pressure	Inlet pressure	Flow orientation
25	Vapour conductivity at saturation	Enthalpy of fluid at inlet	Heated perimeter of channel
26	Enthalpy of vapour at saturation	Vapour specific heat at saturation	Height of channel
27	Latent heat of vapourisation	Liquid viscosity at saturation	Flow area of channel
28	Liquid conductivity at saturation	Heated perimeter of channel	Width of channel
29	Vapour specific heat at saturation	Distance along channel	Vapour specific heat at saturation

Typically, a given number of top-ranking features may be used with the SelectKBest methodology in the feature selection process to train the algorithm. In this case, each algorithm was trained 28 times, beginning with the top 2 highest ranked features and adding the next ranked feature for each iteration of the process. The performance of the algorithm, taken from predictions of the testing data, then allowed for an optimum number of features to be discovered for further development of each model. Each individual flow pattern transition FNN was trained using the same initial FNN architecture and hyperparameters. The initial architecture and hyperparameters are displayed in Table 8.3. In the context of neural networks, hyperparameters control the overall architecture and behaviour of the network through parameters set prior to the learning process [434]. Conversely, model parameters are adjusted

through a learning process, as a poorly tuned network may solely result in the development of a unsuccessful model [435]. The hyperparameters that were tuned for FNN model included the number of layers in the hidden layer, the number of nodes in each of the hidden layers, a parameter referred to as drop out, the activation function in the hidden layer, the optimiser type, the batch size during training and the number of epochs during training.

Table 8.3. Initial hyperparameters for single FNN model.

Hyperparameter	Role in FNN	Initial Value
Number of hidden layers	The number of layers in the hidden layers of the network.	2
Number of neurons in hidden layer(s)	The number of neurons that each layer contains. (Note, this does not need to be the same for each layer.)	32
Activation functions in the hidden layers	This is the function that is applied to the value of the node as it is propagates on to the next node.	Sigmoid
Batch Size	The number of training examples used in a single iteration of optimisation.	32
Number of Epochs	The number of times an entire training set is passed through the network for back propagation.	200
Optimiser	The optimiser adjusts the wrights during training to minimise error between the model predictions and real target values.	Adaptive moment estimation (ADAM)
Dropout rates	The percentage of randomly selected neurons that are ignored during training to reduce overfitting.	0

For clarity, the hyperparameters examined are defined in the following paragraphs. The definition of the numbers of layers in the hidden layer and the number of nodes in the hidden layers is best described by referring back to Figure 8.1. There are two hidden layers in the hidden layer in this simple schematic of an FNN. The first of the hidden layers possesses 3 nodes and the second hidden layer contains 2 nodes. In the remainder of this Chapter, this hidden layer architecture would be represented by [3, 2].

Dropout is a regularisation technique in neural networks that also attempts to prevent overfitting and enhance the model's ability to find general patterns in the data [436]. This process involves randomly "dropping out" (i.e., temporarily disabling) a proportion of neurons during each training iteration. Dropout intends to prevent the neural network from being heavily reliant on specific neurons and forces the model to learn more robust and diverse features. This helps in creating a more resilient and adaptable model that performs well on new data. This process is shown diagrammatically in Figure 8.6.

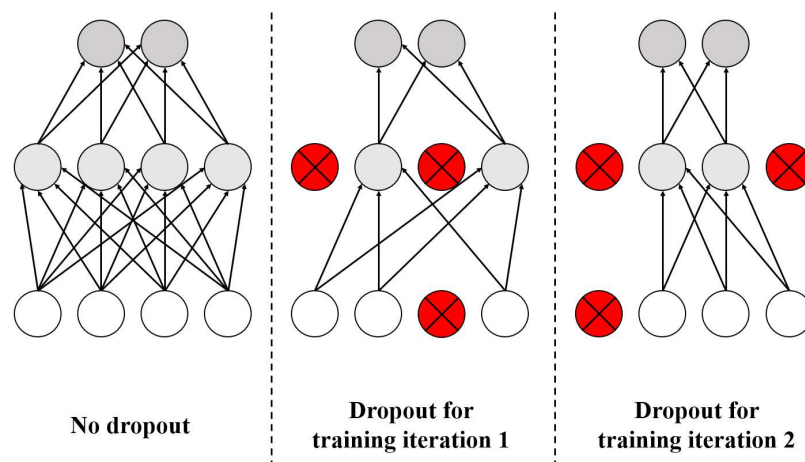


Figure 8.6. Dropout regularisation technique.

The activation functions in the hidden layers of an FNN introduces the non-linearity into the model which is essential to the model's ability to learn the complex patterns in the data [437]. The functions work by changing the neuron outputs based on the nodal values and significantly contribute to the mapping between network input and output variables. Whilst it is possible for custom functions to be developed in a neural network, many application programming interfaces provide a library of activation functions that can be easily swapped into the model. The optimiser, as previously discussed, is responsible for the modification of the weights during algorithm training. Different optimiser types implement contrasting mathematical methodologies for training the algorithm and alter the convergence properties of the algorithm. The selection of a suitable optimiser will avoid local minima in the problem, improve convergence and overall, enhance the overall performance of a neural network.

The performance of the bubbly to slug transition boundary FNN with the initial hyperparameters is shown in Figure 8.7, where the algorithm was trained using the aforementioned feature selection method. The performance of the model with various feature subsets showed that the optimal performance was observed when 16 features were used. These 16 features correlated to an algorithm performance of 93.49 % of the testing data accurately classified, i.e. 93.49 % accurately estimated bubbly and slug data points. The inclusion of additional features to the model were then found to either have no effect on the performance of the transition boundary algorithm or imparted noise or overfitting on the model and was shown to hinder performance.

The final features used to tune and train the bubbly to slug FNN model are displayed in Table 8.4. It must be noted that feature selection in neural networks only provides further interpretability to the model; however, the model must ultimately 'have access' to parameters that are relevant to the problem to facilitate a good performance. Hence, attempting to conduct in depth physical inferences from the feature selection may not be wise but some explanations may be theorised as to their relevance in the

transition. Dimensionless numbers are often favoured by engineers and scientists in the evaluation of thermodynamic problems.

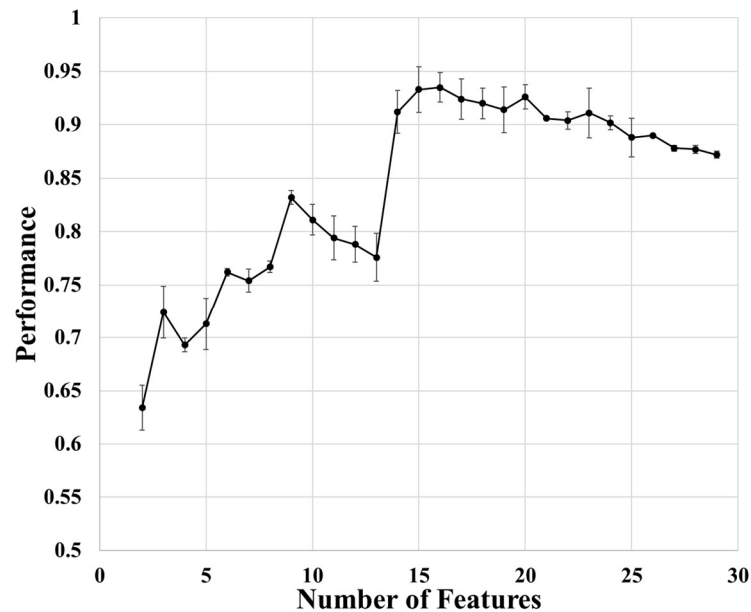


Figure 8.7. Performance of initial bubbly-slug transition boundary FNN with different feature subsets.

The parameters in Table 8.4 can be non-dimensionalised in the form of various phase Reynolds numbers and phase Weber Numbers which represent the ratio of inertial forces to viscous forces and the ratio of inertial to surface tension forces, respectively. The ability for a bubble to depart the surface can be influential in determining either bubbly or slug flow. Relating this to the possible dimensionless parameters, the inertial, surface tension and viscous force will likely all play a role in causing a departure of the bubble from the surface. For example, the inertial force of the liquid phase would be the force that is capable of moving the bubble along the channel. The shear stress acting on the bubble that is capable of shearing it away from the surface directly relates to the viscous forces. Finally, the surface tension forces play a key role in maintaining the liquid vapour interface of the bubble and has shown to be a significant factor affecting bubble departure size in boiling, which would potentially play a role in bubble confinement. The selected features indicate that these may be important mechanisms in determining the transition between bubbly and slug flows, whilst not having the capability to confirm this or provide a further insight into these mechanisms.

Table 8.4. Features used to tune and train the bubbly to slug FNN transition model.

Feature Rank	Feature
1	Mass Flux
2	Superficial liquid velocity
3	Thermodynamic vapour quality
4	Vapour density at saturation
5	Superficial gas velocity
6	Hydraulic diameter of channel
7	Liquid Viscosity
8	Height of channel
9	Distance along channel
10	Width of channel
11	Wall heat flux
12	Flow area of channel
13	Vapour viscosity at saturation
14	Liquid density at saturation
15	Degree of subcooling at inlet
16	Surface tension at saturation

A further final interesting aspect of these results are that there is no dependence on the shape of channel geometry or orientation. Hence, the model does not require knowledge of whether the channel is circular or rectangular and does not require knowledge of whether the flow is vertical or horizontal. The model believes that the flow pattern transition between bubbly and slug flow is not dependent upon these parameters.

Two further dimensionless groups that can be made are the confinement number and the boiling number, which represent the suppression of buoyancy and increase of capillary forces as well as the ratio of vapour generated per mass flux. Two potentially interesting features that may further relate to bubble confinement are height and width of the channel, which are not represented in the confinement number. Again, this may indicate that there is some correlation between aspect ratio and bubble confinement outside the influence of hydraulic diameter.

The same performance plotted against the number of features used in the training subset are displayed in Figure 8.8 for the slug to churn transition. Here, a higher optimal algorithm performance was found at a higher number of 18 features at a lower performance of 88.60%.

The features selected for further tuning and the training of the slug to churn transition boundary FNN are presented in Table 8.5. Here, the transition is said to be dependent on both the number of heated corners of the channel and the flow orientation. A significant caveat to this is that the quality of the data used by the model limits the ability to determine which properties are significant. The Brunel Two-Phase Flow database of high-fidelity experimental contains only data for vertical flows in circular

channels and data for horizontal flows in both single rectangular heat sinks and multichannel heat sinks. Hence, for every channel with 0 heated corners, i.e. a circular channel, the flow orientation is also 90° . Conversely, for every channel with 2 heated corners that represent a rectangular channel with an adiabatic viewing plate the flow orientation is 0° . It is therefore significantly unlikely that the model would be able to perform well for flows between those values and with novel channel cross sections such as triangular channels. Furthermore, it is likely that the model is simply taking circular vertical channels as one subset and treating it differently to the other subset which includes horizontal flows in either singular or multiple rectangular channels. Hence, the effect of channel cross sectional shape or flow orientation cannot be inferred from this model. Further data with a more continuous distribution of the number of heated corners and flow orientation would be required to make this assessment.

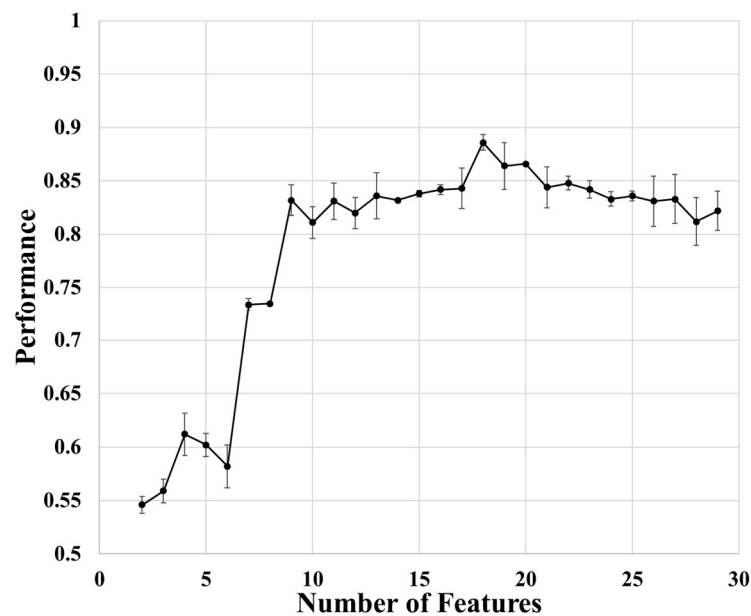


Figure 8.8. Performance of initial slug-churn transition boundary FNN with different feature subsets.

Some unexpected features that are included in the list of features include both the vapour and liquid phase conductivity and the latent heat of vaporisation. Again, further understanding of why these parameters are deemed important to the model cannot be obtained, whilst this may suggest some kind of physical significance to the slug churn transition boundary problem.

Finally, the performance vs features used in the training subset results are shown in Figure 8.9. This model was found to require the fewest number of features in the training subset to achieve an optimal performance; however, it achieved the lowest performance of all three individual FNN transition boundary models with an optimal accuracy of 86.72%. Overall, 13 features were found to be the optimum number of features to obtain the greatest model performance.

The churn to annular transition boundary feature selection parameters are displayed in Table 8.6. Again, there were some unexpected parameters that featured in this list that have previously not been included in churn to annular flow pattern transition boundary correlations. Once more, such features included the liquid phase conductivity and the latent heat of vaporisation; however, there was a new inclusion of the liquid specific heat at constant pressure.

Table 8.5. Features used to tune and train the slug to churn FNN transition model.

Feature Rank	Feature
1	Superficial gas velocity
2	Thermodynamic vapour quality
3	Width of channel
4	Vapour viscosity at saturation
5	Degree of subcooling at inlet
6	Vapour conductivity at saturation
7	Surface tension at saturation
8	Hydraulic diameter of channel
9	Latent heat of vapourisation
10	Liquid conductivity at saturation
11	Vapour density at saturation
12	Liquid density at saturation
13	Superficial liquid velocity
14	Height of channel
15	Mass Flux
16	Flow orientation
17	Flow area of channel
18	Number of heated corners

The final FNN, denoted in this paper as the classification FNN, did not require any form of feature selection for the model input. This is because the network only has 6 input nodes that are the output probabilities for each flow pattern in the three-transition boundary neural networks. It should be noted that assessment of the three transition boundary networks was conducted by analysing the number of correctly estimated flow patterns. As such, the output predictions of the training sets were converted to binary outputs, i.e. 0 for bubbly flow and 1 for slug when considering the bubbly to slug transition boundary correlation based on which flow pattern output possessed the highest probability. The full architecture is explained in further detail later in this section. The outputs themselves were not in such a binary format due to the requirement for training the classification FNN. Three binary inputs to the model would not allow it to perform well or deal with unwanted behaviour of the networks which had not been trained with similar data. In general, neural networks perform better when dealing with continuous data sets that have been either normalised or scaled.

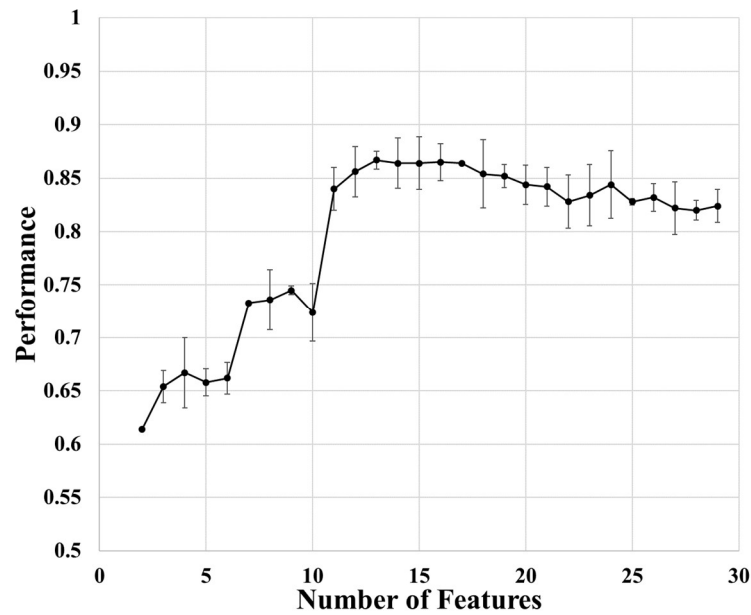


Figure 8.9. Performance of initial churn-annular transition boundary FNN with different feature subsets.

Table 8.6. Features used to tune and train the churn to annular FNN transition model.

Feature Rank	Feature
1	Mass Flux
2	Superficial liquid velocity
3	Liquid viscosity at saturation
4	Vapour density at saturation
5	Superficial gas velocity
6	Thermodynamic vapour quality
7	Hydraulic diameter of channel
8	Surface tension at saturation
9	Vapour viscosity at saturation
10	Degree of subcooling at inlet
11	Liquid specific heat at saturation
12	Latent heat of vapourisation
13	Liquid conductivity at saturation

The completion of a feature selection process permitted the further development of each individual FNN with hyperparameter tuning. Hyperparameter tuning is often conducted empirically through algorithm experimentation. Permutations of different hyperparameters are selected to train and test the model prior to the model performance being evaluated. Important model characteristics include the accuracy of the model, the stability of the model and the time taken to train and test the model. The

time statistic may be significant as a model that provides similar accuracy and stability may be more beneficial than another if it is computationally inexpensive. This is particularly true for a search that involves batch size or epochs as they drastically affect the training time. The values of the hyperparameters tuned for the single FNN are displayed in Table 8.7. The same parameters were also used to tune the classification FNN.

The permutations of hyperparameters are often inserted into the algorithm using a grid search methodology; however, the implementation of all hyperparameters into a single grid search would be highly computationally expensive. For example, the first set of hyperparameters contains 11 different values, the second contains 6 values, the third contains 3 values, the fourth contains 6 values, the fifth contains 4 values and the final set contains 6 values. A single grid search would require the algorithm to be trained 28,512 times without the use of cross validation. Hence, the problem was strategically divided into three different grid search algorithms. The grid searches were divided into a hidden layer configuration and dropout rate grid search, an activation function and optimiser type grid search and finally, a batch size and epochs grid search.

Table 8.7. FNN model hyperparameter tuning variables.

Hyperparameter	Parameters Examined
Hidden Layer Configurations	[32], [64], [128], [32,32], [64,64], [128,128], [128,64], [32,32,32], [64,64,64], [128,128,128], [64,128,64]
Dropout Rates	0.00, 0.01, 0.05, 0.1, 0.2, 0.5
Hidden Layer Activation Functions	Rectified linear unit, hyperbolic tangent, sigmoid
Optimiser types	Stochastic gradient descent (SGD), root mean square propagation (RMSprop), adaptive gradient algorithm (Adagrad), Adadelata, adaptive moment estimation (Adam), Nesterov-accelerated adaptive moment estimation (Nadam)
Batch sizes	16, 32, 64, 124
Epochs	50, 100, 200, 500, 1000, 2500

The results from the hidden layer configuration and dropout rate grid search that was applied to the bubbly to slug transition FNN model are represented by the heatmaps shown in Figure 8.10 and Figure 8.11. These figures present the accuracy and standard deviation metrics, respectively. The optimum performance was observed when a hidden layer configuration of [64,128,64] and no dropout was used in the model. This may indicate that the model is not prone to overfitting or that there is sufficient data in the training set to accurately detect the complex patterns. There was no clear effect of hidden layer configuration or dropout rates on the standard deviation and hence, stability of the model.

The model was then adjusted to keep a dropout rate of 0 and to modify the hidden layers to a [64,128,64] configuration. A grid search process was then conducted on the modified model for the set of activation function and optimisers to be investigated. The performance results from the hidden layer activation function and optimiser type grid search are shown by the heatmaps displayed in Figure 8.12 and Figure 8.13, respectively. The highest performing configuration was a sigmoid activation function used in conjunction with a Nesterov-accelerated adaptive moment estimation optimiser. Again, there was no clear pattern in stability, with the exception of the stochastic gradient descent optimiser. This is likely due to the adaptive learning rates of the other optimiser types, where an initial default learning rate of 0.001 was applied to all models.

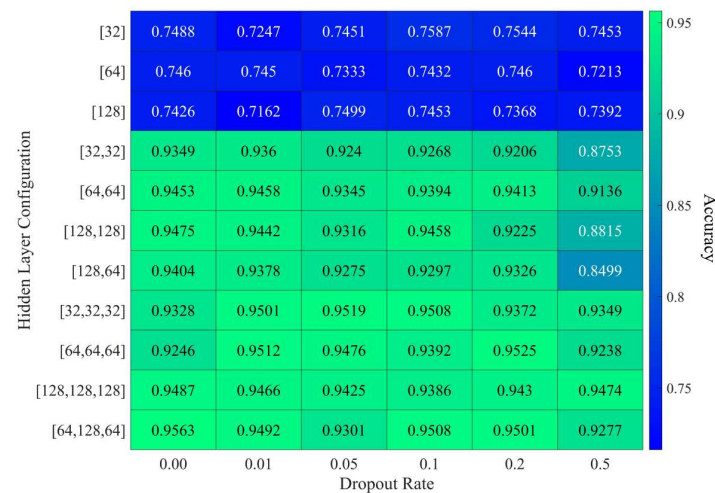


Figure 8.10. Accuracy from hidden layer configuration and dropout rate grid search for bubbly-slug FNN model.

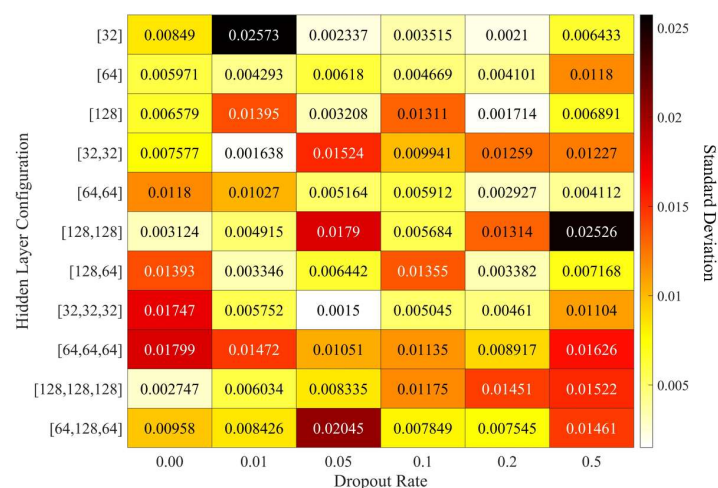


Figure 8.11. Standard deviation from hidden layer configuration and dropout rate grid search for bubbly-slug FNN model.

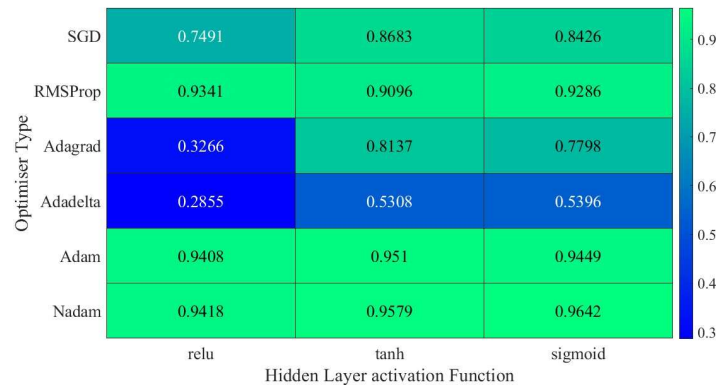


Figure 8.12. Accuracy from hidden layer activation function and optimiser type grid search for bubbly to slug FNN model.

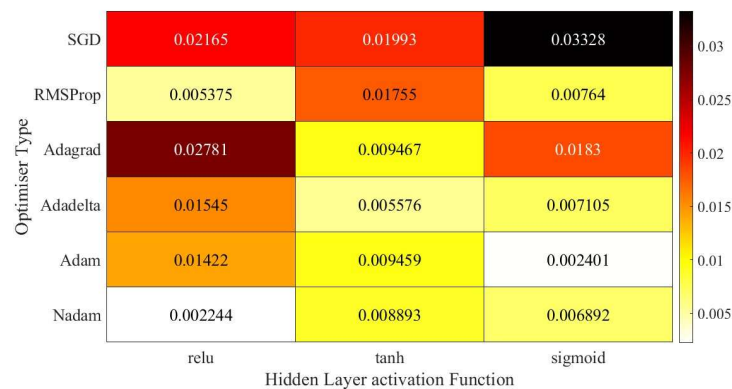


Figure 8.13. Standard deviation from hidden layer activation function and optimiser type grid search for bubbly to slug FNN model.

The model was then re-adjusted to use a Nesterov-accelerated adaptive moment estimation optimiser and a grid search process was implemented for the training parameters, i.e. batch size and epochs. The analysis of the training hyperparameters includes a time for training heat map, as the training time is particularly sensitive to these parameters. The accuracy, standard deviation and time taken to train the models are displayed in Figure 8.14, Figure 8.15 and Figure 8.16, respectively. The optimum accuracy was achieved using a batch size of 64 and 2500 epochs; however, this was found to be highly computationally expensive and took 173.5 seconds to train each fold of the model. A similar level of performance was found using a batch size of 64 and 1000 although training was reduced by 59.98%. Furthermore, this lower number of epochs also showed a more stable algorithm performance. Hence, a final batch size of 64 was used over 1000 epochs.

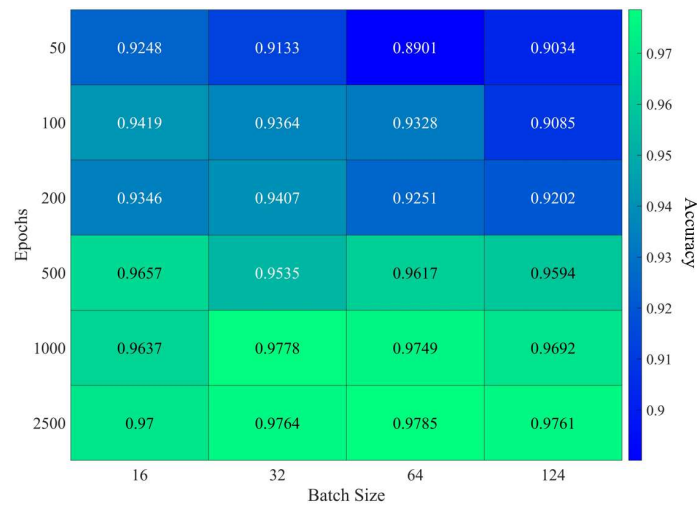


Figure 8.14. Accuracy from batch size and epochs grid search for bubbly-slug FNN model.

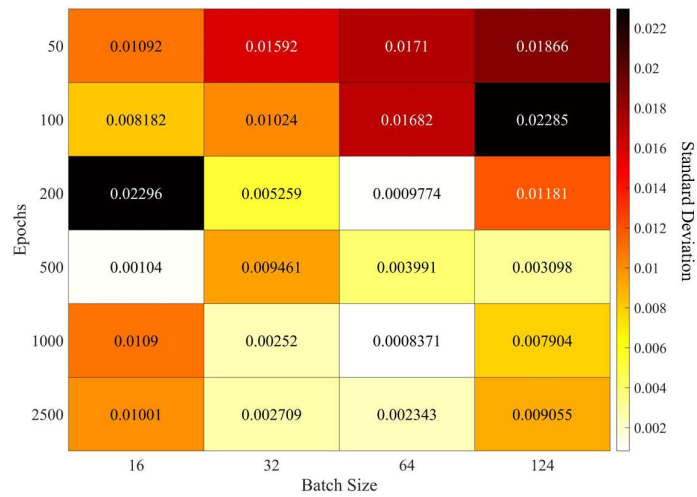


Figure 8.15. Standard deviation from batch size and epochs grid search for bubbly to slug FNN model.

The same hyperparameter tuning process was conducted for the remaining transition boundary networks and the classification FNN. The final parameters used by each individual FNN are displayed in Table 8.8.

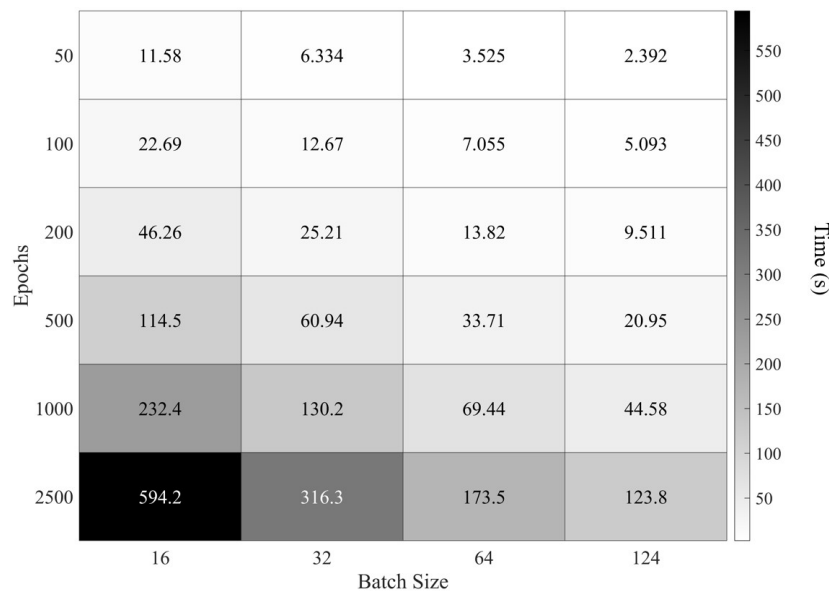


Figure 8.16. Time from batch size and epochs grid search for bubbly to slug FNN model.

Table 8.8. Multiple FNN model hyperparameter tuning variables.

Hyperparameter	Bubbly to Slug Transition FNN	Slug to Churn Transition FNN	Churn to Annular Transition FNN	Classification FNN
Hidden Layer Configurations	[64,128,64]	[64,64]	[64,64,64]	[128,128,128]
Dropout Rates	0	0	0.05	0.2
Hidden Layer Activation Functions	Sigmoid	Sigmoid	Sigmoid	Rectified linear unit
Optimiser types	Nadam	Nadam	Adam	Nadam
Batch sizes	64	128	128	32
Epochs	1000	2500	2500	1000

8.3.5 Final Algorithm Architecture

The final architecture of the bubbly to slug individual transition boundary FNN is presented diagrammatically in Figure 8.17. Here, the 16 features selected from the feature selection process are fed into the input layer before being linearly passed through to the first hidden layer with a weight on each connection between nodes. All input features are then summed at the nodes in the first layer. This is explained in Equation (8.3), which represents the value of the first node in the first hidden layer, $H_{1,1}$.

$$H_{1,1} = W_{X_G H_{1,1}} X_G + W_{X_{U_{ls}} H_{1,1}} X_{U_{ls}} + \cdots + W_{X_{i=16} H_{1,1}} X_{i=16} \quad (8.3)$$

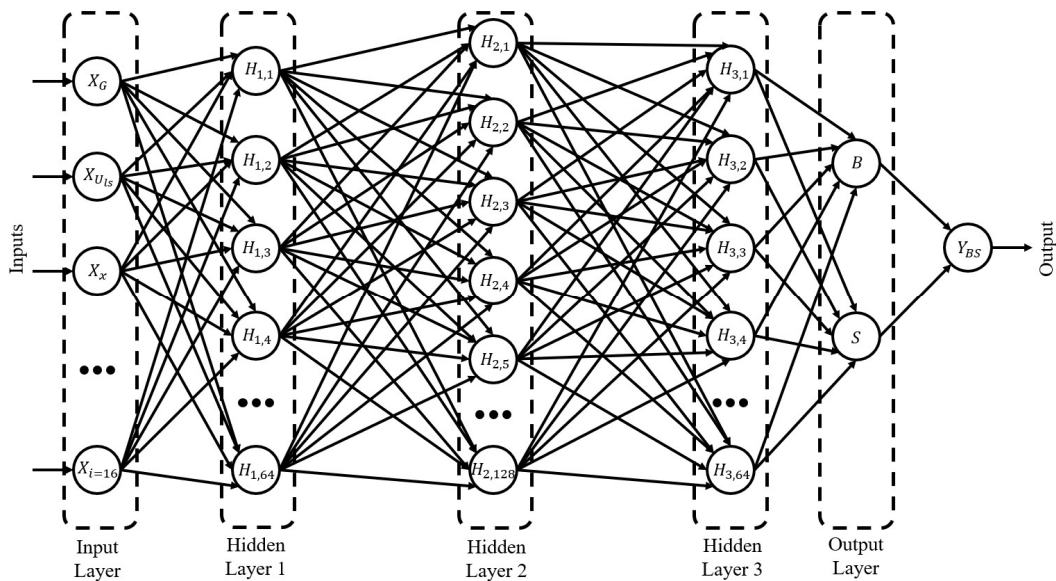


Figure 8.17. Final bubbly to slug transition boundary FNN architecture.

All layers in the model are dense, meaning that all nodes connect to all nodes in the next layer. The hidden layers then pass the values through a weight and the sigmoid function that was selected from the hyperparameter tuning process. The configuration of the hidden layer, also obtained from the hyperparameter process, was the use of 64 nodes in the first layer, 128 in the second layer and 64 in the third layer. The sigmoid activation function takes a real number and compresses it between a value of 0 and 1. The sigmoid function is displayed in Equation (8.4). The summed value of the first node in the second hidden layer is also shown in Equation (8.5).

$$S(X) = \frac{1}{1 + e^{-x}} \quad (8.4)$$

$$H_{2,1} = W_{H_{1,1}H_{2,1}}S(X_{H_{1,1}}) + W_{H_{1,2}H_{2,1}}S(X_{H_{1,2}}) + \dots + W_{H_{1,64}H_{2,1}}S(X_{H_{1,64}}) \quad (8.5)$$

The nodes in the final layer of the hidden layers are then connected to the two nodes in the output layer. The two nodes in the output layer represented each of the flow patterns associated with the transition, In Figure 8.17, this is bubble and slug. The value at each node indicates the probability that the input data correspond to the flow pattern represented by that node. The sum of the values of these two nodes therefore equate to unity. As previously mentioned, the overall algorithm performance required a further decision node that selects the flow pattern with the highest probability and classifies the data under that classification. However, this binary output is not particularly useful in trying to develop a fourth classification FNN. Hence, the values from both nodes were used as inputs for the classification FNN.

The final performance of the bubbly to slug transition boundary FNN is displayed in Figure 8.18. Here, the model was trained over 1000 epochs with a batch size of 64. The blue line represents the

performance of the algorithm with respect to the training data, whilst the orange line represents the algorithm performance in relation to the testing data. Overall, the average performance was 96.42%; however, the algorithm was found to often significantly drop in performance when evaluated after each epoch of the training cycle. One reason for this may again be due to the subjectivity in classifying flow patterns. For example, two researchers may have used similar experimental setups but classified flow patterns slightly differently due to the lack of clear transition criteria. The weights in the model may then get particularly adjusted to account for one of the data sets whilst a large bulk of the opposing data set lies in the testing split. However, the lowest accuracy with regards to the testing data was still above 90% despite the possibility of data where discrepancies are present.

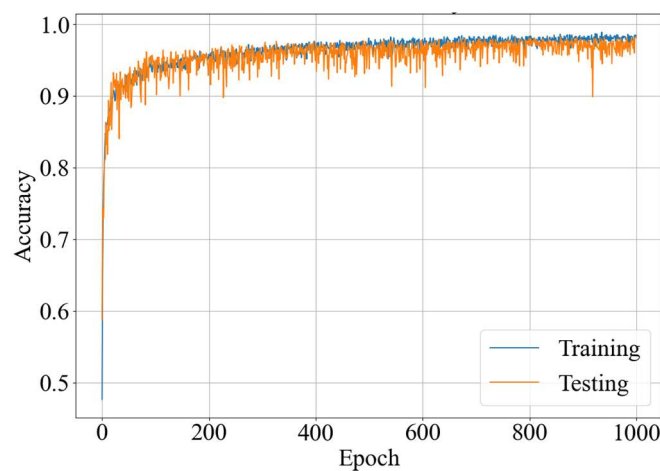


Figure 8.18. Final bubbly to slug transition boundary FNN performance.

The final slug to churn boundary architecture is displayed in Figure 8.19, where 18 features were selected and fewer hidden layers were used. The selected hidden layer configuration consisted of two layers with 64 nodes each. The sigmoid activation function was also used for the slug to churn transition FNN.

The final model accuracy for the slug to churn transition boundary FNN is displayed in Figure 8.20. The average performance was lower than for the bubbly to slug transition boundary at 94.74%. Again, there was significant noise in the results that may be due to disagreements between researchers for similar experimental results.

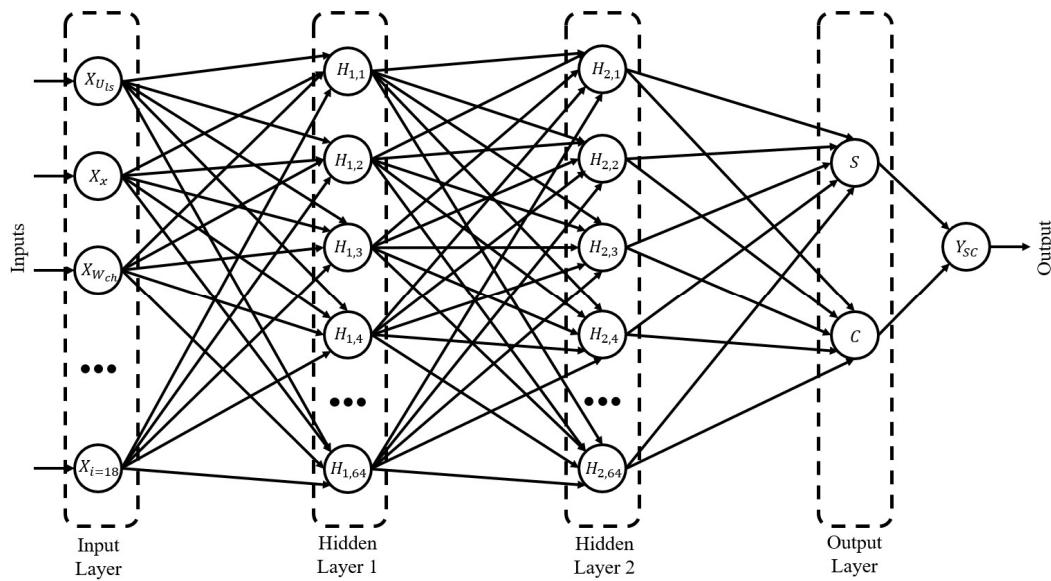


Figure 8.19. Final slug to churn transition boundary FNN architecture.

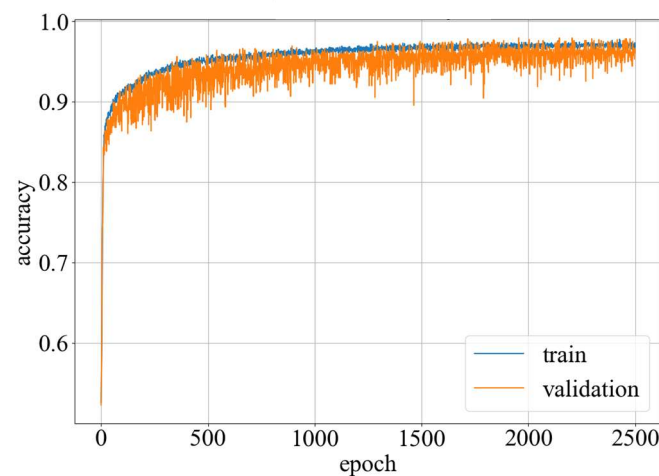


Figure 8.20. Final slug to churn transition boundary FNN performance.

The FNN architecture for the final transition boundary is represented diagrammatically in Figure 8.21. The churn to annular transition boundary FNN utilised the fewest features and contained 3 layers in the hidden layers, each with 64 nodes. This was the only transition boundary FNN model that performed better with a dropout rate, albeit a low one at 5%.

The accuracy of the churn to annular transition boundary is shown in Figure 8.22. Two things that differ from the performances of the other two transition boundary FNNs are the convergence and fluctuations in performance. Firstly, the model was trained with 2500 epochs; although, the model still appears to be moving towards convergence. Despite this, the average performance of the model was 93.96%. Here, a trade off was met between computational expense and model accuracy. The model still provides a satisfactory performance even though the model has not yet fully converged.

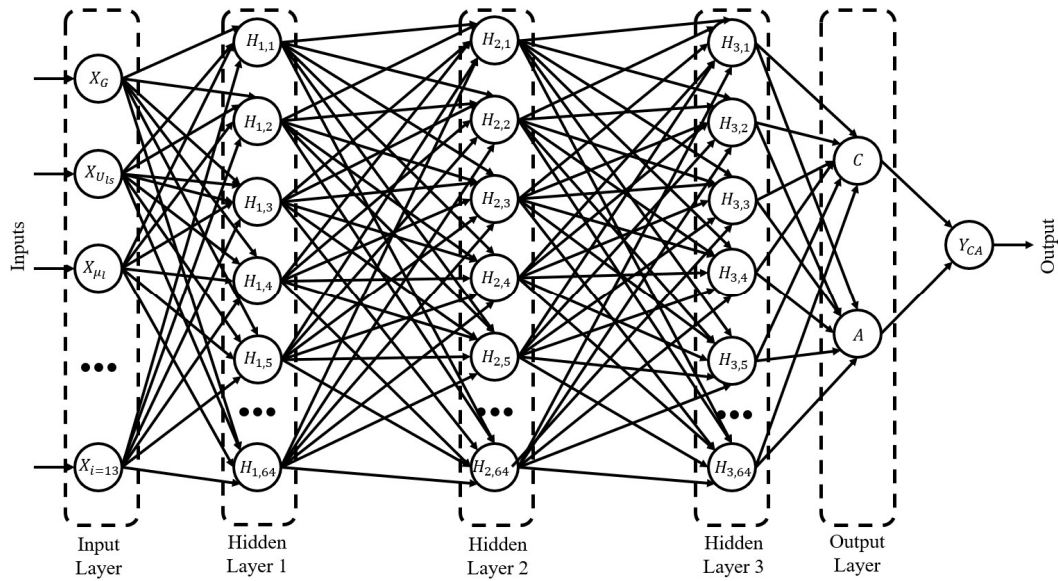


Figure 8.21. Final churn to annular transition boundary FNN architecture.

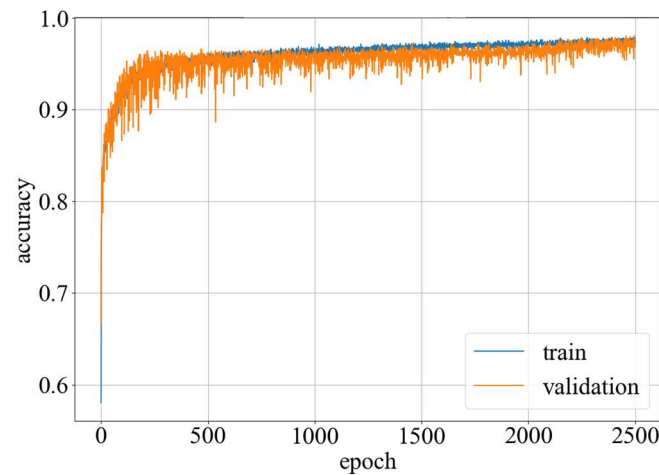


Figure 8.22. Final churn to annular transition boundary FNN performance.

The final architecture of the classification FNN model is displayed in Figure 8.23. The input layer differed to the other FNNs in the fact that the inputs were not physical parameters, but simply the outputs from the other models. The output layer also differed significantly as four nodes were used to represent bubbly, slug, churn and annular flows. The optimal hidden layer configuration was found to be 3 layers each with 128 nodes. However, perhaps the largest difference lies in the activation function where a rectified linear unit function was instead used.

The rectified linear unit function does not compress the data like a sigmoid function but instead treats negative values as 0. This concept is shown in Equation (8.6). The final model performance is displayed in Figure 8.24, where the average accuracy with respect to the training data was 95.53%. It is noticeable that there are no significant gains with the epochs after an initial training. This was to be

expected as the function of the network was to simply handle cases in which the individual FNN algorithms provided unexpected results due to being shown data that is significantly different to that it was trained with, such as bubbly data in the churn to annular transition boundary FNN.

$$\phi(X) = \max(0, X) \quad (8.6)$$

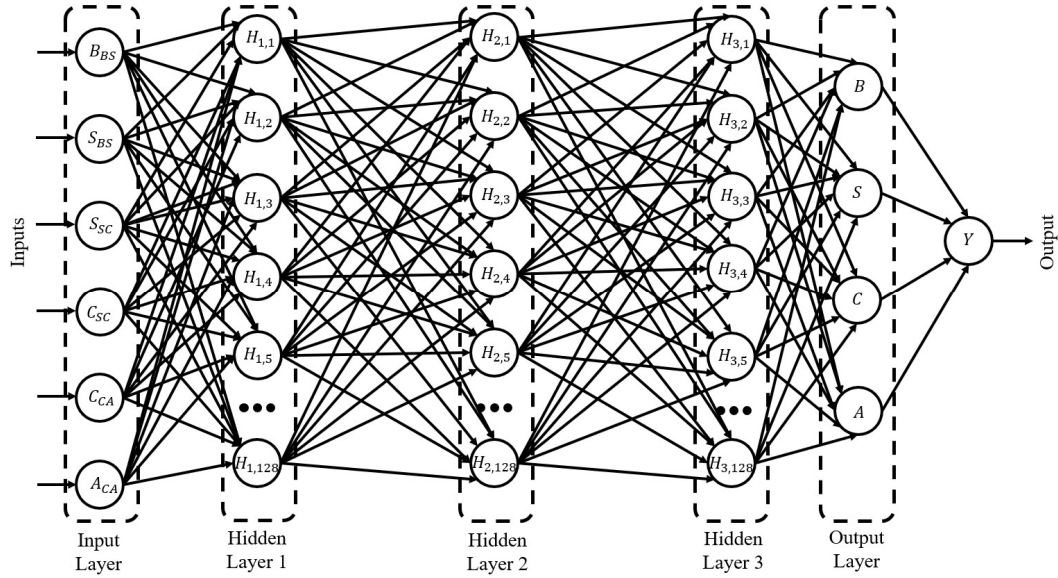


Figure 8.23. Final classification FNN architecture.

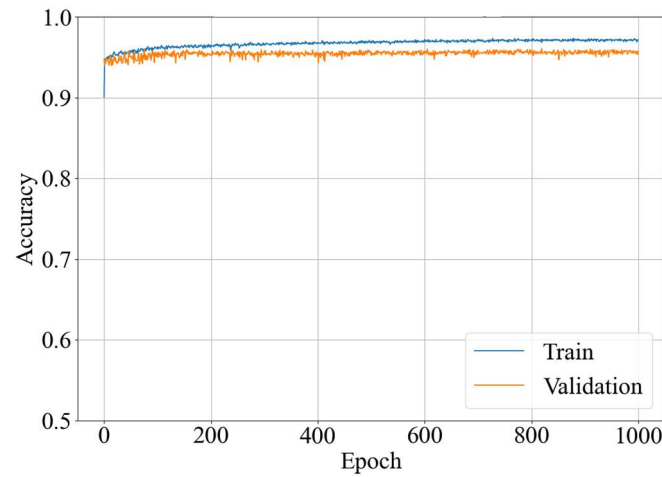


Figure 8.24. Final classification FNN performance.

8.4 COMPARISON TO CURRENT STUDY

The trained multiple FNN model was used to predict the flow pattern regions for the data collected at a 1 bar inlet pressure for the 75 mm long test section in the current study to validate the model against the new experimental data. This was an important step for validation the model, despite the accuracy tests that were conducted and presented in Section 8.3. This is because the algorithm may be capturing idiosyncrasies from the facilities used to obtain the data, as opposed to capturing actual physical trends. The predictions were made by ‘digitising’ a grid of mass flux and vapour quality data points with the corresponding features selected for each individual FNN algorithm. Separating lines were then drawn between the regions. However, this process was required for each location as the bubbly to slug transition boundary network uses the distance along the channel parameter to make predictions. Hence, the transition boundary prediction lines are plotted in Figure 8.25 for each location and compared to the experimental flow pattern data obtained.

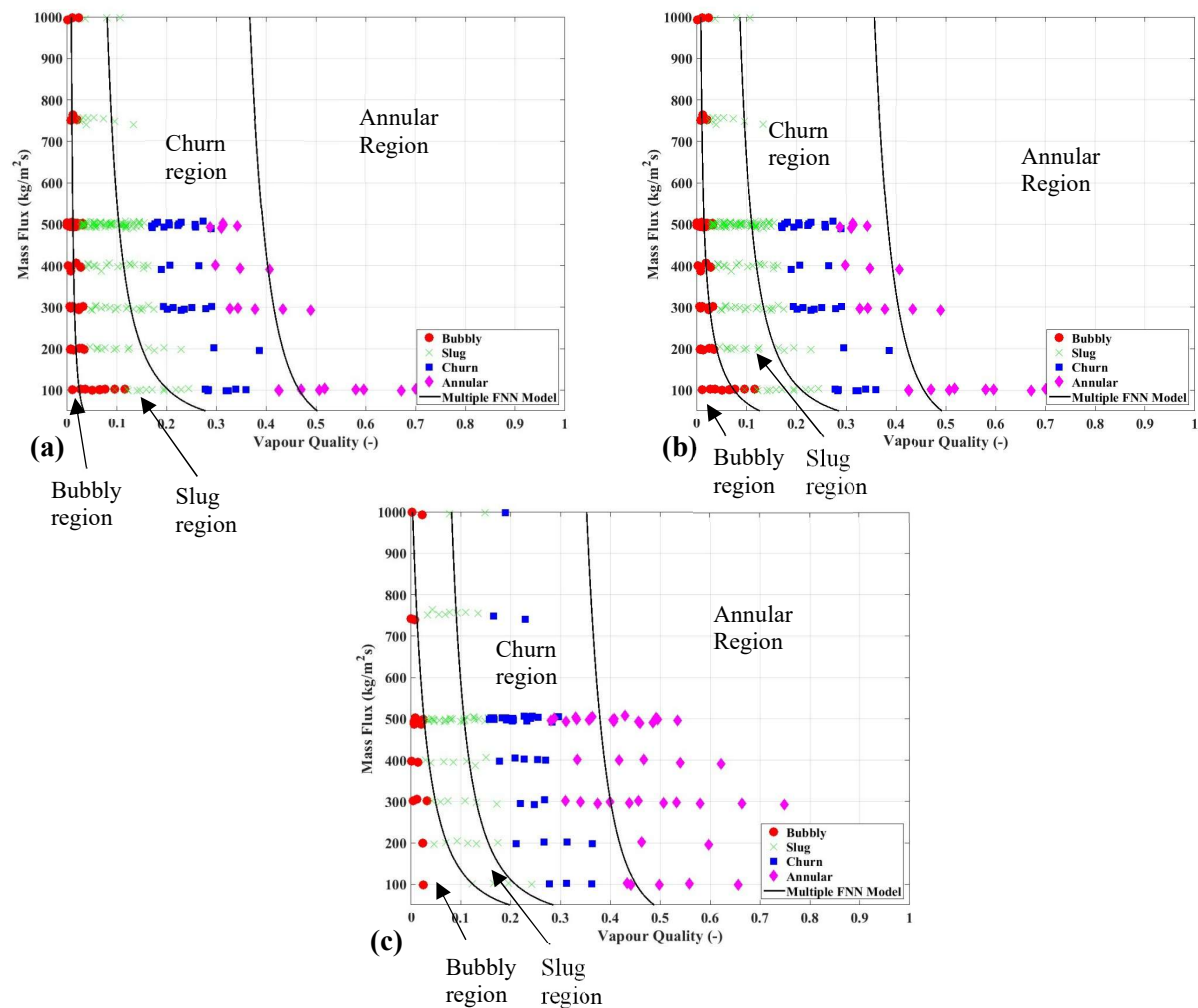


Figure 8.25. Comparison between FNN model predictions and experimental data: (a) at inlet location of channel; (b) at middle location of channel; (c) at outlet location of channel.

Analysis of the experimental data actually showed no dependence of the distance along the channel on the flow pattern transition between bubbly and slug flows. However, Figure 8.25 shows a fairly significant change in the transition boundary prediction when the distance along the channel is changed. It may be that the effect of length is more profound for different operating conditions found in the training data set, such as for other fluids or channel configurations. Thus, the model was seemingly unable to capture the overarching physical effect of heated length. The prediction for the middle of the channel was able to capture the bubbly to slug transition very well.

Overall, the trends were well captured by the multi-FNN algorithm although the vapour qualities at which the slug to churn transition and churn to annular transition occurred were under predicted and over predicted, respectively. However, the actual transition can be considered fairly similar to the model predictions when the propagated uncertainties of the vapour quality and mass flux are considered. Furthermore, the model can also be considered successful when the subjectivity in categorising flow patterns between researchers is also considered. Overall, the FNN performed significantly better than all the existing flow pattern prediction tools compared to in Chapter 4.

The experimental data was also predicted using the FNN algorithm in which all HFE-7100 and HFE-7200 data was removed from the training data set. The purpose of this was to observe if the network is capturing global flow pattern transition patterns for all configurations or if the algorithm is only using patterns taken for fluids with similar properties. The results are displayed in Figure 8.26 and show that the model was not able to accurately predict the flow pattern transitions when the data was removed. Hence, the algorithm is only able to generalise for a given fluid but not for the entire data set.

This represents an important current limitation of the model, as it has not yet demonstrated the ability to capture fully general transition patterns across different fluids and therefore requires training data representative of the fluid to be predicted. One reason for this may simply be the amount of data used to develop the model. Many of the input parameters may not be sufficiently continuous to accurately capture overall effects of certain relevant parameters, despite the use of 11349 data points (8832 with HFE-7100 and HFE-7200 data removed). This is because many data points will come from a single experimental set up. Hence, fluid properties and channel geometry will be very similar for the entire data set from the single experiment. There appears to be a requirement for a larger, more continuous data set to improve the algorithms capability as a universal flow pattern predict tool. This data set would preferably possess a wide range of fluids and channel geometries. However, it does appear to perform well for the fluids it has been trained with.

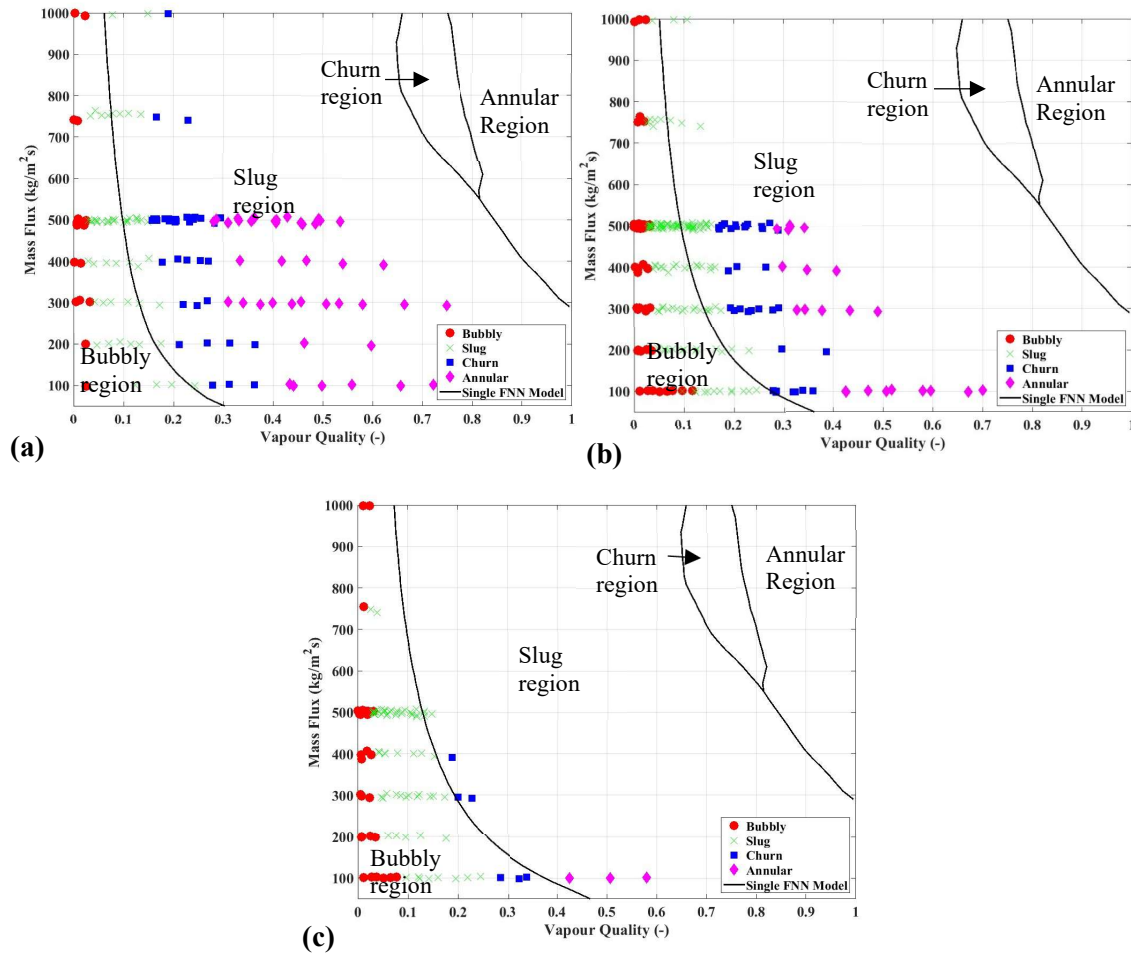


Figure 8.26. Comparison between FNN model predictions with HFE-7100 and HFE-7200 removed from the training data set and experimental data: (a) at inlet location of channel; (b) at middle location of channel; (c) at outlet location of channel.

Overall, the developed FNN demonstrates a clear improvement over existing empirical flow pattern maps discussed in Chapter 4, which exhibited poor predictive accuracy under the studied microscale conditions. The neural network's ability to incorporate a broad range of thermophysical and geometric inputs, combined with data-driven feature selection and rigorous cross-validation, allows it to capture complex, non-linear transition behaviour that conventional maps cannot.

However, the results also highlight that the model's predictive capability is currently limited to fluids represented in its training data. When all HFE-7100 and HFE-7200 data were removed from the training set, the network failed to accurately reproduce the flow pattern transitions, indicating that it has not yet learned fully generalisable physical relationships across different fluids. This reflects a key limitation of data-driven approaches: their performance is strongly dependent on the diversity and representativeness of the training dataset. As such, while the FNN offers a promising alternative or complementary tool to traditional correlations, further work is required to expand the database and improve the model's ability to generalise to new fluids and operating regimes.

8.5 CONCLUSIONS

This study developed and validated a novel neural network based approach for predicting flow pattern transitions in microscale two-phase flows. The research addressed the limitations of conventional empirical methods by integrating data driven modelling with physically informed feature selection and rigorous statistical validation. The conclusions from this study are as follows:

- A new multi-stage FNN architecture was created to predict microscale flow pattern transitions. The framework consisted of three dedicated networks for the key regime boundaries, i.e. bubbly to slug, slug to churn, and churn to annular, followed by a final classification network that integrated these results to provide a complete flow regime map. This structure enabled the model to capture the hierarchical nature of flow transitions in confined geometries.
- The model was trained on the comprehensive Brunel Two-Phase Flow database, which contains over 11,000 high-quality experimental data points covering a wide range of fluids, channel dimensions, and operating conditions. Supervised learning with 80/20 data splitting, input normalisation, and 10-fold cross-validation ensured robust generalisation and minimised overfitting, providing a statistically reliable basis for model training and evaluation.
- Feature importance analysis using the ANOVA F-test and the SelectKBest method identified the most influential thermophysical and geometric parameters governing flow transitions. Hyperparameter optimisation through grid search refined the network architecture, activation functions, and learning parameters, resulting in improved model performance and interpretability.
- The trained networks achieved high predictive accuracy across all transition boundaries, with classification rates of 96.4% for the bubbly to slug, 94.7% for the slug to churn, and 94.0% for churn to annular transitions. The overall classification accuracy was 95.5%, surpassing existing empirical and semi-empirical tools. When applied to new experimental data, the model maintained strong predictive capability, demonstrating its ability to generalise within the trained parameter space.
- The results confirm that data-driven neural architectures can effectively capture the nonlinear and multidimensional behaviour of microscale flow pattern transitions. While generalisation to entirely new fluids remains a challenge that requires a significantly larger databank, the framework establishes a robust foundation for future hybrid models that integrate physics-informed features with machine learning, offering a powerful tool for the design and optimisation of microscale two-phase flow systems.

CHAPTER 9

CONCLUSIONS AND RECOMMENDATIONS

9.1 CONCLUSIONS

Flow boiling of HFE-7100 was investigated in single microchannel test sections with rectangular cross sections. Each test section was designed with the same hydraulic diameter, channel width and height of 0.75 mm, where two heated lengths of 30 mm and 75 mm were investigated. Experiments were conducted at constant inlet pressures of 1 bar, 1.5 bar and 2 bar, at a constant inlet subcooling of 10 K. Mass fluxes between 100 kg/m²s and 1000 kg/m²s were studied at heat fluxes between 0 kW/m² and 435 kW/m², corresponding to outlet vapour qualities between 0 and 1 or the vapour quality at which CHF occurred.

The experimental facility underwent thorough calibration of its instrumentation to ensure precise measurements of pressure, temperature, and mass flow rates. Its accuracy and reliability were then validated through single-phase adiabatic and diabatic flow tests, which showed pressure drop and heat transfer results consistent with theoretical correlations in literature. Repeatability tests further confirmed the stability and consistency of the system, with variations remaining within acceptable error margins. A detailed error propagation was performed and the experimental uncertainties reported.

The experimental investigation examined parametric effects on flow boiling patterns, heat transfer, pressure drop and critical heat flux and linked the quantitative data to the flow behaviour recorded from high-speed, high resolution flow visualisation. Existing predictive tools for flow pattern transition boundaries, two-phase heat transfer coefficients, two-phase pressure drop and critical heat flux were examined against the obtained experimental data.

Finally, a new novel neural network architecture was trained and developed to predict flow pattern transitions in microscale two-phase flows. The approach uses a multi-stage Feed-Forward Neural Network with separate models for key regime boundaries and a final classification network to capture the hierarchical structure of flow transitions. The model was trained on a large experimental dataset with comprehensive pre-processing, cross-validation, and hyperparameter optimisation, and feature analysis was performed to identify the most influential parameters. The results were compared to the newly obtained data to investigate the feasibility of further use of such a model for the prediction of two-phase flow patterns. The results from the investigation are presented below.

9.1.1 Flow Boiling Patterns

The flow visualisation performed at the inlet, mid-length, and outlet of the microchannel provided a clear picture of the dominant boiling regimes over the investigated operating range. Four principal flow patterns were identified: bubbly, slug, churn, and annular flow, with systematic transitions occurring as heat flux and vapour quality increased. Nucleation was consistently observed to originate at the channel corners, where geometric confinement promoted early bubble formation and enhanced bubble stability. The transition from bubbly to slug flow was primarily governed by bubble coalescence under confinement, and this process was strongly influenced by mass flux and system pressure. Increasing the mass flux delayed the bubbly to slug transition to higher heat fluxes but lower boiling numbers for a given dimensionless saturated length, owing to higher phase velocities and increased collision frequency. Similarly, elevated inlet pressure reduced bubble departure diameters and improved bubble stability, shifting the transition to higher vapour qualities.

Heated length also played a significant role in the development of the early regimes. The shorter heated length promoted an earlier onset of slug flow, as higher local heat fluxes generated larger, more unstable bubbles that coalesced more readily. Once established, slug flow transitioned to churn flow through the progressive elongation and merging of confined vapour slugs. Higher heat fluxes intensified interfacial shear and instability, accelerating this transition. The subsequent transition from churn to annular flow occurred with the formation of a continuous vapour core surrounded by a thin liquid film along the channel walls. Increasing the mass flux shifted the churn to annular transition to higher dimensionless saturated length scaled heat fluxes but lower dimensionless saturated length scaled boiling numbers, reflecting the greater vapour momentum required to displace the liquid phase and establish a stable annular structure.

When compared with existing flow pattern maps from the literature, including those of Chen [198], Mahmoud and Karayiannis [82], Ong and Thome [80], and Harirchian and Garimella [203], only partial agreement was observed. The available maps exhibited inconsistent predictive capability, particularly for the present microscale rectangular channels. To address this, a new semi-analytical transition boundary incorporating viscous drag was developed. This approach improved the prediction of the bubbly to slug transition, especially for the 30 mm channel, although it did not fully capture the effects of heated length or local heat flux. Overall, the combined visual and analytical results emphasise the importance of confinement, operating conditions, and channel geometry in governing flow regime development in microchannel flow boiling.

9.1.2 Flow Boiling Heat Transfer

The heat transfer performance of HFE-7100 flow boiling in the single rectangular microchannel was characterised over a range of mass fluxes, heat fluxes, system pressures, and heated lengths. The onset

of nucleate boiling was clearly identifiable by a sharp decrease in wall superheat, observed as temperature overshoot. This phenomenon occurred under nearly all operating conditions, although its magnitude decreased with increasing system pressure due to potential earlier activation of smaller surface cavities. Boiling incipience therefore occurred at lower wall superheats as the system pressure increased, in agreement with Hsu's superheated-boundary-layer model [243] and the gas entrapment criteria proposed by Bankoff [245] and Lorenz [246]. A combined interpretation of the models of Hsu [243], Bankoff [245], Lorenz [246], and Marto and Rohsenow [399] further indicated that smaller cavities become active at reduced superheats but remain stable only when vapour entrapment is maintained. These models helped to provide a physical explanation for the observed trends in temperature overshoot and onset of nucleate boiling with changing system pressure. In a small number of cases, temperature overshoot was not observed; this was attributed to air entrainment prior to testing, which facilitated premature cavity activation and reduced the required wall superheat.

Across all operating conditions, the local heat transfer coefficient increased with wall heat flux, reflecting an increase in nucleation site density and bubble departure frequency at higher heat inputs. Even within the subcooled region, the heat transfer coefficient was affected by wall heat flux due to agitation of the liquid by departing bubbles, caused by the presence of subcooled boiling. As vapour quality increased beyond the bubbly regime, the local heat transfer coefficient generally decreased, because the contribution from discrete bubble agitation diminished. In the slug and churn flow regions, transient dry spots were observed that locally reduced heat transfer; however, continued nucleation within adjacent liquid films maintained relatively high average heat transfer coefficients.

The influence of mass flux on heat transfer was limited within the investigated range, indicating that nucleate boiling dominated the overall heat transfer mechanism rather than convective effects. In contrast, system pressure had a clear and systematic influence. The average two-phase heat transfer coefficient increased with increasing system pressure, particularly at higher wall heat fluxes, due to enhanced nucleation site stability and higher bubble generation frequency. According to the cavity stability criterion of Marto and Rohsenow [399], a greater range of surface cavities remained active at higher pressures, contributing to increased nucleation site density and improved heat transfer performance.

Heated length also had a significant effect on the heat transfer characteristics. The shorter channel produced higher local heat fluxes for the same vapour quality and consequently higher local heat transfer coefficients. Although higher wall superheats were required for boiling incipience in the shorter test section, the overall heat transfer coefficient remained greater because of intensified bubble activity and the more concentrated heat input, i.e. heat flux. Among the tested prediction tools, the correlation of Mahmoud and Karayiannis [270] performed comparatively well, as it incorporated the influence of nucleate boiling, which was found to be consistently present in the experiments. This feature allowed

it to capture part of the dominant heat transfer behaviour observed experimentally, unlike correlations based solely on convective boiling or those that overpredicted the convective contribution.

9.1.3 Flow Boiling Pressure Drop

The pressure drop characteristics of HFE-7100 flow boiling in the single rectangular microchannel were evaluated over a range of mass fluxes, heat fluxes, inlet pressures, and heated lengths, with the measurements interpreted alongside the flow visualisation results. The overall two-phase pressure drop increased with mass flux because of the enhanced liquid and vapour momentum, which produced higher frictional and acceleration pressure losses. In contrast, increasing the inlet pressure reduced the total pressure drop, primarily due to the associated decrease in vapour density ratio and the reduced development of void fraction along the channel. The effect of inlet pressure was also linked to the delayed transition from bubbly to more complex flow regimes, which corresponded to greater interfacial interaction and altered pressure drop behaviour.

The influence of heated length was evident in the distribution of local pressure gradients. The shorter channel exhibited higher local pressure gradients at equivalent vapour qualities because higher local heat fluxes produced more rapid vapour generation, intensifying acceleration losses. Analysis of the pressure drop development suggested that the onset of significant two-phase pressure drop coincided with the appearance of slug flow. At low heat fluxes, lower channel mass fluxes produced higher two-phase pressure drops once slug flow formed, compared with bubbly flow at higher mass fluxes, indicating that the regime transition played a key role in governing the pressure drop behaviour.

Comparison of the experimental data with established predictive models showed that the homogeneous flow model provided the best overall agreement with the measured two-phase pressure drops. This was attributed to a potential confinement mechanism that caused significant acceleration of the vapour phase, resulting in homogeneous two-phase friction factors similar to those suggested for flashing of steam–water mixtures in conventionally sized channels. Correlations developed specifically for microchannels, such as those of Keepaiboon et al. [296] and Moradkhani et al. [302], reproduced some of the observed trends but did not fully capture the effects of the governing parameters. Among the microscale correlations, the model of Gao et al. [299] performed the best, which was attributed to the overlap in experimental parameters and the use of fluid properties similar to those of HFE-7100. Examination of the pressure drop variation with vapour quality further indicated that acceleration losses were dominant in both channels, with frictional losses contributing more significantly in the longer test section.

9.1.4 Critical Heat Flux

The study established the physical mechanisms leading to CHF, examined the influence of mass flux and inlet pressure on the critical condition, and compared the measured data with existing microscale CHF correlations. Flow visualisation was used to identify the sequence of interfacial events preceding CHF, linking local film dynamics to the global thermal response. Critical heat flux in the 30 mm microchannel was governed primarily by liquid film dry out occurring within the annular flow regime. High-speed visualisation confirmed the progression from nucleating annular flow to a fully vapour filled channel, accompanied by cyclic wetting and drying of the heated surface. The onset of dry out was characterised by a marked decline in the two-phase heat transfer coefficient and oscillatory flow pattern behaviour, with fluctuations reflecting unsteady wetting and drying cycles that produced the rapid temperature rise associated with CHF.

Mass flux was found to be the dominant parameter influencing CHF behaviour. Higher mass fluxes required higher wall heat fluxes to generate the necessary vapour production, and it was also proposed to extend the CHF limit to higher wall heat fluxes through increased liquid inertia, which promoted bubble removal and delayed complete film depletion. Inlet pressure also influenced CHF, although to a lesser degree. Elevated pressures produced thicker liquid films that delayed dry out by reducing vapour superficial velocity and interfacial shear stress. The relatively weak pressure dependence observed contrasts with the strong pressure sensitivity typically reported for pool boiling.

The two-phase heat transfer coefficient increased progressively with wall heat flux prior to the onset of dry out, while exhibiting negligible dependence on mass flux in this region. Once dry out commenced, the coefficient declined sharply, consistent with the observed reduction in liquid film coverage and the suppression of nucleation. The dominant CHF mechanism was therefore linked to transient liquid film instability and partial rewetting, rather than gradual depletion of the liquid film or flow reversal, distinguishing the microscale CHF behaviour observed in this study from mechanisms previously reported in the literature.

Comparative evaluation of eight existing CHF correlations showed that the models of Ong and Thome [208] and Tibiri   et al. [186] provided the most accurate predictions, with mean absolute errors of approximately 10–12%. Correlations developed for longer channels or for different working fluids, such as those of Qu and Mudawar [309], substantially over-predicted the measured CHF values.

9.1.5 A Novel Neural Network for Flow Pattern Prediction

Flow pattern prediction in microscale channels was investigated through the development of a new feed-forward neural network model trained on a large, multi-fluid experimental database. The model adopted a multi-stage architecture consisting of three transition-specific subnetworks for the bubbly–

slug, slug–churn, and churn–annular transitions, followed by a final classification network that integrated the outputs into a complete four-regime prediction model. Training was carried out using the Brunel Two-Phase Flow database, which contained more than 11,000 high-fidelity data points covering a range of working fluids, channel geometries, and orientations. Data normalisation and 10-fold cross-validation were applied during training to prevent overfitting and ensure consistent performance on previously unseen data, enabling the feed-forward neural network to achieve genuine generalisation within the trained operating space.

Feature selection was performed using ANOVA F-tests together with the SelectKBest method to identify the most influential physical parameters governing each transition boundary. This process provided interpretability and confirmed the relevance of mass flux, vapour quality, and surface tension in defining regime behaviour. Hyperparameter tuning, including optimisation of hidden-layer configuration, activation functions, optimiser type, batch size, and number of training epochs, produced an optimised network architecture. The final model achieved accuracies of 96.4%, 94.7%, and 94.0% for the three individual transition boundaries, and an overall accuracy of 95.5% for the complete four-regime prediction.

Comparison with the experimental results obtained in the present study showed that the model accurately reproduced the observed flow regime boundaries. However, when fluid-specific data for HFE-7100 and HFE-7200 were removed from the training set, the model performance degraded significantly. This behaviour indicated that the network was not identifying fully general mechanistic transition criteria, but was instead relying on fluid-specific transition patterns present in the training data. This finding highlights the need for broader and more continuous experimental datasets spanning wider ranges of fluid properties and channel geometries in order to improve the generalisation capability of data-driven flow pattern prediction models.

9.2 SUMMARY OF NOVEL CONTRIBUTIONS

The current study has introduced several original insights and advancements to the field, which are outlined below:

1. A new semi-analytical model was proposed for predicting the bubbly to slug flow transition in microscale rectangular channels. This model uniquely introduces a viscous drag term into the force balance on a confined bubble, extending the analytical framework of Mahmoud and Karayiannis [82]. The modified formulation captures the influence of viscous forces acting on confined bubbles within laminar two-phase flow, which had previously been neglected in existing transition criteria. When validated against experimental data, the model significantly improved prediction accuracy for the shorter heated length channel. This suggests that viscous

drag may be a yet previously unrecognised governing mechanism in microscale regime transition. This represents the first semi-analytical flow-pattern transition boundary in the literature to incorporate viscous drag in the confined microscale regime.

2. This research provides, to the author's knowledge, the first experimental investigation into the effect of heated length for flow boiling in a single rectangular microchannel. The heated length was found to exert a direct and systematic influence on both the bubbly to slug and slug to churn flow pattern transitions in confined microscale rectangular channels. The present study demonstrates that shorter heated sections produce earlier transitions at lower vapour qualities due to enhanced evaporation momentum and bubble coalescence dynamics. This establishes heated length as an independent governing parameter for microscale flow boiling in regards to flow patterns, heat transfer and pressure drop.
3. This work is the first to establish a link between microscale flow boiling behaviour and classical low-pressure flashing steam-water studies from the 1930s to 1950s. By demonstrating that the use of a two-phase friction factor value of 0.003 within the homogeneous flow model yields accurate predictions in microscale channels, this study connects current microscale phenomena to historical acceleration dominated flow mechanisms. This novel connection provides new physical insight into why homogeneous models can perform well in confined microscale geometries.
4. The present work provides the first experimental evidence of a cyclic rewetting mechanism governing the onset of CHF in confined microscale rectangular channels. High-speed visualisation revealed that, as the liquid film thinned during annular flow, periodic rewetting of the heated wall occurred due to intermittent liquid inflow following vapour accumulation at the outlet. These cycles of wetting and dry-out produced characteristic wall temperature oscillations immediately prior to CHF, distinguishing this mechanism from the steady dry out or flow reversal processes reported by Qu and Mudawar [309].
5. This study proposes a unique neural network structure comprising three dedicated transition-boundary FNNs (bubbly-slug, slug-churn, churn-annular) followed by a classification FNN. This modular approach allows for targeted feature selection and improved interpretability, representing a significant departure from conventional monolithic classification models in two-phase flow research.
6. The use of ANOVA F-test and SelectKBest to identify key thermophysical, geometric, and operational parameters for each transition boundary enables not only improved accuracy but also deeper understanding of the physical mechanisms influencing flow pattern transitions, bridging the gap between data driven modelling and physical interpretation.

9.3 RECOMMENDATIONS

Based on the findings of this study, there remain areas that warrant additional investigation. The key recommendations for future research are as follows:

1. The results demonstrated that current flow pattern correlations cannot accurately capture microscale flow pattern transition boundaries. Future research should focus on developing unified predictive frameworks that incorporate confinement, evaporation momentum, viscous drag, and interfacial shear effects. Such models would improve the reliability of design calculations for compact two-phase heat exchangers and microelectronic cooling devices.
2. The Mahmoud and Karayiannis [270] heat transfer correlation demonstrated strong predictive performance for microscale flows in which nucleate boiling is the dominant heat transfer mechanism. Future research should aim to clarify the conditions under which nucleate boiling dominates and why it dominates for all vapour qualities in some but not in others. A deeper understanding of these controlling factors will inform the appropriate selection of correlations for engineering design. The Mahmoud and Karayiannis [270] heat transfer correlation is recommended for practical application.
3. The homogeneous flow model exhibited excellent agreement with experimental data, likely due to the dominance of acceleration pressure drop under the studied conditions. By contrast, other researchers, such as Qu and Mudawar [275], have reported better agreement with slip flow models in regimes where frictional pressure drop is dominant. Future work should focus on identifying the flow and geometric conditions under which each modelling approach is most applicable. When acceleration effects dominate in microchannels, the homogeneous flow model is recommended for use.
4. Many mechanisms for critical heat flux have been reported in the literature, and the present study introduces a newly observed cyclic rewetting mechanism in confined microscale channels. Future modelling and experimental efforts should focus on identifying the conditions, working fluid properties, and geometric parameters under which each CHF mechanism dominates. Developing predictive models that can distinguish between these regimes will enable more accurate and generalised CHF correlations for microscale boiling systems.
5. The FNN model's limited ability to generalise to fluids not included in the training set highlights the need for a more comprehensive dataset. Future research should focus on incorporating a wider variety of fluids, channel geometries, and operating conditions to improve the model's robustness and enable universal flow pattern prediction across broader applications.
6. This study demonstrates that neural networks are effective tools for flow pattern prediction in microscale systems. Future research could explore other architectures, such as physics-informed neural networks or hybrid data physics models, that may better capture underlying physical trends and improve predictive accuracy even for previously unseen fluids or conditions.

7. The experimental findings provide direct design guidance for applications employing dielectric working fluids such as HFE-7100 in high heat flux environments. The results indicate that operating within the bubbly and early slug flow regimes at elevated system pressures can enhance overall system efficiency, provided that the associated higher saturation temperatures are below the working limit of the substrate material. This insight is particularly relevant to the thermal management of high-power electronic systems and compact heat exchangers.

REFERENCES

- [1] T.C. Willingham, I. Mudawar, Forced-convection boiling and critical heat flux from a linear array of discrete heat sources, *Int. J. Heat Mass Transf.* 35 (1992) 2879–2890. [https://doi.org/10.1016/0017-9310\(92\)90308-F](https://doi.org/10.1016/0017-9310(92)90308-F).
- [2] S.M. Kim, I. Mudawar, Review of databases and predictive methods for heat transfer in condensing and boiling mini/micro-channel flows, *Int. J. Heat Mass Transf.* 77 (2014) 627–652. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2014.05.036>.
- [3] I. Mudawar, Two-phase microchannel heat sinks: Theory, applications, and limitations, *Journal of Electronic Packaging, Transactions of the ASME* 133 (2011). <https://doi.org/10.1115/1.4005300>.
- [4] S. Hong, B. Zhang, C. Dang, E. Hihara, Development of two-phase flow microchannel heat sink applied to solar-tracking high-concentration photovoltaic thermal hybrid system, *Energy* 212 (2020) 118739. <https://doi.org/10.1016/J.ENERGY.2020.118739>.
- [5] A. Radwan, S. Ookawara, M. Ahmed, Thermal management of concentrator photovoltaic systems using two-phase flow boiling in double-layer microchannel heat sinks, *Appl. Energy* 241 (2019) 404–419. <https://doi.org/10.1016/J.APENERGY.2019.03.017>.
- [6] H. Yu, J. Zhuang, T. Li, W. Li, T. He, N. Mao, Influence of transient heat flux on boiling flow pattern in a straight microchannel applied in concentrator photovoltaic systems, *Int. J. Heat Mass Transf.* 190 (2022) 122792. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2022.122792>.
- [7] S.M. Kim, I. Mudawar, Universal approach to predicting saturated flow boiling heat transfer in mini/micro-channels – Part II. Two-phase heat transfer coefficient, *Int. J. Heat Mass Transf.* 64 (2013) 1239–1256. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2013.04.014>.
- [8] M.H. Mousa, N. Miljkovic, K. Nawaz, Review of heat transfer enhancement techniques for single phase flows, *Renewable and Sustainable Energy Reviews* 137 (2021) 110566. <https://doi.org/10.1016/J.RSER.2020.110566>.
- [9] S. Eiamsa-Ard, V. Kongkai-paiboon, K. Nanan, Thermohydraulics of Turbulent Flow Through Heat Exchanger Tubes Fitted with Circular-rings and Twisted Tapes, *Chin. J. Chem. Eng.* 21 (2013) 585–593. [https://doi.org/10.1016/S1004-9541\(13\)60504-2](https://doi.org/10.1016/S1004-9541(13)60504-2).
- [10] P. Promvong, Thermal augmentation in circular tube with twisted tape and wire coil turbulators, *Energy Convers. Manag.* 49 (2008) 2949–2955. <https://doi.org/10.1016/J.ENCONMAN.2008.06.022>.

- [11] K. Wongcharee, S. Eiamsa-ard, Heat transfer enhancement by twisted tapes with alternate-axes and triangular, rectangular and trapezoidal wings, *Chemical Engineering and Processing: Process Intensification* 50 (2011) 211–219. <https://doi.org/10.1016/J.CEP.2010.11.012>.
- [12] S. Eiamsa-ard, K. Wongcharee, P. Eiamsa-ard, C. Thianpong, Heat transfer enhancement in a tube using delta-winglet twisted tape inserts, *Appl. Therm. Eng.* 30 (2010) 310–318. <https://doi.org/10.1016/J.APPLTHERMALENG.2009.09.006>.
- [13] R.M. Manglik, A.E. Bergles, Heat transfer enhancement and pressure drop in viscous liquid flows in isothermal tubes with twisted-tape inserts, *Wärme- Und Stoffübertragung* 27 (1992) 249–257. <https://doi.org/10.1007/BF01589923>.
- [14] P. Kosky, R. Balmer, W. Keat, G. Wise, *Mechanical Engineering, Exploring Engineering* (2021) 317–340. <https://doi.org/10.1016/B978-0-12-815073-3.00014-4>.
- [15] H. Sakashita, T. Tsuruta, N. Nagai, S. Mori, M. Shoji, Y. Haramura, H. Ohtake, W. Liu, H. Umekawa, Y. Koizumi, S. Morooka, *Boiling*, Elsevier, 2017. <https://doi.org/10.1016/B978-0-08-101010-5.00003-8>.
- [16] A. Prieto-Guerrero, G. Espinosa-Paredes, *Linear and non-linear stability analysis in boiling water reactors: The Design of Real-Time Stability Monitors*, Woodhead Publishing, 2019. <https://doi.org/10.1016/C2017-0-01640-3>.
- [17] X. Ji, X. Ma, X. Yang, J. Wei, B. Sundén, Jet array impingement boiling in compact space for high heat flux cooling, *Appl. Therm. Eng.* 219 (2023) 119538. <https://doi.org/10.1016/J.APPLTHERMALENG.2022.119538>.
- [18] V.S. Devahdhanush, I. Mudawar, Review of Critical Heat Flux (CHF) in Jet Impingement Boiling, *Int. J. Heat Mass Transf.* 169 (2021) 120893. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2020.120893>.
- [19] A.M. Huber, R. Viskanta, Effect of jet-jet spacing on convective heat transfer to confined, impinging arrays of axisymmetric air jets, *Int. J. Heat Mass Transf.* 37 (1994) 2859–2869. [https://doi.org/10.1016/0017-9310\(94\)90340-9](https://doi.org/10.1016/0017-9310(94)90340-9).
- [20] J.Z. Yi, H.X. Hu, Z.B. Wang, Y.G. Zheng, On the critical flow velocity for erosion-corrosion in local eroded regions under liquid-solid jet impingement, *Wear* 422–423 (2019) 94–99. <https://doi.org/10.1016/J.WEAR.2019.01.069>.
- [21] T.G. Karayiannis, M.M. Mahmoud, *Flow boiling in microchannels: Fundamentals and applications*, *Appl. Therm. Eng.* 115 (2017) 1372–1397. <https://doi.org/10.1016/J.APPLTHERMALENG.2016.08.063>.

- [22] M.M. Mahmoud, T.G. Karayiannis, "Flow Boiling in Mini to Microdiameter Channels," in Encyclopedia of Two-Phase Heat Transfer and Flow IV, Ed. World Scientific Publishing, 2018.
- [23] E.M. Fayyadh, M.M. Mahmoud, K. Sefiane, T.G. Karayiannis, Flow boiling heat transfer of R134a in multi microchannels, *Int. J. Heat Mass Transf.* 110 (2017) 422–436. <https://doi.org/10.1016/j.ijheatmasstransfer.2017.03.057>.
- [24] S. Krishnan, S. V. Garimella, G.M. Chrysler, R. V. Mahajan, Towards a thermal Moore's law, *IEEE Transactions on Advanced Packaging* 30 (2007) 462–474. <https://doi.org/10.1109/TADVP.2007.898517>.
- [25] G.E. Moore, Cramming more components onto integrated circuits, Reprinted from *Electronics*, volume 38, number 8, April 19, 1965, pp.114 ff., *IEEE Solid-State Circuits Society Newsletter* 11 (2009) 33–35. <https://doi.org/10.1109/N-SSC.2006.4785860>.
- [26] IBM, IBM System/370 Installation Manual- Physical Planning, Poughkeepsie, USA, 1985.
- [27] S. Paredes, Y. Madhour, G. Schlottig, C.L. Ong, T. Brunschwiler, (Invited) Wafer-Level Integration of Embedded Cooling Approaches, *ECS Trans.* 64 (2014) 253. <https://doi.org/10.1149/06405.0253ECST>.
- [28] J.W. Elliott, M.T. Lebon, A.J. Robinson, Optimising integrated heat spreaders with distributed heat transfer coefficients: A case study for CPU cooling, *Case Studies in Thermal Engineering* 38 (2022) 102354. <https://doi.org/10.1016/J.CSITE.2022.102354>.
- [29] S. kakaç, Introduction to ASI on Cooling of Electronic Systems, *Cooling of Electronic Systems* (1994) 1–15. https://doi.org/10.1007/978-94-011-1090-7_1.
- [30] B. Allard, C. Buttay, C. Martin, H. Morel, Considerations for High Temperature Power Electronics, in: 18th International Symposium on Power Electronics Ee2015, Novi Sad, Serbia, 2015.
- [31] A. Koşar, Y. Peles, Critical Heat Flux of R-123 in Silicon-Based Microchannels, *J. Heat Transfer* 129 (2007) 844–851. <https://doi.org/10.1115/1.2712852>.
- [32] D. Faulkner, M. Khotan, R. Shekariz, Practical design of a 1000 W/cm²/ cooling system [high power electronics], *Nineteenth Annual IEEE Semiconductor Thermal Measurement and Management Symposium*, 2003. (2003) 223–230. <https://doi.org/10.1109/STHERM.2003.1194366>.
- [33] H. Hirshfeld, I. Silverman, A. Arenshtam, D. Kijel, A. Nagler, High heat flux cooling of accelerator targets with micro-channels, *Nucl. Instrum. Methods Phys. Res. A* 562 (2006) 903–905. <https://doi.org/10.1016/J.NIMA.2006.02.104>.

- [34] K.W. Jung, H. Lee, C. Kharangate, F. Zhou, M. Asheghi, E.M. Dede, K.E. Goodson, Experimental Investigation of Single-Phase Cooling in Embedded Microchannels: 3D Manifold Heat Exchanger With R-245fa, ASME 2019 International Technical Conference and Exhibition on Packaging and Integration of Electronic and Photonic Microsystems, InterPACK 2019 (2019). <https://doi.org/10.1115/IPACK2019-6400>.
- [35] R. Mastrullo, A.W. Mauro, J.R. Thome, G.P. Vanoli, L. Viscito, Critical heat flux: Performance of R1234yf, R1234ze and R134a in an aluminum multi-minichannel heat sink at high saturation temperatures, International Journal of Thermal Sciences 106 (2016) 1–17. <https://doi.org/10.1016/J.IJTHERMALSCI.2016.03.011>.
- [36] Y. Madhour, J. Olivier, E. Costa-Patry, S. Paredes, B. Michel, J.R. Thome, Flow boiling of R134a in a multi-microchannel heat sink with hotspot heaters for energy-efficient microelectronic CPU cooling applications, IEEE Trans. Compon. Packaging Manuf. Technol. 1 (2011) 873–883. <https://doi.org/10.1109/TCPMT.2011.2123895>.
- [37] B. Parizad Benam, A.K. Sadaghiani, V. Yağcı, M. Parlak, K. Sefiane, A. Koşar, Review on high heat flux flow boiling of refrigerants and water for electronics cooling, Int. J. Heat Mass Transf. 180 (2021) 121787. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2021.121787>.
- [38] I. Mudawar, Assessment of high-heat-flux thermal management schemes, IEEE Transactions on Components and Packaging Technologies 24 (2001) 122–141. <https://doi.org/10.1109/6144.926375>.
- [39] M. Li, S. Li, Z. Zhang, C. Su, B. Wong, Y. Hu, Advancing Thermal Management Technology for Power Semiconductors through Materials and Interface Engineering, Acc. Mater. Res. 6 (2025) 563–576. <https://doi.org/10.1021/accountsmr.4c00349>.
- [40] A.H. Al-Zaidi, M.M. Mahmoud, T.G. Karayiannis, Flow boiling of HFE-7100 in microchannels: Experimental study and comparison with correlations, Int. J. Heat Mass Transf. 140 (2019) 100–128. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2019.05.095>.
- [41] Y. Wang, S. Liu, S. Liu, L. Liu, Electricity-computility integration of data centers and pumped storage driven by cold energy storage in China, Energy Reports 14 (2025) 4068–4085. <https://doi.org/10.1016/j.egyr.2025.11.048>.
- [42] J. Vidal, “Tsunami of data” could consume one fifth of global electricity by 2025, Climate Home News (2017). <https://www.climatechangenews.com/2017/12/11/tsunami-data-consume-one-fifth-global-electricity-2025/> (accessed June 13, 2023).
- [43] X. Yuan, X. Zhou, Y. Pan, R. Kosonen, H. Cai, Y. Gao, Y. Wang, Phase change cooling in data centers: A review, Energy Build. 236 (2021) 110764. <https://doi.org/10.1016/J.ENBUILD.2021.110764>.

- [44] S. Abedinpour, K. Shenai, Insulated Gate Bipolar Transistor, *Power Electronics Handbook* (2007) 71–88. <https://doi.org/10.1016/B978-012088479-7/50023-7>.
- [45] B.J. Baliga, *The IGBT device : physics, design and applications of the insulated gate bipolar transistor*, n.d.
- [46] C. Qian, A.M. Gheithaghy, J. Fan, H. Tang, B. Sun, H. Ye, G. Zhang, Thermal Management on IGBT Power Electronic Devices and Modules, *IEEE Access* 6 (2018) 12868–12884. <https://doi.org/10.1109/ACCESS.2018.2793300>.
- [47] D. Watts, *ThinkSystem SD650 Direct Water Cooled Server (Xeon SP Gen 1)*, Lenovo Press (2021).
- [48] BP Energy, *Statistical Review of World Energy 2022*, London, 2022.
- [49] IRENA, *Global Energy Transformation: A Roadmap to 2050*, International Renewable Energy Agency (2019) 52 (10–23).
- [50] L. Tan, P. Liu, C. She, P. Xu, L. Yan, H. Quan, Heat Dissipation Characteristics of IGBT Module Based on Flow-Solid Coupling, *Micromachines* 2022, Vol. 13, Page 554 13 (2022) 554. <https://doi.org/10.3390/M113040554>.
- [51] I. International Energy Agency, *Renewable Energy Market Update 2022*, (2022).
- [52] IRENA, *Renewable capacity statistics 2025*, Abu Dhabi, 2025.
- [53] P. Pontes, I. Gonçalves, M. Andredaki, A. Georgoulas, A.L.N. Moreira, A.S. Moita, Fluid flow and heat transfer in microchannel devices for cooling applications: Experimental and numerical approaches, *Appl. Therm. Eng.* 218 (2023) 119358. <https://doi.org/10.1016/J.APPLTHERMALENG.2022.119358>.
- [54] S. Deng, Construction of an Industrial Structure Analysis and Evaluation Model for Oil and Gas Resource-Based Cities Based on Deep Learning Model and Cluster Analysis, *Math. Probl. Eng.* 2022 (2022). <https://doi.org/10.1155/2022/2179494>.
- [55] W. Tian, S. Wang, S. Sun, C. Li, Y. Lin, Intelligent prediction and early warning of abnormal conditions for fluid catalytic cracking process, *Chemical Engineering Research and Design* 181 (2022) 304–320. <https://doi.org/10.1016/J.CHERD.2022.03.031>.
- [56] C. Panjapornpon, S. Bardeeniz, M.A. Hussain, Deep learning approach for energy efficiency prediction with signal monitoring reliability for a vinyl chloride monomer process, *Reliab. Eng. Syst. Saf.* 231 (2023). <https://doi.org/10.1016/J.RESS.2022.109008>.
- [57] M. Castangia, L.M.M. Grajales, A. Aliberti, C. Rossi, A. Macii, E. Macii, E. Patti, Transformer neural networks for interpretable flood forecasting, *Environmental Modelling and Software* 160 (2023). <https://doi.org/10.1016/J.ENVSOFT.2022.105581>.

- [58] K. Nagasree Pujari, S. Soumitri Miriyala, P. Mittal, K. Mitra, Better Wind forecasting using Evolutionary Neural Architecture Search driven Green Deep Learning, *Expert Syst. Appl.* (2022) 119063. <https://doi.org/10.1016/J.ESWA.2022.119063>.
- [59] W. Zhang, Y. Jiang, J. Dong, X. Song, R. Pang, B. Guoan, H. Yu, A deep learning method for real-time bias correction of wind field forecasts in the Western North Pacific, *Atmos. Res.* 284 (2023). <https://doi.org/10.1016/J.ATMOSRES.2022.106586>.
- [60] M. Aslam, J.S. Kim, J. Jung, Multi-step ahead wind power forecasting based on dual-attention mechanism, *Energy Reports* 9 (2023) 239–251. <https://doi.org/10.1016/J.EGYR.2022.11.167>.
- [61] X. Zhou, H. Zeng, C. Chen, H. Xiao, Z. Xiang, An attention-enhanced multi-modal deep learning algorithm for robotic compound fault diagnosis, *Meas. Sci. Technol.* 34 (2023). <https://doi.org/10.1088/1361-6501/AC93A5>.
- [62] M.I. Al-Tameemi, A.A. Hasan, B.K. Oleiwi, Design and implementation monitoring robotic system based on you only look once model using deep learning technique, *IAES International Journal of Artificial Intelligence* 12 (2023) 106–113. <https://doi.org/10.11591/IJAI.V12.I1.PP106-113>.
- [63] H.Z. Imam, H. Al-Musaibeli, Y. Zheng, P. Martinez, R. Ahmad, Vision-based spatial damage localization method for autonomous robotic laser cladding repair processes, *Robot. Comput. Integr. Manuf.* 80 (2023). <https://doi.org/10.1016/J.RCIM.2022.102452>.
- [64] R. Jiménez-Moreno, R.A. Castillo, Deep learning speech recognition for residential assistant robot, *IAES International Journal of Artificial Intelligence* 12 (2023) 585–592. <https://doi.org/10.11591/IJAI.V12.I2.PP585-592>.
- [65] B. Ramsundar, R.B. Zadeh, *TensorFlow for deep learning: from linear regression to reinforcement learning*, 1st ed., O'Reilly, 2018.
- [66] W. Rahman, *AI and machine learning*, 1st ed., SAGE Response, 2020.
- [67] A. Nemade, S. Telang, V. Jumbad, A. Chel, G. Kaushik, M. Hussain, Artificial Intelligence: A tool to resolve thermal behavior issues in disc braking systems, *Artificial Intelligence and Industry 4.0* 1–2 (2022) 117–141. <https://doi.org/10.1016/B978-0-323-88468-6.00005-X>.
- [68] P. Wittek, *Machine Learning, Quantum Machine Learning* (2014) 11–24. <https://doi.org/10.1016/B978-0-12-800953-6.00002-5>.
- [69] V. Kotu, B. Deshpande, *Data Science: Concepts and Practice*, 2nd ed., Morgan Kaufmann, 2019.
- [70] M.Y. Kiang, *Encyclopedia of Information Systems*, Academic Press, 2002.
- [71] K. Gurney, *An Introduction to Neural Networks*, Routledge, 1997.

- [72] F.H. Guenther, *International Encyclopedia of the Social & Behavioral Sciences*, Pergamon, 2001.
- [73] H. Ij, Statistics versus machine learning, *Nat Methods* 15 (2018). <https://doi.org/10.1038/nmeth.4642>.
- [74] S.S. Mehendale, A.M. Jacobi, R.K. Shah, Fluid Flow and Heat Transfer at Micro- and Meso-Scales With Application to Heat Exchanger Design, *Appl. Mech. Rev.* 53 (2000) 175–193. <https://doi.org/10.1115/1.3097347>.
- [75] S.G. Kandlikar, W.J. Grande, Evolution of microchannel flow passages-thermohydraulic performance and fabrication technology, *Heat Transfer Engineering* 24 (2003) 3–17. <https://doi.org/10.1080/01457630304040>.
- [76] M. Kawaji, M.-Y. Chung, Adiabatic Gas-Liquid Flow in Microchannels, ADIABATIC GAS-LIQUID FLOW IN MICROCHANNELS, *Microscale Thermophysical Engineering* 8 (2004) 239–257. <https://doi.org/10.1080/10893950490477518>.
- [77] P.A. Kew, K. Cornwell, Correlations for the prediction of boiling heat transfer in small-diameter channels, *Appl. Therm. Eng.* 17 (1997) 705–715. [https://doi.org/10.1016/s1359-4311\(96\)00071-3](https://doi.org/10.1016/s1359-4311(96)00071-3).
- [78] K. Cornwell, P.A. Kew, Boiling in Small Parallel Channels, *Energy Efficiency in Process Technology* (1993) 624–638. https://doi.org/10.1007/978-94-011-1454-7_56.
- [79] S.G. Kandlikar, Scale effects on flow boiling heat transfer in microchannels: A fundamental perspective, *International Journal of Thermal Sciences* 49 (2010) 1073–1085. <https://doi.org/10.1016/J.IJTHERMALSCI.2009.12.016>.
- [80] C.L. Ong, J.R. Thome, Macro-to-microchannel transition in two-phase flow: Part 1 - Two-phase flow patterns and film thickness measurements, *Exp. Therm. Fluid Sci.* 35 (2011) 37–47. <https://doi.org/10.1016/j.expthermflusci.2010.08.004>.
- [81] C.B. Tibiriçá, G. Ribatski, Flow Boiling Phenomenological Differences Between Micro- and Macroscale Channels, *Heat Transfer Engineering* 36 (2014) 937–942. <https://doi.org/10.1080/01457632.2015.972726>.
- [82] M.M. Mahmoud, T.G. Karayiannis, Flow pattern transition models and correlations for flow boiling in mini-tubes, *Exp. Therm. Fluid Sci.* 70 (2016) 270–282. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2015.09.023>.
- [83] J.R. Thome, A. Cioncolini, Two-Phase Flow Pattern Maps for Microchannels, in: 2015: pp. 47–84. https://doi.org/10.1142/9789814623216_0020.

- [84] J.R. Thome, A. Bar-Cohen, R. Revellin, I. Zun, Unified mechanistic multiscale mapping of two-phase flow patterns in microchannels, *Exp. Therm. Fluid Sci.* 44 (2013) 1–22. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2012.09.012>.
- [85] G.F. Hewitt, “Liquid-Gas Systems,” in: G. Hetsroni (Ed.), *Handbook of Multiphase Systems*, Hemisphere, Washington, D.C. , 1982.
- [86] V.P. Carey, *Liquid Vapour Phase Change Phenomena*, Hemisphere, Washington, D.C., 1992.
- [87] L. Cheng, G. Ribatski, J.R. Thome, Two-Phase Flow Patterns and Flow-Pattern Maps: Fundamentals and Applications, *Appl. Mech. Rev.* 61 (2008) 0508021–05080228. <https://doi.org/10.1115/1.2955990>.
- [88] Y. Taitel, Flow pattern transition in two phase flow, in: 9th International Heat Transfer Conference, Jerusalem, Israel, 1990: pp. 237–254.
- [89] J. Collier, J. Thome, *Convective Boiling and Condensation*, Oxford University Press, New York, 1994.
- [90] G.F. Hewitt, *Gas-Liquid Flow, A-to-Z Guide to Thermodynamics, Heat and Mass Transfer, and Fluids Engineering* (2011). https://doi.org/10.1615/ATOZ.G.GAS-LIQUID_FLOW.
- [91] P.B. Whalley, *Boiling, condensation, and gas-liquid flow*, Clarendon Press, 1987.
- [92] D. Barnea, O. Shoham, Y. Taitel, A.E. Dukler, Flow pattern transition for gas-liquid flow in horizontal and inclined pipes. Comparison of experimental data with theory, *International Journal of Multiphase Flow* 6 (1980) 217–225. [https://doi.org/10.1016/0301-9322\(80\)90012-9](https://doi.org/10.1016/0301-9322(80)90012-9).
- [93] H.S. Han, W.B. Jeong, M.S. Kim, Frequency characteristics of the noise of R600a refrigerant flowing in a pipe with intermittent flow pattern, *International Journal of Refrigeration* 34 (2011) 1497–1506. <https://doi.org/10.1016/J.IJREFRIG.2011.04.004>.
- [94] S. Saini, J. Thaker, J. Banerjee, Transition of Stratified-Wavy Flow to Intermittent Flow Pattern: Nonlinear Analysis of Pressure Fluctuations, *Lecture Notes in Mechanical Engineering* (2021) 35–46. https://doi.org/10.1007/978-981-33-4684-0_5/COVER.
- [95] J.S. Hernandez, C. Valencia, N. Ratkovich, C.F. Torres, F. Muñoz, Data driven methodology for model selection in flow pattern prediction, *Heliyon* 5 (2019) e02718. <https://doi.org/10.1016/J.HELİYON.2019.E02718>.
- [96] A. Arabi, A. Azzi, R. Kadi, A. Al-Sarkhi, B. Hewakandamby, Empirical Modelization of Intermittent Gas/Liquid Flow Hydrodynamic Parameters: The Importance of Distinguishing between Plug and Slug Flows, *SPE Production & Operations* 36 (2021) 703–720. <https://doi.org/10.2118/205481-PA>.

- [97] A. Arabi, Y. Salhi, A. Bouderbail, Y. Zenati, E.K. Si-Ahmed, J. Legrand, Onset of intermittent flow: Visualization of flow structures, *Oil & Gas Science and Technology – Revue d'IFP Energies Nouvelles* 76 (2021) 27. <https://doi.org/10.2516/OGST/2021009>.
- [98] R. Amaya-Gomez, J. López, H. Pineda, D. Urbano-Caguasango, J. Pinilla, N. Ratkovich, F. Muñoz, Probabilistic approach of a flow pattern map for horizontal, vertical, and inclined pipes, *Oil & Gas Science and Technology – Revue d'IFP Energies Nouvelles* 74 (2019) 67. <https://doi.org/10.2516/OGST/2019034>.
- [99] L. Cheng, G. Xia, Flow patterns and flow pattern maps for adiabatic and diabatic gas liquid two phase flow in microchannels: fundamentals, mechanisms and applications, *Exp. Therm. Fluid Sci.* 148 (2023) 110988. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2023.110988>.
- [100] S.A. Tjugum, B.T. Hjertake, G.A. Johansen, Multiphase flow regime identification by multibeam gamma-ray densitometry, *Meas. Sci. Technol.* 13 (2002) 1319. <https://doi.org/10.1088/0957-0233/13/8/321>.
- [101] R. Mosdorf, G. Litak, G. Górski, J. Augustyniak, I. Zaborowska, Classification of two-phase flow patterns using multifractal analysis of pressure oscillations, *Forschung Im Ingenieurwesen/Engineering Research* 85 (2021) 1101–1108. <https://doi.org/10.1007/S10010-021-00549-9/FIGURES/7>.
- [102] N. Kockmann, J. Schuler, B. Oldach, X-Ray-Based Investigations on Multiphase Capillary Flows, *Handbook of Multiphase Flow Science and Technology* (2023) 1145–1181. https://doi.org/10.1007/978-981-287-092-6_29.
- [103] T. (Ted) J. Heindel, X-Ray Flow Visualization: Techniques and Applications, *J. Fluids Eng.* 146 (2024). <https://doi.org/10.1115/1.4064050>.
- [104] N. Takenaka, T. Fujii, K. Akagawa, A. Ono, K. Sonoda, K. Nishizaki, H. Asano, Application of neutron radiography to visualization of multiphase flows, *Flow Measurement and Instrumentation* 1 (1990) 149–156. [https://doi.org/10.1016/0955-5986\(90\)90004-Q](https://doi.org/10.1016/0955-5986(90)90004-Q).
- [105] M. Suo, P. Griffith, Two-Phase Flow in Capillary Tubes, *Journal of Basic Engineering* 86 (1964) 576–582. <https://doi.org/10.1115/1.3653176>.
- [106] J. Frêne, D. Nicolas, B. Degueurce, D. Berthe, M. Godet, Hydrostatic and Hybrid Journal and Thrust Bearings, in: *Lubrification Hydrodynamique- Paliers et Butees*, Editions Eyrolles, Paris, France, 1990: pp. 387–448. [https://doi.org/10.1016/S0167-8922\(97\)80069-7](https://doi.org/10.1016/S0167-8922(97)80069-7).
- [107] J. Biswas, P.F. Greenfield, Two phase flow through vertical capillaries—existence of a stratified flow pattern, *International Journal of Multiphase Flow* 11 (1985) 553–563. [https://doi.org/10.1016/0301-9322\(85\)90075-8](https://doi.org/10.1016/0301-9322(85)90075-8).

- [108] L. Galbiati, P. Andreini, Flow pattern transition for vertical downward two-phase flow in capillary tubes. Inlet mixing effects, *International Communications in Heat and Mass Transfer* 19 (1992) 791–799.
- [109] T. Fukano, A. Kariyasaki, Characteristics of gas-liquid two-phase flow in a capillary tube, *Nuclear Engineering and Design* 141 (1993) 59–68.
- [110] A.B. Barajas, R.L. Panton, Effect of contact angle on two-phase flow in capillary tubes, *Int. J. Multiphase Flow* 19 (1993).
- [111] K. Mishima, T. Hibiki, Some characteristics of air-water two-phase flow in small diameter vertical tubes, *International Journal of Multiphase Flow* 22 (1996) 703–712. [https://doi.org/10.1016/0301-9322\(96\)00010-9](https://doi.org/10.1016/0301-9322(96)00010-9).
- [112] K.A. Triplett, S.M. Ghiaasiaan, S.I. Abdel-Khalik, D.L. Sadowski, Gas-liquid two-phase flow in microchannels part I: Two-phase flow patterns, *International Journal of Multiphase Flow* 25 (1999) 377–394. [https://doi.org/10.1016/S0301-9322\(98\)00054-8](https://doi.org/10.1016/S0301-9322(98)00054-8).
- [113] J.W. Coleman, S. Garimella, Characterization of two-phase flow patterns in small diameter round and rectangular tubes, *Int. J. Heat Mass Transf.* 42 (1999) 2869–2881. [https://doi.org/10.1016/S0017-9310\(98\)00362-7](https://doi.org/10.1016/S0017-9310(98)00362-7).
- [114] C.A. Damianides, J.W. Westwater, Two-phase Flow Patterns in a Compact Heat Exchanger and in Small Tubes, in: 2nd UK Nat. Conf. on Heat Transfer, Glasgow, UK, 1988.
- [115] T.C. Thulasidas, M.A. Abraham, R.L. Cerro, Flow patterns in liquid slugs during bubble-train flow inside capillaries, *Chem. Eng. Sci.* 52 (1997) 2947–2962. [https://doi.org/10.1016/S0009-2509\(97\)00114-0](https://doi.org/10.1016/S0009-2509(97)00114-0).
- [116] L. Cheng, G. Xia, Flow patterns and flow pattern maps for adiabatic and diabatic gas liquid two phase flow in microchannels: fundamentals, mechanisms and applications, (2023). <https://doi.org/10.1016/j.expthermflusci.2023.110988>.
- [117] C.Y. Yang, C.C. Shieh, Flow pattern of air–water and two-phase R-134a in small circular tubes, *International Journal of Multiphase Flow* 27 (2001) 1163–1177. [https://doi.org/10.1016/S0301-9322\(00\)00070-7](https://doi.org/10.1016/S0301-9322(00)00070-7).
- [118] G.E. Alves, Concurrent liquid-gas flow in a pipe-line contactor, Volume 50, Issue 9, Pages 449 - 456 (n.d.).
- [119] F.A. Holland, R. Bragg, Gas–liquid two-phase flow, in: *Fluid Flow for Chemical Engineers*, 2nd ed., Butterworth-Heinemann, 1995: pp. 219–267. <https://doi.org/10.1016/B978-034061058-9.50009-8>.

- [120] J. Pettersen, Flow vaporization of CO₂ in microchannel tubes, *Exp. Therm. Fluid Sci.* 28 (2004) 111–121. [https://doi.org/10.1016/S0894-1777\(03\)00029-3](https://doi.org/10.1016/S0894-1777(03)00029-3).
- [121] L. Chen, Y.S. Tian, T.G. Karayiannis, The effect of tube diameter on vertical two-phase flow regimes in small tubes, *Int. J. Heat Mass Transf.* 49 (2006) 4220–4230. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2006.03.025>.
- [122] J. Kren, B. Zajec, I. Tiselj, S. El Shawish, Ž. Perne, M. Tekavčič, B. Mikuž, Dynamics of Taylor bubble interface in vertical turbulent counter-current flow, *International Journal of Multiphase Flow* 165 (2023) 104482. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2023.104482>.
- [123] R. Revellin, V. Dupont, T. Ursenbacher, J.R. Thome, I. Zun, Characterization of diabatic two-phase flows in microchannels: Flow parameter results for R-134a in a 0.5 mm channel, *International Journal of Multiphase Flow* 32 (2006) 755–774. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2006.02.016>.
- [124] E. Rahim, R. Revellin, J. Thome, A. Bar-Cohen, Characterization and prediction of two-phase flow regimes in miniature tubes, *International Journal of Multiphase Flow* 37 (2011) 12–23. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2010.09.002>.
- [125] R. Revellin, J.R. Thome, A new type of diabatic flow pattern map for boiling heat transfer in microchannels, in: *Journal of Micromechanics and Microengineering*, IOP Publishing, 2007: pp. 788–796. <https://doi.org/10.1088/0960-1317/17/4/016>.
- [126] T. Alam, W. Li, W. Chang, F. Yang, J. Khan, C. Li, A comparative study of flow boiling HFE-7100 in silicon nanowire and plainwall microchannels, *Int. J. Heat Mass Transf.* 124 (2018) 829–840. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2018.04.010>.
- [127] A. Al-Zaidi, *Flow Boiling and Condensation with Refrigerant HFE-7100 for Cooling High Heat Flux Devices*, Brunel University London, 2019.
- [128] A.H. Al-Zaidi, M.M. Mahmoud, T.G. Karayiannis, Flow boiling in copper and aluminium microchannels, *Int. J. Heat Mass Transf.* 194 (2022). <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2022.123101>.
- [129] V.Y.S. Lee, G. Henderson, A. Reip, T.G. Karayiannis, Flow boiling characteristics in plain and porous coated microchannel heat sinks, *Int. J. Heat Mass Transf.* 183 (2022) 122152. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2021.122152>.
- [130] Y.F. Li, G.D. Xia, D.D. Ma, J.L. Yang, W. Li, Experimental investigation of flow boiling characteristics in microchannel with triangular cavities and rectangular fins, *Int. J. Heat Mass Transf.* 148 (2020) 119036. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2019.119036>.

- [131] S. Paul, M. Fernandino, C.A. Dorao, On the scaling of convective boiling heat transfer coefficient, *Int. J. Heat Mass Transf.* 164 (2021) 120589. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2020.120589>.
- [132] S.G. Kandlikar, A General Correlation for Saturated Two-Phase Flow Boiling Heat Transfer Inside Horizontal and Vertical Tubes, *J. Heat Transfer* 112 (1990) 219–228. <https://doi.org/10.1115/1.2910348>.
- [133] H.Y. Wu, P. Cheng, Visualization and measurements of periodic boiling in silicon microchannels, *Int. J. Heat Mass Transf.* 46 (2003) 2603–2614. [https://doi.org/10.1016/S0017-9310\(03\)00039-5](https://doi.org/10.1016/S0017-9310(03)00039-5).
- [134] P. Balasubramanian, S.G. Kandlikar, Experimental Study of Flow Patterns, Pressure Drop, and Flow Instabilities in Parallel Rectangular Minichannels, *Heat Transfer Engineering* 26 (2005) 20–27. <https://doi.org/10.1080/01457630590907167>.
- [135] T. Chen, S. V. Garimella, Measurements and high-speed visualizations of flow boiling of a dielectric fluid in a silicon microchannel heat sink, *International Journal of Multiphase Flow* 32 (2006) 957–971. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2006.03.002>.
- [136] C. Huh, J. Kim, M.H. Kim, Flow pattern transition instability during flow boiling in a single microchannel, *Int. J. Heat Mass Transf.* 50 (2007) 1049–1060. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2006.07.027>.
- [137] C. Huh, M. Hwan Kim, Pressure Drop, Boiling Heat Transfer and Flow Patterns during Flow Boiling in a Single Microchannel, (2010). <https://doi.org/10.1080/01457630701328213>.
- [138] C. Huh, M.H. Kim, An experimental investigation of flow boiling in an asymmetrically heated rectangular microchannel, *Exp. Therm. Fluid Sci.* 30 (2006) 775–784. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2006.03.007>.
- [139] E. Sobierska, R. Kulenovic, R. Mertz, M. Groll, Experimental results of flow boiling of water in a vertical microchannel, *Exp. Therm. Fluid Sci.* 31 (2006) 111–119. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2006.03.022>.
- [140] K.H. Chang, C. Pan, Two-phase flow instability for boiling in a microchannel heat sink, *Int. J. Heat Mass Transf.* 50 (2007) 2078–2088. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2006.11.014>.
- [141] G. Wang, P. Cheng, H. Wu, Unstable and stable flow boiling in parallel microchannels and in a single microchannel, *Int. J. Heat Mass Transf.* 50 (2007) 4297–4310. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2007.01.033>.

- [142] E. Sobierska, R. Kulenovic, R. Mertz, Heat transfer mechanism and flow pattern during flow boiling of water in a vertical narrow channel—experimental results, *International Journal of Thermal Sciences* 46 (2007) 1172–1181. <https://doi.org/10.1016/J.IJTHERMALSCI.2007.06.011>.
- [143] T. Harirchian, S. V. Garimella, The critical role of channel cross-sectional area in microchannel flow boiling heat transfer, *International Journal of Multiphase Flow* 35 (2009) 904–913. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2009.06.005>.
- [144] G.P. Celata, S.K. Saha, G. Zummo, D. Dossevi, Heat transfer characteristics of flow boiling in a single horizontal microchannel, *International Journal of Thermal Sciences* 49 (2010) 1086–1094. <https://doi.org/10.1016/J.IJTHERMALSCI.2010.01.019>.
- [145] E. Galvis, R. Culham, Measurements and flow pattern visualizations of two-phase flow boiling in single channel microevaporators, *International Journal of Multiphase Flow* 42 (2012) 52–61. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2012.01.009>.
- [146] G.P. Celata, M. Cumo, D. Dossevi, R.T.M. Jilisen, S.K. Saha, G. Zummo, Flow pattern analysis of flow boiling inside a 0.48 mm microtube, *International Journal of Thermal Sciences* 58 (2012) 1–8. <https://doi.org/10.1016/J.IJTHERMALSCI.2012.03.014>.
- [147] C.B. Tibirić, G. Ribatski, Flow patterns and bubble departure fundamental characteristics during flow boiling in microscale channels, *Exp. Therm. Fluid Sci.* 59 (2014) 152–165. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2014.02.017>.
- [148] T.W. Lim, S.S. You, J.H. Choi, H.S. Kim, Experimental investigation of heat transfer in two-phase flow boiling, *Experimental Heat Transfer* 28 (2015) 23–36. <https://doi.org/10.1080/08916152.2013.803173>.
- [149] P. Thiangtham, C. Keepaiboon, P. Kiatpachai, L.G. Asirvatham, O. Mahian, A.S. Dalkilic, S. Wongwises, An experimental study on two-phase flow patterns and heat transfer characteristics during boiling of R134a flowing through a multi-microchannel heat sink, *Int. J. Heat Mass Transf.* 98 (2016) 390–400. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2016.02.051>.
- [150] L. Yin, R. Xu, P. Jiang, H. Cai, L. Jia, Subcooled flow boiling of water in a large aspect ratio microchannel, *Int. J. Heat Mass Transf.* 112 (2017) 1081–1089. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2017.05.028>.
- [151] Y.S. Choi, T.W. Lim, S.S. You, H.S. Kim, Two-phase flow boiling heat transfer of FC-72 in parallel micro-channels, *Experimental Heat Transfer* 30 (2017) 284–301. <https://doi.org/10.1080/08916152.2016.1238858>.

- [152] E.M. Fayyadh, M.M. Mahmoud, K. Sefiane, T.G. Karayiannis, Flow boiling heat transfer of R134a in multi microchannels, *Int. J. Heat Mass Transf.* 110 (2017) 422–436. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2017.03.057>.
- [153] Y.W. Kuang, W. Wang, J.Y. Miao, X.G. Yu, H.X. Zhang, R. Zhuan, Flow boiling of ammonia and flow instabilities in mini-channels, *Appl. Therm. Eng.* 113 (2017) 831–842. <https://doi.org/10.1016/J.APPLTHERMALENG.2016.11.093>.
- [154] A.H. Al-Zaidi, M.M. Mahmoud, T.G. Karayiannis, Flow boiling of HFE-7100 in microchannels: Experimental study and comparison with correlations, *Int. J. Heat Mass Transf.* 140 (2019) 100–128. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2019.05.095>.
- [155] S.A. Mohammed, E.M. Fayyadh, Experimental Investigation of Heat Transfer and Flow Characteristics in Different Inlet Subcooled Flow Boiling in Microchannel, in: *IOP Conf. Ser. Mater. Sci. Eng.*, Institute of Physics Publishing, 2020: p. 012023. <https://doi.org/10.1088/1757-899X/671/1/012023>.
- [156] A.M. Sahar, M.R. Özdemir, E.M. Fayyadh, J. Wissink, M.M. Mahmoud, T.G. Karayiannis, Single phase flow pressure drop and heat transfer in rectangular metallic microchannels, *Appl. Therm. Eng.* 93 (2016) 1324–1336. <https://doi.org/10.1016/J.APPLTHERMALENG.2015.08.087>.
- [157] K. Sakalkale, R. Majumdar, S.K. Saha, An Investigation of Two-Phase Flow Regimes for Microchannels Based on Void Fraction, *IEEE Trans. Compon. Packaging Manuf. Technol.* 9 (2019) 2189–2199. <https://doi.org/10.1109/TCPMT.2019.2918925>.
- [158] Y. Lv, G. Xia, L. Cheng, D. Ma, Experimental study on the pressure drop oscillation characteristics of the flow boiling instability with FC-72 in parallel rectangle microchannels, *International Communications in Heat and Mass Transfer* 108 (2019). <https://doi.org/10.1016/J.ICHEATMASSTRANSFER.2019.104289>.
- [159] R. Ajith Krishnan, K. Balasubramanian, S. Suresh, An experimental investigation on the effect of gravitational orientation on flow boiling performance in different channel sizes ranges from minichannels to microchannels, *Heat and Mass Transfer/Waerme- Und Stoffuebertragung* 56 (2020) 1391–1420. <https://doi.org/10.1007/S00231-019-02802-Y>.
- [160] M. Li, Y. Luo, Y. Jiang, W. Wei, M. Wang, Experimental research on flow and heat transfer in microchannel with refrigerant hfo1234yf, *J. Thermophys. Heat Trans.* 35 (2021) 225–233. <https://doi.org/10.2514/1.T6064>.
- [161] M.R. Özdemir, M.M. Mahmoud, T.G. Karayiannis, Flow Boiling of Water in a Rectangular Metallic Microchannel, *Heat Transfer Engineering* 42 (2021) 492–516. <https://doi.org/10.1080/01457632.2019.1707390>.

- [162] S. Zhou, B. Shu, Z. Yu, Y. Huang, Y. Zhang, Experimental study and mechanism analysis of the flow boiling and heat transfer characteristics in microchannels with different surface wettability, *Micromachines* (Basel). 12 (2021). <https://doi.org/10.3390/MI12080881>.
- [163] V.Y.S. Lee, A. Al-Zaidi, G. Henderson, T.G. Karayiannis, Flow Boiling Results of HFE-7200 in a Multi-Microchannel Evaporator and Comparison with HFE-7100, *Proceedings of the 4th World Congress on Momentum, Heat and Mass Transfer* (2019). <https://doi.org/10.11159/ICMFHT19.120>.
- [164] H. Nalbandian, C.Y. Yang, K.T. Chen, Flow Boiling Heat Transfer of Refrigerants HFO-1234yf and HFC-134a in an Extruded Aluminum Tube with Multi-Port microchannels, *International Journal of Refrigeration* 142 (2022) 37–47. <https://doi.org/10.1016/J.IJREFRIG.2022.06.013>.
- [165] Y. Zhang, H. Wu, L. Zhang, Y. Yang, X. Niu, Z. Zeng, B. Shu, Flow Pattern Study and Pressure Drop Prediction of Two-Phase Boiling Process in Different Surface Wettability Microchannel, *Micromachines* 2023, Vol. 14, Page 958 14 (2023) 958. <https://doi.org/10.3390/MI14050958>.
- [166] G. Wang, P. Cheng, An experimental study of flow boiling instability in a single microchannel, *International Communications in Heat and Mass Transfer* 35 (2008) 1229–1234. <https://doi.org/10.1016/J.ICHEATMASSTRANSFER.2008.07.019>.
- [167] J. Lee, I. Mudawar, Low-temperature two-phase microchannel cooling for high-heat-flux thermal management of defense electronics, *IEEE Transactions on Components and Packaging Technologies* 32 (2009) 453–465. <https://doi.org/10.1109/TCAPT.2008.2005783>.
- [168] D. Wang, D. Wang, F. Hong, C. Zhang, J. Xu, Improved flow boiling performance and temperature uniformity in counter-flow interconnected microchannel heat sink, *Appl. Therm. Eng.* 241 (2024) 122370. <https://doi.org/10.1016/J.APPLTHERMALENG.2024.122370>.
- [169] C. Martín-Callizo, B. Palm, W. Owhaib, R. Ali, Flow boiling visualization of R-134a in a vertical channel of small diameter, *J. Heat Transfer* 132 (2010) 1–8. <https://doi.org/10.1115/1.4000012/467866>.
- [170] R. Ali, B. Palm, C. Martin-Callizo, M.H. Maqbool, Study of flow boiling characteristics of a microchannel using high speed visualization, *Journal of Dynamic Systems, Measurement and Control, Transactions of the ASME* 135 (2013). <https://doi.org/10.1115/1.4023879>.
- [171] E. Costa-Patry, J. Olivier, J.R. Thome, Heat transfer characteristics in a copper micro-evaporator and flowpattern-based prediction method for flowboiling in microchannels, *Frontiers in Heat and Mass Transfer* 3 (2012). <https://doi.org/10.5098/HMT.V3.1.3002>.

- [172] W. Li, Z. Chen, J. Li, K. Zhou, Z. Feng, Effects of heat flux, mass flux, inlet fluid subcooling, and channel orientation on subcooled flow boiling in a high-aspect-ratio rectangular microchannel, *J. Heat Transfer* 142 (2020). <https://doi.org/10.1115/1.4046969>.
- [173] Y. Lin, J. Li, J. Sun, W. Li, Y. Cao, Flow Pattern Analysis and Heat Transfer Characteristics During Subcooled Flow Boiling in a Rectangular Microchannel on ZnO Microrod Surface, *J. Heat Transfer* 143 (2021). <https://doi.org/10.1115/1.4052434/1120399>.
- [174] H. Yu, J. Zhuang, T. Li, W. Li, T. He, N. Mao, Influence of transient heat flux on boiling flow pattern in a straight microchannel applied in concentrator photovoltaic systems, *Int. J. Heat Mass Transf.* 190 (2022). <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2022.122792>.
- [175] T.Y. Liu, P.L. Li, C.W. Liu, C. Gau, Boiling flow characteristics in microchannels with very hydrophobic surface to super-hydrophilic surface, *Int. J. Heat Mass Transf.* 54 (2011) 126–134. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2010.09.060>.
- [176] M. Lee, Y.Y. Wong, M. Wong, Y. Zohar, Size and shape effects on two-phase flow patterns in microchannel forced convection boiling, *Journal of Micromechanics and Microengineering* 13 (2002) 155. <https://doi.org/10.1088/0960-1317/13/1/322>.
- [177] J. Xu, Y. Gan, D. Zhang, X. Li, Microscale boiling heat transfer in a micro-timescale at high heat fluxes, *Journal of Micromechanics and Microengineering* 15 (2004) 362. <https://doi.org/10.1088/0960-1317/15/2/017>.
- [178] J. Xu, S. Shen, Y. Gan, Y. Li, W. Zhang, Q. Su, Transient flow pattern based microscale boiling heat transfer mechanisms, *Journal of Micromechanics and Microengineering* 15 (2005) 1344. <https://doi.org/10.1088/0960-1317/15/6/028>.
- [179] K. Zhou, C. Coyle, J. Li, J. Buongiorno, W. Li, Flow boiling in vertical narrow microchannels of different surface wettability characteristics, *Int. J. Heat Mass Transf.* 109 (2017) 103–114. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2017.01.111>.
- [180] H. Li, P. Hrnjak, Heat transfer coefficient, pressure gradient, and flow patterns of R1233zd(E) and R1336mzz(Z) evaporating in a microchannel tube, *Int. J. Heat Mass Transf.* 182 (2022) 121992. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2021.121992>.
- [181] H. Li, P. Hrnjak, Flow visualization of R32 in parallel-port microchannel tube, *Int. J. Heat Mass Transf.* 128 (2019) 1–11. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2018.08.120>.
- [182] K. Zhou, H. Zhu, W. Li, J. Li, K. Sheng, S. Shao, H. Li, Z. Tao, Heat transfer characteristics and flow pattern visualization for flow boiling in a vertical narrow microchannel, *Journal of Electronic Packaging, Transactions of the ASME* 141 (2019). <https://doi.org/10.1115/1.4043476>.

- [183] M. Habibi Matin, S. Moghaddam, Mechanism of transition from elongated bubbles to wavy-annular regime in flow boiling through microchannels, *Int. J. Heat Mass Transf.* 176 (2021) 121464. <https://doi.org/10.1016/j.ijheatmasstransfer.2021.121464>.
- [184] A. Widyatama, M. Venter, D. Orejon, K. Sefiane, Experimental investigation of bubble dynamics and flow patterns during flow boiling in high aspect ratio microchannels with the effect of flow orientation, *International Journal of Thermal Sciences* 189 (2023) 108238. <https://doi.org/10.1016/J.IJTHERMALSCI.2023.108238>.
- [185] L.X. Yang, A. Guo, D. Liu, Experimental Investigation of Subcooled Vertical Upward Flow Boiling in a Narrow Rectangular Channel, *Experimental Heat Transfer* 29 (2016) 221–243. <https://doi.org/10.1080/08916152.2014.973978>.
- [186] C.B. Tibiriçá, D.M. Rocha, I.L.S. Sueth, G. Bochio, G.K.K. Shimizu, M.C. Barbosa, S. dos S. Ferreira, A complete set of simple and optimized correlations for microchannel flow boiling and two-phase flow applications, *Appl. Therm. Eng.* 126 (2017) 774–795. <https://doi.org/10.1016/j.applthermaleng.2017.07.161>.
- [187] J.L. Gasche, Carbon dioxide evaporation in a single microchannel, *Journal of the Brazilian Society of Mechanical Sciences and Engineering* 28 (2006) 69–83. <https://doi.org/10.1590/S1678-58782006000100008>.
- [188] L. Yin, L. Jia, Confined bubble growth and heat transfer characteristics during flow boiling in microchannel, *Int. J. Heat Mass Transf.* 98 (2016) 114–123. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2016.02.063>.
- [189] S.T. Kadam, R. Kumar, Twenty first century cooling solution: Microchannel heat sinks, (2014). <https://doi.org/10.1016/j.ijthermalsci.2014.06.013>.
- [190] S. Kang, H. Kim, D. Kim, Mitigation of pressure fluctuation in two-phase slug flow by rib geometry, *International Journal of Multiphase Flow* 143 (2021) 103753. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2021.103753>.
- [191] J.R. Thome, V. Dupont, A.M. Jacobi, Heat transfer model for evaporation in microchannels. Part I: presentation of the model, *Int. J. Heat Mass Transf.* 47 (2004) 3375–3385. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2004.01.006>.
- [192] N. Borhani, B. Agostini, J.R. Thome, A novel time strip flow visualisation technique for investigation of intermittent dewetting and dryout in elongated bubble flow in a microchannel evaporator, *Int. J. Heat Mass Transf.* 53 (2010) 4809–4818. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2010.06.011>.

- [193] Y. Han, N. Shikazono, Measurement of the liquid film thickness in micro tube slug flow, *Int. J. Heat Fluid Flow* 30 (2009) 842–853. <https://doi.org/10.1016/J.IJHEATFLUIDFLOW.2009.02.019>.
- [194] L. Consolini, J.R. Thome, Micro-channel flow boiling heat transfer of R-134a, R-236fa, and R-245fa, *Microfluid. Nanofluidics* 6 (2009) 731–746. <https://doi.org/10.1007/S10404-008-0348-7/FIGURES/21>.
- [195] R.S. Patel, J.A. Weibel, S. V. Garimella, Mechanistic modeling of the liquid film shape and heat transfer coefficient in annular-regime microchannel flow boiling, *Int. J. Heat Mass Transf.* 114 (2017) 841–851. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2017.06.121>.
- [196] H. Li, P. Hrnjak, A mechanistic model in annular flow in microchannel tube for predicting heat transfer coefficient and pressure gradient, *Int. J. Heat Mass Transf.* 203 (2023) 123805. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2022.123805>.
- [197] N. Borhani, J.R. Thome, Intermittent dewetting and dryout of annular flows, *International Journal of Multiphase Flow* 67 (2014) 144–152. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2014.04.009>.
- [198] L. Chen, Flow patterns in upward two-phase flow in small diameter tubes, Brunel University London, 2006.
- [199] T. Karayiannis, E. Pike-Wilson, L. Chen, M. Mahmoud, Y. Tian, Flow Patterns and Comparison with Correlations for Vertical Flow Boiling of R 245 fa in Small to Micro Tubes, (2014).
- [200] M. Kaichiro, M. Ishii, Flow regime transition criteria for upward two-phase flow in vertical tubes, *Int. J. Heat Mass Transf.* 27 (1984) 723–737. [https://doi.org/10.1016/0017-9310\(84\)90142-X](https://doi.org/10.1016/0017-9310(84)90142-X).
- [201] S. Jayanti, G.F. Hewitt, Prediction of the slug-to-churn flow transition in vertical two-phase flow, *International Journal of Multiphase Flow* 18 (1992) 847–860. [https://doi.org/10.1016/0301-9322\(92\)90063-M](https://doi.org/10.1016/0301-9322(92)90063-M).
- [202] T. Harirchian, S. V. Garimella, A comprehensive flow regime map for microchannel flow boiling with quantitative transition criteria, *Int. J. Heat Mass Transf.* 53 (2010) 2694–2702. <https://doi.org/10.1016/j.ijheatmasstransfer.2010.02.039>.
- [203] T. Harirchian, S. V. Garimella, Flow regime-based modeling of heat transfer and pressure drop in microchannel flow boiling, *Int. J. Heat Mass Transf.* 55 (2012) 1246–1260. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2011.09.024>.

- [204] P. Saha, M. Ishii, N. Zuber, An Experimental Investigation of the Thermally Induced Flow Oscillations in Two-Phase Systems, *J. Heat Transfer* 98 (1976) 616–622. <https://doi.org/10.1115/1.3450609>.
- [205] D. Jige, S. Kikuchi, H. Eda, N. Inoue, S. Koyama, Two-phase flow characteristics of R32 in horizontal multiport minichannels: Flow visualization and development of flow regime map, *International Journal of Refrigeration* 95 (2018) 156–164. <https://doi.org/10.1016/J.IJREFRIG.2018.09.005>.
- [206] T.G. Karayiannis, M.M. Mahmoud, Flow boiling in microchannels: Fundamentals and applications, *Appl. Therm. Eng.* 115 (2017) 1372–1397. <https://doi.org/10.1016/j.applthermaleng.2016.08.063>.
- [207] S. In, S. Jeong, Flow boiling heat transfer characteristics of R123 and R134a in a micro-channel, *International Journal of Multiphase Flow* 35 (2009) 987–1000. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2009.07.003>.
- [208] C.L. Ong, J.R. Thome, Macro-to-microchannel transition in two-phase flow: Part 2 – Flow boiling heat transfer and critical heat flux, *Exp. Therm. Fluid Sci.* 35 (2011) 873–886. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2010.12.003>.
- [209] S.M. Kim, I. Mudawar, Universal approach to predicting saturated flow boiling heat transfer in mini/micro-channels – Part I. Dryout incipience quality, *Int. J. Heat Mass Transf.* 64 (2013) 1226–1238. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2013.04.016>.
- [210] Z. Zhang, G. Zhang, Y. Zhang, M. Tian, Experimental study on the effect of hydraulic diameter on the flow boiling characteristics in microchannels, *Int. J. Heat Mass Transf.* 241 (2025) 126736. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2025.126736>.
- [211] T.C.B. Roldão, C.B. Tibiriçá, Flow boiling pressure drop and heat transfer at high mass fluxes for R514a in a single microchannel, *Appl. Therm. Eng.* 254 (2024) 123807. <https://doi.org/10.1016/J.APPLTHERMALENG.2024.123807>.
- [212] T.C.B. Roldão, C.B. Tibiriçá, Critical heat flux at high mass velocities in a microchannel and surface tension measurements for R514a, *Int. J. Heat Mass Transf.* 227 (2024) 125575. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2024.125575>.
- [213] D.B. Marchetto, R. Revellin, R. Rullière, G. Ribatski, An experimental investigation on R245fa and R1233zd(E) flow boiling at high saturation temperatures in a horizontal small diameter channel, *Int. J. Heat Mass Transf.* 220 (2024) 124986. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2023.124986>.

- [214] B. Markal, O. Aydin, M. Avci, An experimental investigation of saturated flow boiling heat transfer and pressure drop in square microchannels, *International Journal of Refrigeration* 65 (2016) 1–11. <https://doi.org/10.1016/J.IJREFRIG.2015.12.013>.
- [215] D. Deng, W. Wan, Y. Qin, J. Zhang, X. Chu, Flow boiling enhancement of structured microchannels with micro pin fins, *Int. J. Heat Mass Transf.* 105 (2017) 338–349. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2016.09.086>.
- [216] C.T. Lu, C. Pan, Convective boiling in a parallel microchannel heat sink with a diverging cross section and artificial nucleation sites, *Exp. Therm. Fluid Sci.* 35 (2011) 810–815. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2010.08.018>.
- [217] P.S. Lee, S. V. Garimella, Saturated flow boiling heat transfer and pressure drop in silicon microchannel arrays, *Int. J. Heat Mass Transf.* 51 (2008) 789–806. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2007.04.019>.
- [218] B.J. Jones, S. V. Garimella, Surface roughness effects on flow boiling in microchannels, *J. Therm. Sci. Eng. Appl.* 1 (2009) 1–9. <https://doi.org/10.1115/1.4001804/469344>.
- [219] Y. Sun, Z. Zhang, G. Zhang, Y. Jiang, J. Zheng, Effect of surface wettability on bubble dynamics and heat transfer in microchannel flow boiling, *Int. J. Heat Mass Transf.* 241 (2025) 126729. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2025.126729>.
- [220] E. Costa-Patry, J. Olivier, B. Michel, J.R. Thome, Two-phase flow of refrigerants in 85 μm -wide multi-microchannels: Part II – Heat transfer with 35 local heaters, *Int. J. Heat Fluid Flow* 32 (2011) 464–476. <https://doi.org/10.1016/J.IJHEATFLUIDFLOW.2011.01.006>.
- [221] W. Qu, I. Mudawar, Flow boiling heat transfer in two-phase micro-channel heat sinks—I. Experimental investigation and assessment of correlation methods, *Int. J. Heat Mass Transf.* 46 (2003) 2755–2771. [https://doi.org/10.1016/S0017-9310\(03\)00041-3](https://doi.org/10.1016/S0017-9310(03)00041-3).
- [222] B. Agostini, J.R. Thome, M. Fabbri, B. Michel, D. Calmi, U. Kloter, High heat flux flow boiling in silicon multi-microchannels – Part I: Heat transfer characteristics of refrigerant R236fa, *Int. J. Heat Mass Transf.* 51 (2008) 5400–5414. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2008.03.006>.
- [223] E.J. Choi, J.Y. Park, M.S. Kim, Two-phase cooling using HFE-7100 for polymer electrolyte membrane fuel cell application, *Appl. Therm. Eng.* 148 (2019) 868–877. <https://doi.org/10.1016/J.APPLTHERMALENG.2018.11.103>.
- [224] H. Yu, Q. Xu, X. Tang, C. Pei, L. Guo, Experimental study on flow boiling heat transfer in manifold microchannel heat sinks with different numbers and widths of manifolds, *Int. J. Heat*

- Mass Transf. 244 (2025) 126925. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2025.126925>.
- [225] C.K. Patra, A. Bhattacharya, P.K. Das, Heat transfer characteristics of acetone and methanol flow boiling in parallel microchannel heat sink: A parametric study and assessment of predictive correlations, *Appl. Therm. Eng.* 270 (2025) 126217. <https://doi.org/10.1016/J.APPLTHERMALENG.2025.126217>.
- [226] A. Widyatama, G. Duursma, D. Orejon, K. Sefiane, Flow boiling characteristics of HFE-7000 in high aspect ratio microchannels with the effect of flow orientation, *Appl. Therm. Eng.* 257 (2024) 124094. <https://doi.org/10.1016/J.APPLTHERMALENG.2024.124094>.
- [227] Z. Zhang, L. Jia, C. Dang, Y. Ding, Experimental study on flow boiling characteristics of R134a inside high-heat-flux microchannels, *Int. J. Heat Fluid Flow* 107 (2024) 109372. <https://doi.org/10.1016/J.IJHEATFLUIDFLOW.2024.109372>.
- [228] B. Sumith, F. Kaminaga, K. Matsumura, Saturated flow boiling of water in a vertical small diameter tube, *Exp. Therm. Fluid Sci.* 27 (2003) 789–801. [https://doi.org/10.1016/S0894-1777\(02\)00317-5](https://doi.org/10.1016/S0894-1777(02)00317-5).
- [229] H.J. Lee, S.Y. Lee, Heat transfer correlation for boiling flows in small rectangular horizontal channels with low aspect ratios, *International Journal of Multiphase Flow* 27 (2001) 2043–2062. [https://doi.org/10.1016/S0301-9322\(01\)00054-4](https://doi.org/10.1016/S0301-9322(01)00054-4).
- [230] T. Harirchian, S. V. Garimella, Microchannel size effects on local flow boiling heat transfer to a dielectric fluid, *Int. J. Heat Mass Transf.* 51 (2008) 3724–3735. <https://doi.org/10.1016/j.ijheatmasstransfer.2008.03.013>.
- [231] H. Huang, N. Borhani, J.R. Thome, Experimental investigation on flow boiling pressure drop and heat transfer of R1233zd(E) in a multi-microchannel evaporator, *Int. J. Heat Mass Transf.* 98 (2016) 596–610. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2016.03.051>.
- [232] X. Li, L. Jia, The investigation on flow boiling heat transfer of R134a in micro-channels, *Journal of Thermal Science* 24 (2015) 452–462. <https://doi.org/10.1007/S11630-015-0808-9/METRICS>.
- [233] G.M. Lazarek, S.H. Black, Evaporative heat transfer, pressure drop and critical heat flux in a small vertical tube with R-113, *Int. J. Heat Mass Transf.* 25 (1982) 945–960. [https://doi.org/10.1016/0017-9310\(82\)90070-9](https://doi.org/10.1016/0017-9310(82)90070-9).
- [234] Z.Y. Bao, D.F. Fletcher, B.S. Haynes, Flow boiling heat transfer of Freon R11 and HCFC123 in narrow passages, *Int. J. Heat Mass Transf.* 43 (2000) 3347–3358. [https://doi.org/10.1016/S0017-9310\(99\)00379-8](https://doi.org/10.1016/S0017-9310(99)00379-8).

- [235] B. Agostini, J.R. Thome, M. Fabbri, B. Michel, D. Calmi, U. Kloter, High heat flux flow boiling in silicon multi-microchannels – Part II: Heat transfer characteristics of refrigerant R245fa, *Int. J. Heat Mass Transf.* 51 (2008) 5415–5425. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2008.03.007>.
- [236] W. Owhaib, C. Martín-Callizo, B. Palm, Evaporative heat transfer in vertical circular microchannels, *Appl. Therm. Eng.* 24 (2004) 1241–1253. <https://doi.org/10.1016/J.APPLTHERMALENG.2003.12.030>.
- [237] R. Ali, B. Palm, M.H. Maqbool, Flow boiling heat transfer characteristics of a minichannel up to dryout conditiona, *J. Heat Transfer* 133 (2011). <https://doi.org/10.1115/1.4003669/467322>.
- [238] S. Basu, S. Ndao, G.J. Michna, Y. Peles, M.K. Jensen, Flow boiling of R134a in circular microtubes - Part I: Study of heat transfer characteristics, *J. Heat Transfer* 133 (2011). <https://doi.org/10.1115/1.4003159/468043>.
- [239] S.S. Bertsch, E.A. Groll, S. V. Garimella, Effects of heat flux, mass flux, vapor quality, and saturation temperature on flow boiling heat transfer in microchannels, *International Journal of Multiphase Flow* 35 (2009) 142–154. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2008.10.004>.
- [240] Y. Xu, X. Fang, D. Li, G. Li, Y. Yuan, A. Xu, An experimental study of flow boiling frictional pressure drop of R134a and evaluation of existing correlations, *Int. J. Heat Mass Transf. C* (2016) 150–163. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2016.03.018>.
- [241] V.Y.S. Lee, T.G. Karayiannis, Influence of system pressure on flow boiling in microchannels, *Int. J. Heat Mass Transf.* 215 (2023) 124470. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2023.124470>.
- [242] A.H. Al-Zaidi, M.M. Mahmoud, A. Ivanov, T.G. Karayiannis, Bubble nucleation site density, generation frequency and departure diameter in flow boiling of HFE-7100, *Int. J. Heat Mass Transf.* 242 (2025) 126830. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2025.126830>.
- [243] Y.Y. Hsu, On the Size Range of Active Nucleation Cavities on a Heating Surface, *J. Heat Transfer* 84 (1962) 207–213. <https://doi.org/10.1115/1.3684339>.
- [244] W.M. Rohsenow, Pool Boiling , in: G. Hestroni (Ed.), *Handbook of Multiphase Systems*, McGraw-Hill Hemisphere, New York, USA, 1982.
- [245] S.G. Bankoff, Entrapment of gas in the spreading of a liquid over a rough surface, *AIChE Journal* 4 (1958) 24–26. <https://doi.org/10.1002/AIC.690040105>.
- [246] J.J. Lorenz, The effects of surface conditions on boiling characteristics, PhD Thesis, Massachusetts Institute of Technology, 1972.

- [247] W. Tong, A. Bar-Cohen, T.W. Simon, S.M. You, Contact angle effects on boiling incipience of highly-wetting liquids, *Int. J. Heat Mass Transf.* 33 (1990) 91–103. [https://doi.org/10.1016/0017-9310\(90\)90144-J](https://doi.org/10.1016/0017-9310(90)90144-J).
- [248] C.H. Wang, V.K. Dhir, On the Gas Entrapment and Nucleation Site Density During Pool Boiling of Saturated Water, *J. Heat Transfer* 115 (1993) 670–679. <https://doi.org/10.1115/1.2910738>.
- [249] R.H.S. Winterton, Nucleation of boiling and cavitation, *J. Phys. D Appl. Phys.* 10 (1977) 2041. <https://doi.org/10.1088/0022-3727/10/15/008>.
- [250] S. Ming-Heng, J. Ma, W. Bu-Xuan, Analysis on hysteresis in nucleate pool boiling heat transfer, *Int. J. Heat Mass Transf.* 36 (1993) 4461–4466. [https://doi.org/10.1016/0017-9310\(93\)90130-X](https://doi.org/10.1016/0017-9310(93)90130-X).
- [251] D. EUH, B. OZAR, T. HIBIKI, M. ISHII, C.-H. SONG, Characteristics of Bubble Departure Frequency in a Low-Pressure Subcooled Boiling Flow, *J. Nucl. Sci. Technol.* 47 (2010) 608–617. <https://doi.org/10.1080/18811248.2010.9720958>.
- [252] R.M. Podowski, D.A. Drew, R.T.Jr. Lahey, M.Z. Podowski, A mechanistic model of the ebullition cycle in forced convection subcooled boiling, (1997) 1535–1542.
- [253] R. Situ, M. Ishii, T. Hibiki, J.Y. Tu, G.H. Yeoh, M. Mori, Bubble departure frequency in forced convective subcooled boiling flow, *Int. J. Heat Mass Transf.* 51 (2008) 6268–6282. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2008.04.028>.
- [254] J.F. Klausner, R. Mei, D.M. Bernhard, L.Z. Zeng, Vapor bubble departure in forced convection boiling, *Int. J. Heat Mass Transf.* 36 (1993) 651–662. [https://doi.org/10.1016/0017-9310\(93\)80041-R](https://doi.org/10.1016/0017-9310(93)80041-R).
- [255] L. Zhang, C. Yang, Z.S. Mao, Unsteady motion of a single bubble in highly viscous liquid and empirical correlation of drag coefficient, *Chem. Eng. Sci.* 63 (2008) 2099–2106. <https://doi.org/10.1016/J.CES.2008.01.010>.
- [256] G.L. Chahine, B.E. Schmidt, X. Deng, C.T. Hsiao, Q. Liu, Bubble dynamics in a pressure gradient with reentrant jet break through and energy loss, *Ultrason. Sonochem.* 99 (2023) 106541. <https://doi.org/10.1016/J.ULTSONCH.2023.106541>.
- [257] E. Loth, Quasi-steady shape and drag of deformable bubbles and drops, *International Journal of Multiphase Flow* 34 (2008) 523–546. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2007.08.010>.
- [258] N. Hidman, H. Ström, G. Sardina, S. Sasic, A comprehensive lift force model for deformable bubbles rising in moderate shear flows, *International Journal of Multiphase Flow* 187 (2025) 105166. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2025.105166>.

- [259] X. Huo, L. Chen, Y.S. Tian, T.G. Karayiannis, Flow boiling and flow regimes in small diameter tubes, *Appl. Therm. Eng.* 24 (2004) 1225–1239. <https://doi.org/10.1016/J.APPLTHERMALENG.2003.11.027>.
- [260] T. Wen, H. Zhan, D. Zhang, Flow boiling heat transfer in mini channel with serrated fins: Experimental investigation and development of new correlation, *Int. J. Heat Mass Transf.* 128 (2019) 1081–1094. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2018.09.071>.
- [261] T.G. Karayiannis, M.M. Mahmoud, D.B.R. Kenning, A study of discrepancies in flow boiling results in small to microdiameter metallic tubes, *Exp. Therm. Fluid Sci.* 36 (2012) 126–142. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2011.09.005>.
- [262] S. Saisorn, J. Kaew-On, S. Wongwises, Flow pattern and heat transfer characteristics of R-134a refrigerant during flow boiling in a horizontal circular mini-channel, *Int. J. Heat Mass Transf.* 53 (2010) 4023–4038. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2010.05.022>.
- [263] M. Özdemir, *Single-Phase Flow and Flow Boiling of Water in Rectangular Metallic Microchannels*, Brunel University London, 2016.
- [264] Y. Zhang, H. Tan, Y. Li, S. Shan, Y. Liu, A modified heat transfer correlation for flow boiling in small channels based on the boundary layer theory, *Int. J. Heat Mass Transf.* 132 (2019) 107–117. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2018.11.148>.
- [265] J.C. Chen, Correlation for boiling heat transfer to saturated fluids in convective flow, *Industrial and Engineering Chemistry Process Design and Development* 5 (1966) 322–329. https://doi.org/10.1021/I260019A023/ASSET/I260019A023.FP.PNG_V03.
- [266] J. Lee, I. Mudawar, Two-phase flow in high-heat-flux micro-channel heat sink for refrigeration cooling applications: Part II—heat transfer characteristics, *Int. J. Heat Mass Transf.* 48 (2005) 941–955. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2004.09.019>.
- [267] S.M. Kim, I. Mudawar, Review of two-phase critical flow models and investigation of the relationship between choking, premature CHF, and CHF in micro-channel heat sinks, *Int. J. Heat Mass Transf.* 87 (2015) 497–511. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2015.04.012>.
- [268] M.M. Shah, Chart correlation for saturated boiling heat transfer: Equations and Further Study, *ASHRAE Trans.* 88 (1982) 185–195.
- [269] G.R. Warrier, V.K. Dhir, L.A. Momoda, Heat transfer and pressure drop in narrow rectangular channels, *Exp. Therm. Fluid Sci.* 26 (2002) 53–64. [https://doi.org/10.1016/S0894-1777\(02\)00107-3](https://doi.org/10.1016/S0894-1777(02)00107-3).

- [270] M.M. Mahmoud, T.G. Karayiannis, Heat transfer correlation for flow boiling in small to micro tubes, *Int. J. Heat Mass Transf.* 66 (2013) 553–574. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2013.07.042>.
- [271] J.C. Chen, Correlation for boiling heat transfer to saturated fluids in convective flow, *Industrial and Engineering Chemistry Process Design and Development* 5 (1966) 322–329. https://doi.org/10.1021/I260019A023/ASSET/I260019A023.FP.PNG_V03.
- [272] M.G. Cooper, Heat Flow Rates in Saturated Nucleate Pool Boiling-A Wide-Ranging Examination Using Reduced Properties, *Adv. Heat Transf.* 16 (1984) 157–239. [https://doi.org/10.1016/S0065-2717\(08\)70205-3](https://doi.org/10.1016/S0065-2717(08)70205-3).
- [273] M.M. Shah, Unified correlation for heat transfer during boiling in plain mini/micro and conventional channels, *International Journal of Refrigeration* 74 (2017) 604–624. <https://doi.org/10.1016/j.ijrefrig.2016.11.023>.
- [274] W. Tong, A.E. Bergles, M.K. Jensen, Pressure drop with highly subcooled flow boiling in small-diameter tubes, *Exp. Therm. Fluid Sci.* 15 (1997) 202–212. [https://doi.org/10.1016/S0894-1777\(97\)00018-6](https://doi.org/10.1016/S0894-1777(97)00018-6).
- [275] W. Qu, I. Mudawar, Measurement and prediction of pressure drop in two-phase micro-channel heat sinks, *Int. J. Heat Mass Transf.* 46 (2003) 2737–2753. [https://doi.org/10.1016/S0017-9310\(03\)00044-9](https://doi.org/10.1016/S0017-9310(03)00044-9).
- [276] E. Costa-Patry, J. Olivier, B.A. Nichita, B. Michel, J.R. Thome, Two-phase flow of refrigerants in 85 μ m-wide multi-microchannels: Part I – Pressure drop, *Int. J. Heat Fluid Flow* 32 (2011) 451–463. <https://doi.org/10.1016/J.IJHEATFLUIDFLOW.2011.01.005>.
- [277] K. Balasubramanian, P.S. Lee, L.W. Jin, S.K. Chou, C.J. Teo, S. Gao, Experimental investigations of flow boiling heat transfer and pressure drop in straight and expanding microchannels – A comparative study, *International Journal of Thermal Sciences* 50 (2011) 2413–2421. <https://doi.org/10.1016/J.IJTHERMALSCI.2011.07.007>.
- [278] J. Zeng, S. Zhang, Y. Tang, Y. Sun, W. Yuan, Flow boiling characteristics of micro-grooved channels with reentrant cavity array at different operational conditions, *Int. J. Heat Mass Transf.* 114 (2017) 1001–1012. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2017.06.128>.
- [279] J. Lee, I. Mudawar, Two-phase flow in high-heat-flux micro-channel heat sink for refrigeration cooling applications: Part I—pressure drop characteristics, *Int. J. Heat Mass Transf.* 48 (2005) 928–940. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2004.09.018>.

- [280] U. Soupremanien, S. Le Person, M. Favre-Marinet, Y. Bultel, Influence of the aspect ratio on boiling flows in rectangular mini-channels, *Exp. Therm. Fluid Sci.* 35 (2011) 797–809. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2010.06.014>.
- [281] S.G. Singh, A. Kulkarni, S.P. Duttagupta, B.P. Puranik, A. Agrawal, Impact of aspect ratio on flow boiling of water in rectangular microchannels, *Exp. Therm. Fluid Sci.* 33 (2008) 153–160. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2008.07.014>.
- [282] C.B. Tibiriçá, G. Ribatski, Two-Phase Frictional Pressure Drop and Flow Boiling Heat Transfer for R245fa in a 2.32-mm Tube, <Http://Dx.Doi.Org/10.1080/01457632.2011.562725> 32 (2011) 1139–1149. <https://doi.org/10.1080/01457632.2011.562725>.
- [283] L. Yin, P. Jiang, R. Xu, H. Hu, L. Jia, Heat transfer and pressure drop characteristics of water flow boiling in open microchannels, *Int. J. Heat Mass Transf.* 137 (2019) 204–215. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2019.03.108>.
- [284] T.G. Karayiannis, D. Shiferaw, D.B. Kenning, Saturated flow boiling in small- to micro-diameter metallic tubes: Experimental results and modeling, in: *Engineering Conferences International*, 2008.
- [285] B. Agostini, R. Revellin, J.R. Thome, M. Fabbri, B. Michel, D. Calmi, U. Kloter, High heat flux flow boiling in silicon multi-microchannels – Part III: Saturated critical heat flux of R236fa and two-phase pressure drops, *Int. J. Heat Mass Transf.* 51 (2008) 5426–5442. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2008.03.005>.
- [286] T.C.B. Roldão, C.B. Tibiriçá, Critical heat flux, heat transfer and pressure drop at high mass fluxes for R123 in a single microchannel, *Int. J. Heat Mass Transf.* 222 (2024) 125143. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2023.125143>.
- [287] D. Deng, W. Wan, H. Shao, Y. Tang, J. Feng, J. Zeng, Effects of operation parameters on flow boiling characteristics of heat sink cooling systems with reentrant porous microchannels, *Undefined* 96 (2015) 340–351. <https://doi.org/10.1016/J.ENCONMAN.2015.02.072>.
- [288] B. Markal, O. Aydin, M. Avci, Effect of aspect ratio on saturated flow boiling in microchannels, *Int. J. Heat Mass Transf.* 93 (2016) 130–143. <https://doi.org/10.1016/j.ijheatmasstransfer.2015.10.024>.
- [289] X. Chen, S. Chen, J. Chen, J. Li, X. Liu, L. Chen, Y. Hou, Two-phase flow boiling frictional pressure drop of liquid nitrogen in horizontal circular mini-tubes: Experimental investigation and comparison with correlations, *Cryogenics (Guildf)*. 83 (2017) 85–94. <https://doi.org/10.1016/J.CRYOGENICS.2017.03.002>.

- [290] Francesco. D'Auria, Thermal-hydraulics of water cooled nuclear reactors, Woodhead Publishing, 2017.
- [291] H. McAdams, W. Woods, L. Heroman, Vaporization inside horizontal tubes-II Benzene-oil mixtures, *Trans. ASME* 39 (1949) 39–48.
- [292] M. Ducoulombier, S. Colasson, J. Bonjour, P. Haberschill, Carbon dioxide flow boiling in a single microchannel – Part I: Pressure drops, *Exp. Therm. Fluid Sci.* 35 (2011) 581–596. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2010.12.010>.
- [293] A.E. Dukler, M. Wicks, R.G. Cleveland, Frictional pressure drop in two-phase flow: A. A comparison of existing correlations for pressure loss and holdup, *AIChE Journal* 10 (1964) 38–43. <https://doi.org/10.1002/AIC.690100117>;WEBSITE:WEBSITE:AICHE;WGROU:STRING:PUBLICATION.
- [294] R. Lockhart, R. Martinelli, Proposed correlation of data for isothermal two-phase, two-component flow in pipes, *Chem. Eng. Prog.* 45 (1949) 39–48.
- [295] D. Chisholm, A theoretical basis for the Lockhart-Martinelli correlation for two-phase flow, *Int. J. Heat Mass Transf.* 10 (1967) 1767–1778. [https://doi.org/10.1016/0017-9310\(67\)90047-6](https://doi.org/10.1016/0017-9310(67)90047-6).
- [296] C. Keepaiboon, P. Thiangtham, O. Mahian, A.S. Dalkılıç, S. Wongwises, Pressure drop characteristics of R134a during flow boiling in a single rectangular micro-channel, *International Communications in Heat and Mass Transfer C* (2016) 245–253. <https://doi.org/10.1016/J.ICHEATMASSTRANSFER.2015.12.013>.
- [297] H. Huang, J.R. Thome, An experimental study on flow boiling pressure drop in multi-microchannel evaporators with different refrigerants, *Exp. Therm. Fluid Sci.* 80 (2017) 391–407. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2016.08.030>.
- [298] B. Markal, O. Aydin, M. Avci, Prediction of Pressure Drop for Flow Boiling in Rectangular Multi-Microchannel Heat Sinks, *Heat Transfer Engineering* 40 (2019) 26–38. <https://doi.org/10.1080/01457632.2017.1404552>.
- [299] Y. Gao, Y. Feng, S. Shao, C. Tian, Two-phase pressure drop of ammonia in horizontal small diameter tubes: Experiments and correlation, *International Journal of Refrigeration* 98 (2019) 283–293. <https://doi.org/10.1016/J.IJREFRIG.2018.11.018>.
- [300] H. Müller-Steinhagen, K. Heck, A simple friction pressure drop correlation for two-phase flow in pipes, *Chemical Engineering and Processing: Process Intensification* 20 (1986) 297–308. [https://doi.org/10.1016/0255-2701\(86\)80008-3](https://doi.org/10.1016/0255-2701(86)80008-3).

- [301] M.A. Moradkhani, S.H. Hosseini, P. Morshedi, M. Rahimi, S. Mengjie, Saturated flow boiling inside conventional and mini/micro channels: A new general model for frictional pressure drop using genetic programming, *International Journal of Refrigeration* 132 (2021) 197–212. <https://doi.org/10.1016/J.IJREFRIG.2021.09.022>.
- [302] M.A. Moradkhani, S.H. Hosseini, M. Valizadeh, A. Zendehboudi, G. Ahmadi, A general correlation for the frictional pressure drop during condensation in mini/micro and macro channels, *Int. J. Heat Mass Transf.* 163 (2020) 120475. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2020.120475>.
- [303] A.H. Al-Zaidi, M.M. Mahmoud, T.G. Karayiannis, Flow boiling pressure drop correlation in small to micro passages, *Int. J. Heat Mass Transf.* 224 (2024) 125376. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2024.125376>.
- [304] Y. Katto, Critical Heat Flux, *Adv. Heat Transf.* 17 (1985) 1–64. [https://doi.org/10.1016/S0065-2717\(08\)70284-3](https://doi.org/10.1016/S0065-2717(08)70284-3).
- [305] G. Liang, I. Mudawar, Review of spray cooling – Part 1: Single-phase and nucleate boiling regimes, and critical heat flux, *Int. J. Heat Mass Transf.* 115 (2017) 1174–1205. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2017.06.029>.
- [306] N. Mascarenhas, I. Mudawar, Analytical and computational methodology for modeling spray quenching of solid alloy cylinders, *Int. J. Heat Mass Transf.* 53 (2010) 5871–5883. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2010.06.055>.
- [307] G. Liang, I. Mudawar, Pool boiling critical heat flux (CHF) – Part 1: Review of mechanisms, models, and correlations, *Int. J. Heat Mass Transf.* 117 (2018) 1352–1367. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2017.09.134>.
- [308] L. Guo, J. Camm, T. Hirokawa, H. Li, L. Huang, A detailed review on the role of critical heat flux in micro-channel dryout phenomena and strategies for heat transfer enhancement, *Int. J. Heat Mass Transf.* 241 (2025) 126740. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2025.126740>.
- [309] W. Qu, I. Mudawar, Measurement and correlation of critical heat flux in two-phase micro-channel heat sinks, *Int. J. Heat Mass Transf.* 47 (2004) 2045–2059. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2003.12.006>.
- [310] S.M. Kim, I. Mudawar, Thermal design and operational limits of two-phase micro-channel heat sinks, *Int. J. Heat Mass Transf.* 106 (2017) 861–876. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2016.10.020>.

- [311] H. Zhang, I. Mudawar, M.M. Hasan, Flow boiling CHF in microgravity, *Int. J. Heat Mass Transf.* 48 (2005) 3107–3118. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2005.02.015>.
- [312] D.D. Hall, I. Mudawar, Ultra-high critical heat flux (CHF) for subcooled water flow boiling—II: high-CHF database and design equations, *Int. J. Heat Mass Transf.* 42 (1999) 1429–1456. [https://doi.org/10.1016/S0017-9310\(98\)00242-7](https://doi.org/10.1016/S0017-9310(98)00242-7).
- [313] T.C. Willingham, I. Mudawar, Forced-convection boiling and critical heat flux from a linear array of discrete heat sources, *Int. J. Heat Mass Transf.* 35 (1992) 2879–2890. [https://doi.org/10.1016/0017-9310\(92\)90308-F](https://doi.org/10.1016/0017-9310(92)90308-F).
- [314] J.C. Sturgis, I. Mudawar, Critical heat flux in a long, rectangular channel subjected to one-sided heating—II. Analysis of critical heat flux data, *Int. J. Heat Mass Transf.* 42 (1999) 1849–1862. [https://doi.org/10.1016/S0017-9310\(98\)00275-0](https://doi.org/10.1016/S0017-9310(98)00275-0).
- [315] W. Liu, S. Peng, G. Jiang, Y. Liu, Development and assessment of a new CHF mechanistic model for subcooled and low quality flow boiling, *Int. J. Heat Mass Transf.* 165 (2021) 120641. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2020.120641>.
- [316] M. Yan, Q. He, Z. Ma, H. Liu, L. Wan, Z. Li, L. ming Pan, Experimental investigation and a mechanical model of critical heat flux in a narrow rectangular channel, *Exp. Therm. Fluid Sci.* 128 (2021) 110432. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2021.110432>.
- [317] J. Weisman, B.S. Pei, Prediction of critical heat flux in flow boiling at low qualities, *Int. J. Heat Mass Transf.* 26 (1983) 1463–1477. [https://doi.org/10.1016/S0017-9310\(83\)80047-7](https://doi.org/10.1016/S0017-9310(83)80047-7).
- [318] S.S. Kutateladze, A.I. Leont'ev, Some applications of the asymptotic theory of the turbulent boundary layer, in: *Proceedings of the Third International Heat Transfer Conference*, New York, 1966.
- [319] C.H. Lee, I. Mudawwar, A mechanistic critical heat flux model for subcooled flow boiling based on local bulk flow conditions, *International Journal of Multiphase Flow* 14 (1988) 711–728. [https://doi.org/10.1016/0301-9322\(88\)90070-5](https://doi.org/10.1016/0301-9322(88)90070-5).
- [320] J.G. Kirby, R. Stainforth, L.H. Kinnier, A visual study of forced convective boiling. Part II: Flow patterns and burnout for a round test section, 1967.
- [321] L.S. Tong, G.F. Hewitt, Overall viewpoint of flow boiling CHF mechanisms, 1972.
- [322] W.X. Liu, W.X. Tian, Y.W. Wu, G.H. Su, S.Z. Qiu, X. Yan, Y.P. Huang, D.X. Du, An improved mechanistic critical heat flux model and its application to motion conditions, *Progress in Nuclear Energy* 61 (2012) 88–101. <https://doi.org/10.1016/J.PNUCENE.2012.07.002>.

- [323] H.S. Ahn, S.H. Kang, C. Lee, J. Kim, M.H. Kim, The effect of liquid spreading due to microstructures of flow boiling critical heat flux, *International Journal of Multiphase Flow* 43 (2012) 1–12. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2012.02.003>.
- [324] V.Y.S. Lee, T.G. Karayiannis, Effect of inlet subcooling on flow boiling in microchannels, *Appl. Therm. Eng.* 181 (2020) 115966. <https://doi.org/10.1016/j.applthermaleng.2020.115966>.
- [325] M.B. Bowers, I. Mudawar, High flux boiling in low flow rate, low pressure drop mini-channel and micro-channel heat sinks, *Int. J. Heat Mass Transf.* 37 (1994) 321–332. [https://doi.org/10.1016/0017-9310\(94\)90103-1](https://doi.org/10.1016/0017-9310(94)90103-1).
- [326] W.K. Kuan, Experimental study of flow boiling heat transfer and critical heat flux in microchannels, Rochester Institute of Technology, 2006.
- [327] W.K. Kuan, S.G. Kandlikar, Experimental study and model on critical heat flux of refrigerant-123 and water in microchannels, *J. Heat Transfer* 130 (2008). <https://doi.org/10.1115/1.2804936/475427>.
- [328] L. Wojtan, R. Revellin, J.R. Thome, Investigation of saturated critical heat flux in a single, uniformly heated microchannel, *Exp. Therm. Fluid Sci.* 30 (2006) 765–774. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2006.03.006>.
- [329] J.E. Park, Critical heat flux in multi-microchannel copper elements with low pressure refrigerants, Thesis, École Polytechnique Fédérale de Lausanne, 2008.
- [330] Z. Anwar, B.E. Palm, R. Khodabandeh, Dryout characteristics of natural and synthetic refrigerants in single vertical mini-channels, *Exp. Therm. Fluid Sci.* 68 (2015) 257–267. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2015.04.016>.
- [331] C.B. Tibiriçá, S. Szczukiewicz, G. Ribatski, J.R. Thome, Critical heat flux of R134a and R245fa inside small-diameter tubes, *Heat Transfer Engineering* 34 (2013) 492–499. <https://doi.org/10.1080/01457632.2012.722457>; WEBSITE: WEBSITE: TFOPB; PAGEGROUP :STRING: PUBLICATION.
- [332] X. Zhou, H. Liu, F. Pourpanah, T. Zeng, X. Wang, A survey on epistemic (model) uncertainty in supervised learning: Recent advances and applications, *Neurocomputing* 489 (2022) 449–465. <https://doi.org/10.1016/J.NEUCOM.2021.10.119>.
- [333] İ. Yazıcı, I. Shayea, J. Din, A survey of applications of artificial intelligence and machine learning in future mobile networks-enabled systems, *Engineering Science and Technology, an International Journal* 44 (2023) 101455. <https://doi.org/10.1016/J.JESTCH.2023.101455>.
- [334] J.M. Duarte, L. Berton, A review of semi-supervised learning for text classification, *Artif. Intell. Rev.* 56 (2023) 9401–9469. <https://doi.org/10.1007/S10462-023-10393-8/TABLES/13>.

- [335] A.G. Barto, R.S. Sutton, Reinforcement Learning, Second Edition, The MIT Press, Cambridge, MA, USA, 2018.
- [336] I. Zaborowska, H. Grzybowski, R. Mosdorf, Boiling flow pattern identification using a self-organizing map, *Applied Sciences* (Switzerland) 10 (2020). <https://doi.org/10.3390/APP10082792>.
- [337] T. Kohonen, Self-organized formation of topologically correct feature maps, *Biol. Cybern.* 43 (1982) 59–69. <https://doi.org/10.1007/BF00337288/METRICS>.
- [338] A.W. Chu, B.Y. Liu, C.L. Pan, D.H. Zhu, E.X. Yang, Identification of boiling flow pattern in narrow rectangular channel based on TFA-CNN combined method, *Flow Measurement and Instrumentation* 83 (2022) 102086. <https://doi.org/10.1016/J.FLOWMEASINST.2021.102086>.
- [339] X. Zhao, L. Wang, Y. Zhang, X. Han, M. Deveci, M. Parmar, A review of convolutional neural networks in computer vision, *Artif. Intell. Rev.* 57 (2024) 1–43. <https://doi.org/10.1007/S10462-024-10721-6/FIGURES/33>.
- [340] H. Xu, T. Tang, Two-phase flow pattern online monitoring system based on convolutional neural network and transfer learning, *Nuclear Engineering and Technology* (2022). <https://doi.org/10.1016/J.NET.2022.07.016>.
- [341] L. OuYang, N. Jin, W. Ren, A new deep neural network framework with multivariate time series for two-phase flow pattern identification, *Expert Syst. Appl.* 205 (2022). <https://doi.org/10.1016/J.ESWA.2022.117704>.
- [342] M. Schepperle, S. Junaid, P. Woias, Computer-Vision-and Deep-Learning-Based Determination of Flow Regimes, Void Fraction, and Resistance Sensor Data in Microchannel Flow Boiling, (2024). <https://doi.org/10.3390/s24113363>.
- [343] C. Paz, A. Cabarcos, M. Concheiro, M. Conde-Fontenla, Classification of boiling regimes, fluids, and heating surfaces through deep learning algorithms and image analysis, *Int. J. Heat Mass Transf.* 242 (2025) 126829. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2025.126829>.
- [344] M. Moiz, R. Prajapat, A. Srivastava, Prediction of bubble dynamics parameters in nucleate flow boiling using artificial neural networks (ANN), *International Communications in Heat and Mass Transfer* 162 (2025) 108668. <https://doi.org/10.1016/J.ICHEATMASSTRANSFER.2025.108668>.
- [345] L.S. Dias, J. Schmith, J.D. de Oliveira, J.B. Copetti, Classification of two-phase flow pattern in microtubes by image processing and machine learning, *International Journal of Multiphase Flow* 185 (2025) 105122. <https://doi.org/10.1016/J.IJMULTIPHASEFLOW.2024.105122>.

- [346] Y. Shi, K. Yang, Z. Yang, Y. Zhou, *Mobile Edge Artificial Intelligence*, Academic Press, 2021.
- [347] G. Scalabrin, M. Condosta, P. Marchi, Flow boiling of pure fluids: local heat transfer and flow pattern modeling through artificial neural networks, *International Journal of Thermal Sciences* 45 (2006) 739–751. <https://doi.org/10.1016/J.IJTHERMALSCI.2005.09.010>.
- [348] J.R. Thome, J. El Hajal, Two-Phase Flow Pattern Map for Evaporation in Horizontal Tubes: Latest Version, (2010). <https://doi.org/10.1080/714044410>.
- [349] R.M. Rizk-Allah, A.E. Hassanien, COVID-19 forecasting based on an improved interior search algorithm and multi-layer feed forward neural network, *Studies in Computational Intelligence* 1005 (2020) 129–152. https://doi.org/10.1007/978-3-030-91103-4_8.
- [350] P.G. Benardos, G.C. Vosniakos, Optimizing feedforward artificial neural network architecture, *Eng. Appl. Artif. Intell.* 20 (2007) 365–382. <https://doi.org/10.1016/J.ENGAPPAL.2006.06.005>.
- [351] L. Guo, S. Zhang, Y. Chen, L. Cheng, Identification of boiling two-phase flow patterns in water wall tube based on BP neural network, *Proceedings - International Conference on Electrical and Control Engineering, ICECE 2010* (2010) 1150–1153. <https://doi.org/10.1109/ICECE.2010.1415>.
- [352] S. Timung, T.K. Mandal, Prediction of flow pattern of gas–liquid flow through circular microchannel using probabilistic neural network, *Appl. Soft Comput.* 13 (2013) 1674–1685. <https://doi.org/10.1016/J.ASOC.2013.01.011>.
- [353] N. Brauner, *The Onset of Drops Atomization and the Prediction of Annular Flow Boundaries in Two-Phase Pipe Flow*, Tel-Aviv, Israel, 2000.
- [354] M.K. Akbar, D.A. Plummer, S.M. Ghiaasiaan, On gas-liquid two-phase flow regimes in microchannels, *International Journal of Multiphase Flow* 29 (2003) 855–865. [https://doi.org/10.1016/S0301-9322\(03\)00043-0](https://doi.org/10.1016/S0301-9322(03)00043-0).
- [355] N. Parveen, S. Zaidi, M. Danish, Development and analyses of artificial intelligence (Ai)-based models for the flow boiling heat transfer coefficient of r600a in a mini-channel, *ChemEngineering* 2 (2018) 1–13. <https://doi.org/10.3390/CHEMENGINEERING2020027>.
- [356] J. Zhang, Y. Ma, M. Wang, D. Zhang, S. Qiu, W. Tian, G. Su, Prediction of flow boiling heat transfer coefficient in horizontal channels varying from conventional to small-diameter scales by genetic neural network, *Nuclear Engineering and Technology* 51 (2019) 1897–1904. <https://doi.org/10.1016/J.NET.2019.06.009>.
- [357] Y. Qiu, D. Garg, L. Zhou, C.R. Kharangate, S.M. Kim, I. Mudawar, An artificial neural network model to predict mini/micro-channels saturated flow boiling heat transfer coefficient based on

- universal consolidated data, *Int. J. Heat Mass Transf.* 149 (2020). <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2019.119211>.
- [358] N. Agustiarini, H.N. Hoang, J.T. Oh, N. Mohd-Ghazali, Application of machine learning to the prediction of the boiling heat transfer coefficient of R32 inside a multiport mini-channel tube, *J. Therm. Anal. Calorim.* (2022) 1–17. <https://doi.org/10.1007/S10973-022-11602-2/FIGURES/12>.
- [359] N. Agustiarini, H.N. Hoang, J.T. Oh, J.K. Kim, Predictive modeling for the boiling heat transfer coefficient of R1234yf inside a multiport minichannel tube, *Int. J. Heat Mass Transf.* 211 (2023) 124188. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2023.124188>.
- [360] B.L. Chen, T.F. Yang, U. Sajjad, H.M. Ali, W.M. Yan, Deep learning-based assessment of saturated flow boiling heat transfer and two-phase pressure drop for evaporating flow, *Eng. Anal. Bound. Elem.* 151 (2023) 519–537. <https://doi.org/10.1016/J.ENGANABOUND.2023.03.016>.
- [361] E.S. Galicia, A.H. Matamoros, A. Miyara, Prediction of heat transfer coefficient and pressure drop of flow boiling and condensation using machine learning, *J. Phys. Conf. Ser.* 2766 (2024) 012152. <https://doi.org/10.1088/1742-6596/2766/1/012152>.
- [362] M. Nalenz, J. Rodemann, T. Augustin, Learning de-biased regression trees and forests from complex samples, *Mach. Learn.* 113 (2024) 3379–3398. <https://doi.org/10.1007/S10994-023-06439-1/FIGURES/5>.
- [363] J.A. Montañez-Barrera, J.M. Barroso-Maldonado, A.F. Bedoya-Santacruz, A. Mota-Babiloni, Correlated-informed neural networks: A new machine learning framework to predict pressure drop in micro-channels, *Int. J. Heat Mass Transf.* 194 (2022) 123017. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2022.123017>.
- [364] L. Sun, K. Mishima, An evaluation of prediction methods for saturated flow boiling heat transfer in mini-channels, *Int. J. Heat Mass Transf.* 52 (2009) 5323–5329. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2009.06.041>.
- [365] S. Zheng, J. Yan, S. Wang, S. Wang, Z. Liu, F. Zhu, Pressure drop for subcooled water boiling in microchannels under high heat fluxes: Experiments and predictions from artificial neural networks, *Appl. Therm. Eng.* 236 (2024) 121498. <https://doi.org/10.1016/J.APPLTHERMALENG.2023.121498>.
- [366] J. Xiao, J. Zhang, Using deep learning algorithm to predict flow boiling pressure drop under ultrasound fields in mini-channels, *International Journal of Thermal Sciences* 215 (2025) 110023. <https://doi.org/10.1016/J.IJTHERMALSCI.2025.110023>.

- [367] S. Korniliou, T.G. Karayiannis, Flow Boiling of Water in Square Cross Section Microchannel at Different Inlet Subcooling Conditions, Proceedings of the 5th World Congress on Momentum, Heat and Mass Transfer (2020) 1–9. <https://doi.org/10.11159/icmfht20.154>.
- [368] Society of Automotive Engineers, AS568F: Aerospace Size Standard for O-Rings , (2020). <https://www.sae.org/standards/content/as568f/> (accessed November 15, 2021).
- [369] 3M Novec 7100 Engineered Fluid Product Description, 2009.
- [370] A.H. Al-Zaidi, M.M. Mahmoud, T.G. Karayiannis, Effect of aspect ratio on flow boiling characteristics in microchannels, *Int. J. Heat Mass Transf.* 164 (2021) 120587. <https://doi.org/10.1016/j.ijheatmasstransfer.2020.120587>.
- [371] A.H. Al-Zaidi, M.M. Mahmoud, T.G. Karayiannis, Flow boiling of HFE-7100 in microchannels: Experimental study and comparison with correlations, *Int. J. Heat Mass Transf.* 140 (2019) 100–128. <https://doi.org/10.1016/j.ijheatmasstransfer.2019.05.095>.
- [372] E.J. Choi, J.Y. Park, M.S. Kim, Two-phase cooling using HFE-7100 for polymer electrolyte membrane fuel cell application, *Appl. Therm. Eng.* 148 (2019) 868–877. <https://doi.org/10.1016/J.APPLTHERMALENG.2018.11.103>.
- [373] W. Li, J. Ma, T. Alam, F. Yang, J. Khan, C. Li, Flow boiling of HFE-7100 in silicon microchannels integrated with multiple micro-nozzles and reentry micro-cavities, *Int. J. Heat Mass Transf.* 123 (2018) 354–366. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2018.02.108>.
- [374] K.P. Drummond, D. Back, M.D. Sinanis, D.B. Janes, D. Peroulis, J.A. Weibel, S. V. Garimella, A hierarchical manifold microchannel heat sink array for high-heat-flux two-phase cooling of electronics, *Int. J. Heat Mass Transf.* 117 (2018) 319–330. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2017.10.015>.
- [375] B.R. Fu, S.Y. Chung, W.J. Lin, L. Wang, C. Pan, Critical heat flux enhancement of HFE-7100 flow boiling in a minichannel heat sink with saw-tooth structures, *Advances in Mechanical Engineering* 9 (2017). https://doi.org/10.1177/1687814016689022/ASSET/1EE43FE3-6D5A-4211-AB54-9A5C3BB63A58/ASSETS/IMAGES/LARGE/10.1177_1687814016689022-FIG12.JPG.
- [376] F. Yang, W. Li, X. Dai, C. Li, Flow boiling heat transfer of HFE-7000 in nanowire-coated microchannels, *Appl. Therm. Eng.* 93 (2016) 260–268. <https://doi.org/10.1016/J.APPLTHERMALENG.2015.09.097>.

- [377] Y. Sun, L. Zhang, H. Xu, X. Zhong, Subcooled flow boiling heat transfer from microporous surfaces in a small channel, *Undefined* 50 (2011) 881–889. <https://doi.org/10.1016/J.IJTHERMALSCI.2011.01.019>.
- [378] X. Luo, Q. Hou, M. Wang, W. Yang, Flow boiling heat transfer characteristics of microchannels with different corrugated wall surfaces, *Zhongnan Daxue Xuebao (Ziran Kexue Ban)/Journal of Central South University (Science and Technology)* 52 (2021) 3341–3352. <https://doi.org/10.11817/J.ISSN.1672-7207.2021.09.036>.
- [379] Ralph. Remsburg, Thermal design of electronic equipment, CRC Press, 2001.
- [380] W.M. Kays, A.L. London, Compact Heat Exchangers, 3rd ed., Krieger Publishing, 1998, 1960. <https://doi.org/10.1115/1.3644004>.
- [381] R.J. Phillips, Forced-convection, liquid-cooled, microchannel heat sinks, Massachusetts Institute of Technology, 1987.
- [382] R.K. Shah, A.L. London, Laminar flow forced convection in ducts : a source book for compact heat exchanger analytical data, 1st ed., Academic Press, New York, 1978.
- [383] D. Chisholm, Two-phase flow in bends, *International Journal of Multiphase Flow* 6 (1980) 363–367. [https://doi.org/10.1016/0301-9322\(80\)90028-2](https://doi.org/10.1016/0301-9322(80)90028-2).
- [384] A. Kawahara, P.Y. Chung, M. Kawaji, Investigation of two-phase flow pattern, void fraction and pressure drop in a microchannel, *International Journal of Multiphase Flow* 28 (2002) 1411–1435. [https://doi.org/10.1016/S0301-9322\(02\)00037-X](https://doi.org/10.1016/S0301-9322(02)00037-X).
- [385] D. Chisholm, Two-Phase Flow in Heat Exchangers and Pipelines, *Heat Transfer Engineering* 6 (1985) 48–57. <https://doi.org/10.1080/01457638508939624>.
- [386] H.W. Coleman, W.G. Steele, Experimentation, Validation, and Uncertainty Analysis for Engineers: Third Edition, John Wiley and Sons, 2009. <https://doi.org/10.1002/9780470485682>.
- [387] H. Müller-steinhausen, N. Epstein, A.P. Watkinson, Effect of dissolved gases on subcooled flow boiling heat transfer, *Chemical Engineering and Processing: Process Intensification* 23 (1988) 115–124. [https://doi.org/10.1016/0255-2701\(88\)80005-9](https://doi.org/10.1016/0255-2701(88)80005-9).
- [388] F.P. Incropera, D.P. DeWitt, T.L. Bergman, A.S. Lavine, Incropera's Principles of Heat and Mass Transfer, 1st Edition, Global Edition, 1st ed., John Wiley & Sons, Singapore, 2017.
- [389] P.X. Jiang, M.H. Fan, G.S. Si, Z.P. Ren, Thermal–hydraulic performance of small scale micro-channel and porous-media heat-exchangers, *Int. J. Heat Mass Transf.* 44 (2001) 1039–1051. [https://doi.org/10.1016/S0017-9310\(00\)00169-1](https://doi.org/10.1016/S0017-9310(00)00169-1).

- [390] P.S. Lee, S. V. Garimella, Thermally Developing Flow and Heat Transfer in Rectangular Microchannels of Different Aspect Ratios, *CTRC Research Publications* 49 (2006) 3060–3067. <https://doi.org/10.1016/j.ijheatmasstransfer.2006.02.011>.
- [391] T. Ren, Z. Zhu, R. Zhang, J. Shi, C. Yan, Visualization Experiment of Bubble Coalescence in a Narrow Vertical Rectangular Channel, *Front. Energy Res.* 8 (2020) 539526. <https://doi.org/10.3389/FENRG.2020.00096/BIBTEX>.
- [392] H. Chen, S. Wei, W. Ding, H. Wei, L. Li, H. Saxén, H. Long, Y. Yu, Interfacial Area Transport Equation for Bubble Coalescence and Breakup: Developments and Comparisons, *Entropy* 2021, Vol. 23, Page 1106 23 (2021) 1106. <https://doi.org/10.3390/E23091106>.
- [393] J. Ni, T. Cao, L. Hou, F. Dong, Experimental study on bubble size distribution of subcooled flow boiling in cylinder head of internal combustion engines, *Journal of Mechanical Science and Technology* 33 (2019) 913–921. <https://doi.org/10.1007/S12206-019-0148-X/METRICS>.
- [394] J. Happel, H. Brenner, *Low Reynolds number hydrodynamics*, Springer Netherlands, Dordrecht, 1983. <https://doi.org/10.1007/978-94-009-8352-6>.
- [395] M. Piasecka, B. Maciejewska, D. Michalski, N. Dadas, A. Piasecki, Investigations of Flow Boiling in Mini-Channels: Heat Transfer Calculations with Temperature Uncertainty Analyses, *Energies* 2024, Vol. 17, Page 791 17 (2024) 791. <https://doi.org/10.3390/EN17040791>.
- [396] X. Zhuang, Y. Xie, X. Li, S. Yue, H. Wang, H. Wang, P. Yu, Experimental investigation on flow boiling of HFE-7100 in a microchannel with pin fin array, *Appl. Therm. Eng.* 225 (2023) 120180. <https://doi.org/10.1016/J.APPLTHERMALENG.2023.120180>.
- [397] M.M. Mahmoud, T.G. Karayiannis, Pool boiling review: Part I – Fundamentals of boiling and relation to surface design, *Thermal Science and Engineering Progress* 25 (2021) 101024. <https://doi.org/10.1016/J.TSEP.2021.101024>.
- [398] A. Al-Gaheeshi, M. Mahmoud, T. Karayiannis, Flow Boiling Heat Transfer in a Vertical Small Diameter Tube: Effect of Different Fluids and Surface Characteristics, in: *Conference: Proceedings of the 4th International Forum on Heat Transfer*, Sendai, Japan, 2016.
- [399] P.J. Marto, W.M. Rohsenow, Nucleate Boiling Instability of Alkali Metals, *ASME J. Heat Transfer* 88 (1966) 183–193.
- [400] P.G. Kosky, Nucleation site instability in nucleate boiling, *Int. J. Heat Mass Transf.* 11 (1968) 929–932. [https://doi.org/10.1016/0017-9310\(68\)90135-X](https://doi.org/10.1016/0017-9310(68)90135-X).
- [401] S.T. Hsu, F.W. Schmidt, Measured Variations in Local Surface Temperatures in Pool Boiling of Water, *J. Heat Transfer* 83 (1961) 254–260. <https://doi.org/10.1115/1.3682252>.

- [402] S. Wang, X. Bi, S. Wang, Boiling heat transfer in small rectangular channels at low Reynolds number and mass flux, *Exp. Therm. Fluid Sci.* 77 (2016) 234–245. <https://doi.org/10.1016/j.expthermflusci.2016.04.022>.
- [403] G. Criscuolo, W.B. Markussen, K.E. Meyer, M.R. Kærn, High heat flux flow boiling of R1234yf, R1234ze(E) and R134a in high aspect ratio microchannels, *Int. J. Heat Mass Transf.* 186 (2022) 122215. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2021.122215>.
- [404] Z. Anwar, B. Palm, R. Khodabandeh, Flow boiling heat transfer and dryout characteristics of R152a in a vertical mini-channel, *Exp. Therm. Fluid Sci.* 53 (2014) 207–217. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2013.12.009>.
- [405] M. Asim, Z. Anwar, M. Farooq, R. Shaukat, S. Imran, M. Abbas, Q. Ali, Flow Boiling Heat Transfer Characteristics of Low Global, *Thermal Science* 26 (2022) 63–76. <https://doi.org/10.2298/TSCI200601327A>.
- [406] E.R. Dário, J.C. Passos, M.L. Sánchez Simón, L. Tadrist, Pressure drop during flow boiling inside parallel microchannels, *International Journal of Refrigeration* 72 (2016) 111–123. <https://doi.org/10.1016/J.IJREFRIG.2016.08.002>.
- [407] S.Z. Rouhani, Modified Correlations for Void Fraction and Two-Phase Pressure Drop, AB Atomenergi Sweden (1969).
- [408] V. Lee, Flow Boiling of HFE-7200 in Multi-Microchannel Heat Sinks for High-Heat Flux Applications, PhD, Brunel University London, 2020.
- [409] D. Shiferaw, T.G. Karayiannis, D.B.R. Kenning, Flow boiling in a 1.1 mm tube with R134a: Experimental results and comparison with model, *International Journal of Thermal Sciences* 48 (2009) 331–341. <https://doi.org/10.1016/J.IJTHERMALSCI.2008.02.009>.
- [410] Mirmanto, Single-Phase Flow and Flow Boiling of Water in Horizontal Rectangular Microchannels, Brunel University London, 2013.
- [411] W.T. Bottomley, Flow of Boiling Water Through Orifices and Pipes, *Trans. North-East Coast Institution of Engineers and Shipbuilders* 53 (1936) 65–100.
- [412] M.W. Benjamin, J.G. Miller, The Flow of a Flashing Mixture of Water and Steam Through Pipes, *J. Fluids Eng.* 64 (1942) 657–664. <https://doi.org/10.1115/1.4019232>.
- [413] W.F. Allen, Flow of a flashing mixture of water and steam through pipes and valves, *Trans. AMSE* (1951) 257–265.
- [414] J. Kaew-On, S. Wongwises, Experimental investigation of evaporation heat transfer coefficient and pressure drop of R-410A in a multiport mini-channel, *International Journal of Refrigeration* 32 (2009) 124–137. <https://doi.org/10.1016/j.ijrefrig.2008.06.010>.

- [415] D.F. Sempértegui-Tapia, G. Ribatski, Two-phase frictional pressure drop in horizontal micro-scale channels: Experimental data analysis and prediction method development, *International Journal of Refrigeration* 79 (2017) 143–163. <https://doi.org/10.1016/j.ijrefrig.2017.03.024>.
- [416] A. Donniacuo, R. Charnay, R. Mastrullo, A.W. Mauro, R. Revellin, Film thickness measurements for annular flow in minichannels: Description of the optical technique and experimental results, *Exp. Therm. Fluid Sci.* 69 (2015) 73–85. <https://doi.org/10.1016/J.EXPTHERMFLUSCI.2015.07.005>.
- [417] R. Charnay, J. Bonjour, R. Revellin, Experimental investigation of R-245fa flow boiling in minichannels at high saturation temperatures: Flow patterns and flow pattern maps, *Int. J. Heat Fluid Flow* 46 (2014) 1–16. <https://doi.org/10.1016/J.IJHEATFLUIDFLOW.2013.12.002>.
- [418] H. Liu, Z. Ma, L. Zhang, K. Li, Q. Lian, L. ming Pan, Minimum entropy production principle-based semi-empirical model for void fraction prediction in vertical upward dispersed gas-liquid flows, *Int. J. Heat Mass Transf.* 247 (2025) 127120. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2025.127120>.
- [419] S.M. Zivi, Estimation of Steady-State Steam Void-Fraction by Means of the Principle of Minimum Entropy Production, *J. Heat Transfer* 86 (1964) 247–251. <https://doi.org/10.1115/1.3687113>.
- [420] E.O. Moeck, Annular-Dispersed Two-Phase Flow and Critical Heat Flux , PhD Thesis, McGill University, 1970.
- [421] A. Al Gaheeshi, Flow Boiling in Vertical Small to Micro Scale Tubes, Brunel University London, 2018.
- [422] E. Pike-Wilson, Flow Boiling of R245fa in Vertical Small Metallic Tubes, Brunel University London, 2014.
- [423] M. Mahmoud, Flow Boiling of R134a in Vertical Mini-Diameter Tubes, Brunel University London, 2011.
- [424] D. Shiferaw, Two-Phase Flow Boiling In Small to Micro Diameter Tubes, Brunel University London, 2008.
- [425] R. AL-Janabi, Flow Boiling in Multi Microchannels Using Refrigerant R134a and R1234yf, Brunel University London, 2021.
- [426] R.J. May, H.R. Maier, G.C. Dandy, Data splitting for artificial neural networks using SOM-based stratified sampling, *Neural Networks* 23 (2010) 283–294. <https://doi.org/10.1016/J.NEUNET.2009.11.009>.

- [427] I. Ben Ayed, *High-Order Models in Semantic Image Segmentation*, Elsevier, 2023. <https://doi.org/10.1016/C2015-0-04313-1>.
- [428] A. Subasi, *Practical Machine Learning for Data Analysis Using Python*, Elsevier, 2020. <https://doi.org/10.1016/B978-0-12-821379-7.00008-4>.
- [429] S. Nesmachnow, S. Gallardo Saavedra, D. Eskin, J. Loyola-Fuentes, L. Pietrasanta, M. Marengo, F. Coletti, *Machine Learning Algorithms for Flow Pattern Classification in Pulsating Heat Pipes*, (2022). <https://doi.org/10.3390/en15061970>.
- [430] J. Ha, M. Kambe, J. Pe, *Data Mining: Concepts and Techniques*, Elsevier, 2011. <https://doi.org/10.1016/C2009-0-61819-5>.
- [431] J. Miao, L. Niu, A Survey on Feature Selection, *Procedia Comput. Sci.* 91 (2016) 919–926. <https://doi.org/10.1016/J.PROCS.2016.07.111>.
- [432] Rand.R. Wilcox, *Applying Contemporary Statistical Techniques*, Elsevier, 2003. <https://doi.org/10.1016/B978-0-12-751541-0.X5021-4>.
- [433] J. Cai, J. Luo, S. Wang, S. Yang, Feature selection in machine learning: A new perspective, *Neurocomputing* 300 (2018) 70–79. <https://doi.org/10.1016/J.NEUCOM.2017.11.077>.
- [434] J. Wu, X.Y. Chen, H. Zhang, L.D. Xiong, H. Lei, S.H. Deng, Hyperparameter Optimization for Machine Learning Models Based on Bayesian Optimization, *Journal of Electronic Science and Technology* 17 (2019) 26–40. <https://doi.org/10.11989/JEST.1674-862X.80904120>.
- [435] P.P. Ippolito, Hyperparameter Tuning: The Art of Fine-Tuning Machine and Deep Learning Models to Improve Metric Results, *Tourism on the Verge Part F1051* (2022) 231–251. https://doi.org/10.1007/978-3-030-88389-8_12/COVER.
- [436] O. İrsoy, E. Alpaydın, Dropout regularization in hierarchical mixture of experts, *Neurocomputing* 419 (2021) 148–156. <https://doi.org/10.1016/J.NEUCOM.2020.08.052>.
- [437] S.M. Roy, T. Goswami, C.K. Nara, *Statistical Modeling in Machine Learning*, Academic Press, 2022. <https://doi.org/10.1016/B978-0-323-91776-6.00001-4>.

APPENDIX A

FLOW BOILING HEAT TRANSFER CORRELATIONS

Author(s)	Correlation	Operating Conditions
Shah [268]	$h_{tp} = \text{Max}(E, S)h_{sp}$ $S = 1.8/N^{0.8}$ For $1 < N$: $E = 230Bo^{0.5} \quad (\text{for } Bo > 3 \times 10^{-5})$ $E = 1 + 46Bo^{0.5} \quad (\text{for } Bo < 3 \times 10^{-5})$ For $0.1 < N \leq 1$: $E = FBo^{0.5}e^{2.74N^{-0.1}}$ For $N > 1$: $E = FBo^{0.5}e^{2.47N^{-0.15}}$ $F = 14.7 \quad (\text{for } Bo \geq 11 \times 10^{-4})$ $F = 15.43 \quad (\text{for } Bo < 11 \times 10^{-4})$ $N = N_{co} \quad (\text{for } Fr_{lo} \geq 0.04)$ $N = 0.38Fr_{lo}^{-0.3}N_{co} \quad (\text{for } Fr_{lo} < 0.04)$	$D_h = 6\text{--}25.4 \text{ mm}$ Fluids (water, R11, R12, R22, R113, cyclohexane) For saturated boiling in vertical and horizontal channels
Warrier et al. [269]	$h_{tp} = \left[1 + 6Bo^{\frac{1}{16}} - 5.3(1 - 855Bo)x^{0.65} \right] h_{sp}$	$D_h = 0.75 \text{ mm}$ Fluid (FC-84) $G = 557\text{--}1600 \text{ kg/m}^2\text{s}$ $q'' = \text{up to } 59.9 \text{ kW/m}^2$ For saturated boiling in horizontal rectangular multi-channels

Lee and Mudawar [266]	$h_{tp} = 3.856\chi^{0.267}h_{sp} \quad (for\ 0 < x < 0.05)$ $h_{tp} = 436.48Bo^{0.522}We_{lo}^{0.351}\chi^{0.665}h_{sp} \quad (for\ 0.05 < x < 0.55)$ $h_{tp} = Max\{(108.6\chi^{1.665}h_{sp,g})h_{sp,g}\}, \quad (for\ 0.55 < x < 1)$ $h_{sp,g} = \frac{Nu_3k_g}{D_h} \quad (for\ laminar\ vapour\ flow)$ $h_{sp,g} = 0.023Re_{gs}^{0.8}Pr_g^{0.4}\frac{k_g}{D_h} \quad (for\ turbulent\ vapour\ flow)$	$D_h = 0.348\text{ mm}$ Fluids (water, R134a) $G = 127\text{--}654\text{ kg/m}^2\text{s}$ $q'' = 159\text{--}938\text{ kW/m}^2$ For saturated boiling in horizontal rectangular multi-channels
Mahmoud and Karayiannis [270]	$h_{tp} = Sh_{coop} + Fh_{sp}$ $h_{coop} = 55P_R^{0.12-0.2\log_{10}Rp}(-\log_{10}(P_R))^{-0.55}M^{-0.5}q''_w^{0.67}$ $S = \frac{1}{1 + 2.56 \times 10^{-6}(Re_{ls}F^{1.25})^{1.17}}$ $F = \left(1 + \frac{A}{\chi}\right)^{0.64}$ $A = 2.812C_o^{-0.408}$	$D_h = 0.52\text{--}4.26\text{ mm}$ Fluid (R134a) $G = 100\text{--}700\text{ kg/m}^2\text{s}$ $q'' = 1.7\text{--}158\text{ kW/m}^2$ For saturated boiling in vertical single tubes.
Li and Jia [232]	For nucleate boiling regime: $h_{tp} = 189Bo^{0.178}Bd^{0.4}Re_{ls}^{0.12}\frac{k_l}{D_h}$ For convective boiling regime: $h_{tp} = 277.3Bo^{0.978}Re_{ls}^{0.94}(1-x)^{0.47}\frac{k_l}{D_h}$	$D_h = 0.5\text{ mm}$ Fluid (R134a) $G = 373.33\text{--}1244.44\text{ kg/m}^2\text{s}$ $q'' = \text{up to } 802.12\text{ kW/m}^2$ For saturated boiling in horizontal rectangular multi-channels

Thiangtham et al. [149]	$h_{tp} = \frac{10^{16.7} Bo^{0.3} We_{lo}^{2.7} k_l}{Re_{lo}^{4.95} \left(\frac{\rho_l}{\rho_g} \right)^{0.9} D_h}$	$D_h = 0.421 \text{ mm}$ Fluid (R134a) $G = 150\text{--}600 \text{ kg/m}^2\text{s}$ $q'' = 3\text{--}127 \text{ kW/m}^2$ $T_{\text{sat}} = 13, 18, 23 \text{ }^\circ\text{C}$ For saturated boiling in horizontal rectangular multi-channels
Shah [273]	$h_{tp} = F h_{Shah(1982)}$ $F = 2.1 - 0.008 We_{go} - 110 Bo$ If $F < 1$, Use $F = 1$	$D_h = 0.38\text{--}27.1 \text{ mm}$ Fluids (Water, CO₂, R11, R12, R22, R32, R113, R123, R114, R123, R134a, R152a, R1234yf, R236fa, R245fa, ammonia, propane, isobutane, carbon tetrachloride, isopropyl alcohol, ethanol, methanol, n butanol, cyclohexane, benzene, heptane, pentane, argon, hydrogen, nitrogen, helium) $G = 15\text{--}2437 \text{ kg/m}^2\text{s}$ For saturated boiling in vertical and horizontal (single and multi) channels

Wen et al. [260]	$h_{tp} = h_{sp} + \frac{1}{1 + e^{\frac{x-F}{12}}}$ $h_{sp} = xh_{sp,g} + (1 - x)h_{sp,l}$ $h_{nb} = 2115.4P_R^{0.12}(-04343\log P_R)^{-0.55}M^{-0.5}q''_w^{0.27}\left(\frac{Fr}{We}\right)^{0.18}$ $F = (BoWe)^{0.07}$ $h_{sp,k} = 0.023Re_k^{0.8}Pr_k^{0.4}\frac{k_k}{D_h}$ where k refers to g or l	$D_h = 1.28\text{--}1.59$ mm Fluid (R-134a) $G=60\text{--}308$ kg/m²s $q'' = 6\text{--}48$ kW/m² For flow boiling in horizontal rectangular multi-channels with serrated fins
Zhang et al. [210]	$h_{tp} = \frac{0.00344Re_{lo}^{1.23629}e^{0.6715x_{out}}k_l}{Bo^{0.32148}We_{lo}^{0.467}}\frac{k_l}{D_h}$	$D_h = 0.25 - 1.5$ mm Fluid (water) $x < 0.4$ For saturated boiling in horizontal rectangular multi-channels
Patra et al. [225]	$h_{tp} = (C_f Bo^{0.095} We_{lo}^{0.256} Fr_{lo}^{-0.007})h_{sp,l}$ where $C_f = \begin{cases} 10.3 & \text{for Acetone} \\ 15.4 & \text{for Methanol} \end{cases}$	$D_h=0.5$ mm Fluids (Acetone, methanol) $G=28\text{--}84$ kg/m²s For saturated boiling in horizontal rectangular multi-channels

APPENDIX B

FLOW BOILING PRESSURE DROP CORRELATIONS

Author(s)	Correlation	Operating Conditions
Lockhart and Martinelli [294]	$\Delta P_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{out}^2}{\alpha_{out}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{out})^2}{1 - \alpha_{out}} - 1 \right]$ $\alpha_{out} = \left[1 + 0.28 \left(\frac{1 - x_{out}}{x_{out}} \right)^{0.64} \left(\frac{\rho_g}{\rho_l} \right)^{0.36} \left(\frac{\mu_l}{\mu_g} \right)^{0.07} \right]^{-1}$ $\Delta P_{fric} = \frac{L_{tp}}{x_{out}} \int_0^{x_{out}} \frac{2f_l G^2 (1 - x)^2}{\rho_l D_h} \phi^2 dx$ $\phi^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ $X = \sqrt{\frac{(dp/dz)_l}{(dp/dz)_g}}$ $C = \begin{cases} 5 & \text{for laminar liquid and laminar vapour flows} \\ 12 & \text{for laminar liquid and turbulent vapour flows} \\ 10 & \text{for turbulent liquid and laminar vapour flows} \\ 20 & \text{for turbulent liquid and turbulent vapour flows} \end{cases}$	$D_h = 1.5\text{--}25.8 \text{ mm}$ Fluids (air/water, air/benzene, air/kerosene and air/oils) For circular horizontal channels
Homogeneous Flow Model	$\Delta P_{acc} = G^2 \Delta v x_{out}$ $\Delta P_{fric} = \frac{2f_{tp} G^2 v_l L_{tp}}{D_h} \left[1 + \frac{x_{out}}{2} \left(\frac{\Delta v}{v_l} \right) \right]$ $\Delta v = v_g - v_l$ $f_{tp} = 0.003 \text{ (In accordance with Qu and Mudawar [275])}$	

Qu and Mudawar [275]	$\Delta P_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{out}^2}{\alpha_{out}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{out})^2}{1 - \alpha_{out}} - 1 \right]$ $\alpha_{out} = \left[1 + \frac{1 - x_{out}}{x_{out}} \left(\frac{\rho_g}{\rho_l} \right)^{0.67} \right]^{-1} \text{ (proposed by [419])}$ $\Delta P_{fric} = \frac{L_{tp}}{x_{out}} \int_0^{x_{out}} \frac{2f_l G^2 (1 - x)^2}{\rho_l D_h} \phi^2 dx$ $\phi^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ $C = 21(1 - e^{-319D_h})(0.00418G + 0.0613)$	$D_h = 0.35 \text{ mm}$ Fluids (water) For horizontal rectangular multi-channel heat sinks
Lee and Garimella [217]	$\Delta P_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{out}^2}{\alpha_{out}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{out})^2}{1 - \alpha_{out}} - 1 \right]$ $\alpha_{out} = \left[1 + \frac{1 - x_{out}}{x_{out}} \left(\frac{\rho_g}{\rho_l} \right)^{0.67} \right]^{-1} \text{ (proposed by [419])}$ $\Delta P_{fric} = \frac{L_{tp}}{x_{out}} \int_0^{x_{out}} \frac{2f_l G^2 (1 - x)^2}{\rho_l D_h} \phi^2 dx$ $\phi^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ $C = 2566G^{0.5466}D_h^{0.8819}(1 - e^{-319D_h})$	$D_h = 0.162\text{--}0.571 \text{ mm}$ Fluids (water) For horizontal rectangular multi-channel heat sinks
Keepaiboon et al. [296]	$\Delta P_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{out}^2}{\alpha_{out}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{out})^2}{1 - \alpha_{out}} - 1 \right]$ $\alpha_{out} = \left[1 + \frac{1 - x_{out}}{x_{out}} \left(\frac{\rho_g}{\rho_l} \right)^{0.67} \right]^{-1} \text{ (proposed by [419])}$ $\Delta P_{fric} = \frac{L_{tp}}{x_{out}} \int_0^{x_{out}} \frac{2f_l G^2 (1 - x)^2}{\rho_l D_h} \phi^2 dx$ $\phi^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ $C = 1.93 \times 10^5 Re_{lo}^{-1.18} Co^{-27.99} X^{0.93}$	$D_h = 0.68 \text{ mm}$ Fluids (R-134a) For horizontal single rectangular channel

Huang and Thome [297]	$\Delta P_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{out}^2}{\alpha_{out}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{out})^2}{1 - \alpha_{out}} - 1 \right]$ $\alpha_{out} = \left[1 + \frac{1 - x_{out}}{x_{out}} \left(\frac{\rho_g}{\rho_l} \right)^{0.67} \right]^{-1} \text{ (proposed by [419])}$ $\Delta P_{fric} = \frac{L_{tp}}{x_{out}} \int_0^{x_{out}} \frac{2f_l G^2 (1 - x)^2}{\rho_l D_h} \phi^2 dx$ $\phi^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ For $Re_{ls} \leq 2000$ and R1233zd(E) $f_l = \frac{8.058}{Re_{ls}}$ For $Re_{ls} \leq 2000$ and other fluids f_l is taken from Shah and London [382] For $Re_{ls} > 2000$ $f_l = \frac{0.079}{Re_{ls}^{-0.25}}$ For $Re_{ls} \leq 2000$ and $Re_{gs} \leq 2000$ $C = 0.0037 Re_{gs}^{1.7} Re_{lo}^{-0.83}$ For $Re_{ls} \leq 2000$ and $Re_{gs} > 2000$ $C = 0.9 Re_{gs}^{0.034} Re_{lo}^{0.2}$	$D_h = 0.1 \text{ mm}$ Fluids (R-1233zd(E) R-245fa and R-236fa) For horizontal rectangular multi-channel heat sinks
------------------------------	--	---

Markal et al. [298]	$\Delta P_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{out}^2}{\alpha_{out}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{out})^2}{1 - \alpha_{out}} - 1 \right]$ $\alpha_{out} = \left[1 + \frac{1 - x_{out}}{x_{out}} \left(\frac{\rho_g}{\rho_l} \right)^{0.67} \right]^{-1} \text{ (proposed by [419])}$ $\Delta P_{fric} = \frac{L_{tp}}{x_{out}} \int_0^{x_{out}} \frac{2f_l G^2 (1 - x)^2}{\rho_l D_h} \phi^2 dx$ $\phi^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ $C = -7.1 + \left(\frac{(1 - x_{out})^{1.766}}{Re_{lo}^{0.12} \beta^{0.031} Bo^{0.165} We_{lo}^{0.074} \left(\frac{\rho_l}{\rho_g} \right)^{0.233} \left(\frac{L_{tp}}{D_h} \right)^{0.247}} \right)$	$D_h = 0.1\text{-}0.25 \text{ mm}$ Fluids (DI water) For horizontal rectangular multi-channel heat sinks
Gao et al. [299]	$\Delta P_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{out}^2}{\alpha_{out}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{out})^2}{1 - \alpha_{out}} - 1 \right]$ $\alpha_{out} = \left[1 + 0.28 \left(\frac{1 - x_{out}}{x_{out}} \right)^{0.64} \left(\frac{\rho_g}{\rho_l} \right)^{0.36} \left(\frac{\mu_l}{\mu_g} \right)^{0.07} \right]^{-1}$ $\Delta P_{fric} = \frac{L_{tp}}{x_{out}} \int_0^{x_{out}} F (1 - x)^{\frac{1}{3}} + B x^3 dx$ $F = A + (1 + 0.07695 B d^{-0.03573} Re_l^{0.394}) (B - A) x$ $A = 2 \frac{f_l G^2}{D_h \rho_l}$ $B = 2 \frac{f_g G^2}{D_h \rho_g}$ f_k is taken from Shah and London [382] if $Re_{ko} < 2000$ $f_k = \frac{0.079}{Re_{ko}^{0.25}} \text{ (for } 2000 \leq Re_{ko} < 20000 \text{)}$ $f_k = \frac{0.046}{Re_{ko}^{0.2}} \text{ (for } Re_{ko} \geq 20000 \text{)}$ where k refers to g or l	$D_h = 0.81\text{-}14 \text{ mm}$ Fluids (Ammonia, R-22, R-134a, CO₂) For horizontal and vertical tubes

Maradkhani et al. [301]	$\Delta P_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{out}^2}{\alpha_{out}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1 - x_{out})^2}{1 - \alpha_{out}} - 1 \right]$ $\alpha_{out} = \left[1 + \frac{1 - x_{out}}{x_{out}} \left(\frac{\rho_g}{\rho_l} \right)^{0.67} \right]^{-1} \text{ (proposed by [419])}$ $\Delta P_{fric} = \frac{2f_{lo}L_{tp}G^2}{\rho_l x_{out} D_h} \int_0^{x_{out}} \phi^2 dx$ For $Bd < 3$: $\phi^2 = \left(\frac{P_{in}}{P_c} \right) + \frac{1.352x}{H} + \min(A_1, A_2)$ For $Bd \geq 3$: $\phi^2 = x + 7 \left(\frac{P_{in}}{P_c} \right) + 1.456 \frac{\sin(\sin(x))}{H} + \frac{2.67 \times 10^{-3} Bd + 227.68 \left[0.651^{\sin\left(\frac{7.42}{P_c H} \frac{in}{H}\right)} \right]}{x}$ $A_1 = \frac{8x}{Bd^2 H} + e^{2 \tan\left(\frac{P_{in}}{P_c}\right)}$ $A_2 = 1.0937 \frac{Re_{go}}{Re_{lo}} Bd$ $H = \left(\frac{f_{lo}}{f_{go}} \right) \left(\frac{\rho_g}{\rho_l} \right)$	$D_h = 0.07\text{-}14 \text{ mm}$ Fluids (R-717, R-245fa, R-290, R-600a, R-744, R-1234yf, R-134a, R-718, R-32, R-410A, R-1234ze(E), R-1270, R-404A, R-22, R-12, Ethanol) For rectangular and circular channel in horizontal and vertical flow
-------------------------	---	---

Al-Zaidi et al. [303]	$\Delta P_{acc} = \frac{G^2}{\rho_l} \left[\frac{x_{out}^2}{\alpha_{out}} \left(\frac{\rho_l}{\rho_g} \right) + \frac{(1-x_{out})^2}{1-\alpha_{out}} - 1 \right]$																							
	$\alpha_{out} = \left[1 + \frac{1-x_{out}}{x_{out}} \left(\frac{\rho_g}{\rho_l} \right)^{0.67} \right]^{-1} \text{ (proposed by [419])}$																							
	$\Delta P_{fric} = \frac{2f_k L_{tp} G^2}{\rho_l x_{out} D_h} \int_0^{x_{out}} \phi^2 \, dx$																							
	$\phi^2 = E (Bo^a X^b Re_{gs}^c P_R^d)$																							
	For horizontal rectangular multi-channels:																							
	<table><tr><td></td><td><i>E</i></td><td><i>a</i></td><td><i>b</i></td><td><i>c</i></td><td><i>d</i></td></tr><tr><td>$Re_{ls} < 2000$ and $Re_{gs} < 2000$</td><td>0.65</td><td>−0.6</td><td>−0.45</td><td>0.55</td><td>−0.208</td></tr><tr><td>$Re_{ls} < 2000$ and $Re_{gs} \geq 2000$</td><td>4.3×10^{-6}</td><td>−1.63</td><td>−0.16</td><td>1.2</td><td>1.72</td></tr></table>			<i>E</i>	<i>a</i>	<i>b</i>	<i>c</i>	<i>d</i>	$Re_{ls} < 2000$ and $Re_{gs} < 2000$	0.65	−0.6	−0.45	0.55	−0.208	$Re_{ls} < 2000$ and $Re_{gs} \geq 2000$	4.3×10^{-6}	−1.63	−0.16	1.2	1.72				
			<i>E</i>	<i>a</i>	<i>b</i>	<i>c</i>	<i>d</i>																	
	$Re_{ls} < 2000$ and $Re_{gs} < 2000$		0.65	−0.6	−0.45	0.55	−0.208																	
	$Re_{ls} < 2000$ and $Re_{gs} \geq 2000$		4.3×10^{-6}	−1.63	−0.16	1.2	1.72																	
For vertical circular channels:																								
<table><tr><td></td><td><i>E</i></td><td><i>a</i></td><td><i>b</i></td><td><i>c</i></td><td><i>d</i></td></tr><tr><td>$Re_{ls} < 2000$ and $Re_{gs} < 2000$</td><td>5×10^{12}</td><td>2.5</td><td>−0.2</td><td>−2</td><td>2</td></tr><tr><td>$Re_{ls} < 2000$ and $Re_{gs} \geq 2000$ or $Re_{ls} \geq 2000$ and $Re_{gs} < 2000$</td><td>$7 \times 10^3 \times We_{gs}^{0.67}$</td><td>0.45</td><td>0.085</td><td>−0.8</td><td>−0.7</td></tr><tr><td>$Re_{ls} < 2000$ and $Re_{gs} < 2000$</td><td>7.6×10^5</td><td>1.1</td><td>−0.3</td><td>−0.47</td><td>−0.35</td></tr></table>		<i>E</i>	<i>a</i>	<i>b</i>	<i>c</i>	<i>d</i>	$Re_{ls} < 2000$ and $Re_{gs} < 2000$	5×10^{12}	2.5	−0.2	−2	2	$Re_{ls} < 2000$ and $Re_{gs} \geq 2000$ or $Re_{ls} \geq 2000$ and $Re_{gs} < 2000$	$7 \times 10^3 \times We_{gs}^{0.67}$	0.45	0.085	−0.8	−0.7	$Re_{ls} < 2000$ and $Re_{gs} < 2000$	7.6×10^5	1.1	−0.3	−0.47	−0.35
	<i>E</i>	<i>a</i>	<i>b</i>	<i>c</i>	<i>d</i>																			
$Re_{ls} < 2000$ and $Re_{gs} < 2000$	5×10^{12}	2.5	−0.2	−2	2																			
$Re_{ls} < 2000$ and $Re_{gs} \geq 2000$ or $Re_{ls} \geq 2000$ and $Re_{gs} < 2000$	$7 \times 10^3 \times We_{gs}^{0.67}$	0.45	0.085	−0.8	−0.7																			
$Re_{ls} < 2000$ and $Re_{gs} < 2000$	7.6×10^5	1.1	−0.3	−0.47	−0.35																			
where f_k is the same as Gao et al. [299]																								

APPENDIX C

MICROSCALE CRITICAL HEAT FLUX CORRELATIONS

Author(s)	Correlation	Operating Conditions	Mechanism
Bowers and Mudawar [325]	$q''_{CHF} = 0.16 We_{lo}^{-0.19} Gi_{fg} \left(\frac{L_{ch}}{D_h} \right)^{-0.54}$	$D_h = 0.51 - 2.54$ mm Fluids (R-113) $G = 31 - 480$ kg/m²s Saturated boiling in multichannel heat sinks (Tubes and Channels)	N/A
Qu and Mudawar [309]	$q''_{CHF} = 33.43 Gi_{fg} \left(\frac{\rho_g}{\rho_l} \right)^{1.11} We_L^{-0.21} \left(\frac{L_{ch}}{D_{heat}} \right)^{-0.36}$	$D_h = 0.34 - 2.54$ mm Fluids (DI Water, R-113) $G = 86 - 1118$ kg/m²s Saturated boiling in multichannel heat sinks	Flow Reversal

Kuan [326]	$q''_{CHF} = 0.2305 G i_{fg} \left(\frac{L_{ch}}{D_{heat}} \right)^{-0.9056}$	$D_h = 0.27 - 0.33$ mm Fluids (DI Water, R-123) $G = 33.4 - 535.1$ kg/m ² s Saturated boiling in multichannel heat sinks	Dry-out spot formation during annular flow
Koşar and Peles [31]	$q''_{CHF} = G i_{fg} \left\{ \left[9.34 \times 10^{-2} \frac{P_{out}}{P_{cr}} - 0.34 \left(\frac{P_{out}}{P_c} \right)^2 - 1.3 \times 10^{-4} \right] x_{out}^{0.59} \right\}^{\frac{1}{1.08}}$	$D_h = 0.228$ mm Fluids (R-123) $G = 291 - 1118$ kg/m ² s Saturated boiling in multichannel heat sinks	Dry-out spot formation during annular flow
Kuan and Kandlikar [327]	$q''_{CHF} = C i_{fg} \sqrt{\rho_g} \sqrt{\frac{2\sigma \cos\theta}{H_{ch}} + \frac{G^2}{2\bar{\rho}}}$ Where $\bar{\rho} = \frac{x}{\rho_g} + \frac{1-x}{\rho_l}$	$D_h = 0.27 - 0.33$ mm Fluids (DI Water, R-123) $G = 33.4 - 535.1$ kg/m ² s Saturated boiling in multichannel heat sinks	Dry-out spot formation during annular flow

Ong and Thome [208]	$q''_{CHF} = 0.12 Gi_{fg} \left(\frac{\mu_l}{\mu_v} \right)^{0.183} \left(\frac{\rho_v}{\rho_l} \right)^{0.062} We_{lo}^{-0.141} \left(\frac{L_{ch}}{D_h} \right)^{-0.7} \left(\frac{D_h}{D_{th}} \right)^{0.11}$ Where $D_{th} = 2 \sqrt{\frac{\sigma}{g(\rho_l - \rho_g)}}$	$D_h = 1.03 - 3.04$ mm Fluids (R-134a, R-236fa, R-245fa) $G = 100 - 1500$ kg/m ² s Single Horizontal Channels	N/A
Anwar et al. [330]	$q''_{CHF} = 0.27 Gi_{fg} \left(\frac{D_h}{L_{heat}} \right)$	$D_h = 0.64 - 1.70$ mm Fluids (R-134a, R-1234yf, R-152a, R-22, R-245fa, R-290, R-600a) $G = 50 - 525$ kg/m ² s Stainless Steel Tubes	Dryout of Annular Film
Tibiriça et al. [186]	$q''_{CHF} = 0.242 Gi_{fg} \left(\frac{\rho_g}{\rho_l} \right)^{0.0598} We_{lo}^{-0.1635} \left(\frac{L_{ch}}{D_h} \right)^{-0.6834} (1 - x_{in})^{0.881} La_l^{-0.0714}$	$D_h = 0.24 - 6.92$ mm Fluids (Water, R-12, R-123, R-134a, R-236fa, R-245fa, R-1234ze(E), nitrogen) $G = 50 - 525$ kg/m ² s Tubes and microchannel heat sinks	N/A

APPENDIX D

75 MM LONG CHANNEL TEST SECTION ENGINEERING DRAWINGS

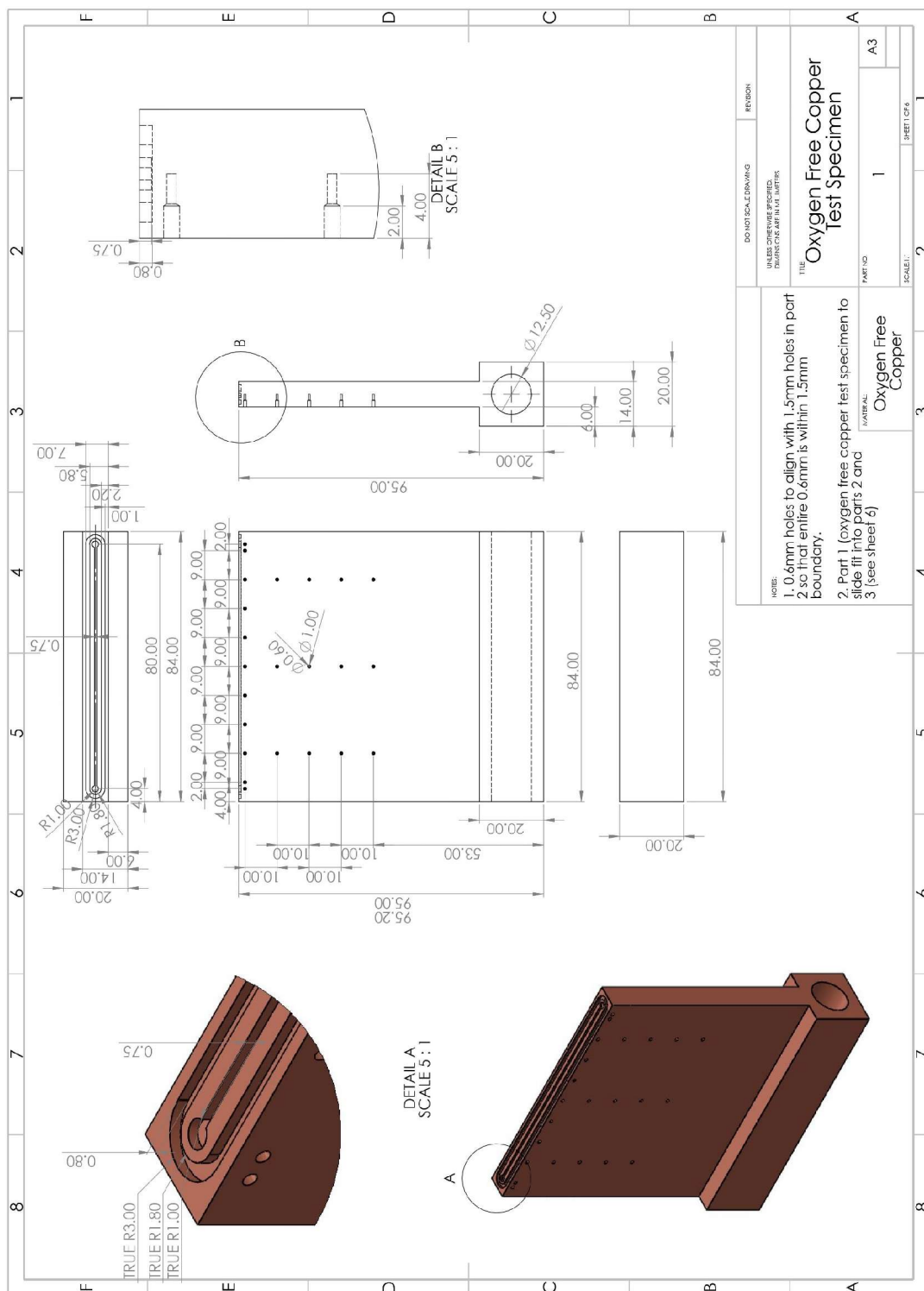


Figure D.1. Engineering drawing of oxygen free copper test section for 75 mm long channel test section (not to scale indicated on drawing).

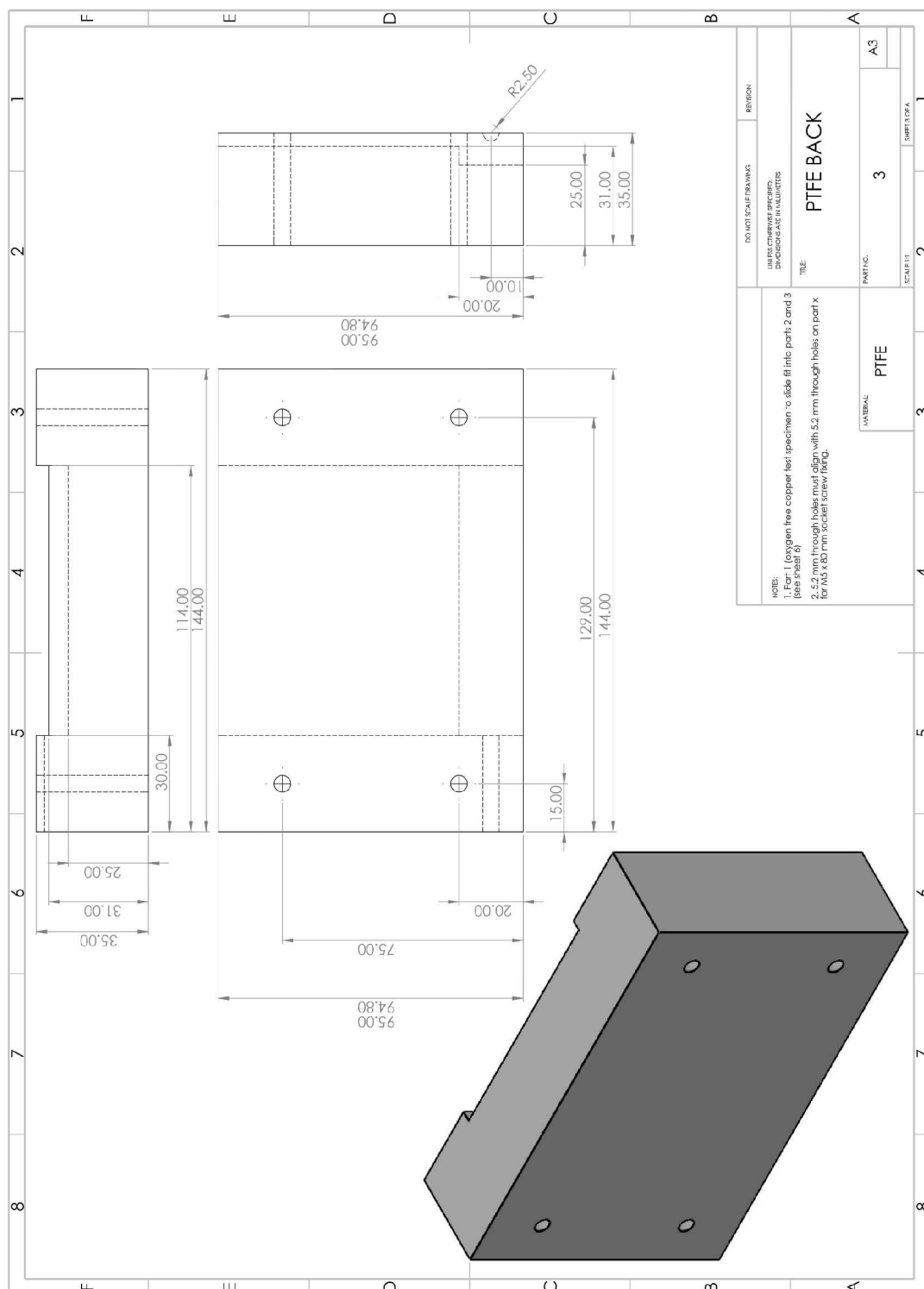


Figure D.3. Engineering drawing of PTFE back housing for 75 mm long test section (not to scale indicated on drawing).

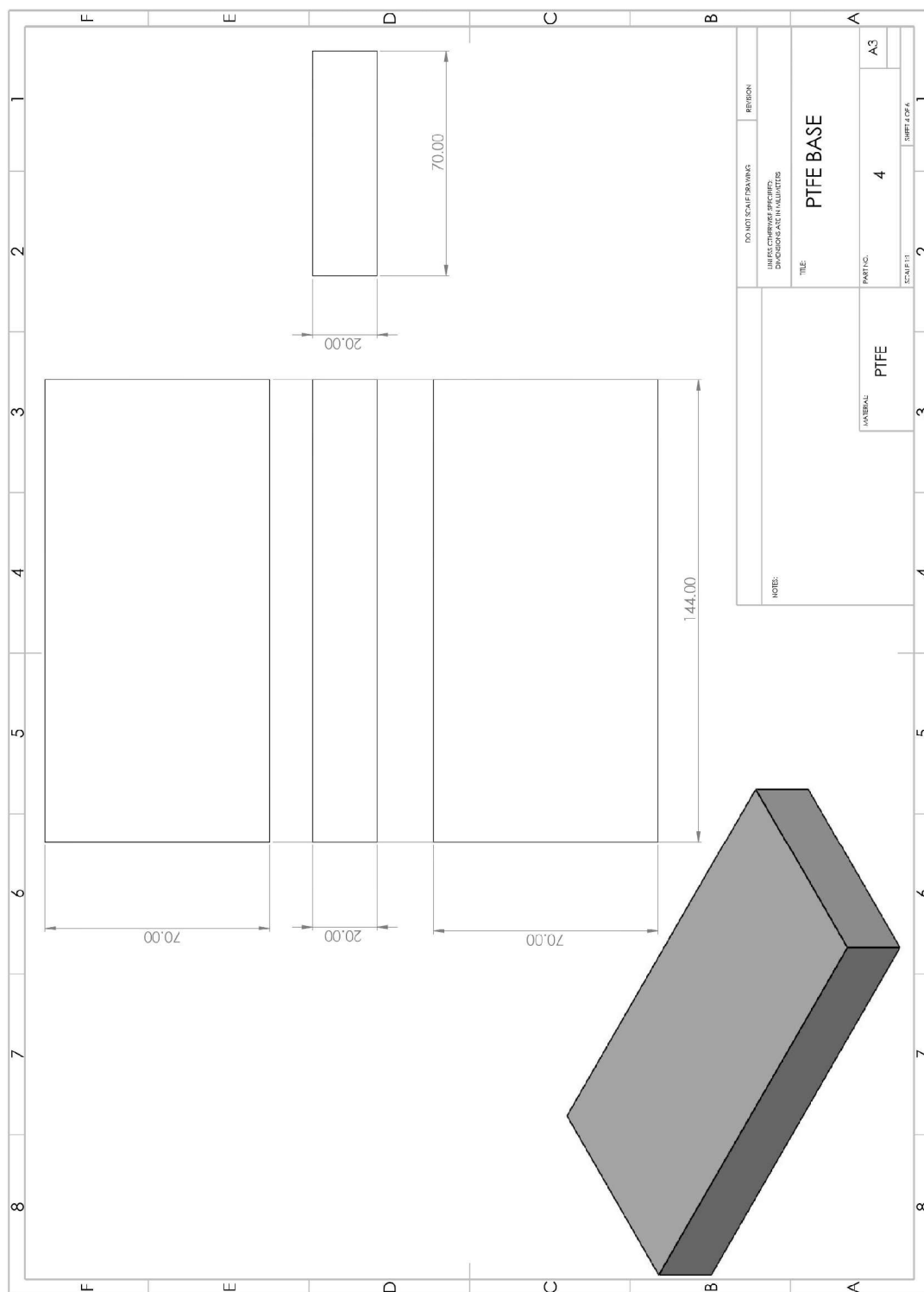


Figure D.4. Engineering drawing of PTFE bottom housing for 75 mm long test section (not to scale indicated on drawing).

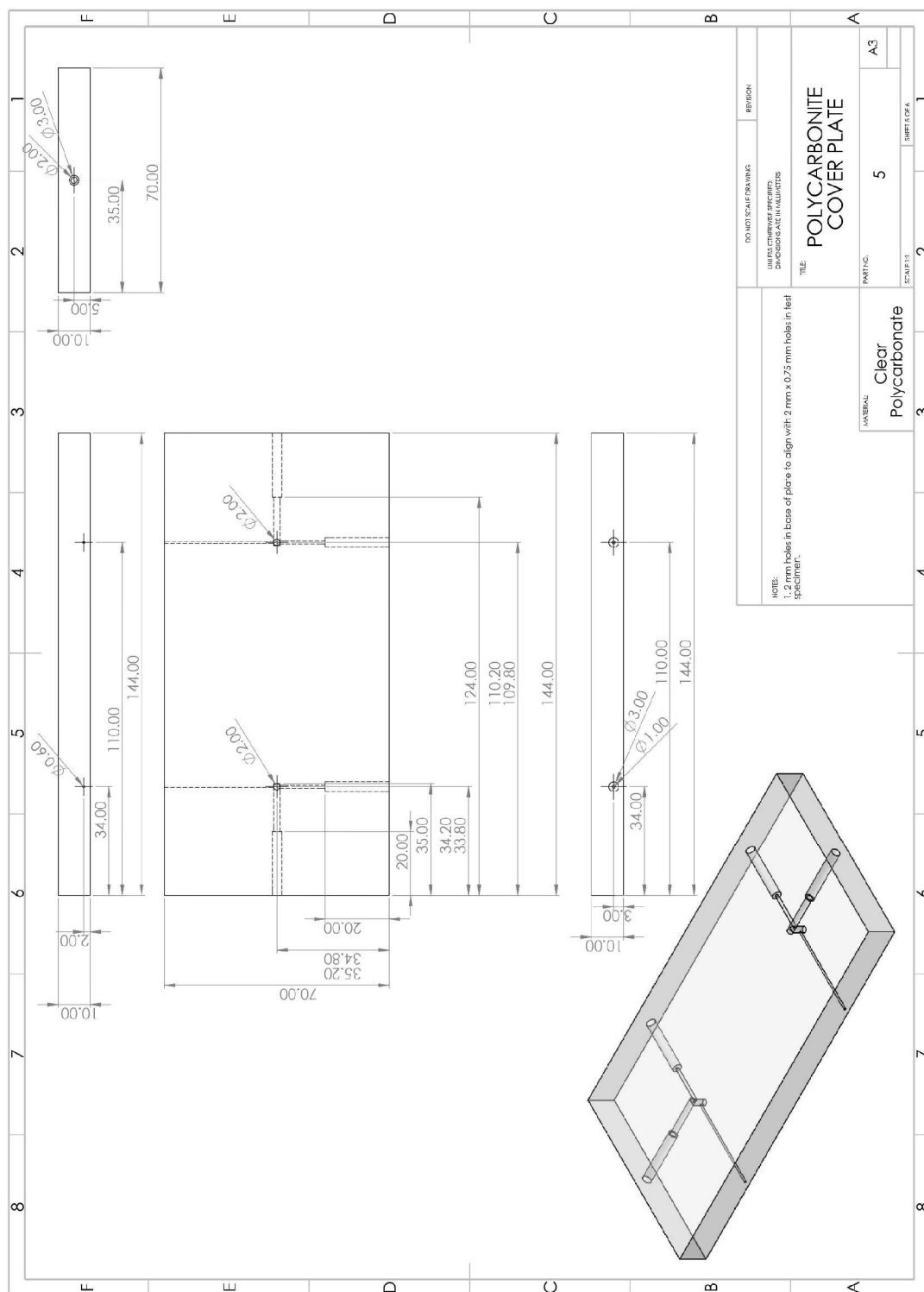


Figure D.5. Engineering drawing of clear polycarbonate cover plate for 75 mm long test section (not to scale indicated on drawing).

APPENDIX E

30 MM LONG CHANNEL TEST SECTION ENGINEERING DRAWINGS

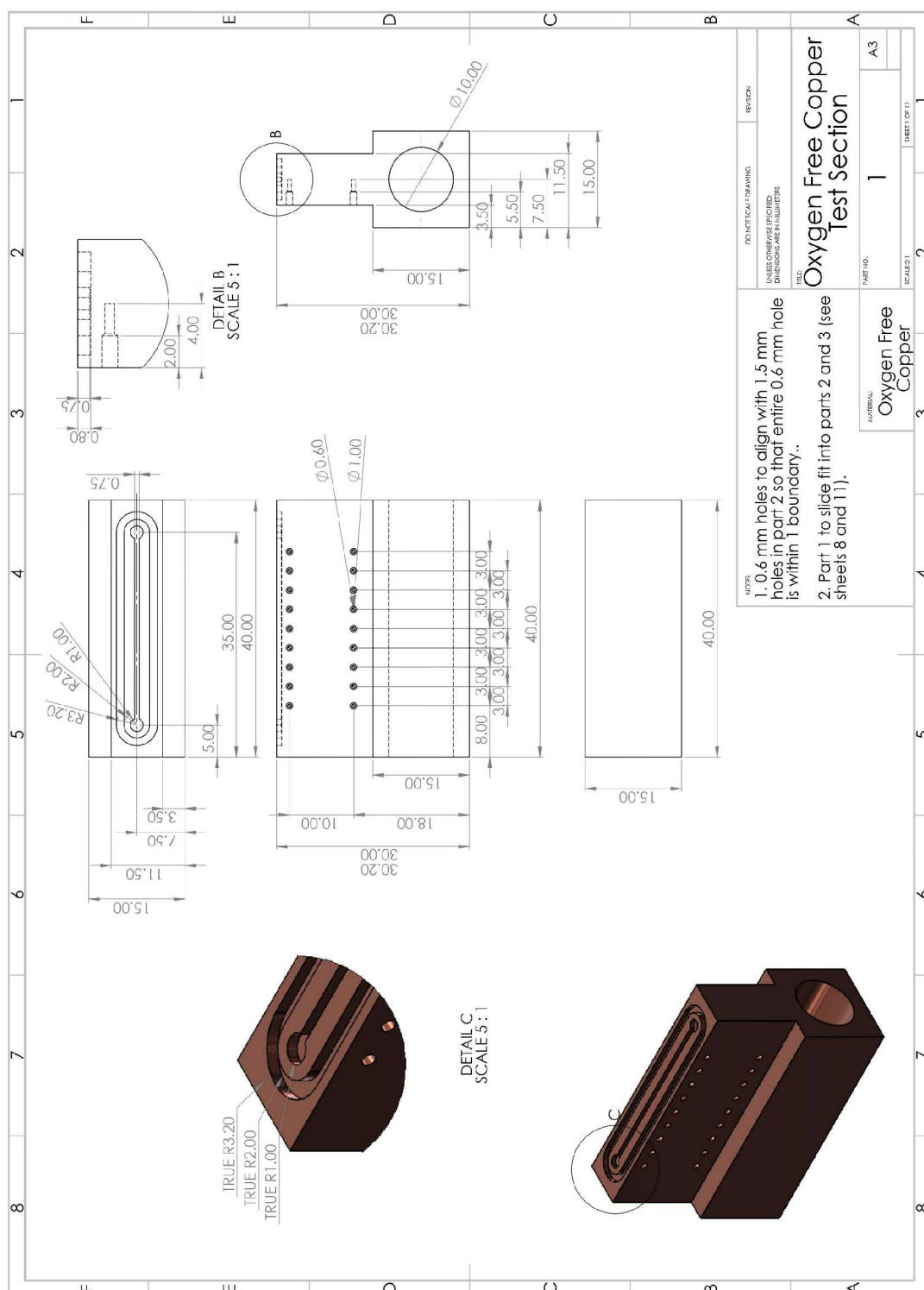


Figure E.1. Engineering drawing of oxygen free copper test section for 30 mm long test section (not to scale indicated on drawing).

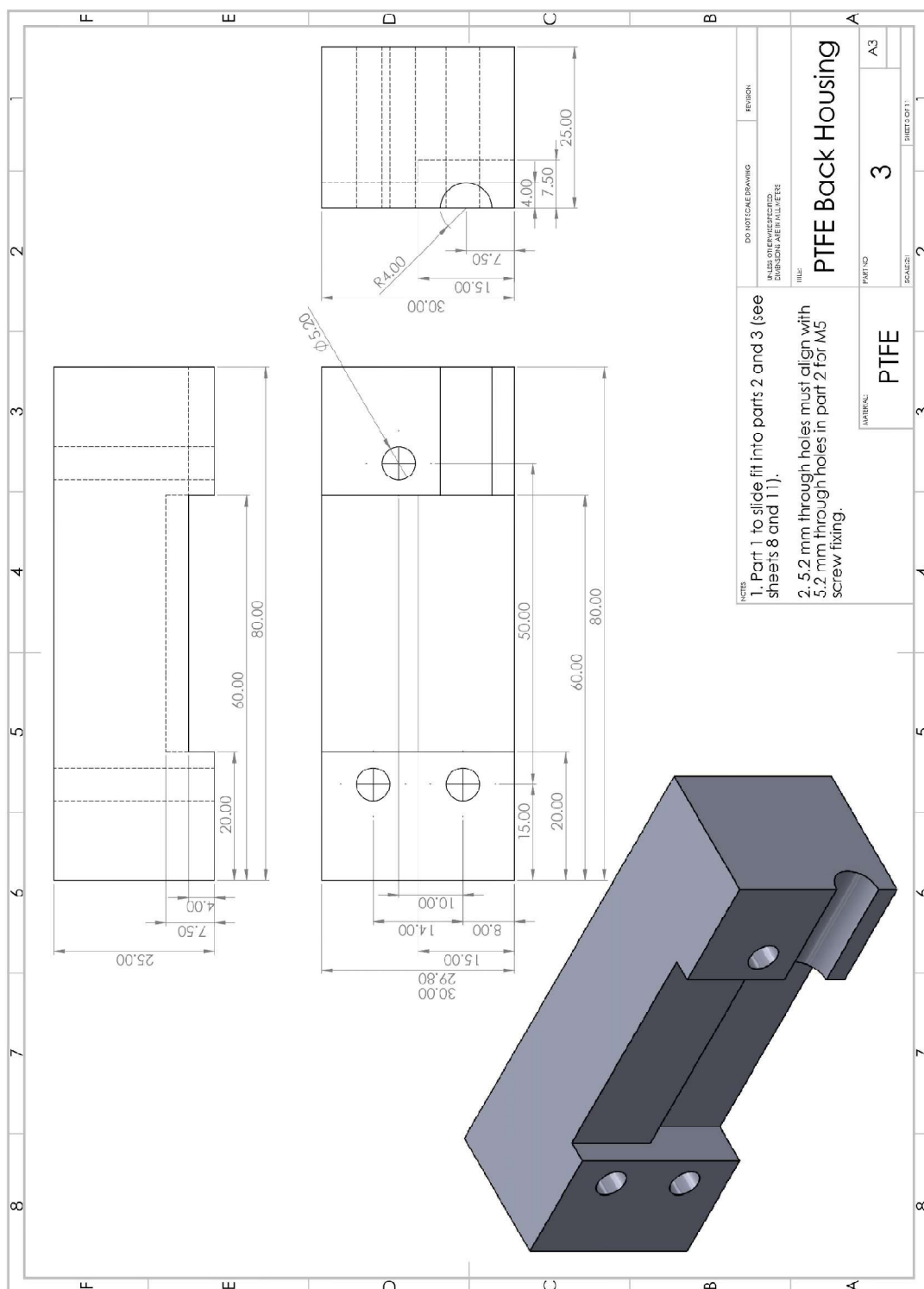


Figure E.3. Engineering drawing of PTFE back housing for 30 mm long test section (not to scale indicated on drawing).

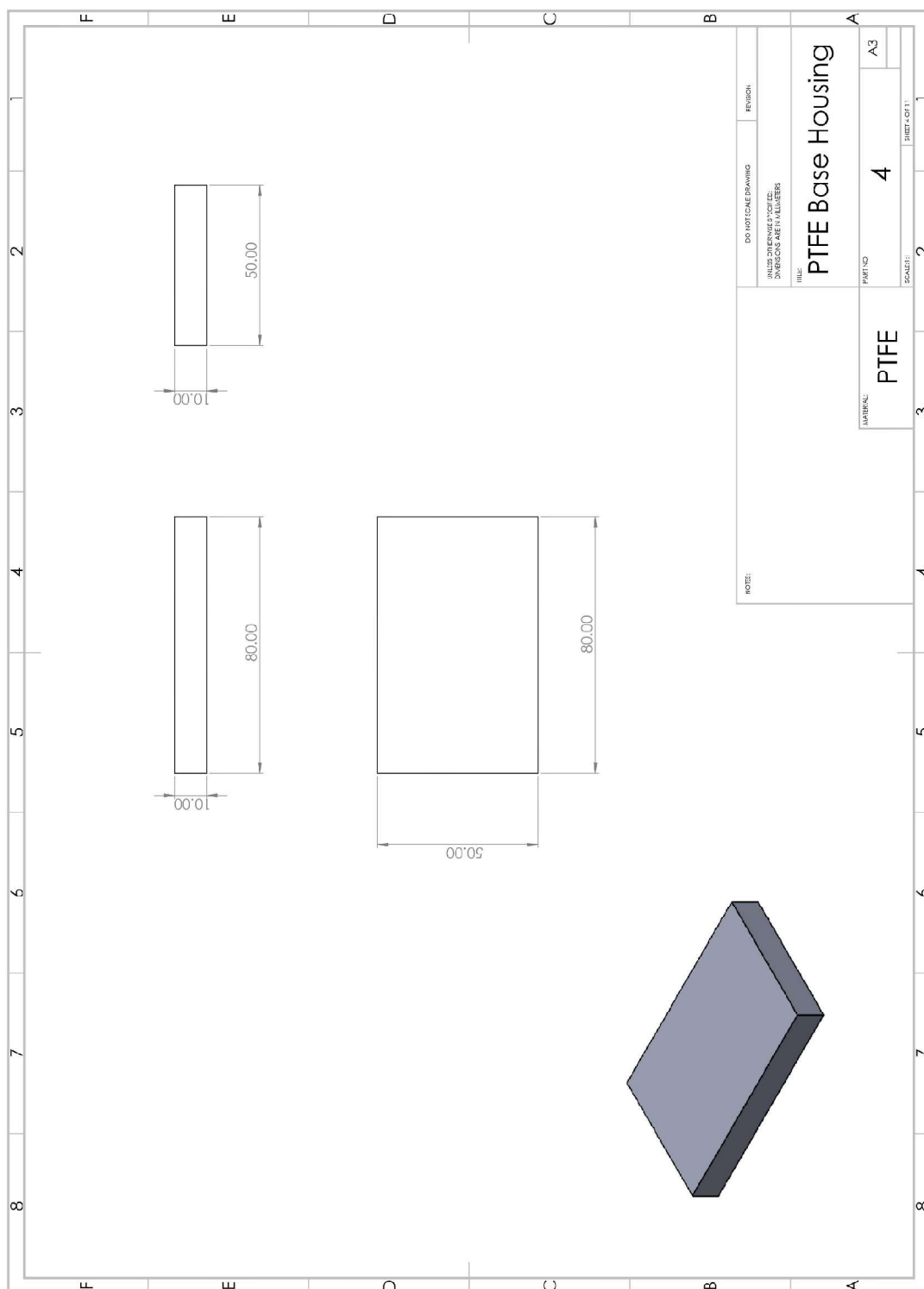


Figure E.4. Engineering drawing of PTFE bottom housing for 30 mm long test section (not to scale indicated on drawing).

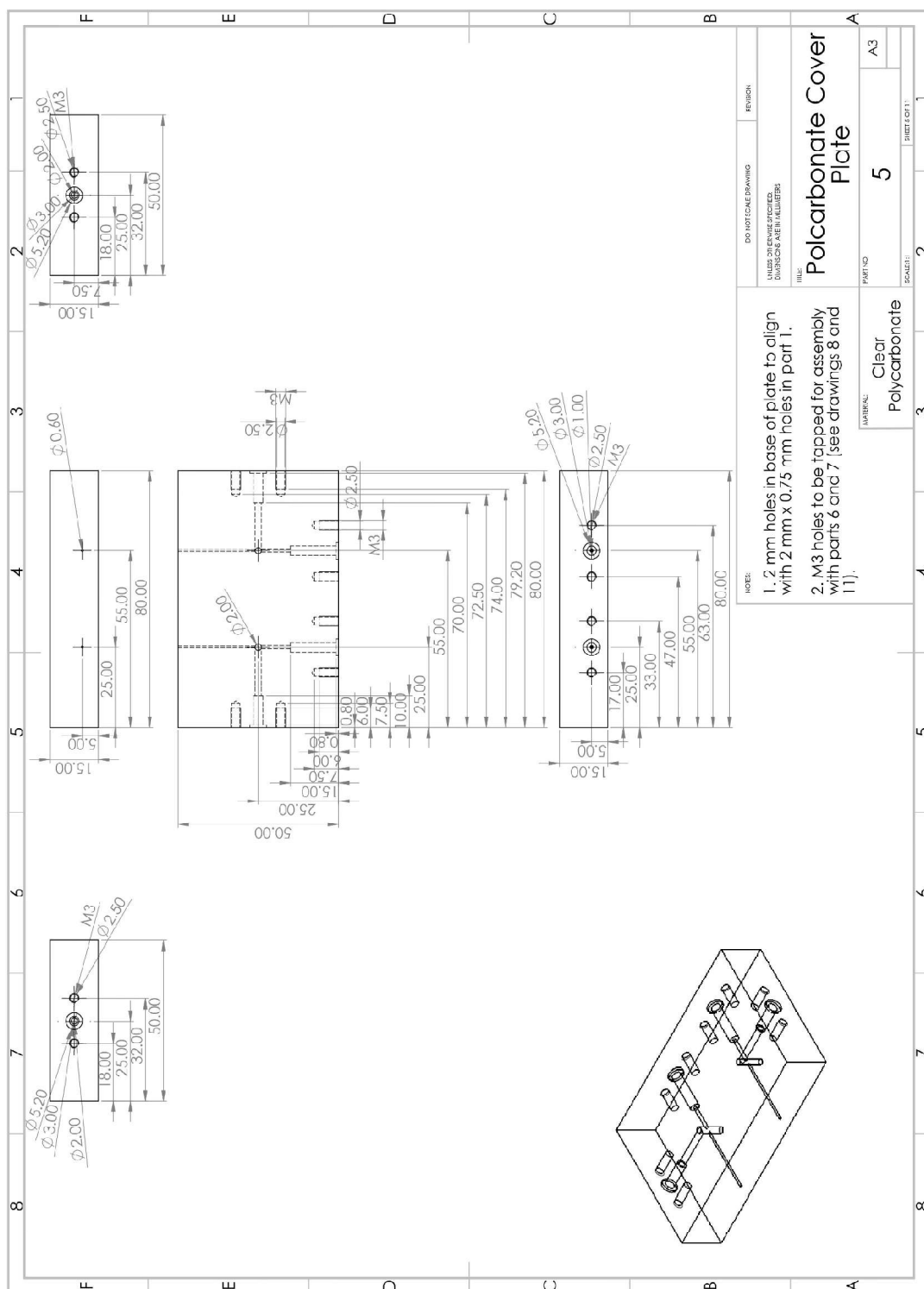


Figure E.5. Engineering drawing of clear polycarbonate cover plate for 30 mm long test section (not to scale indicated on drawing).

APPENDIX F

UNCERTAINTY PROPAGATION ANALYSIS

Uncertainty in the collected data arises from two primary sources: systematic errors linked to measurement instruments and random errors associated with inherent variability during experimentation. Systematic errors are addressed through careful calibration of all measuring devices, as explained in Section 3.7. Random errors, on the other hand, are mitigated by increasing the volume of recorded data to improve statistical reliability. For this investigation, data were sampled at 1 kHz over a 60-second interval, yielding 60,000 individual measurements for each data point. These measurements were then averaged to obtain representative values. Assuming that the data follow a Gaussian distribution, the random uncertainty, U_{random} was quantified using statistical techniques based on a 95 % confidence interval, following the methodology outlined by Coleman and Steele [386].

$$U_{random} = t_{95\%} S_x \quad (F.1)$$

$$S_x = \sqrt{\frac{1}{n-1} \sum_{i=1}^n X_i^2 - \bar{X}^2} \quad (F.2)$$

$$\bar{X} = \frac{1}{n} \sum_{i=1}^n X_i \quad (F.3)$$

where $t_{95\%}$ represents the critical value from the t-distribution corresponding to a 95 % confidence level, S_x denotes the standard deviation of the sample population, X_i is an individual measurement within the sample, and $\bar{X}$ is the sample mean.

Systematic uncertainty accounts for the bias uncertainty, B_x , associated with each component k . This value is typically determined using the equipment's technical specifications or calibration certificates. The corresponding standard deviation of the systematic uncertainty, S_{Bx} , is calculated as half of the bias uncertainty:

$$B_X = \sqrt{\sum_{k=1}^M (B_X)^2_k} \quad (\text{F.4})$$

$$S_{BX} = \frac{B_X}{2} \quad (\text{F.5})$$

The total uncertainty, U , is determined at a 95 % confidence level based on the combined standard uncertainty, u_c , which incorporates contributions from both random and systematic errors:

$$u_c = \sqrt{S_{BX}^2 + S_X^2} \quad (\text{F.6})$$

$$U = t_{95\%} u_c \quad (\text{F.7})$$

Consider an experimental variable r that is expressed as a function of j independent variables, X :

$$r = f(X_1, X_2, X_3 \dots X_j) \quad (\text{F.8})$$

The absolute and relative propagated uncertainties of the experimental variable, U_r , are given respectively by:

$$U_r^2 = \left(\frac{\partial r}{\partial X_1}\right) U_{X_1}^2 + \left(\frac{\partial r}{\partial X_2}\right) U_{X_2}^2 + \left(\frac{\partial r}{\partial X_3}\right) U_{X_3}^2 + \dots + \left(\frac{\partial r}{\partial X_j}\right) U_{X_j}^2 \quad (\text{F.9})$$

$$\frac{U_r^2}{r^2} = \left(\frac{X_1}{r} \frac{\partial r}{\partial X_1}\right) \left(\frac{U_{X_1}}{X_1}\right)^2 + \left(\frac{X_2}{r} \frac{\partial r}{\partial X_2}\right) \left(\frac{U_{X_2}}{X_2}\right)^2 + \left(\frac{X_3}{r} \frac{\partial r}{\partial X_3}\right) \left(\frac{U_{X_3}}{X_3}\right)^2 + \dots + \left(\frac{X_j}{r} \frac{\partial r}{\partial X_j}\right) \left(\frac{U_{X_j}}{X_j}\right)^2 \quad (\text{F.10})$$

The uncertainty analysis of the experimental variables in this study was conducted following the methodology described above. The resulting uncertainty values for the measured variables, derived from technical datasheets and calibration certificates, are summarised in Table 3.4.

The relative uncertainty of the channel hydraulic diameter is defined in Equation (F.11), where $U_{W_{ch}}$ and $U_{H_{ch}}$ are the absolute uncertainty of the channel width and height, respectively.

$$\frac{U_{D_h}}{D_h} = \sqrt{\left(\frac{H_{ch}}{H_{ch} + W_{ch}} \cdot \frac{U_{H_{ch}}}{H_{ch}}\right)^2 + \left(\frac{W_{ch}}{H_{ch} + W_{ch}} \cdot \frac{U_{W_{ch}}}{W_{ch}}\right)^2} \quad (\text{F.11})$$

The relative uncertainty of the mass flux is defined in Equation (F.12).

$$\frac{U_G}{G} = \sqrt{\left(\frac{U_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{U_{H_{ch}}}{U_{H_{ch}}}\right)^2 + \left(\frac{U_{W_{ch}}}{U_{W_{ch}}}\right)^2} \quad (\text{F.12})$$

The absolute uncertainty for the heat loss through the top plate, $U_{Q_{loss}}$, arises from the temperature difference between the top surface and the ambient environment. Since the product of $h_{top}A_{top}$ is treated as a measured exact coefficient, the uncertainty originates solely from the temperature measurements. This relationship is given by Equation (F.13).

$$U_{Q_{loss}} = h_{top}A_{top}\sqrt{(U_{T_{ts}})^2 + (U_{T_{\infty}})^2} \quad (\text{F.13})$$

The relative uncertainty of the base heat flux, q''_{base} , is evaluated from the conduction method, where the heat flux is proportional to the temperature gradient through the copper block. Considering the uncertainties in the temperature measurements and thermocouple spacing, the relative uncertainty is expressed by Equation (F.14).

$$\frac{U_{q''_{base}}}{q''_{base}} = \sqrt{\left(\frac{U_T}{\Delta T_{th}}\right)^2 + \left(\frac{U_{\Delta y}}{\Delta y}\right)^2} \quad (\text{F.14})$$

To determine the overall uncertainty in the wall heat flux, the uncertainties associated with the base heat flux, the heated area, and the top losses must be propagated. The uncertainty in the heated area of the base plate, A_{base} , is first evaluated using the measured height and width uncertainties, as shown in Equation (F.15).

$$\frac{U_{A_{base}}}{A_{base}} = \sqrt{\left(\frac{U_{H_{base}}}{H_{base}}\right)^2 + \left(\frac{U_{W_{base}}}{W_{base}}\right)^2} \quad (\text{F.15})$$

The relative uncertainty in the total input heat, Q_{in} , which is defined as $q''_{base}A_{base}$, is determined by combining the uncertainties in both the base heat flux and the base area, as given in Equation (F.16).

$$\frac{U_{Q_{in}}}{Q_{in}} = \sqrt{\left(\frac{U_{q''_{base}}}{q''_{base}}\right)^2 + \left(\frac{U_{A_{base}}}{A_{base}}\right)^2} \quad (\text{F.16})$$

The net heat transferred to the channel, Q_{ch} , is obtained by subtracting the top loss, Q_{loss} , from the total heat input, Q_{in} . The corresponding absolute uncertainty is determined by combining the uncertainties in these two quantities, as presented in Equation (F.17).

$$U_{Q_{ch}} = \sqrt{Q_{in}^2 \left\{ \left(\frac{U_{q''_{base}}}{q''_{base}} \right)^2 + \left(\frac{U_{A_{base}}}{A_{base}} \right)^2 \right\} + (h_{top} A_{top})^2 \left\{ (U_{T_{ts}})^2 + (U_{T_{\infty}})^2 \right\}} \quad (F.17)$$

$$\text{where } Q_{ch} = q''_{base} A_{base} - Q_{loss}$$

The relative uncertainty in the wall heat flux, q''_w , is then obtained by propagating the uncertainties in the net heat input and the geometric parameters of the channel; namely, the channel height, width, and length. This yields Equation (F.18), which represents the uncertainty in the wall heat flux based on the test section dimensions

$$\frac{U_{q''_w}}{q''_w} = \sqrt{\left(\frac{U_{Q_{ch}}}{Q_{ch}} \right)^2 + \left(\frac{2H_{ch}}{2H_{ch} + W_{ch}} \cdot \frac{U_{H_{ch}}}{H_{ch}} \right)^2 + \left(\frac{W_{ch}}{2H_{ch} + W_{ch}} \cdot \frac{U_{W_{ch}}}{W_{ch}} \right)^2 + \left(\frac{U_{L_{ch}}}{L_{ch}} \right)^2} \quad (F.18)$$

The uncertainty in the wall temperature is determined by propagating the uncertainties in the thermal gradient through the copper wall and the thermal conductivity of copper. Equation (F.19) expresses this relationship.

$$U_{T_{w(z)}} = \sqrt{(U_T)^2 + \left(\frac{\Delta y U_{q''_{base}}}{k_{cu}} \right)^2 + \left(\frac{q''_{base} U_{\Delta y}}{k_{cu}} \right)^2} \quad (F.19)$$

The inlet fluid temperature is corrected for the additional heat gained between the temperature measurement location and the actual channel inlet. The corresponding absolute uncertainty is obtained by combining the uncertainties in the measured inlet temperature, wall heat flux, and mass flow rate, as shown in Equation (F.20).

$$U_{T_{in}} = \sqrt{(U_{T_{in,meas}})^2 + \left(\frac{q''_w A_{pl,heat}}{\dot{m} C_{p,l}} \right)^2 \left\{ \left(\frac{U_{q''_w}}{q''_w} \right)^2 + \left(\frac{U_{\dot{m}}}{\dot{m}} \right)^2 \right\}} \quad (F.20)$$

The bulk fluid temperature along the channel is determined from an energy balance using the corrected inlet temperature and the local heat input. The uncertainty in arises from the uncertainties in the inlet temperature, wall heat flux, channel dimensions, mass flow rate, and axial position. The complete expression is presented in Equation (F.21).

$$U_{T_f(z)} = \left((U_{T_{in}})^2 + \left\{ \frac{q''_w (2H_{ch} + W_{ch})z}{\dot{m} C_{p,l}} \right\}^2 \left\{ \left(\frac{2H_{ch}}{2H_{ch} + W_{ch}} \cdot \frac{U_{H_{ch}}}{H_{ch}} \right)^2 + \left(\frac{W_{ch}}{2H_{ch} + W_{ch}} \cdot \frac{U_{W_{ch}}}{W_{ch}} \right)^2 + \left(\frac{U_z}{z} \right)^2 + \left(\frac{U_{q''_w}}{q''_w} \right)^2 + \left(\frac{U_{\dot{m}}}{\dot{m}} \right)^2 \right\} \right)^{0.5} \quad (F.21)$$

The uncertainty in the local temperature difference is obtained by combining the uncertainties in the wall and fluid temperatures, as given in Equation (F.22).

$$U_{\Delta T(z)} = \sqrt{U_{T_w(z)}^2 + U_{T_f(z)}^2} \quad (F.22)$$

$$\text{where } \Delta T(z) = T_w(z) - T_f(z)$$

The uncertainty in the local heat transfer coefficient is determined from the propagation of the uncertainties in the wall heat flux and the local temperature difference. This is expressed in Equation (F.23).

$$\frac{U_{h(z)}}{h(z)} = \sqrt{\left(\frac{U_{\Delta T(z)}}{\Delta T(z)} \right)^2 + \left(\frac{U_{q''_w}}{q''_w} \right)^2} \quad (F.23)$$

The uncertainty in the subcooled boiling length is calculated by combining the uncertainties in mass flow rate, channel dimensions, wall heat flux, and the temperature difference between the saturation and inlet temperatures. The resulting relationship is given by Equation (F.24).

$$\frac{U_{L_{sub}}}{L_{sub}} = \sqrt{\left(\frac{U_{\dot{m}}}{\dot{m}} \right)^2 + \left(\frac{2H_{ch}}{2H_{ch} + W_{ch}} \cdot \frac{U_{H_{ch}}}{H_{ch}} \right)^2 + \left(\frac{W_{ch}}{2H_{ch} + W_{ch}} \cdot \frac{U_{W_{ch}}}{W_{ch}} \right)^2 + \left(\frac{U_{q''_w}}{q''_w} \right)^2 + \left(\frac{U_{T_{in}}}{T_{sat,sub} - T_{in}} \right)^2} \quad (F.24)$$

The uncertainty in the local enthalpy is computed by propagating the uncertainties in the wall heat flux, channel geometry, and mass flow rate, as described in Equation (F.25).

$$U_{i(z)} = \left(\left\{ \frac{(2H_{ch} + W_{ch})z}{\dot{m}} U_{q''_w} \right\}^2 + \left\{ \frac{2q''_w z}{\dot{m}} U_{H_{ch}} \right\}^2 + \left\{ \frac{2q''_w z}{\dot{m}} U_{W_{ch}} \right\}^2 + \left\{ \frac{(2H_{ch} + W_{ch})q''_w}{\dot{m}} U_z \right\}^2 + \left\{ \frac{q''_w (2H_{ch} + W_{ch})z}{\dot{m}^2} U_{\dot{m}} \right\}^2 \right)^{0.5} \quad (F.25)$$

The relative uncertainty in the vapour quality is determined from the propagated uncertainty in the local enthalpy, as shown in Equation (F.26).

$$U_{x(z)} = \frac{U_{i(z)}}{i_{fg}(z)} \quad (\text{F.26})$$

The relative uncertainties in the superficial liquid and gas velocities are computed by combining the uncertainties in mass flux and vapour quality, as described in Equations (F.27) and (F.28), respectively.

$$\frac{U_{U_{ls}(z)}}{U_{ls}(z)} = \sqrt{\left(\frac{U_G}{G}\right)^2 + \left(\frac{U_{x(z)}}{1-x(z)}\right)^2} \quad (\text{F.27})$$

$$\frac{U_{U_{ls}(z)}}{U_{ls}(z)} = \sqrt{\left(\frac{U_G}{G}\right)^2 + \left(\frac{U_{x(z)}}{x(z)}\right)^2} \quad (\text{F.28})$$

The uncertainty in the two-phase pressure drop is obtained by propagating the uncertainties in the hydraulic diameter, mass flux, and subcooled boiling length, as shown in Equation (F.29).

$$\frac{U_{\Delta P_{sp}}}{\Delta P_{sp}} = \sqrt{\left(\frac{U_{D_h}}{D_h}\right)^2 + \left(\frac{2U_G}{G}\right)^2 + \left(\frac{U_{L_{sub}}}{L_{sub}}\right)^2} \quad (\text{F.29})$$

The total uncertainty in the measured pressure drop is determined by combining the uncertainties from the various flow regions (single-phase and two-phase), as expressed in Equation (F.30).

$$U_{\Delta P_{tp}} = \sqrt{U_{\Delta P_{ch}}^2 + U_{\Delta P_{sp}}^2} \quad (\text{F.30})$$

The overall pressure-drop uncertainty for the channel accounts for contributions from the inlet and outlet manifolds which is calculated based on the propagated uncertainties of Equations (3.2) to (3.5), (3.27) and (3.33).

$$U_{\Delta P_{ch}} = \sqrt{U_{\Delta P_{meas}}^2 + U_{\Delta P_{in,90}}^2 + U_{\Delta P_{sc}}^2 + U_{\Delta P_{se}}^2 + U_{\Delta P_{out,90}}^2} \quad (\text{F.31})$$

Finally, the uncertainty in the friction factor is obtained from the propagation of uncertainties in the hydraulic diameter, mass flux, channel length, and total pressure drop. This relationship is shown in Equation (F.32).

$$\frac{U_f}{f} = \sqrt{\left(\frac{U_{D_h}}{D_h}\right)^2 + \left(\frac{2U_G}{G}\right)^2 + \left(\frac{U_{L_{ch}}}{L_{ch}}\right)^2 + \left(\frac{U_{\Delta P_{ch}}}{\Delta P_{ch}}\right)^2} \quad (\text{F.32})$$

The overall resulting propagated uncertainties are presented in Table 3.5.

PUBLICATIONS

Conference Publications:

1. Widington, J. J., Ivanov, A. and Karayiannis, T.G. (2022), 'Review on Flow Boiling Patterns in Microchannels', *Therminic 2022*, Dublin, Ireland.- (Chapter 2)
2. Widington. J. J., Wang. F, Ivanov. A and Karayiannis. T.G. (2023), 'Predicting Microscale Bubble to Slug Transition Boundaries using an Artificial Neural Network', *International Conference on Boiling and Condensation Heat Transfer 11*, Edinburgh, UK. - (Chapter 8)
3. Widington. J. J., Ivanov. A, Karayiannis. T. G. (2023), 'Flow Boiling with HFE-7100 in a Single Microchannel', *8th Micro and Nano Flows Conference*, Padova, Italy. - (Chapter 5)
4. Widington. J. J., Karayiannis. T. G. (2026), 'Flow Boiling Pressure Drop in a Single Microchanel and Comparison with Correlations', *18th International Heat Transfer Conference*, Rio de Janeiro, Brazil. - (Chapter 6)

Journal Publications:

1. Widington. J. J, Ivanov. A, Karayiannis. T. G. (2025), 'Flow boiling heat transfer in a single microchannel and comparison with correlations', *Int. J. Heat Mass Transf.* 251 (2025) 127388. <https://doi.org/10.1016/J.IJHEATMASSTRANSFER.2025.127388>. - (Chapters 4 and 5)
2. Widington. J. J, Ali H. Al-Zaidi, Karayiannis. T. G. (2026), 'Flow Boiling Critical Heat Flux in Micro-scale Heat Sinks and Comparison with Correlations', *Applied Thermal Engineering (Under Review)*. - (Chapter 7)