

INFORMATION TO USERS

This manuscript has been reproduced from the microfilm master. UMI films the text directly from the original or copy submitted. Thus, some thesis and dissertation copies are in typewriter face, while others may be from any type of computer printer.

The quality of this reproduction is dependent upon the quality of the copy submitted. Broken or indistinct print, colored or poor quality illustrations and photographs, print bleedthrough, substandard margins, and improper alignment can adversely affect reproduction.

In the unlikely event that the author did not send UMI a complete manuscript and there are missing pages, these will be noted. Also, if unauthorized copyright material had to be removed, a note will indicate the deletion.

Oversize materials (e.g., maps, drawings, charts) are reproduced by sectioning the original, beginning at the upper left-hand corner and continuing from left to right in equal sections with small overlaps.

Photographs included in the original manuscript have been reproduced xerographically in this copy. Higher quality 6" x 9" black and white photographic prints are available for any photographs or illustrations appearing in this copy for an additional charge. Contact UMI directly to order.

ProQuest Information and Learning
300 North Zeeb Road, Ann Arbor, MI 48106-1346 USA
800-521-0600

UMI[®]

NOTE TO USERS

This reproduction is the best copy available.

UMI[®]



Université d'Ottawa • University of Ottawa

ANALYTICAL AND EXPERIMENTAL STUDIES OF CRITICAL HEAT FLUX IN COMPLEX GEOMETRY

by

GENG CHEN

A dissertation submitted to the School of Graduate Studies and Research of the Ottawa-Carleton Institute for Mechanical and Aeronautical Engineering at the University of Ottawa in partial fulfillment of the requirements for the degree of Doctor of Philosophy in Mechanical Engineering.

© Geng Chen

Ottawa-Carleton Institute for Mechanical and Aeronautical Engineering

University of Ottawa, Ottawa, Ontario, Canada K1N 6N5

2001 September



National Library
of Canada

Acquisitions and
Bibliographic Services

395 Wellington Street
Ottawa ON K1A 0N4
Canada

Bibliothèque nationale
du Canada

Acquisitions et
services bibliographiques

395, rue Wellington
Ottawa ON K1A 0N4
Canada

Your file Votre référence

Our file Notre référence

0-612-66137-7

The author has granted a non-exclusive licence allowing the National Library of Canada to reproduce, loan, distribute or sell copies of this thesis in microform, paper or electronic formats.

The author retains ownership of the copyright in this thesis. Neither the thesis nor substantial extracts from it may be printed or otherwise reproduced without the author's permission.

L'auteur a accordé une licence non exclusive permettant à la Bibliothèque nationale du Canada de reproduire, prêter, distribuer ou vendre des copies de cette thèse sous la forme de microfiche/film, de reproduction sur papier ou sur format électronique.

L'auteur conserve la propriété du droit d'auteur qui protège cette thèse. Ni la thèse ni des extraits substantiels de celle-ci ne doivent être imprimés ou autrement reproduits sans son autorisation.

Canada

ABSTRACT

A systematic study on the CHF in complex geometry has been carried out using a dumbbell shaped tube. The dumbbell shaped flow channel consists of two subchannels interconnected via a narrow gap, which simulates the two adjacent subchannels in a nuclear fuel bundle. The study was divided into two parts: mechanistic modeling and experimental investigation.

Over 620 CHF data were obtained in a dumbbell-shaped tube with R-134a as working fluid. A wide range of flow conditions was covered. The gap size of dumbbell shaped tube was varied from 0, 0.15, 0.3, 0.5, 0.7, to 1 mm. This experimental investigation is one of the most systematic researches on the gap effect. It is found that the CHF trends against thermalhydraulic parameters in the dumbbell shaped tube are similar to those in circular tubes. No severe adverse effect was detected in the dumbbell-shaped tube under the test range of this study. The CHF decreases with decreasing gap size. On the average, as gap size decreases from 1 mm to zero CHF drops about 17.2% at a fixed inlet condition and about 11.5% at a fixed local condition.

An analytical model for complex geometry, the partition model, was developed based on the annular flow assumption. The partition model divides a complex geometry cross section into two logical channels. Instead of making a complex two-dimensional description, the partition model calculates the complex geometry CHF by analyzing flow and heat transfer in two interactive one-dimensional flow channels. A new approach for the geometry effect on deposition was developed. A software package CHFA++ was developed based on the partition model to predict the CHF in complex geometries. The partition model predicts the CHF in dumbbell-shaped tubes and the gap effect with good accuracy.

Several recommendations were made for further research on complex geometry CHF, including the optimization of the partition model, extending the application of the partition model to other complex geometries, and conducting experiments on the liquid film distribution.

ACKNOWLEDGEMENT

I wish to express my sincere appreciation to Professor S.C. Cheng, my co-supervisor, for his motivating guidance and his continuous availability for constructive discussions. His quick and careful editing of the manuscript and his many comments and suggestions contributed greatly to the quality of the thesis.

I am deeply grateful to Dr. D.C. Groeneveld, my co-supervisor, for his valuable advice, recommendations, and encouragement during this study. His world class expertise in the critical heat flux, two-phase flow and heat transfer contributed significantly to the success of this research project.

My appreciation is extended to many of my colleagues and friends at the Thermalhydraulic Research Team in the University of Ottawa, for their help and for providing an enjoyable working atmosphere. A special thank is addressed to Dr. D. Vaitekunas. His valuable advice, timely suggestions and careful editing of the manuscript are very helpful. A special word of appreciation is also due to our research fellows, A. Vasic and Dr. I. Pioro, for their assistance in experimental investigation.

I would like to take this opportunity to thank the Natural Sciences and Engineering Research Council of Canada for its financial support of this study and the University of Ottawa for its post graduate scholarship.

The comments and suggestions provided by the Advisory Committee for my Ph.D. program (consisting of Dr. C. Snoek of National Resources Canada, Dr. E. Plett of Carleton University, Dr. W. Hallett and Dr. R. Milane of University of Ottawa) are also appreciated.

Finally, I am indebted to my family for their patience, understanding, and constant moral support through these many years of Ph.D. study.

TABLE OF CONTENTS

ABSTRACT	i
ACKNOWLEDGEMENT	ii
TABLE OF CONTENTS	iii
LIST OF FIGURES.....	x
LIST OF TABLES	xiv
NOMENCLATURE.....	xv
Chapter 1	1-1
INTRODUCTION.....	1-1
1.1 Background.....	1-1
1.2 Statement of Problem.....	1-3
1.3 Objectives and Scope.....	1-6
Chapter 2	2-1
CRITICAL HEAT FLUX IN FLOW BOILING	2-1
2.1 Flow Boiling	2-1
2.1.1 Regimes of Heat Transfer in a Vertical Heated Tube.....	2-1
2.1.2 The Boiling Map.....	2-5
2.2 Flow Boiling CHF.....	2-7
2.2.1 CHF in a Subcooled or Low-Quality Region	2-8
2.2.2 CHF in a High-Quality Region.....	2-10
2.3 Parametric Effects on CHF	2-11
2.3.1 General.....	2-11
2.3.2 Effects of Pressure	2-13
2.3.3 Effect of Mass Flux.....	2-14
2.3.4 Effect of Subcooling	2-14
2.3.5 Effect of Quality	2-14
2.3.6 Effect of Geometry	2-15
2.3.6.1 Effect of Heated Length on CHF	2-15
2.3.6.2 Effect of Tube Diameter on CHF.....	2-15
2.4 Prediction of CHF	2-15
2.4.1 Empirical Methods.....	2-16
2.4.1.1 Lookup Table Approach.....	2-16
2.4.1.2 Correlation.....	2-16
2.4.2 Mechanistic Methods	2-17
2.4.2.1 Dryout-Type CHF Modeling.....	2-17
2.4.2.2 DNB Type CHF Modeling.....	2-18

Chapter 3	3-1
LITERATURE REVIEW.....	3-1
3.1 Experimental Studies	3-1
3.1.1 Concentric Annuli	3-2
3.1.1.1 Gap Effect	3-2
3.1.1.1.1 General Trend	3-2
3.1.1.1.2 Mechanism of Gap Effect.....	3-4
3.1.1.2 Curvature Effect	3-5
3.1.1.3 Bilateral Heating	3-5
3.1.1.4 Prediction of CHF	3-5
3.1.1.4.1 Lookup Table Approach.....	3-5
3.1.1.4.2 Correlation Method.....	3-6
3.1.2 Eccentric Annuli	3-8
3.1.2.1 Eccentricity Effect.....	3-9
3.1.2.2 Minimum Gap Effect	3-9
3.1.2.3 Void Migration Effect	3-10
3.1.2.4 Prediction of CHF	3-11
3.1.3 Rectangular Ducts	3-12
3.1.3.1 General Trend.....	3-12
3.1.3.2 Prediction of CHF	3-13
3.1.4 Dumbbell-Shaped Channels.....	3-14
3.1.5 Fuel bundles.....	3-18
3.2 Modeling Studies	3-19
3.2.1 Whalley's Model.....	3-20
3.2.2 Saito's Model	3-26
3.2.2.1 Physical Model	3-26
3.2.2.2 Mechanisms Review of CHF in Annular Two-phase Regime	3-27
3.2.2.3 Formulation	3-27
3.2.2.4 Modeling Results.....	3-29
3.2.3 Levy's Model	3-30
3.2.4 Govan's Model.....	3-30
3.2.5 Lim's model.....	3-31
3.2.6 Sugawara's Model	3-32
Chapter 4	4-1
EXPERIMENTAL STUDY	4-1
4.1 General	4-1
4.2 Dumbbell-Shaped Tube	4-2
4.2.1 Geometry Parameters.....	4-2
4.2.2 Flow Area.....	4-3
4.3 Working Fluid.....	4-4
4.4 Fluid-to-Fluid Modeling of CHF	4-6
4.5 Flow Loop.....	4-7
4.5.1 General	4-8
4.5.2 System Parameters and Measurements	4-8

4.5.2.1	Pressure	4-8
4.5.2.2	Mass Flux	4-8
4.5.2.3	Inlet Subcooling	4-8
4.5.2.4	Heat Flux	4-10
4.5.3	Test section	4-10
4.5.3.1	General	4-10
4.5.3.2	Minimum Gap Size Control and Measurement.....	4-10
4.5.3.3	Monitoring of CHF	4-12
4.5.4	Contamination Control.....	4-14
4.5.5	System Leak Testing.....	4-14
4.5.6	System Charging and Discharging.....	4-15
4.5.7	Heat Balance Testing	4-15
4.6	Data Acquisition System.....	4-15
4.7	Test Procedure	4-16
4.8	Test Conditions	4-16
4.9	Flow Visualization	4-17
4.10	Data Reduction	4-17
4.11	Error Analysis.....	4-18
Chapter 5		5-1
MODELING STUDY		5-1
5.1	Partition Model	5-1
5.1.1	Introduction.....	5-1
5.1.2	Basic Assumptions.....	5-2
5.1.3	Partition Assumptions.....	5-4
5.1.4	Partitioning Flow Channels.....	5-4
5.1.4.1	Partitioning a Circular Tube	5-4
5.1.4.2	Partitioning a Dumbbell-Shaped Tube	5-5
5.1.5	Dumbbell Flow Channel Assumptions	5-6
5.1.5.1	Two-phase Flow Regimes in Dumbbell-Shaped Tube	5-7
5.1.5.2	Circumferential Distribution of Liquid Film Thickness	5-8
5.1.5.3	Zero Gap Case.....	5-11
5.2	Formulation.....	5-12
5.2.1	Conservation of Mass	5-12
5.2.2	Conservation of momentum.....	5-14
5.2.3	Conservation of Energy	5-15
5.2.4	Entrainment and Deposition Models	5-16
5.2.4.1	Entrainment Process in Simple Geometry	5-16
5.2.4.2	Deposition Process in Simple Geometry.....	5-19
5.2.4.3	Deposition Model for Complex Geometry.....	5-20
5.2.4.4	Entrainment Model for Complex Geometry	5-21
5.2.4.5	Deposition Factor and Shape Factor	5-22
5.2.5	Interfacial Shear Stress	5-25
5.2.6	Triangular relationship.....	5-25
5.2.7	Frictional Pressure Gradient	5-27
5.3	Computing Model	5-27

5.3.1	General.....	5-27
5.3.2	Two-Phase Flow and Heat Transfer System.....	5-27
5.3.3	Subsystems.....	5-28
5.3.3.1	Kernel.....	5-30
5.3.3.1.1	Partition Model.....	5-30
5.3.3.1.2	Node.....	5-30
5.3.3.1.3	Log.....	5-30
5.3.3.1.4	Flow Pattern.....	5-31
5.3.3.2	Application.....	5-31
5.3.3.2.1	Geometry.....	5-32
5.3.3.2.2	Fluid.....	5-32
5.3.4	Scheme of CHF Computing.....	5-33
5.3.4.1	General.....	5-33
5.3.4.2	Increments.....	5-35
Chapter 6		6-1
EXPERIMENTAL RESULTS AND ANALYSIS		6-1
6.1	General Trend.....	6-1
6.1.1	Inlet Condition.....	6-2
6.1.1.1	Inlet Enthalpy/Subcooling Effect.....	6-2
6.1.1.2	Mass Flux Effect at Fixed Inlet Condition.....	6-5
6.1.1.3	Pressure Effect at Fixed Inlet Condition.....	6-8
6.1.2	Local Condition.....	6-10
6.1.2.1	Quality Effect at Fixed Local Condition.....	6-10
6.1.2.2	Mass Flux Effect at Fixed Local Condition.....	6-17
6.1.2.3	Pressure Effect at Fixed Local Condition.....	6-17
6.2	Gap Effect.....	6-21
6.2.1	Overall Geometry Effect on CHF.....	6-21
6.2.1.1	Overall Geometry Effect at Fixed Inlet Condition.....	6-21
6.2.1.2	Geometry Effect on CHF at Fixed Mass Flow Rate.....	6-24
6.2.1.3	Overall Geometry Effect at Fixed Local Condition.....	6-27
6.2.2	Gap Effect on CHF.....	6-29
6.2.2.1	Approach to Isolate Gap Effect.....	6-29
6.2.2.2	Flow Area Effect.....	6-29
6.2.2.2.1	Hydraulic Diameter Effect on CHF at Fixed Local Condition.....	6-30
6.2.2.2.2	Hydraulic Diameter Effect on CHF at Fixed Inlet Condition.....	6-31
6.2.2.3	Gap Effect.....	6-33
6.2.2.3.1	Gap Effect on CHF at Fixed Inlet Condition.....	6-33
6.2.2.3.2	Gap Effect on CHF at Fixed Local Condition.....	6-35
6.2.3	CHF Location.....	6-38
6.3	Comparison Against Circular Tube Data.....	6-38
6.3.1	Inlet Condition.....	6-38
6.3.2	Local Condition.....	6-41
6.4	Flow Visualization.....	6-41
6.5	Summary.....	6-42
Chapter 7		7-1

MODELING RESULTS AND ANALYSIS	7-1
7.1 Partition Model Application to Circular Tube	7-1
7.1.1 Steam-Water Flow Case	7-1
7.1.1.1 Whalley's Annular Flow Model Results	7-2
7.1.1.2 Partition Model Results.....	7-6
7.1.2 R-134a Flow Cases	7-7
7.1.2.1 General Trend.....	7-7
7.1.2.2 Mass Flux Influence on Model Accuracy	7-9
7.1.2.3 Inlet Subcooling Influence on Model Accuracy	7-10
7.1.2.4 Variations of Flow Parameters.....	7-11
7.2 Partition Model Application to Dumbbell-Shaped Tube	7-15
7.2.1 Deposition Factors for Dumbbell-Shaped Tube	7-15
7.2.2 General Trend	7-16
7.2.2.1 Fixed Inlet Condition	7-16
7.2.2.1.1 Inlet Enthalpy (Subcooling) Effect.....	7-16
7.2.2.1.2 Mass Flux Effect at Fixed Inlet Condition	7-16
7.2.2.1.3 Pressure Effect at Fixed Inlet Condition.....	7-17
7.2.2.2 Fixed Local Condition.....	7-23
7.2.2.2.1 Quality Effect	7-23
7.2.2.2.2 Mass Flux Effect at Fixed Local Condition.....	7-23
7.2.2.2.3 Pressure Effect at Fixed Local Condition.....	7-26
7.2.3 Gap Effect.....	7-27
7.2.3.1 Inlet Condition.....	7-27
7.2.3.2 Local Condition.....	7-31
7.2.3.3 CHF Location.....	7-34
7.2.4 Comparison Against Test Results.....	7-34
7.2.4.1 Boundary Condition	7-35
7.2.4.2 CHF Location.....	7-37
7.2.4.3 General Trend.....	7-37
7.2.4.4 Gap Effect	7-39
7.3 Partition Model Application to Non-uniform Heating Case.....	7-44
7.3.1.1 Partition Model Approach for Non-uniform Heating	7-44
7.3.1.2 Results and Analysis	7-44
7.3.1.2.1 CHF Location	7-44
7.3.1.2.2 General Trend	7-45
7.3.1.2.3 Gap Effect.....	7-45
7.4 Summary	7-46
Chapter 8	8-1
CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE RESEARCH. 8-1	
8.1 Conclusions.....	8-1
8.1.1 Experimental Investigation	8-1
8.1.2 Modeling study	8-2
8.2 Recommendations for Future Research.....	8-2
8.2.1 Experimental Investigation	8-2
8.2.2 Modeling Study.....	8-3

8.2.2.1	Kernel	8-3
8.2.2.2	Application	8-4
REFERENCES		1
Appendix I		I-1
Dumbbell-Shaped Test Section		I-1
I.1	Construction of Dumbbell-Shaped Test Section	I-1
I.2	Gap Size	I-1
I.3	Cross Section of Dumbbell-Shaped Tube	I-4
I.4	Temperature and Pressure Effects on the Shape of Test Section	I-5
Appendix II		II-1
ERROR ANALYSIS		II-1
II.1	Introduction	II-1
II.2	Uncertainties in Basic Measurements	II-1
II.2.1	Temperature	II-1
II.2.2	Pressure	II-2
II.2.3	Mass Flow Rate	II-2
II.2.4	Power	II-3
II.2.4	Geometry Parameters	II-4
II.2.4.1	Heated Length	II-4
II.2.4.2	Perimeter	II-4
II.2.4.3	Gap Size	II-4
II.2.4.4	Flow Area	II-5
II.3	Inlet Subcooling	II-5
II.4	Mass Flux	II-6
II.5	Quality	II-7
II.6	Critical Heat Flux	II-8
Appendix III		III-1
DUMBBELL-SHAPED TUBE CHF EXPERIMENTAL DATA		III-1
Appendix IV		IV-1
DUMBBELL-SHAPED TUBE CHF MODELING DATA		IV-1
Appendix V		V-1
PREDICTED CHF FOR NON-UNIFORM HEATING AND COMPARISON WITH THE RESULTS OF TEST AND UNIFORM HEATING		V-1
Appendix VI		VI-1
CIRCULAR TUBE CHF EXPERIMENTAL AND MODELING DATA		VI-1

LIST OF FIGURES

Figure 1.1	Schematic of CANDU system (from http://www.ncf.carleton.ca/~cz725/).....	1-2
Figure 1.2	Terms used to denote CHF condition phenomenon.	1-3
Figure 1.3	Candu fuel bundle (from http://www.ncf.carleton.ca/~cz725/).....	1-5
Figure 1.4	Subchannels and rods in a CANDU fuel bundle.	1-5
Figure 1.5	Dumbbell-shaped tube.....	1-6
Figure 2.1	Regions of heat transfer in convective boiling (from Collier and Thome, 1994).	2-2
Figure 2.2	Variation of heat transfer coefficient with quality with increasing heat flux as parameter (from Collier and Thome, 1994).....	2-5
Figure 2.3	Forced convective boiling surface (from Collier and Thome, 1994)	2-6
Figure 2.4	Regions of two-phase convective heat transfer as a function of quality with increasing heat flux as ordinate (from Collier and Thome, 1994).....	2-7
Figure 2.5	Comparison of CHF mechanisms in various flow patterns (from Tong and Tang, 1997).....	2-8
Figure 2.6	Mechanisms of DNB in bubbly flows (from Tong and Tang, 1997).	2-9
Figure 2.7	Mechanisms of dryout annular flows (from Tong and Tang, 1997).	2-10
Figure 2.8	Parametric effects on CHF at fixed inlet condition (from Whalley, 1987).	2-12
Figure 2.9	Parametric effects on CHF at fixed local condition (from Whalley, 1987).	2-13
Figure 2.10	Annular flow in a circular tube (from Whalley, 1977).....	2-18
Figure 3.1	Effect of gap size on CHF in concentric annuli (from Doerffer et al., 1994).....	3-3
Figure 3.2	Annular flow (from Whalley, 1977).....	3-20
Figure 3.3	Variation of deposition coefficient with surface tension (from Whalley, 1977).....	3-22
Figure 3.4	Variation of equilibrium concentration of entrained droplets with $\tau_i \delta / \sigma$ (from Whalley, 1977).	3-23
Figure 3.5	Film dryout model in FIDAS code (from Sugawara, 1990a).	3-33
Figure 4.1	Dumbbell-shaped tube cross section.	4-1
Figure 4.2	Dumbbell-shaped tube geometry parameters.	4-2
Figure 4.3	Dumbbell-shaped flow area calculation.	4-4
Figure 4.4	Schematic of the test loop.....	4-9

Figure 4.5	Sketch of the dumbbell-shaped test section.....	4-11
Figure 4.6	The chain for measuring minimum gap size.	4-12
Figure 4.7	Thermocouples for monitoring CHF.	4-13
Figure 4.8	Assembly sketch of rod, clamps and thermocouples.....	4-14
Figure 5.1	Partition of circular tube.	5-5
Figure 5.2	Partition of dumbbell-shaped tube.....	5-6
Figure 5.3	Zero gap case.	5-11
Figure 5.4	Mass balance on a control volume of a pseudo-channel.	5-12
Figure 5.5	Entrainment process in a side view (from Whalley, 1987).	5-17
Figure 5.6	Entrainment process in an axial view (from Whalley, 1987).	5-18
Figure 5.7	Droplet trajectories (redraw based on Whalley, 1987).	5-19
Figure 5.8	Calculation of shape factor.	5-23
Figure 5.9	Object diagram for two-phase flow and heat transfer.	5-28
Figure 5.10	Partition model subsystem hierarchy.....	5-29
Figure 5.11	Node states.....	5-31
Figure 5.12	Fluids.	5-33
Figure 5.13	General scheme of CHF computing.	5-34
Figure 6.1	Inlet enthalpy effect on CHF in dumbbell-shaped tube.....	6-3
Figure 6.2	Inlet enthalpy effect on CHF in dumbbell-shaped tube (continued).	6-4
Figure 6.3	Mass flux effect on CHF in dumbbell-shaped tube (inlet condition, 1130kPa). ..	6-6
Figure 6.4	Mass flux effect on CHF in dumbbell-shaped tube (inlet condition, 1660kPa). ..	6-7
Figure 6.5	Pressure effect on CHF in dumbbell-shaped tube (inlet condition).	6-9
Figure 6.6	Exit quality effect on CHF in dumbbell-shaped tube (1130kPa, 1000kg/m ² s). .	6-11
Figure 6.7	Exit quality effect on CHF in dumbbell-shaped tube (1130kPa, 2000kg/m ² s). .	6-12
Figure 6.8	Exit quality effect on CHF in dumbbell-shaped tube (1130kPa, 3000kg/m ² s). .	6-13
Figure 6.9	Exit quality effect on CHF in dumbbell-shaped tube (1660kPa, 1000kg/m ² s). .	6-14
Figure 6.10	Exit quality effect on CHF in dumbbell-shaped tube (1660kPa, 2000kg/m ² s). .	6-15
Figure 6.11	Exit quality effect on CHF in dumbbell-shaped tube (1660kPa, 3000kg/m ² s). .	6-16
Figure 6.12	Mass flux effect on CHF in dumbbell-shaped tube (local condition, 1130kPa). .	6-18
Figure 6.13	Mass flux effect on CHF in dumbbell-shaped tube (local condition, 1660kPa). .	6-19
Figure 6.14	Pressure effect on CHF in dumbbell-shaped tube at fixed local condition.	6-20

Figure 6.15	Geometry effect on CHF at fixed inlet condition (1130kPa).	6-22
Figure 6.16	Geometry effect on CHF at fixed inlet condition (1660kPa).	6-23
Figure 6.17	Geometry effect on CHF in dumbbell-shaped tube at fixed mass flow rate.	6-25
Figure 6.18	Geometry effect on CHF in dumbbell-shaped tube at fixed local condition.	6-27
Figure 6.19	Hydraulic diameter effect on CHF at fixed local condition.	6-30
Figure 6.20	Flow chart for computing the CHF at a given inlet condition.	6-31
Figure 6.21	Gap effect on CHF in dumbbell-shaped tube at fixed local condition.	6-35
Figure 6.22	Variation of CHF ratio with gap size at fixed inlet condition.	6-37
Figure 6.23	Variation of CHF ratio with gap size at fixed local condition.	6-38
Figure 7.1	Variation of liquid film flow rate along the tube axis.	7-3
Figure 7.2	Variation of liquid film thickness along the tube axis.	7-4
Figure 7.3	Variation of interfacial shear stress along the tube axis.	7-5
Figure 7.4	Variation of deposition rate along the tube axis.	7-5
Figure 7.5	Variation of entrainment rate along the tube axis.	7-6
Figure 7.6	Predicted CHF for circular tube ($G = 1000 \text{ kg/m}^2\text{s}$).	7-8
Figure 7.7	Predicted CHF for circular tube ($G = 2000 \text{ kg/m}^2\text{s}$).	7-8
Figure 7.8	Predicted CHF for circular tube ($G = 3000 \text{ kg/m}^2\text{s}$).	7-9
Figure 7.9	Variation of deposition rate along the tube axis (R-134a).	7-12
Figure 7.10	Variation of entrainment rate along the tube axis (R-134a).	7-13
Figure 7.11	Variation of liquid film thickness along the tube axis (R-134a).	7-14
Figure 7.12	Variation of liquid film mass flow rate along the tube axis (R-134a).	7-14
Figure 7.13	Inlet enthalpy/subcooling effect on the CHF in dumbbell-shaped tube ($G = 1000 \text{ kg/m}^2\text{s}$).	7-18
Figure 7.14	Inlet enthalpy/subcooling effect on the CHF in dumbbell-shaped tube ($G = 2000 \text{ kg/m}^2\text{s}$).	7-19
Figure 7.15	Inlet enthalpy/subcooling effect on the CHF in dumbbell-shaped tube ($G = 3000 \text{ kg/m}^2\text{s}$).	7-20
Figure 7.16	Mass flux effect on the CHF in dumbbell-shaped tube at fixed inlet condition.	7-21
Figure 7.17	Pressure effect on the CHF in dumbbell-shaped tube at fixed inlet condition.	7-22
Figure 7.18	Exit quality effect on the CHF in dumbbell-shaped tube.	7-24
Figure 7.19	Mass flux effect on CHF in dumbbell-shaped tube at fixed local condition.	7-25

Figure 7.20	Pressure effect on CHF in dumbbell-shaped tube at fixed local condition.	7-26
Figure 7.21	Geometry effect on CHF in dumbbell-shaped tube (inlet condition, $1000\text{kg/m}^2\text{s}$).	7-28
Figure 7.22	Geometry effect on CHF in dumbbell-shaped tube (inlet condition, $2000\text{kg/m}^2\text{s}$).	7-29
Figure 7.23	Geometry effect on CHF in dumbbell-shaped tube (inlet condition, $3000\text{kg/m}^2\text{s}$).	7-30
Figure 7.24	Geometry effect on CHF in dumbbell-shaped tube (local condition, 1130kPa).	7-32
Figure 7.25	Geometry effect on CHF in dumbbell-shaped tube (local condition, 1660kPa).	7-33
Figure 7.26	Variation of liquid film flow rate along the tube axis.	7-34
Figure 7.27	Difference of heat flux in the dumbbell-shaped tube.	7-35
Figure 7.28	Comparison against experimental data at fixed inlet condition.	7-38
Figure 7.29	Comparison of predicted and measured geometry effect at fixed inlet condition.	7-41
Figure 7.30	Comparison of predicted and measured gap effect at fixed local condition.	7-42
Figure I.1	Dumbbell-shaped test section.	I-2
Figure I.2	Die for dumbbell-shaped tube.	I-3
Figure I.3	Variation of gap size against turning angle.	I-4

LIST OF TABLES

Table 3.1	CHF correction factors for concentric annuli	3-6
Table 3.2	Correction factors for eccentric annuli (from Doerffer et al., 1994)	3-11
Table 3.3	Coefficients for Macbeth (1963) correlation	3-14
Table 3.4	CHF data for gap size of 0.76 mm (from Green et al., 1969).....	3-15
Table 3.5	CHF data for gap size of 0.38 mm (from Green et al., 1969).....	3-16
Table 3.6	Variation of deposition coefficient with surface tension	3-22
Table 4.1	Test matrix	4-17
Table 6.1	$CHF_{gap=0}/CHF_{gap=1mm}$ at pressure of 1130 kPa	6-24
Table 6.2	$CHF_{gap=0}/CHF_{gap=1mm}$ at pressure of 1660 kPa	6-24
Table 6.3	Water equivalent flow parameters	6-29
Table 6.4	$CHF_{d=3.43mm}/CHF_{d=4.30mm}$ at pressure of 1130 kPa	6-32
Table 6.5	$CHF_{d=3.43mm}/CHF_{d=4.30mm}$ at pressure of 1660 kPa	6-32
Table 6.6	Summary of experimental results	6-40
Table 7.1	Comparison of CHF prediction data at fixed inlet condition.....	7-2
Table 7.2	Deposition factors for the dumbbell-shaped tube	7-15
Table 7.3	Heat fluxes of the dumbbell-shaped tube.....	7-36
Table 7.4	Prediction errors of partition model.....	7-45
Table 7.5	Summary of modeling results for uniform heating	7-45
Table 7.6	Summary of modeling results for non-uniform heating	7-46
Table I.1	Dumbbell-shaped tube characteristic geometric parameters.	I-4
Table I.2	Temperature and pressure effects on the shaped of dumbbell tube.	I-5

NOMENCLATURE

Symbol	Description	Unit
A	Flow area	m ²
A	Correlating parameter	-
B	Correlating parameter	-
C	Correlating parameter	-
C	Droplet concentration	kg/m ³
C _E	Equilibrium droplet concentration	kg/m ³
C _{eq}	Equilibrium droplet concentration	kg/m ³
CHF	Critical heat flux	W/m ²
CHF _o	Circular tube CHF	W/m ²
C _p	Specific heat	J/kg°C
D	Diameter	m
D	Deposition rate of liquid droplets	kg/m ² s
D _{eq}	Equivalent diameter	m
D _H	Hydraulic/heated diameter	m
D _{he}	Heated diameter	m
D _{hy}	Hydraulic diameter	m
D _i	Inner tube diameter of annulus	m
D _o	Outer tube diameter of annulus	m
d	Diameter	m
d _{eq}	Equivalent diameter	m
d _{he}	Heated diameter	m
d _{hy}	Hydraulic diameter	m
d _o	Circular tube diameter	m
E	Entrainment rate	kg/m ² s
F	Shape factor	-
f	Fanning friction factor	-
G	Mass flux	kg/m ² s
g	Constant of gravitational acceleration	m/s ²
h	Enthalpy	J/kg
h _{exit}	Enthalpy at the exit of test section	J/kg
h _f	Saturate liquid enthalpy	J/kg
h _{in}	Enthalpy at the inlet of test section	J/kg
h _{fg}	Latent heat	J/kg
h _i	Inlet enthalpy	J/kg
Δh _{sub}	Inlet subcooling	J/kg
i	Inlet enthalpy	J/kg
Δi _{sub}	Inlet subcooling	J/kg
K	Correlating parameter	-
k	Deposition rate coefficient	m/s
k _p	Pressure correction factor	-
k _x	Quality correction factor	-
k _δ	Gap size correction factor	-

L	Length of test section	m
Le	Lewis number	-
Δl_{eq}	Equivalent height of roll waves	
l_p	Perimeter	m
M	Model	
P	Pressure	Pa
Pr	Prandtl number	-
q''	Heat flux	kg/m ²
R	Radius	m
Re	Reynolds number	
r	Radius	m
Sc	Schmidt number	-
T	Temperature	°C
u	Actual velocity	m/s
V	Superficial velocity	m/s
W	Mass flow rate	kg/s
x	Quality	
$y_0, y_1, \dots, y_5$	Correlating parameter	-
z	Location	m
Greek		
α	Void fraction	-
α	Angle extended to a surface	Deg
β	Entrainment parameter	-
γ	Void migration parameter	-
δ	Gap size	m
δ	Film thickness	m
ε	Eccentricity	-
θ	Angle	
μ	Dynamic viscosity	kg/ms
ν	Kinematic viscosity	m ² /s
ρ	Density	kg/m ³
σ	Surface tension	N/m
τ	Shear stress	Pa
ϕ^2	Two-phase frictional multiplier	
ϕ	Deposition factor	
Subscripts		
1	Logical channel 1	
2	Logical channel 2	
C	Main flow channel region	
conc	Concentric	
D	Deposition	
D=8mm	Reference tube of 8 mm in diameter	

E	Equilibrium, entrainment, entrained droplets
eccn	Eccentric
eq	Equivalent, equilibrium
exit	Exit
F	Liquid film
G	Gap region
he	Heated
hy	Hydraulic
i	Internal; inner, interfacial
in	Inlet
L	Saturated liquid
LE	Entrained liquid droplets
LF	Liquid film
M	Model
p	Perimeter
o	Outlet, outer
sat	Saturation
sub	Subcooled condition
W	Wall
W	Water

Abbreviations

AECL	Atomic Energy of Canada Limited
CANDU	CANada Deuterium Uranium
CHF	Critical Heat Flux
DNB	Departure from Nucleate Boiling
I.D.	Inner Diameter
NOM	NOMinal
O.D.	Outer Diameter
OMT	Object Modeling Technique
ONAF	ONset of Annular Flow
RMS	Root Mean Square

Chapter 1

INTRODUCTION

1.1 Background

The boiling heat transfer process, by which a liquid phase is converted to a vapor phase, is of key importance in a wide range of applications such as water-cooled nuclear reactors and power plant boilers. In general, boiling is an extremely efficient mode of heat transfer. However, if the surface temperature or heat flux is high enough, the boiling heat-transfer process breaks down, and a severe deterioration in heat transfer can occur. The term ‘critical heat flux condition’ is used to denote the state of the system at which the characteristic reduction in heat transfer coefficient has just occurred. The maximum heat flux just before this state is called the “critical heat flux (CHF)”. For a system in which the surface heat flux is controlled, the critical heat flux condition is characterized by a sharp increase in surface temperature. Likewise, for a system in which the surface temperature is controlled, a considerable reduction in heat flux will result when the CHF condition is reached.

Critical heat flux plays a critical role in the design of thermal power systems. It is essentially the maximum power at which the system can be operated. Exceeding this limit may result in a sudden rise in surface temperature due to the sharp drop in heat transfer coefficient. In many practical cases this large temperature jump is sufficient to cause failure of the heating surface.

One such thermal power system is the CANDU system illustrated in Figure 1.1. In this system, heavy water, which works as both primary coolant and moderator, is forced to circulate through the CANDU reactor core (fuel bundle) and steam generator (boiler). The pressurized heavy water transfers the heat produced by uranium fission in the reactor core to ordinary light water flowing in secondary loop. Heat applied to ordinary water in the secondary loop produces steam, which drives the turbine, which in turn drives the generator to produce electricity. The

evaluation of CHF is crucial in water-cooled reactor design since it limits the operating power. In order to maintain a proper safety margin during reactor operations, accurate prediction of the CHF value is required.

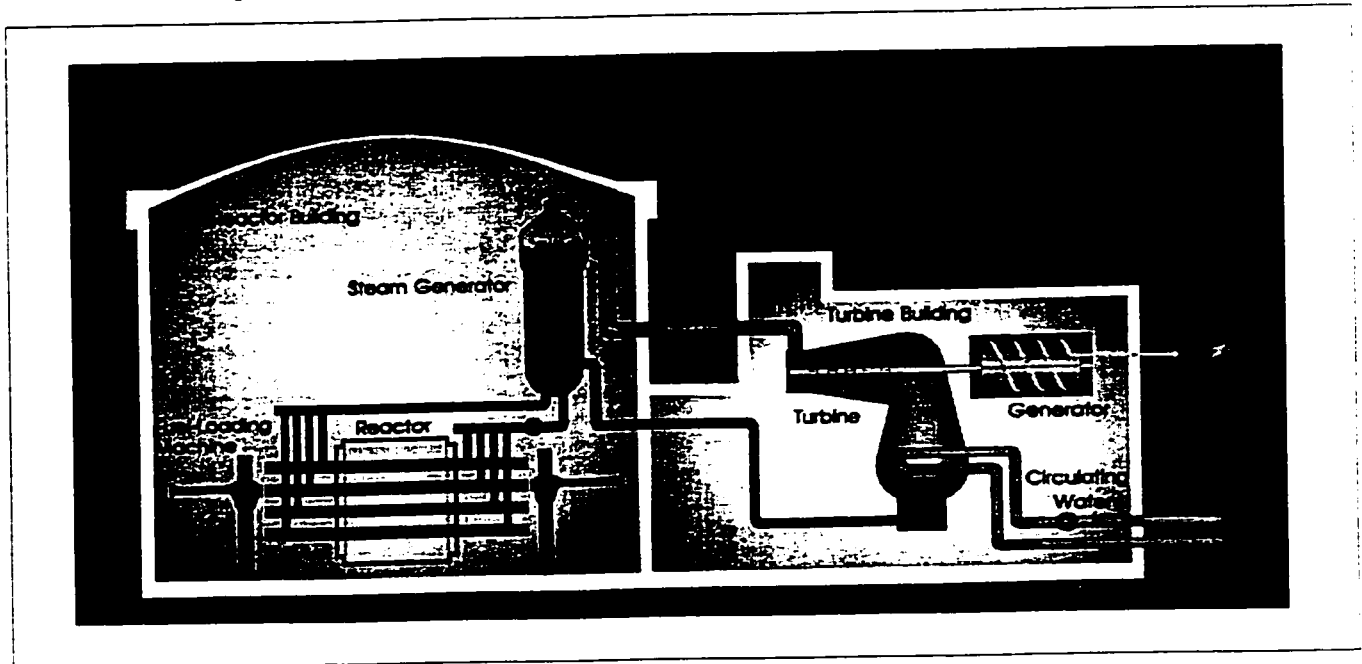


Figure 1.1 Schematic of CANDU system (from <http://www.ncf.carleton.ca/~cz725/>).

The considerable economic importance of predicting CHF has resulted in a vast amount of literature on the subject. Many terms other than CHF have been used to denote this phenomenon. These include boiling crisis, burnout, dryout, and DNB (departure from nucleate boiling). CHF condition is the most widely accepted term because it is a general description. In the non-English language literature, 'boiling crisis' is extensively used to refer to this phenomenon. Burnout is the common name but is consequence-focused. This implies a physical destruction of the heated surface. In fact, failure of the heated surface may not always occur once the CHF is exceeded.

Dryout and DNB are mechanism-oriented and imply definite mechanisms for particular flow patterns. A critical heat flux condition occurring in an annular flow is usually called dryout or dryout-type CHF; a critical heat flux condition occurring in a bubbly flow is usually called departure from nucleate boiling (DNB) or DNB type CHF. Figure 1.2 summarizes these terms.

The terms in the top layer are general while the terms in bottom layer denote a particular type of the CHF.

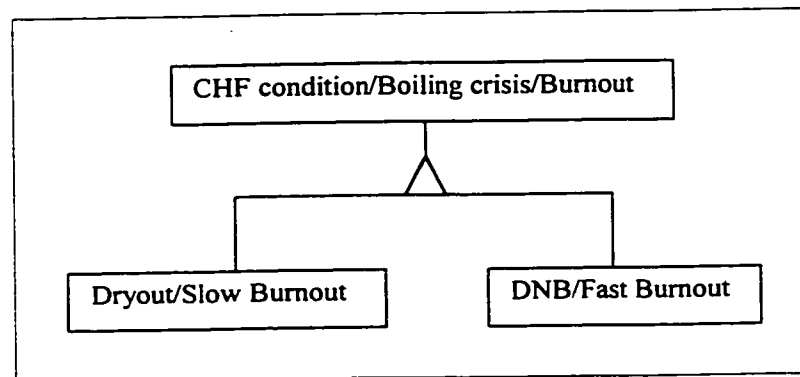


Figure 1.2 Terms used to denote CHF condition phenomenon.

1.2 Statement of Problem

In the last four decades, with the increased demand in the thermal loading of water-cooled nuclear reactors and power stations, the CHF has been studied extensively. Hundreds of correlations and numerous models to predict CHF have been developed. Because of the complexity of the CHF, most prediction methods are based on a round tube. For other geometries of interest, such as subchannel shapes or rod bundles, current CHF prediction methods are often based on extensions to the round tube or empirical correlations derived from circular tubes. The effects of the geometry difference are largely ignored.

In the nuclear industry, the application of CHF tube correlations to the subchannel geometry is a possible source of large errors in CHF prediction; hence large safety margins are required to compensate for these potential errors. The effect of geometry on CHF becomes more important if considering geometry changes due to pressure-tube creep, fuel strain and fuel-element bowing during normal operation or postulated loss-of-regulation accident conditions. However, no universal method is yet available for the precise prediction of CHF for industrial applications.

Current difficulties in developing a reliable prediction method for CHF in a complex geometry like the rod-bundle are mainly attributed to a lack of understanding of the geometry effect. The geometric configuration of the flow channel can significantly affect the flow regime of the two-

phase mixture, which in turn affects the rate of inter-phase heat, mass, and momentum transfer, and thus affects CHF. Different from the effect of thermalhydraulic parameters (e.g., quality, mass flux, etc.) on CHF, which can be investigated in simple tubular geometry, the geometry effect is much more difficult to study due to the large number of parameters involved. For the case of rod-bundle geometries, any variation in each of the characteristic dimensions (e.g., inter-element gap size, element/pressure-tube gap size, curvature of the surface, subchannel shape) must be considered. Moreover, the resultant effect of such a geometry may change with different flow conditions.

In addition, full-scale CHF experiments on each fuel-bundle geometry would typically cost from \$2 to \$5 million. The cost of experimentally investigating the geometry effects in full-scale simulated fuel-bundle geometries is also prohibitive. Moreover, these full-scale CHF tests are sometimes risky. In practice, CHF measurements are normally performed in out-of-reactor loops, using a similar test-section geometry. To obtain an accurate value of the CHF at reactor conditions of interest, these test sections must simulate the reactor as closely as possible.

A 37-rod fuel bundle from the CANDU reactor is shown in Figure 1.3. The rod bundle's cross section is usually divided into a series of subchannels. A traditional approach to define the subchannel is to join lines between adjacent rod centers. Figure 1.4 shows two adjacent subchannels. In this study, a dumbbell-shaped tube is used to investigate the effect of such geometry on CHF. As shown in Figure 1.5, the dumbbell-shaped channel consists of two subchannels interconnected via a narrow gap. Its shape is close to the two adjacent subchannels shown in the nuclear fuel-rod bundle. The narrow gap region simulates the gap between two adjacent rods.



Figure 1.3 CANDU fuel bundle (from <http://www.ncf.carleton.ca/~cz725/>).

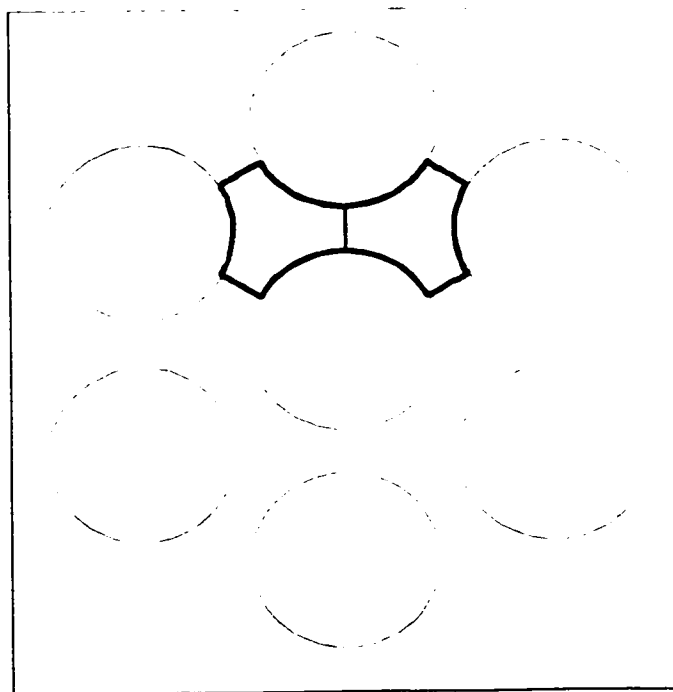


Figure 1.4 Subchannels and rods in a CANDU fuel bundle.

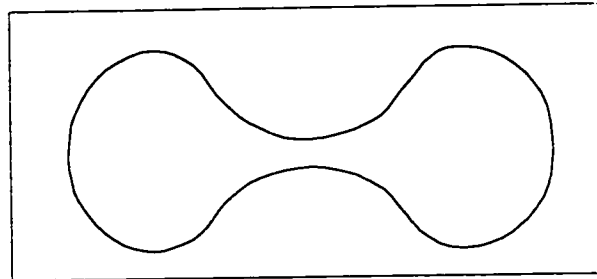


Figure 1.5 Dumbbell-shaped tube.

1.3 Objectives and Scope

The objective of this work is to investigate CHF in a complex flow geometry utilizing a dumbbell-shaped tube. The study includes experimental investigation and modeling of CHF. To accomplish this goal, the following interrelated areas of endeavor are investigated.

1. Analysis of CHF mechanisms

The CHF prediction methods are based on the understanding and analysis of mechanisms producing CHF. It is necessary to investigate CHF mechanisms at various flow conditions, using different CHF prediction methods and CHF modeling techniques.

2. Survey of previous research on complex flow geometry CHF

A considerable amount of research work has been carried out to determine CHF values for various flow geometries. This task will focus on reviewing the previous studies related to complex flow geometry CHF and dryout-type CHF modeling. Also, the effect of all the dominating geometry parameters, such as tube diameter and gap size, will be examined and summarized.

3. Experimental investigation of CHF in a dumbbell-shaped tube

The results from previous studies show that the gap size between two adjacent surfaces is a fundamental parameter in defining CHF. To understand the gap size effect on CHF and obtain systematic database for a mechanistic analysis, CHF experiments are performed with a series of dumbbell-shaped tubes using different narrow gap sizes. This task focuses on acquiring narrow gap CHF data and analyzing the gap effect on CHF.

4. Development of a complex flow geometry dryout-type CHF model

This task aims to derive a theoretical model to predict dryout-type CHF in a complex flow geometry. This model extends the circular tube CHF model by considering the effect of geometry on flow, heat and mass transfer.

Chapter 2 provides a summary of the CHF mechanisms at various flow conditions, illustrates the parameter trends, and discusses various prediction methods. Chapter 3 reviews the literature on complex flow geometry CHF and models used to predict CHF. Chapters 4 and 5 outline the experimental investigation and modeling of CHF in the dumbbell geometry. Chapter 6 analyzes the experimental data, whereby comparisons are made between the model predictions and experimental data in Chapter 7.

The conclusions of this study are presented in Chapter 8: both experiment and model results are summarized, the limitations of the present study are discussed, and recommendations for future work are provided.

Chapter 2

CRITICAL HEAT FLUX IN FLOW BOILING

An understanding of boiling and CHF mechanisms is important for accurate prediction of CHF. There are two basic types of boiling: pool boiling and flow boiling. Pool boiling is the process in which the heating surface is submerged in a large body of stagnant liquid. The relative motion of the vapor produced and of the surrounding liquid near the heating surface is due primarily to the buoyancy effect of the vapor. Nevertheless, the body of the liquid as a whole is essentially at rest. Flow boiling is boiling in a flow stream of fluid, where the heating surface may be the channel wall confining the flow. Detailed reviews of the CHF mechanisms in Pool Boiling are given by Hewitt (1997), Katto (1992), Dhir (1996), and Begles (1992). This study is restricted to flow boiling CHF because the most important applications are those where boiling occurs in a channel, such as flow through a rod bundle as in the fuel elements of a nuclear reactor.

The CHF mechanisms depend strongly on flow regimes. This chapter focuses on the description of flow regimes, heat transfer regions, and finally analyzing the CHF mechanisms.

2.1 Flow Boiling

2.1.1 Regimes of Heat Transfer in a Vertical Heated Tube

In the majority of boiling systems, evaporation occurs in the fluid flow through vertical channels of one form or another. Let us consider the case of a vertical tube heated uniformly over its length with a relatively low heat flux and with a pure single-phase upward-flowing liquid entering at its inlet. This prescribed heat flux boundary condition for a one-dimensional situation is the simplest case. The presence of a two-phase fluid flow has a profound effect on the heat transfer characteristics of the system. Referring to Figure 2.1, the following definitions can be adopted for various heat transfer regions.

- Single-phase convective heat transfer to the liquid phase (region A)

As liquid enters the channel, if subcooling is high enough, there is a section of single phase forced convection until incipient boiling sets in. The wall temperature remains below that necessary for nucleation. In this region, the heat transfer coefficient changes slightly due to the effect of temperature on the liquid properties

- Onset of nucleate boiling (ONB)

This is the point at which boiling is initiated in a vertical heated tube. The wall temperature must be some what above the saturation temperature before bubble growth can occur.

- Subcooled boiling (region B)

Bubble nucleation begins when the bulk flow quality is less than zero. From the onset of bubbles to the saturation condition is called the subcooled nucleate boiling region. But even after boiling incipience, the degree of subcooling is still very high, so that bubbles collapse before leaving the heating surface. Thus the void is concentrated at the wall. As the fluid proceeds farther downstream and loses more subcooling, the bubbles grow bigger and detach from the surface. The void fraction increases with distance. In the subcooled boiling, the wall temperature remains a few degrees above the saturation temperature and the mean bulk fluid temperature increases towards the saturation temperature. In this region, the temperature difference between the wall and the bulk fluid decreases linearly with length. At fixed heat flux, the heat transfer coefficient increases linearly with length.

- Saturation Point

This is the point at which the bulk fluid inside the channel reaches the saturation temperature (thermodynamic equilibrium quality $x = 0$) found on the basis of simple heat balance calculations.

- Saturated nucleate boiling (regions C and D)

Beyond the saturation point, bulk void increases even more rapidly. The flow pattern changes from bubble or slug to annular type. In this region, the temperature difference and the heat transfer coefficient remain constant.

- Two-phase forced convective region of heat transfer (regions E and F)

As the quality increases through the saturated nucleate boiling region, a point may be reached where a fundamental transition in the mechanism of heat transfer takes place: the process of boiling is replaced by the process of evaporation. The region beyond the transition has been referred to as the two-phase forced convective region of heat transfer. In this region, the thickness of the thin liquid film on the heating surface is often such that the effective thermal conductivity is sufficient to prevent the liquid in contact with the wall from being superheated to a temperature which would allow bubble nucleation. Heat is carried away from the wall by forced convection in the film to the liquid-vapor core interface, where evaporation can no longer be called boiling. Before dryout occurs, the difference between wall temperature and the saturation temperature gradually decreases. Therefore, the boiling heat transfer coefficient is increasing; this is a consequence of the liquid film in annular flow becoming thinner as the quality increases.

- Dryout point

For the case described above, the critical heat flux condition occurs at some critical value of the quality, where complete evaporation of the liquid film occurs. This particular CHF transition is known as "dryout" and is accompanied by a rise in the wall temperature for channels operating with a controlled surface heat flux. At this point, the heat transfer coefficient suddenly drops from a very high value in the forced convection region to a value near to that of forced convection to dry saturated vapor.

- Liquid deficient regime (region G)

The area between the dryout point and the transition to dry saturated vapor (region H) has been termed the liquid deficient region (corresponding to the drop flow pattern), where post-CHF heat transfer occurs. In this region, the vapor velocity increases and the heat transfer coefficient rises correspondingly.

- Single-phase convective heat transfer to the vapor (region H)

In the single-phase vapor region, the heat transfer coefficient levels out to that corresponding to convective heat transfer to a single-phase vapor flow.

2.1.2 The Boiling Map

Figure 2.1 shows a conceptual picture of the various flow patterns over the tube length, the qualitative temperature profile, and the corresponding heat transfer regimes. The heat transfer coefficient changes with the length in different heat transfer regions are described in section 2.1.1. However, the case discussed in the section 2.1.1 is restricted to a relatively low heat flux supplied. When heat flux is increased, the flow and heat transfer may change accordingly. Figure 2.2 shows the heat transfer coefficient plotted against quality with increasing heat flux as a parameter (curves i to vii).

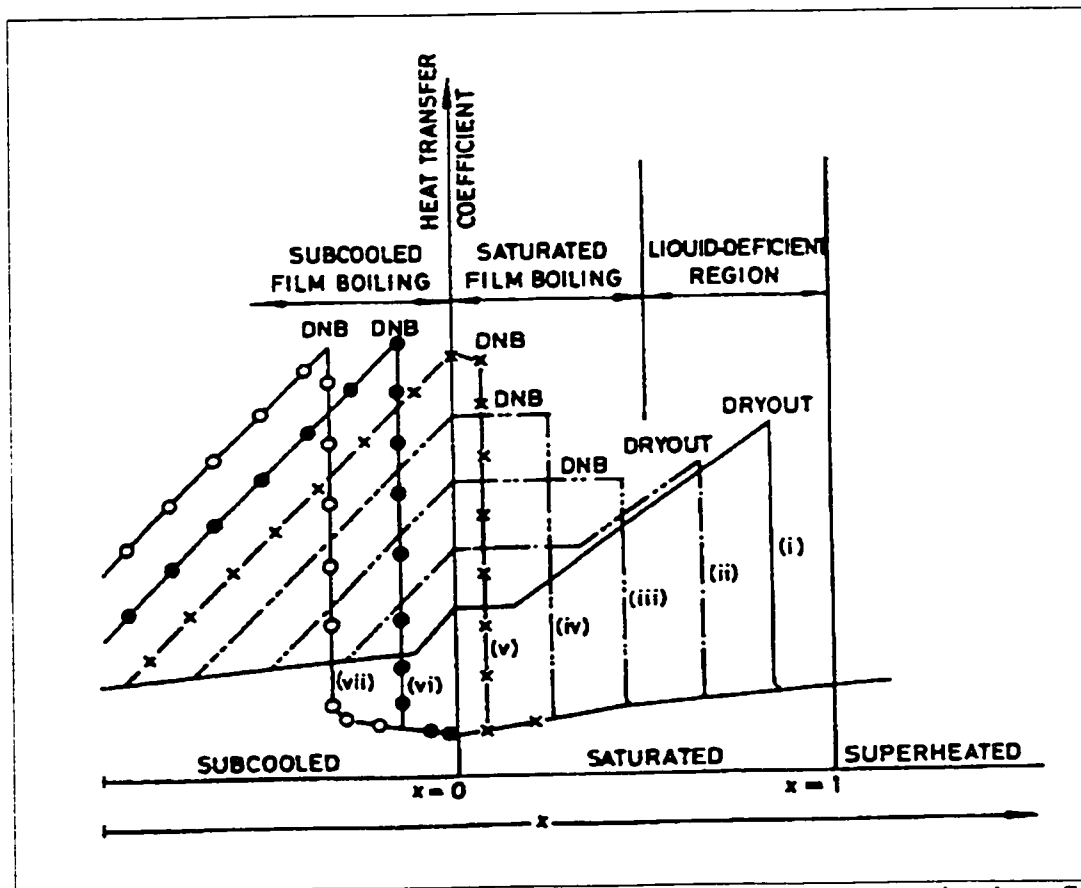


Figure 2.2 Variation of heat transfer coefficient with quality with increasing heat flux as parameter (from Collier and Thome, 1994).

Figure 2.3 shows the various regions of two-phase heat transfer in forced convective boiling on a three-dimensional diagram with heat flux, mass quality, and temperature as co-ordinates. Figure 2.4 shows the regions of two-phase forced convective heat transfer as a function of quality with

increasing heat flux as ordinate. These diagrams, which illustrate the relationship of heat transfer coefficient, heat flux and quality, are called the boiling maps.

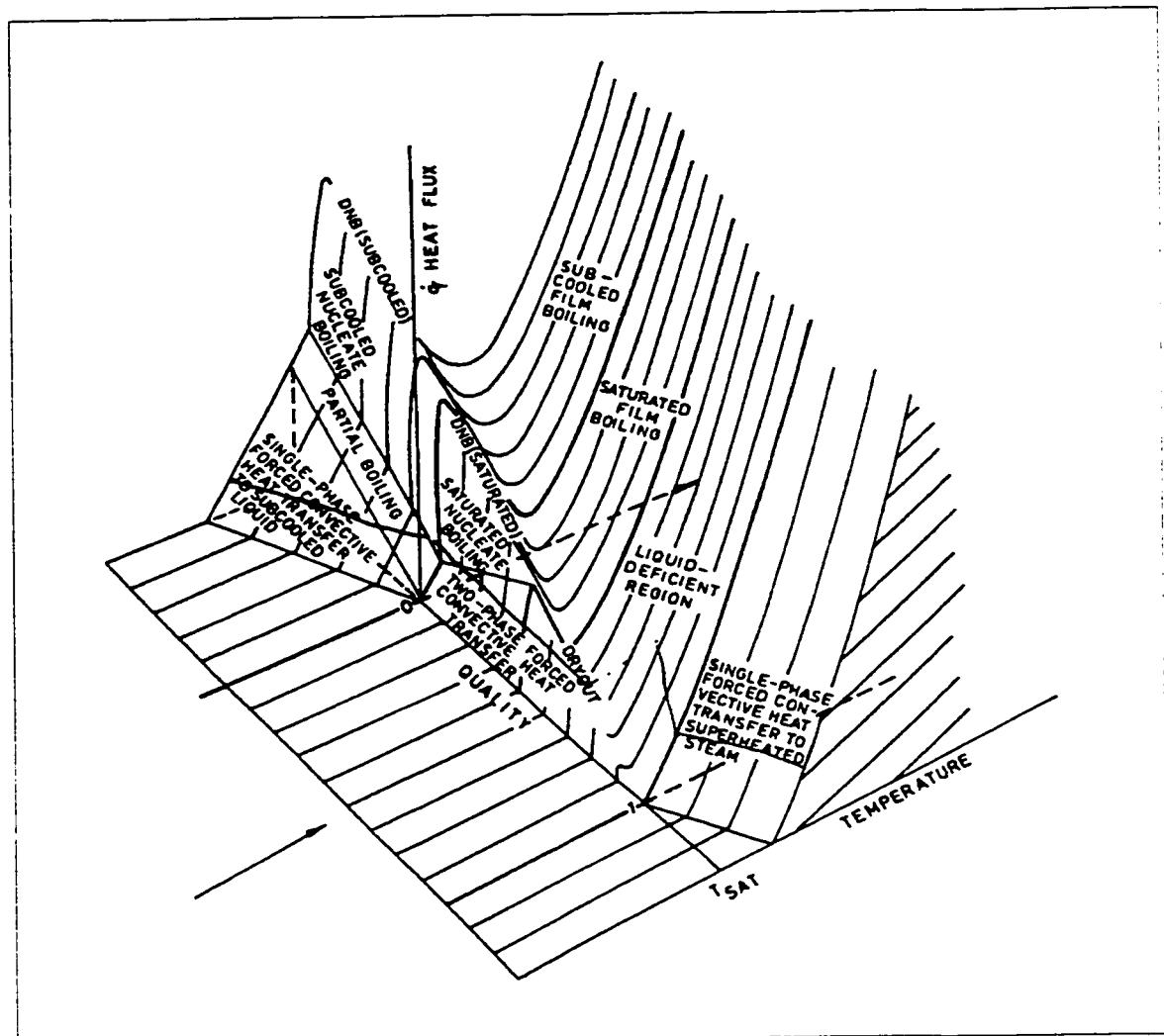


Figure 2.3 Forced convective boiling surface (from Collier and Thome, 1994)

Figure 2.2 curve i represents the variation of heat transfer coefficient along the tube for the conditions given in Figure 2.1. The temperature pattern shown in Figure 2.1 can be recognized as the projection in plan view of Figure 2.3. Curve ii shows the influence of increasing the heat flux. Subcooled boiling is initiated sooner; the heat transfer coefficient in the nucleate boiling region is higher but unaffected in the two-phase forced convection region. Dryout occurs at a lower mass quality. Curve iii shows the effect of a further increase in heat flux. Subcooled boiling is initiated earlier and the heat transfer is again higher in the nucleate boiling region. As the mass quality increase, before the two-phase forced convective region is initiated, and while

bubble nucleation is still occurring, an abrupt deterioration in the cooling process takes place. This transition is essentially similar to the critical heat flux phenomenon in saturated pool boiling and is termed departure from nucleate boiling (DNB).

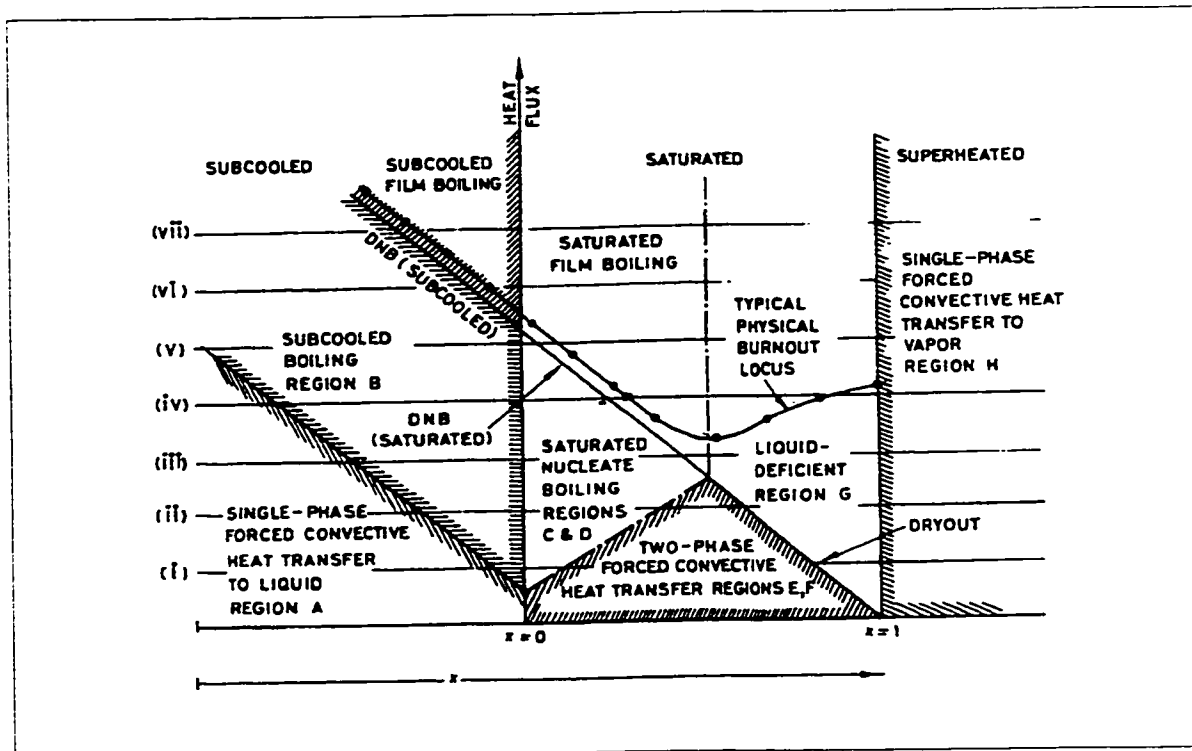


Figure 2.4 Regions of two-phase convective heat transfer as a function of quality with increasing heat flux as ordinate (from Collier and Thome, 1994).

The mechanism of heat transfer under conditions where the critical heat flux has been exceeded is dependent on whether the initial condition was the process of “boiling” or the process of “evaporation”. In the later case, the liquid-deficient region is initiated; in the former case, the resulting mechanism is one of film boiling. Further increases in heat flux cause the condition of DNB to occur in the subcooled region (curve vi and vii).

2.2 Flow Boiling CHF

Flow boiling CHF is closely related to the type of flow pattern and phase distribution present at its occurrence. A large number of CHF mechanisms with their regions of operation have been proposed. In this section, CHF mechanisms are discussed according to Tong and Tang (1997).

As illustrated in Figure 2.5, the flow boiling CHF mechanisms are roughly classified into the following two categories:

- CHF in a subcooled or low-quality region (DNB).
- CHF in a high-quality region (dryout).

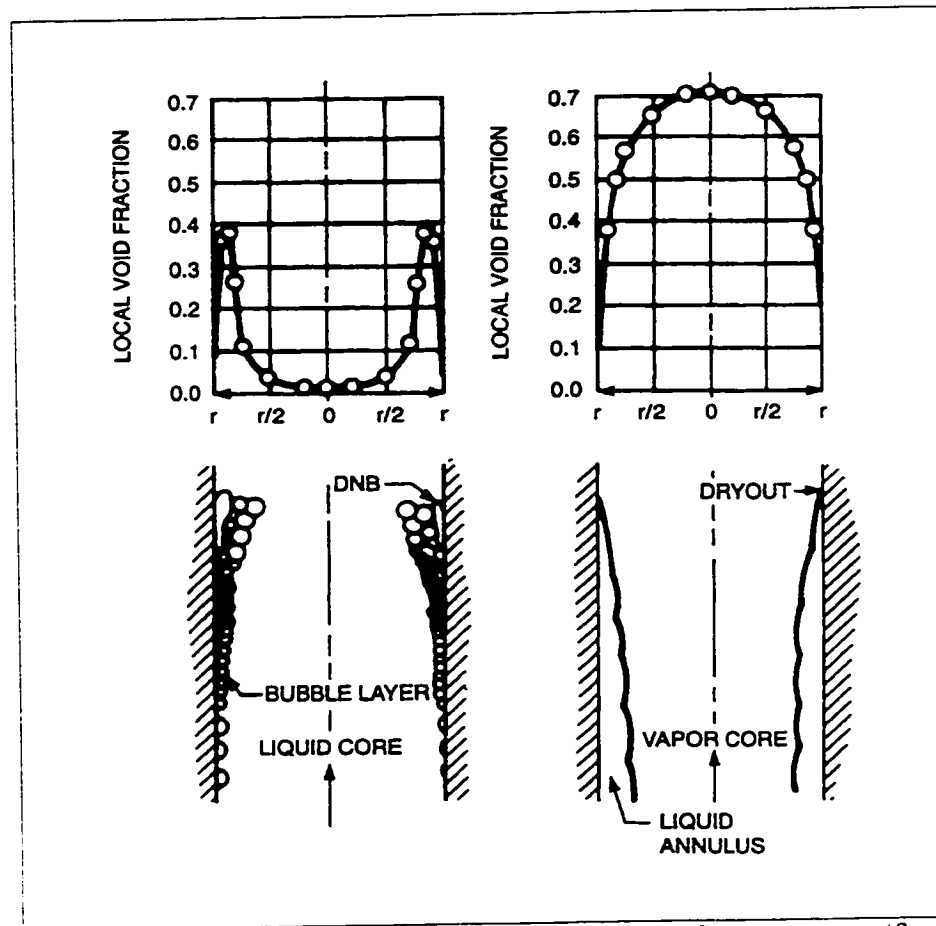


Figure 2.5 Comparison of CHF mechanisms in various flow patterns (from Tong and Tang, 1997).

2.2.1 CHF in a Subcooled or Low-Quality Region

In general, CHF occurs in a subcooled or low-quality region in case of high heat flux. The high heat flux causes intensive boiling, so the bubbles are crowded in a layer near the heated surface. The local voidage impairs surface cooling by reducing the amount of incoming liquid. At CHF conditions, the local void spreads as a vapor blanket on the heating surface and the nucleate boiling disappears, replaced by a film boiling. Thus was the name, departure from nucleate

boiling (DNB), established. This type of CHF condition usually occurs at a high flow rate, and the flow pattern is called inverted annular flow. The observed CHF mechanisms in the subcooled or low quality, bubbly flows are sketched in Figure 2.6, listed in the order of decreasing subcooling.

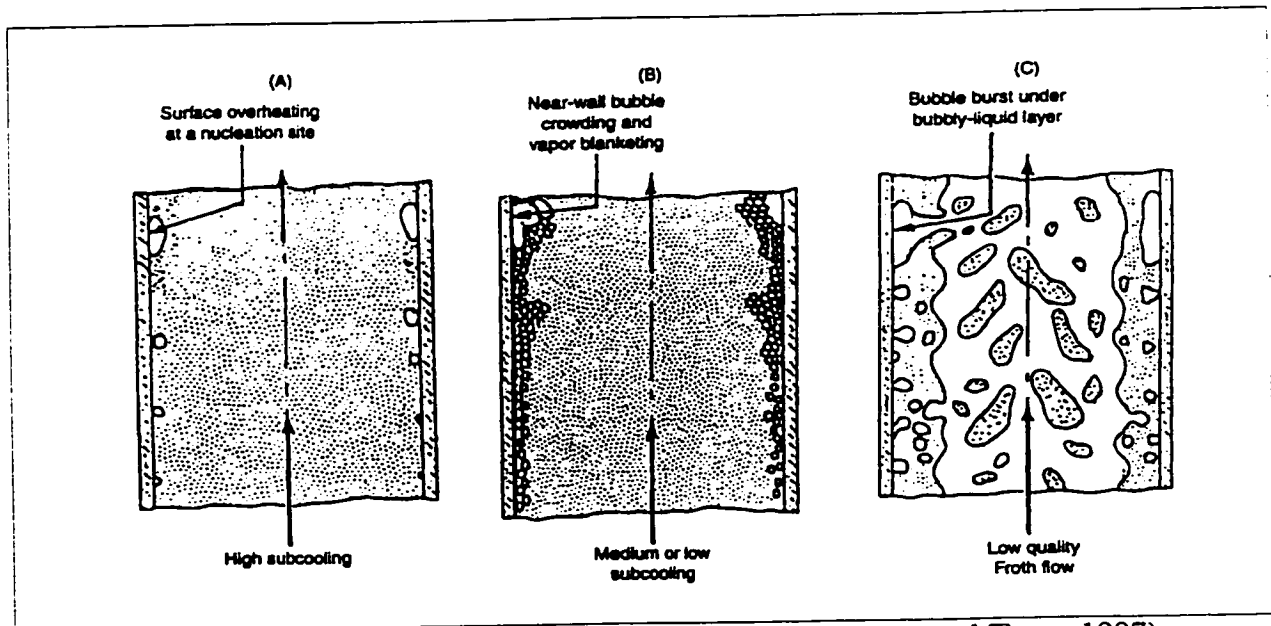


Figure 2.6 Mechanisms of DNB in bubbly flows (from Tong and Tang, 1997).

- (A) The surface overheating at CHF in a highly subcooled flow is caused by a poor heat transfer capability of the liquid core.
- (B) The DNB in a medium- or low-subcooling bubbly flow is caused by near-wall bubble crowding and vapor blanketing.
- (C) The DNB in a low-quality froth flow is caused by a bubble burst under the bubbly liquid layer.

The two-phase mixture in the boundary layer near the heating surface can hardly be in thermodynamic equilibrium with the bulk stream. Thus, the magnitude of the CHF depends on the surface proximity parameter as well as the local flow patterns. The surface proximity parameters include the surface heat flux, local voidage, boundary-layer behavior, and its upstream affects. When this type of CHF occurs, the internal heating source causes the surface temperature to rise rapidly to a very high value. The high temperature may physically burn out the surface. In this case, it is also described as fast burnout.

2.2.2 CHF in a High-Quality Region

This category of CHF occurs at a heat flux lower than the one described previously. The total mass flow rate can be small, but the vapor velocity may still be high owing to the high void fraction. The flow pattern is generally annular. In each of these configurations a liquid annulus normally covers the heated surface as a liquid layer and acts as a cooling medium. If there is excessive evaporation due to boiling, this liquid layer breaks down and the surface becomes dry. Thus this CHF condition is often termed dryout. The magnitude of the CHF in such a high-void-fraction region depends strongly on the flow pattern parameters, which can include flow quality, average void fraction, slip ratio, vapor velocity, liquid-layer thickness, and boiling length. It has only a weak dependence on the surface parameters. The observed CHF mechanisms in medium- or high-quality annular flows are sketched in Figure 2.7.

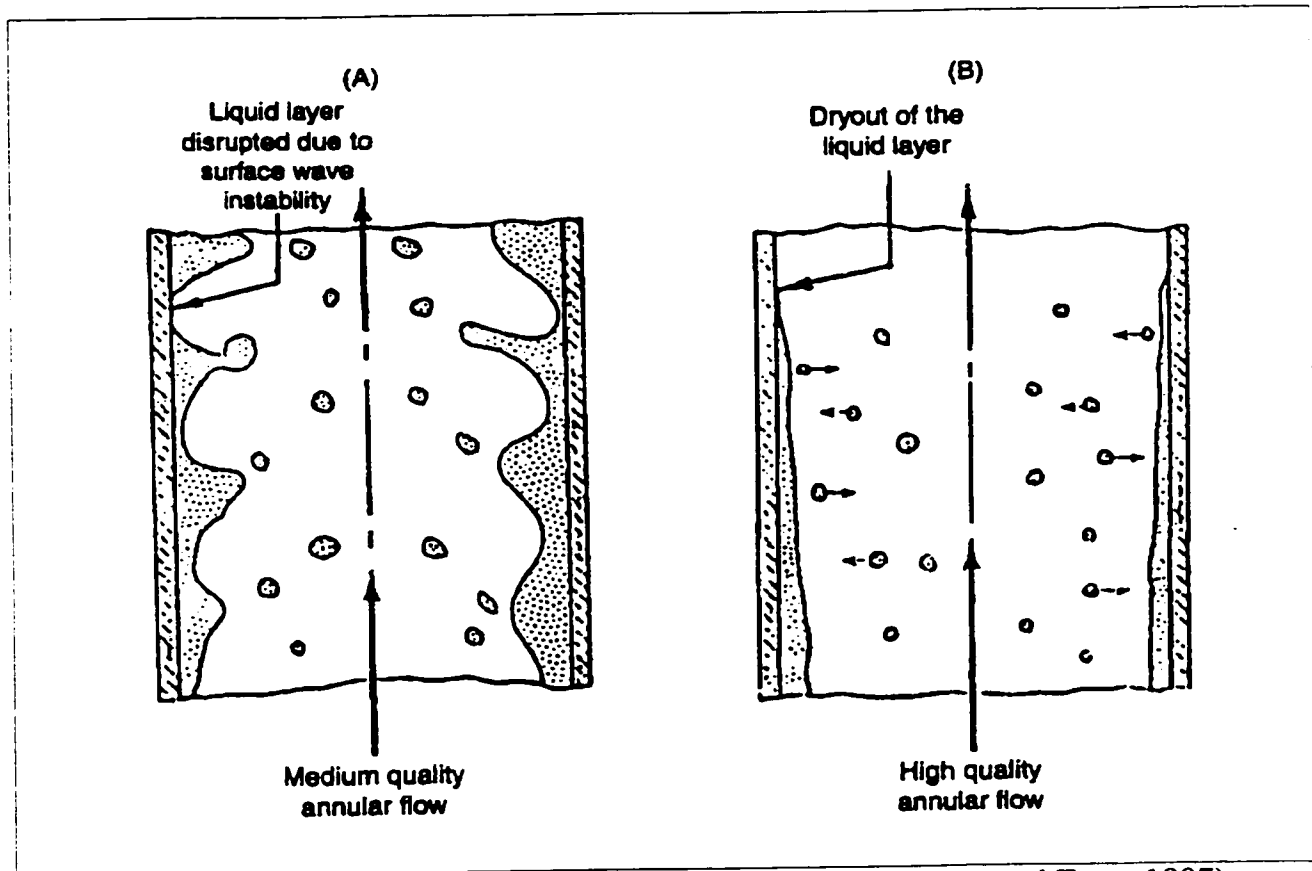


Figure 2.7 Mechanisms of dryout annular flows (from Tong and Tang, 1997).

(A) The dryout in a medium-quality annular flow is caused by liquid-layer disruption due to surface wave instability.

(B) The dryout in a high-quality annular flow is caused by the disappearance of the liquid layer on the heated wall.

When the CHF phenomenon occurs, the surface temperature rises. Because of the fairly good heat transfer coefficient of a fast-moving liquid-vapor core in an annular flow, the wall temperature rise after a dryout in the high-quality region is usually smaller than that in subcooled CHF. It is even possible to establish a steady-state condition at moderate wall temperatures, so that physical burnout may not occur immediately – this form of dryout is also described as slow burnout.

2.3 Parametric Effects on CHF

2.3.1 General

A large amount of work has been done on the determination of CHF under a wide variety of conditions. This section focuses on a brief summary of the important parameter trends in CHF. Extensive reviews are given by Collier (1994) and Tong (1997).

Parameter effects on CHF are usually investigated by examining the effect of each independent variable separately. System parameters, such as inlet flow-rate, inlet temperature, system pressure, tube geometry, and the tube length, are considered as possible inlet conditions which can influence the critical heat flux. Thus, CHF can be expressed as a function of these inlet conditions:

$$\dot{q}_{CHF} = f(p, G, \Delta i_{sub}, D_H, L)$$

As shown, a CHF correlation based on inlet enthalpy must be accompanied by a length term. Figure 2.8 shows conceptually the effect of various system parameters on CHF based on fixed inlet condition.

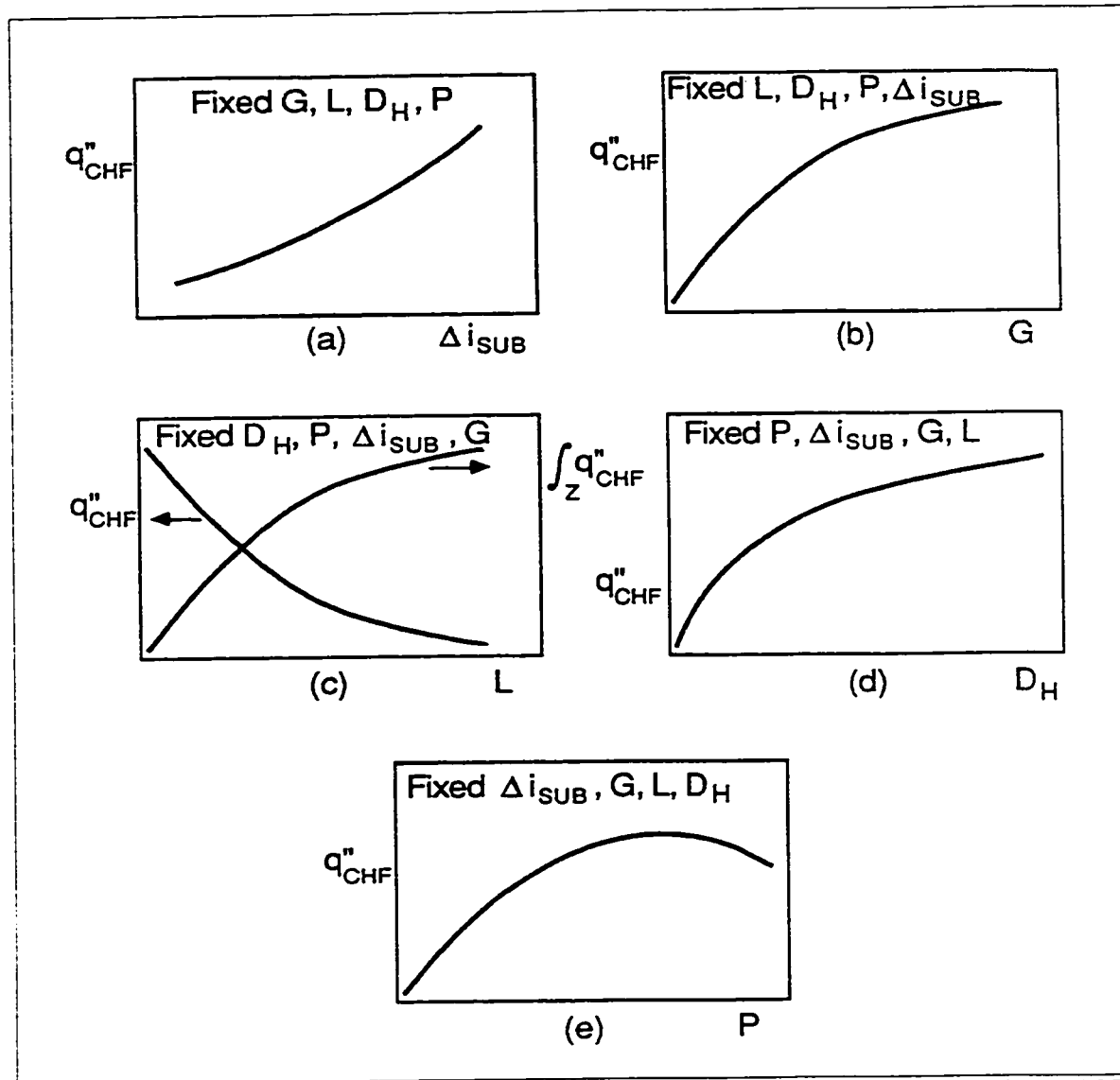


Figure 2.8 Parametric effects on CHF at fixed inlet condition (from Whalley, 1987).

In a typical system with a uniform heat flux, CHF first occurs at the exit of the heated section. The CHF correlation is usually expressed in terms of fixed local condition as follows:

$$\dot{q}_{CHF} = f(p, G, x, D_H)$$

Figure 2.9 illustrates conceptually the effect of various parameters based on fixed local condition.

Furthermore, it is necessary to point out that the effects of these parameters on CHF are not entirely independent, but are mutually related. There are also many other parameters, such as

channel orientation, local flow obstruction, surface property, hot-patch, rod bowing, rod spacing, flow instability, dissolved gas, and volatile additive, which can affect the CHF.

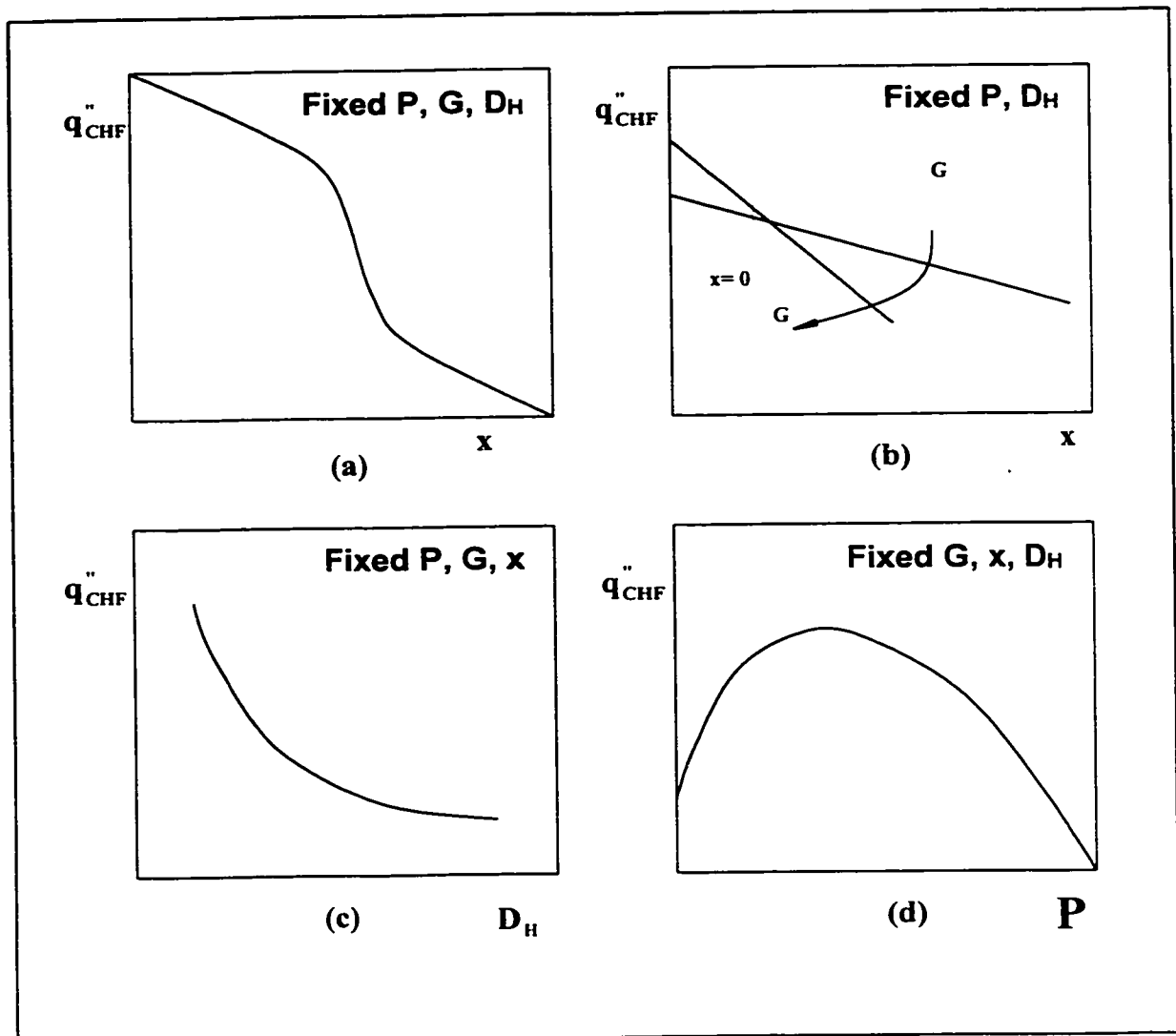


Figure 2.9 Parametric effects on CHF at fixed local condition (from Whalley, 1987).

2.3.2 Effects of Pressure

Pressure has a strong effect on CHF primarily because of its effect on thermodynamic properties. As shown in Figure 2.8 (e) and Figure 2.9 (d), the critical heat flux initially increases with increasing pressure and then decreases with pressure. When the pressure increases to the critical pressure, the CHF approaches zero. At the critical point, there is no property difference between the vapor and liquid phase.

2.3.3 Effect of Mass Flux

As shown in Figure 2.8 (b), at fixed inlet subcooling, pressure, diameter and length, CHF increases linearly with mass flux at low value of G but then rises much less rapidly at high values of mass flux.

Figure 2.9 (b) illustrates the effect of mass flux on CHF at local conditions. The effect of mass flux is coupled with quality at a given pressure. In the subcooled region, the CHF increases with increased mass flux at fixed local condition. In the saturated region, the CHF vs. X plot shows a crossover of constant G lines. In the low quality region, the CHF increases with G while in the high quality region, the trend is opposite. This crossover may be explained by considering the CHF mechanisms. At low quality, CHF is caused by bubble clouding or microlayer evaporation - an increase in mass flux results in a more efficient removal of vapor nucleate from the wall or in sweeping away the bubble boundary layer, and this tends to delay CHF. At high qualities, dryout occurs due to liquid film depletion - higher flow results in a wavier liquid film and hence increased entrainment, which tends to promote an earlier CHF.

2.3.4 Effect of Subcooling

For a given pressure, fixed mass flux, tube length and diameter, the critical heat flux increases approximately linearly with increasing inlet subcooling as illustrated in Figure 2.8 (a). This relationship occurs over fairly wide range of subcooling, but has no fundamental significance, except to indicate that more energy is required to saturate the fluid. If a very wide range of inlet subcooling is used, then departure from linearity is observed.

2.3.5 Effect of Quality

The effect of quality or local enthalpy is due primarily to the wall voidage, which impairs the critical heat flux, and secondarily to the bulk voidage, which affects the flow pattern. Figure 2.9 (a) shows the generally decreasing trend of CHF with an increase in quality. This trend continues until the CHF is zero at 100% quality.

2.3.6 Effect of Geometry

2.3.6.1 Effect of Heated Length on CHF

The critical heat flux decreases with tube length as shown in Figure 2.8 (c). The total power input increases with length. Some data show an asymptotic value of power input over a wide range of tube length, corresponding to a constant outlet quality for the CHF condition.

All the studies show an agreement on the effect of heated length (Tong, 1997). For fixed inlet condition, the CHF increases as tube length is decreased, while for fixed exit conditions, the effect of tube length on CHF is small.

2.3.6.2 Effect of Tube Diameter on CHF

The tube diameter is shown to have a strong effect on CHF. The diameter effect becomes more important as the tube diameter decreases. As shown in Figure 2.8 (d), the diameter effect becomes less important at large diameter. For fixed mass flux, length, pressure, and inlet subcooling, the critical heat flux increases with tube diameter, eventually reaching an almost constant value for large tube diameters.

On the other hand, for a fixed quality, CHF decreases with increasing channel diameter as shown in Figure 2.9 (c). Hewitt and Hall-Taylor (1970) indicated that no satisfactory explanation is available, but that the resultant decrease in shear stress near the wall and higher entrainment rates in larger diameters would be contributing factors.

2.4 Prediction of CHF

The accurate prediction of critical heat flux in flow boiling is of importance in a wide variety of process equipment. The CHF prediction methods can be classified into two categories: measured or empirical and mechanistic models.

2.4.1 Empirical Methods

The CHF has been measured extensively, especially in directly heated circular tubes. Currently, for design purposes, CHF predictions are still largely performed using a lookup table or empirical correlation which is based on direct CHF measurement.

2.4.1.1 Lookup Table Approach

The lookup table is a general technique that can cover a wide range of applications. To summarize the previous P-G-X effects concisely and predict CHF accurately, Groeneveld, Cheng and Doan (1986) developed the “AECL-UO” lookup table for CHF. The table uses AECL’s CHF databank, which contains over 15,000 CHF data points for water over a wide range of operating conditions. This reference table is derived specifically for upward water flow in a uniformly heated 8-mm tube, and shows a matrix of combined P-G-X effects on CHF. The table predicts the general trend of P-G-X quite accurately. Groeneveld et al. (1996) updated the CHF lookup table by using a more extensive data base containing more than 30,000 CHF data points.

The updated lookup table covers the range: pressures from 0.1 MPa to 20 MPa, mass fluxes from 6 to 8000 kg/m²s, qualities from –0.5 to 1. To apply them to reactor design, proper adjustments for detailed geometrical effects for a prototype rod bundle are needed. Such adjustments are usually expressed in empirical formulas that are as complicated as the existing CHF design correlations.

2.4.1.2 Correlation

Although correlations are quite valuable, their extrapolation to conditions outside the original data set usually results in very large errors. CHF correlations can be divided into two main categories:

- Based on local conditions:

These correlations are based on the local condition hypothesis suggested by Barnett (1963) and Macbeth (1963). This hypothesis implies that CHF depends only on the local conditions.

$$\dot{q}_{CHF} = f(p, G, x, \text{cross-section geometry})$$

- Based on inlet conditions:

These correlations predict the CHF as a function of the inlet condition:

$$\dot{q}_{CHF} = f(p, G, \Delta i_{sub}, \text{cross-section geometry}, L)$$

This approach does not predict the axial location of CHF and assumes steady state conditions with a uniform axial flux distribution. The Bowring correlation (1972) is generally accepted as one of the most reliable correlations of this type for water. It is noted that these two types of correlation (local versus inlet) are inter-related by a simple heat balance equation.

2.4.2 Mechanistic Methods

These models use an analytical approach to characterize the physical phenomena of CHF, and may still require the use of some optimized empirical correlations obtained using experimental data. Theoretical models normally predict CHF in a system within a set of conditions independent of the fluid type, but are often valid for one flow regime.

2.4.2.1 Dryout-Type CHF Modeling

There is a general agreement that the mechanism leading to CHF depends on the flow quality. At high quality, the flow is annular and CHF is assumed to be caused by dryout of the liquid film on the heated surface. It is also generally agreed that such dryout can be predicted by appropriate modeling of the vaporization, entrainment, and deposition mechanisms operating on the annular liquid film.

The fundamental tool for dryout-type CHF modeling is an annular two-phase flow model. For the case of annular flow in a circular tube as shown in Figure 2.10, the liquid film is characterized as a thin liquid film flowing along the tube walls with vapor and liquid droplets flowing through the central core of the tube. In addition, the film and vapor flow are generally assumed turbulent.

There are two important processes which any annular flow model must describe: entrainment and deposition. The liquid film will have large amplitude waves on its surface, from which liquid droplets are entrained into the vapor core, usually termed “roll-waves”. Subsequently, these droplets are highly turbulent and are eventually re-deposited onto the liquid film.

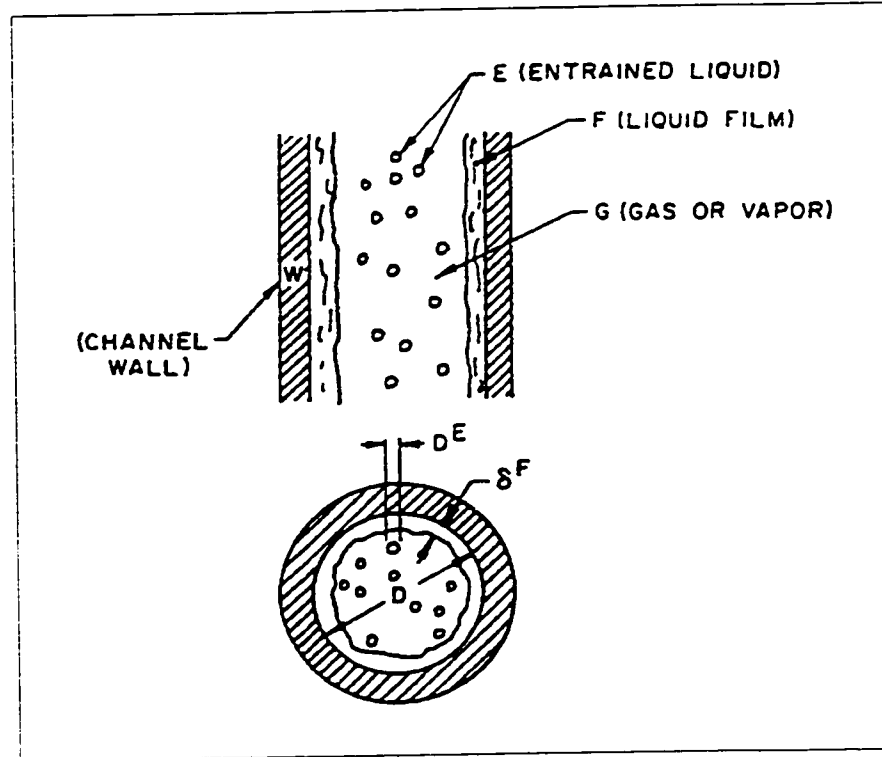


Figure 2.10 Annular flow in a circular tube (from Whalley, 1977).

Most dryout CHF models predict the onset of CHF in an evaporating vertical flow channel by calculating the point at which the liquid film flow rate reduces to zero, while allowing for evaporation, droplet entrainment and droplet deposition. The models were initially developed using empirical correlations of entrainment and deposition in adiabatic equilibrium flows, which were later improved for application to non-equilibrium (diabatic) flows.

2.4.2.2 DNB Type CHF Modeling

CHF at low quality occurs when the bubbles near the wall coalesce to form a vapor film. Weisman and Pei (1983) presented a theoretical CHF model for subcooled and low quality flow boiling in a tube. The CHF condition occurs when the “volume fraction of vapor” in the bubbly

layer just exceeds a certain critical value, which is brought about through the balance of outward flowing vapor from bubbly layer and the inward flowing liquid from the core liquid region.

Chapter 3

LITERATURE REVIEW

This literature survey is divided into two sections: experimental studies and modeling studies. The experimental studies focus on the previous measurement results and analysis of CHF in complex geometries. Complex geometry here denotes a non-circular channel, while simple geometry denotes a circular tube. Many complex geometries represent or are similar to the fuel channels present in different types of water-cooled nuclear reactors. CHF in the following five geometries is surveyed:

- Concentric annulus
- Eccentric annulus
- Rectangular duct and parallel plates
- Dumbbell-shaped channel
- Fuel-rod bundles

Geometry effects on CHF as well as the influence of flow and heating conditions are discussed.

The modeling studies describe numerous existing dryout-type CHF models and their applications. The following models are reviewed:

- Whalley (1974)
- Saito (1978)
- Levy (1980)
- Lim (1988)
- Govan (1990)
- Sugawara (1990)

3.1 Experimental Studies

A considerable amount of research has been carried out to determine the CHF in complex flow geometries, such as annuli, rectangular and triangular ducts, parallel plates, dumbbell-

shaped channels, and fuel bundles. Many research papers and survey reports related to the geometry effect on CHF are available. As part of the first phase of study into the effect of geometry on CHF in the University of Ottawa, Joobar (1993) conducted an extensive literature survey. This section is based on his review. The geometry effect on CHF is usually studied by examining the effect of a few key geometric parameters, such as minimum gap size, curvature, eccentricity, and channel shape.

3.1.1 Concentric Annuli

Concentric annuli are considered the simplest case of a fuel-rod bundle (i.e., a rod in a pressure tube). Most of the annulus CHF data pertain to uniform heating on the internal surface, since these most closely match the boundary conditions present in the nuclear industry. The geometry parameters for concentric annuli include gap size and curvature of the heated surface.

Annulus CHF data are usually compared with circular tube data for the same local condition and equivalent hydraulic diameter. Generally, the annulus CHF is found to be considerably lower than that in an equivalent circular tube, especially in the case of narrow gaps.

3.1.1.1 Gap Effect

3.1.1.1.1 General Trend

A large number of experimental investigations on CHF in internally heated annuli have been performed. Barnett (1966, 1968) compiled more than 800 experimental CHF data for water in annuli at 6.9 MPa from 11 different sources. Doerffer et al. (1994,1995) and Joobar (1993) summarized the gap effect on CHF in annuli. The global presentation of all data, such as CHF_{AN} or $CHF_{AN}/CHF_{D=8}$ versus gap size indicates a general trend. As illustrated in Figure 3.1, CHF increases with increased gap size from the smallest gaps to some value in the middle of the gap range, and then decreases again for larger gap sizes. Most data sets show the same gap effect: an increase in CHF for gap sizes from 1-4 mm, a maximum CHF value for gap sizes in the range 4-6 mm, and a decrease in CHF for larger gap sizes.

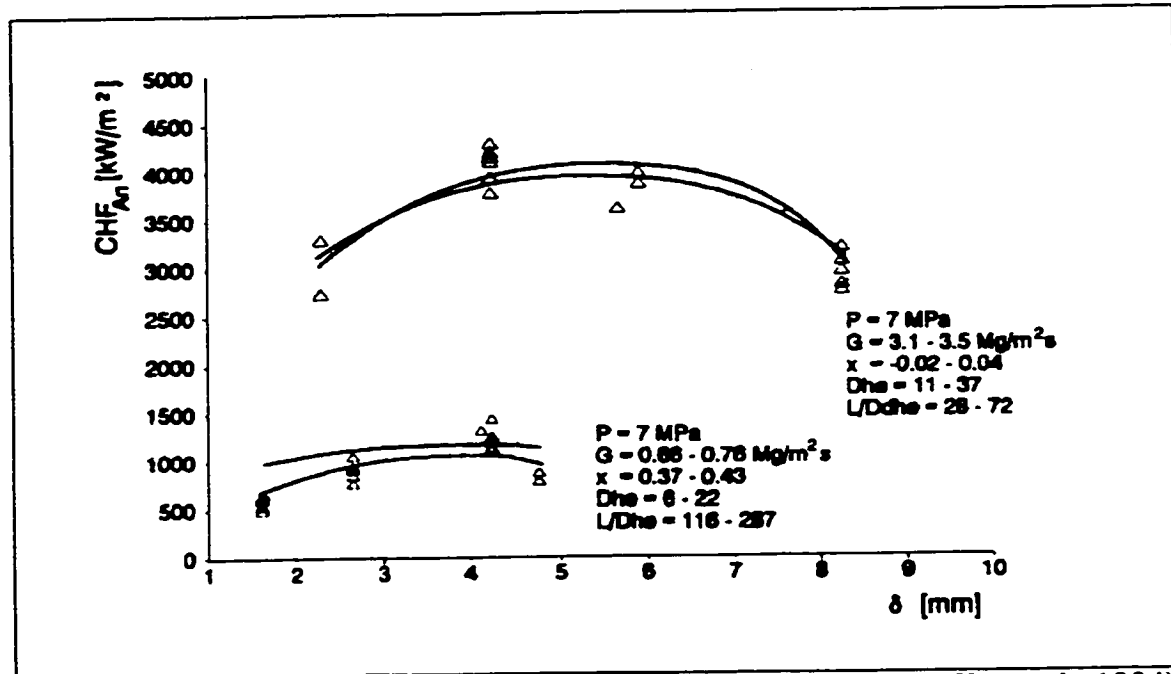


Figure 3.1 Effect of gap size on CHF in concentric annuli (from Doerffer et al., 1994).

Janssen et al. (1963) compared internally heated annuli CHF data in different gap sizes. They found that for $3.20 \text{ mm} \leq \delta \leq 6.35 \text{ mm}$, the gap size had little effect on the CHF. Moeck et al. (1966) concluded that at a constant mass flux and exit quality, the CHF increased as δ was increased from 1 mm to about 3.7 mm, and decreased as δ was further increased.

Tolubinskiy (1969) found that the gap size had little effect on CHF when the gap was greater than 1 mm, but the CHF decreased sharply when the gap became less than 1 mm. Tolubinskiy (1970a) investigated CHF in internally heated annuli and found that CHF increased as the gap size increased from 0.5 mm to 4.0 mm. Another study of Tolubinskiy et al. (1970b) indicates that with the increase of gap size from 0.5 mm to 1.5 mm, the CHF increased. Moreover, when the concentric gap size was more than 1.0 mm, the increase in the CHF was insignificant. A further increase in gap size beyond 1.5 mm resulted in no effect on CHF.

Ornatskiy (1969a) obtained the same trend as Tolubinskiy et al. (1969, 1970a). The CHF increases as the gap size increases from 0.1 mm to 1-2 mm. For a larger gap, the gap effect on CHF is little. Ornatskiy also observed a pressure influence on the gap effect. He found

that the gap size limit for which a further increase in the annular gap did not affect the CHF, increased with a decreasing pressure.

Hung and Yao (1983) observed the following behavior of boiling for Freon-113 in an annulus with a quartz outer tube near the CHF condition. For a gap size of 2.58 mm, bubbles became large and slightly deformed when the heat flux was gradually increased to the CHF. When the gap size was reduced to 0.8 mm, bubbles started out spherical and as they grew they were squeezed due to the narrow gap between opposite walls. The flattened bubbles hence coalesced and formed vapor plugs. The thin liquid film between the vapor plug or bubble and the heated surface (microlayer) easily evaporated, causing CHF to occur prematurely. A further reduction in gap size to 0.32 mm deformed the bubbles even more, leading to more coalescence and resulting in a further reduction in CHF. Hung and Yao (1983) concluded that the CHF decreased with a reduction in gap size for gap sizes less than 1.5 mm.

Joober (1993) arrived at the following conclusions about the gap effect on CHF for concentric annuli based on an extensive literature survey. He found that the gap size has an important effect on CHF if the gap size is reduced below a certain value. This value varies with the geometry and the flow condition. Depending on the flow pattern and the heat transfer regime, either the droplet or bubble size may play an important role in determining the effect of gap size on CHF. The effect of gap size on CHF becomes important when its size is comparable to the maximum bubble or droplet diameter.

3.1.1.1.2 Mechanism of Gap Effect

For dryout in annular flow, Moeck et al. (1966) presented two possible mechanisms to explain the gap effect on CHF. The first is that the interfacial shear stress on the heated wall increases with decreasing gap size. This increase in shear stress reduces the liquid-film flow rate and thus increases the entrainment rate. The second is that for narrow gaps, the “roll” surface waves from the liquid film of the unheated wall will interact with the liquid film of the heated wall, thus increasing entrainment and reducing the CHF. The effect becomes negligible for large gap sizes.

In the case of subcooled boiling, Joobar (1993) made a conclusion based on the literature review. If the gap size becomes relatively small, compared to bubble size, the bubble can completely block the gap, and thus result in a premature CHF condition.

3.1.1.2 Curvature Effect

In general, the literature on the curvature effect on CHF is very scarce, compared to that of the gap effect. Tolubinskiy et al. (1973) investigated the effect of curvature on the CHF in annuli for the same gap size (δ). For internally heated annuli, reducing the internal diameter (increasing the curvature) results in increased CHF. However, if the radius of the heated surface is less than 1 mm the CHF value will decrease. This has been attributed to the effect that heat transfer is reduced along the heated surface, and the fact that the radius of the heated surface becomes commensurable with that of the coalescing bubbles.

3.1.1.3 Bilateral Heating

Doerffer et al (1995) investigated the effect of bilateral heating on dryout-type CHF in annuli. They concluded that due to an enhancement of droplet deposition the heat addition to one surface of an annulus enhances CHF on the other surface. Thus, the CHF values in bilaterally heated annuli are always higher than those in unilaterally heated annuli. Furthermore, the CHF occurring on the outer annulus surface is less sensitive to the heat addition on the inner surface. During simultaneous CHF occurrence at both annulus surfaces, the CHF on the outer surface is always larger than that on the inner surface, and the difference is a strong function of pressure. The location of the CHF in bilaterally heated annuli is depended on the heat flux ratio between the two surfaces.

3.1.1.4 Prediction of CHF

3.1.1.4.1 Lookup Table Approach

Groeneveld et al. (1992) examined Barnett (1966, 1968) CHF data for internally heated annuli and found that critical quality and gap size have the strongest influence on CHF. They applied the AECL-UO lookup table to internally heated concentric annuli by using two

correction factors: a quality correction factor (k_x), and a gap-size correction factor (k_δ). The CHF in internally heated concentric annuli are predicted by

$$CHF_{ann,conc} = CHF_{D=8mm} k_x k_\delta$$

Doerffer et al. (1994) modified this approach by including a pressure effect :

$$CHF_{ann,conc} = CHF_{D=8mm} k_x k_\delta k_p$$

where, $CHF_{D=8mm}$ is the CHF for a circular tube with an inner diameter of 8 mm, as predicted by the CHF look-up table (Groeneveld et al., 1986, 1995). k_x , k_δ , and k_p are given by Table 3.1.

Table 3.1 CHF correction factors for concentric annuli

Correction Factor	Value	Range
K_x	0.81	$x \leq 0.025$
	$0.859 - 16.179x^{1.5} + 15.6x^2 - 7.185x^2 \ln(x)$	$x > 0.025$
K_δ	$0.2872 + 1.209\delta^2 - 1.156\delta^{2.5} + 0.2873\delta^3$	$\delta < 4.26 \text{ mm}$
	$1.2672 - 0.0298\delta$	$4.26 \text{ mm} \leq \delta \leq 6.27 \text{ mm}$
	$0.6663 + 64.374 \exp(-\delta/1.242)$	$6.27 \text{ mm} < \delta \leq 8.26 \text{ mm}$
	0.75	$\delta > 8.26 \text{ mm}$
K_p	0.9	$P < 3.3 \text{ MPa}$
	$0.808 + 0.0278p$	$3.3 \leq p \leq 10.5 \text{ MPa}$
	1.1	$P > 10.5 \text{ MPa}$

3.1.1.4.2 Correlation Method

The other parameter effects on CHF in concentric annuli appear to be the same as for the round tube. Barnett (1966) correlated annuli CHF data using a Macbeth-type correlation.

$$CHF \times 10^{-6} = \frac{A + B \Delta h_i}{C + L}$$

where for a pressure of 1000 psia (6.9 MPa)

$$A = 67.45 D_{hy}^{0.68} (G \times 10^{-6})^{0.196} \left[1 - 0.744 e^{-6.512 D_{he} (G \times 10^{-6})} \right]$$

$$B = 0.2587 D_{hy}^{1.261} (G \times 10^{-6})^{0.817}$$

$$C = 185.0 D_{hy}^{1.415} (G \times 10^{-6})^{0.212}$$

D_{hy} and D_{he} are the hydraulic diameter and heated diameter, respectively.

$$D_{hy} = D_o - D_i$$

$$D_{he} = \frac{D_o^2 - D_i^2}{D_i}$$

The correlation covers the following range of parameters:

D_i : 0.375 to 3.798 inch (9.52 to 96.5 mm)

D_o : 0.551 to 4.006 inch (14 to 101.6 mm)

L : 24.0 to 108.0 inch (0.61 to 2.74 m)

G : 0.14 to 6.20 ($\times 10^{-6}$) lb/h ft² (190 to 8430 kg/m²s)

Δh_i : 0 to 412 Btu/lb (0 to 0.958 MJ/kg)

Tolubinskiy et al. (1969) proposed an annulus CHF correlation for small gap sizes, but its application is limited by the range of short heated lengths (20 mm to 200 mm). Tong (1972) presented a correlation for predicting CHF in annuli, but the quality range is somewhat limited ($x \leq 0.1$).

Katto (1979a) extended his circular tube approach to annuli. CHF was classified into four regimes: L-regime, H-regime, N-regime, and HP-regime. The L-regime corresponds to low mass velocity, the N-regime to very high mass velocity, H-regime is for an intermediate range between the L- and N-regimes. The HP-regime replaces the N-regime when the system pressure is very high. These regimes are distinguished from each other by the following boundary criteria.

L-regime:

$$\frac{L}{D_{he}} \geq \frac{1}{0.48 \left(\frac{\rho_G}{\rho_L} \right)^{0.133} \left(\frac{\sigma \rho_L}{G^2 L} \right)^{0.29} - 0.0081}$$

H-regime:

$$\frac{0.0206}{\left(\frac{\sigma \rho_L}{G^2 L} \right)^{0.584}} \leq \frac{L}{D_{he}} < \frac{1}{0.48 \left(\frac{\rho_G}{\rho_L} \right)^{0.133} \left(\frac{\sigma \rho_L}{G^2 L} \right)^{0.29} - 0.0081}$$

N-regime:

$$\frac{L}{D_{he}} < \frac{0.0206}{\left(\frac{\sigma \rho_L}{G^2 L}\right)^{0.584}}$$

In case of zero inlet subcooling, $\Delta h_i = 0$, Katto suggested calculating the CHF in annuli as follows:

L-regime:

$$\frac{CHF_0}{G h_{fg}} = 0.25 \left(\frac{\sigma \rho_L}{G^2 L}\right)^{10.043} \frac{1}{L / D_{he}}$$

H-regime:

$$\frac{CHF_0}{G h_{fg}} = 0.12 \left(\frac{\rho_G}{\rho_L}\right)^{0.133} \left(\frac{\sigma \rho_L}{G^2 L}\right)^{1/3} \frac{1}{1 + 0.0081 L / D_{he}}$$

N-regime:

$$\frac{CHF_0}{G h_{fg}} = 0.22 \left(\frac{\rho_G}{\rho_L}\right)^{0.133} \left(\frac{\sigma \rho_L}{G^2 L}\right)^{0.433} \frac{(L / D_{he})^{0.171}}{1 + 0.0081 L / D_{he}}$$

When the inlet to the annulus is subcooled, the CHF is given by

$$CHF = CHF_0 \left(1 + K \frac{\Delta h_i}{h_{fg}}\right)$$

where K is a dimensionless parameter depending on the regime.

3.1.2 Eccentric Annuli

The important geometry parameters for eccentric annuli are minimum gap size and eccentricity. The eccentricity of an annulus is defined by

$$\varepsilon = 1 - \frac{\delta_{\min}}{\delta_{\text{nom}}}$$

where $\delta_{\min}$ is the minimum gap size, and δ_{nom} is the nominal gap size.

$$\delta_{\text{nom}} = \frac{D_o - D_i}{2}$$

$\varepsilon = 0$ refers to a concentric annulus, while $\varepsilon = 1$ refers to a physical contact between the two surfaces.

3.1.2.1 Eccentricity Effect

Ornatskiy et al. (1969c) investigated the CHF in eccentric annuli and found that the CHF decreases with increasing eccentricity at the same local conditions, but that its dependence on local subcooling diminished. As the eccentricity exceeds 0.66, the CHF becomes almost insensitive to local subcooling.

Anderson et al. (1974) examined the effect of eccentricity on CHF. Their results also show that CHF decreases with increasing eccentricity, but that the effect depends on flow conditions. The effect of eccentricity increases at higher flow ($G \geq 0.5 \text{ Mg/m}^2\text{s}$), while at mass fluxes near $0.5 \text{ Mg/m}^2\text{s}$, the effect of eccentricity becomes insignificant. Anderson et al. (1974) also found that “the greater the eccentricity, the more uneven the film flow rate”. The maximum film flow rate occurs in the maximum gap region. In the narrow gap region, the minimum film flow rate is close to zero, and this can be the cause of premature CHF occurring in the narrow gap region.

Tolubinskiy et al. (1977) concluded that increasing eccentricity reduces the CHF. In eccentric annuli, the axial velocity distribution around the annular cross section is non-uniform. The flow in the narrow gap region is slower than that in the wide gap region. This non-uniform velocity distribution in turn causes uneven temperature distribution. Hence, the CHF will occur first in the narrow gap region at a lower heat flux than the concentric case. Due to eccentricity, the closer proximity of the walls in the narrow gap region makes it difficult for the bubbles to detach from the heated surface. Hence, the narrow gap may become vapor locked and result in premature CHF. Moreover, increasing the equivalent diameter of an eccentric annulus will always reduce the detrimental effect of eccentricity.

3.1.2.2 Minimum Gap Effect

Levy et al. (1962) examined four gap sizes: 4.2 mm (concentric case), 2.4 mm, 1.5 mm and 0.8 mm, and found that for minimum gap sizes of 2.4 mm and 1.5 mm, the CHF results are similar to that of the concentric case. As the minimum gap size was reduced to 0.8 mm, the CHF decreased by only 15% to 30%. As quality increased, the CHF in the eccentric annuli with a 0.8 mm gap size converged towards the CHF of other gap size (4.2, 2.4 and 1.5 mm).

The decrease in CHF is somewhat mass-flux dependent, where the reduction in CHF at low mass flow is much smaller than that at high mass flow. Janssen et al. (1963) obtained similar results.

Moeck et al. (1966) investigated gap effect on CHF in eccentric annuli. As the gap size was reduced from the concentric case of 2.6 mm to 0.8 mm, the CHF decreased. When compared to the concentric case, it was found that at fixed inlet condition, the CHF decreased by 37, 50, and 60% for mass fluxes of 0.8, 1.4 and 4.1 $\text{Mg/m}^2\text{s}$, respectively.

Rudzinski (1992) carried out an experimental investigation of CHF in horizontal by bowed annuli. Four gap sizes were tested (1.25, 0.708, 0.367 and 0.068 mm) with a concentric gap size of 2.2 mm. Results from these experiments have shown that for a range of mass flux (3.0 to 7.0 $\text{Mg/m}^2\text{s}$), the effect of flow on CHF is not very important within each gap size. The CHF decreased with decreasing gap size. Although the range of outlet quality tested was quite small (-2 to 22%), it still showed the decrease to be more important at lower quality (near 0%). As the quality increased, the CHF for different gap sizes tended to converge to similar values, meaning that the CHF in concentric and eccentric annuli yields similar results at high quality. The decrease in CHF from a 1.25 mm gap size to a 0.708 mm gap size was less important than the decrease in CHF from a gap size of 0.708 mm to 0.367 mm. However, a further reduction in the gap size from 0.367 to 0.068 mm did not result in any further reduction in CHF. Again, the gap size effect is more pronounced at higher mass flux over a specific range of quality.

At high quality the CHF occurrence is controlled by film dryout. At these conditions, when the gap size is reduced, the outer surface liquid film may come into contact with the inner surface liquid film, thus filling the gap with liquid. This could have an effect of locally enhancing CHF for the eccentric annuli, moving the CHF point to a comparable value for the concentric annuli.

3.1.2.3 Void Migration Effect

Doerffer et al. (1994) introduced a new parameter to account for the mixing between a

narrow gap region and a wider gap region. A void migration parameter (γ) was defined as

$$\gamma = \frac{\delta_{nom}}{D_o + D_i}$$

where γ represents the void transport from the narrow gap region to the wider gap region in eccentric annuli. A comparison of annuli with similar pressure, mass flux, eccentricity, and nominal gap size, but different diameters showed the CHF to be a factor of 2 or more larger in an annulus of a smaller size or larger γ value.

3.1.2.4 Prediction of CHF

In general, the prediction of CHF in eccentric annuli is based on concentric annuli. Tolubinskiy et al. (1971,1977) noted that the dependence of CHF on flow parameters such as quality, pressure, and mass flux diminishes with increasing eccentricity, and practically vanishes with full contact. Tolubinskiy et al. (1977) derived a correlation, which considered the effect of eccentricity:

$$\frac{CHF_{ann,eccn}}{CHF_{ann,conc}} = 1 - 0.58 G^{-0.37} \left(\frac{D_{hy}}{D_{he}} \right)^{0.5} L^{0.29} \left(\frac{D_{hy}}{D_{he}} \right) \epsilon^{0.5} (1 - 0.3 x)$$

Doerffer et al. (1994) examined the CHF data and found that eccentricity, the void migration parameter, and quality all have to be considered:

$$CHF_{ann,eccn} = CHF_{ann,conc} [1 - (1 - k_\epsilon) k_\gamma k_x]$$

where values of k_ϵ , k_γ and k_x are given in Table 3.2.

Table 3.2 Correction factors for eccentric annuli (from Doerffer et al., 1994)

Correction Factor	Value	Range
k_ϵ	$0.991 - 2.771\epsilon^{2.5} + 2.32\epsilon^3$	
k_γ	2.07	$\gamma < 0.0133$
	$-3.403 + 5.7588\exp(-\gamma/0.2615)$	$0.0133 \leq \gamma \leq 0.1183$
	0.26	$\gamma > 0.1183$
k_x	1.05	$x \leq 0.05$
	$1.2516 - 0.9111x^{0.5}$	$0.05 < x < 0.5$
	0.6	$x \geq 0.6$

3.1.3 Rectangular Ducts

The geometry parameters for rectangular ducts include equivalent diameter (hydraulic diameter or heated diameter), gap size (short size length), and the ratio of the two sides.

3.1.3.1 General Trend

In general, the analysis of experimental data has shown that the values of CHF in uniformly heated rectangular and triangular shaped cross sections are lower than those in round tubes. This effect is more significant in the subcooled region, where the CHF condition starts to develop in the corner regions and spreads to the sides.

De Bortoli et al. (1958) presented the CHF data in rectangular ducts. The long side was fixed at 25.4 mm long, while the gap was changed from 1.27, 1.40, 1.50, 2.46 to 2.56 mm. Only the two long sides of the channel were heated. The rectangular CHF data were compared with a circular tube. It was found that the CHF for round tubes is as much as 45% lower than that of a rectangular channel at the same inlet condition. However, a comparison for the same local condition was not performed. Pexton (1961) reanalyzed the De Bortoli's data and concluded that the CHF appears to increase with decreasing D_{hy} for fixed local condition.

Jacket et al. (1958) investigated the CHF in rectangular channels with both long sides heated under conditions of approximately zero exit quality. The heated width (22.3 mm) was slightly less than the channel width (25.4 mm) to eliminate corner effects. The gap sides were varied from 1.27, 1.40 to 2.46 mm. There was good agreement between CHF for the rectangular channel with a 2.5 mm gap and that of a round tube for the same local quality and equivalent diameter (D_{hy} and D_{he}).

Levy et al. (1959) investigated the CHF in a rectangular channel (63.5mm × 2.54 mm) with the entire wetted perimeter heated. It was found that CHF with vapor blanketing first occurred at the corners of the test section. The CHF value was about one third of that in a circular tube.

Chernobay (1970) investigated the CHF of a rectangular channel (3.6 mm × 1.8 mm). The fraction of heated perimeter was varied from 16.7% to 100%. In the case of uniform heating ($D_{hy} = D_{he} = 2.4$ mm), the CHF values of the rectangular channel were much lower than those of a round tube ($D = 2$ mm) with the same inlet subcooling. Joobar (1993) analyzed these results and concluded that the low CHF in the rectangular duct was due to the corner effect. This effect becomes less important at very high mass flux due to better mixing. Chernobay found that the CHF increased as the fraction of the perimeter being heated was decreased for the same inlet subcooling. The highest values of CHF were obtained when only 17% of the perimeter was heated. These results suggest that as the fraction of the perimeter being heated is decreased the CHF values for a rectangular channel approach those of a round tube. In case of only one short side heated, the CHF results were close to round tube CHF data with uniform heating and the same hydraulic diameter.

Zeigarnik et al. (1981) investigated the CHF in their rectangular channel (3.5 mm × 3.75 mm, all sides heated), and found that the CHF was about 20% lower than that of a circular tube with an equivalent diameter. Poor mixing at the corners resulted in local over-heating and thus reduced the CHF.

Katto (1981) investigated uniformly heated rectangular channels and applied his CHF characteristic regime to the rectangular ducts. A set of empirical correlations were developed for rectangular channels with two long sides heated and one long side heated.

3.1.3.2 Prediction of CHF

For CHF in a uniformly heated rectangular channel, Macbeth (1963) presented an empirical correlation based on the water data obtained by DeBortoli et al. (1958). For high mass flux ($G > 760$ kg/m² s):

$$CHF = 3.15 \frac{C_1 + 6.39 \times 10^{-6} C_2 \delta G \Delta h_{sub}}{1 + 39.37 C_2 L}$$

where CHF is in [MW/m²], δ and L are in [m], Δh is in [J/kg], G is in [kg/m²s], and C_1 and C_2 are given by

$$C_1 = y_0 (39.37 \delta)^{y_1} (7.37 \times 10^{-4} G)^{y_2}$$

$$C_2 = y_3 (39.37\delta)^{y_4} (7.37 \times 10^{-4} G)^{y_5}$$

y_i is given by Table 3.3.

Table 3.3 Coefficients for Macbeth (1963) correlation

Pressure (MPa)	Coefficient					
	y_0	y_1	y_2	y_3	y_4	y_5
4.15	23.4	-0.472	-3.29	0.123	-1.4	-3.93
5.5	0.445	-1.01	0.384	0.0096	-1.4	-0.0067
8.3	1.88	-0.081	-0.526	0.0035	-1.4	-1.29
13.8	0.546	-0.315	-0.056	0.0027	-1.4	-0.725

For low mass flux ($G < 760 \text{ kg/m}^2\text{s}$)

$$CHF = 0.56 \times 10^{-6} G \frac{h_{fg} + \Delta h_{sub}}{1.31 \times 10^{-6} \delta^{-1.73} G^{1.1} + \frac{L}{\delta}}$$

3.1.4 Dumbbell-Shaped Channels

Green et al. (1969) conducted CHF tests with vertical upflow of water in a special shaped channel to investigate the CHF characteristics of a coolant channel with small clearance dimensions. The channel, which is currently called a dumbbell-shaped channel, was used to simulate a typical reactor core coolant flow in a bundle of closely spaced rods. Two gap sizes, 0.03 and 0.015 inches (0.76 and 0.38 mm), were tested, simulating the actual rod-to-rod spacing. The measurement conditions were as follows:

- Pressure: 1200, 1600, and 2000 psi (8.3, 11.0, 13.8 MPa)
- Mass Flux: 0.25×10^6 to 4.0×10^6 lb/hr-ft² (339 to 542.5 kg/m²s)
- Inlet Temperature: 200, 400, 500, and 600 F (93 to 316 °C)

Table 3.4 and Table 3.5 list their CHF test results. Based on their 90 CHF data points (50 for the 0.03 inch gap, 40 for the 0.015 inch gap) and their rectangular channel data, obtained using 0.059 and 0.097 inch channel spacings, Green concluded that there is a negligible effect of channel spacing on the critical heat flux over the range of 0.015 to 0.097 inch (0.38

to 2.46 mm). For low mass flux ($G < 0.7 \text{ Mg/m}^2\text{s}$), dryout was detected simultaneously at the gap and at the main flow channel regions for the large gap channel (0.76 mm) at all pressures, and for the small gap size (0.38 mm) at a pressure of 8.2 MPa. For all other conditions, dryout first appeared in the gap region.

Table 3.4 CHF data for gap size of 0.76 mm (from Green et al., 1969)

P (psia)	h_l (Btu/lb)	$G \times 10^{-6}$ (lb/hr ft ²)	CHF $\times 10^{-6}$ (Btu/hr ft ²)	h_c (Btu/lb)
2000	612	0.247	0.0897	1081
2000	485	0.252	0.1142	1069
2000	377	0.253	0.1336	1058
2000	173	0.254	0.1682	1029
2000	613	0.499	0.1345	962
2000	482	0.501	0.1831	953
2000	377	0.504	0.2263	954
2000	174	0.505	0.3009	944
2000	613	1.047	0.2147	878
2000	485	1.046	0.3033	860
2000	377	1.045	0.3862	852
2000	174	1.044	0.5243	823
2000	614	2.078	0.3134	810
2000	485	2.094	0.4483	762
2000	377	2.1	0.5919	740
2000	173	2.091	0.8191	680
2000	614	4.067	0.4324	752
2000	487	4.183	0.6773	697
2000	377	4.176	0.9294	662
2000	171	4.19	1.4214	610
1600	483	0.251	0.1266	1134
1600	377	0.252	0.1438	1112
1600	172	0.254	0.1836	1106
1600	483	0.5	0.2064	1016
1600	376	0.502	0.2462	1007
1600	174	0.504	0.3209	994
1600	488	1.041	0.317	880
1600	377	1.045	0.3938	862

P (psia)	h_l (Btu/lb)	$G \times 10^{-6}$ (lb/hr ft ²)	CHF $\times 10^{-6}$ (Btu/hr ft ²)	h_c (Btu/lb)
1600	173	1.043	0.5388	840
1600	488	2.079	0.4565	771
1600	377	2.088	0.6175	758
1600	175	2.088	0.8644	709
1600	486	4.156	0.6608	692
1600	377	4.174	0.9225	661
1600	173	4.169	1.3592	594
1200	485	0.25	0.1321	1166
1200	376	0.252	0.1528	1158
1200	172	0.254	0.1913	1144
1200	484	0.5	0.245	1116
1200	374	0.504	0.2884	1112
1200	172	0.505	0.3662	1108
1200	488	1.038	0.3653	942
1200	376	1.042	0.445	926
1200	172	1.041	0.5881	900
1200	488	2.079	0.4759	783
1200	376	2.086	0.6171	757
1200	176	2.093	0.9004	731
1200	488	4.136	0.6348	686
1200	376	4.171	0.8779	647
1200	171	4.178	1.3523	589

Table 3.5 CHF data for gap size of 0.38 mm (from Green et al., 1969)

P (psia)	h_l (Btu/lb)	$G \times 10^{-6}$ (lb/hr ft ²)	CHF $\times 10^{-6}$ (Btu/hr ft ²)	h_c (Btu/lb)
2000	614	0.247	0.0869	1061
2000	488	0.248	0.1126	1066
2000	379	0.248	0.131	1050
2000	176	0.247	0.169	1045
2000	612	0.496	0.1341	956
2000	487	0.493	0.1828	958
2000	378	0.495	0.2203	943

P (psia)	h_l (Btu/lb)	$G \times 10^{-6}$ (lb/hr ft ²)	CHF $\times 10^{-6}$ (Btu/hr ft ²)	h_c (Btu/lb)
2000	176	0.494	0.2918	927
2000	616	0.995	0.202	874
2000	487	0.988	0.282	849
2000	376	0.994	0.3512	825
2000	615	1.986	0.3024	809
2000	486	1.975	0.4452	773
2000	379	1.983	0.556	736
2000	614	3.964	0.4306	752
2000	484	3.949	0.6911	707
2000	376	3.954	0.9327	677
2000	171	3.949	1.416	627
1600	378	0.992	0.3657	846
1600	179	0.991	0.5099	833
1600	178	1.972	0.8186	706
1600	486	3.932	0.6514	697
1600	377	3.946	0.9094	671
1600	173	3.951	1.4071	625
1200	489	0.2455	0.1312	1171
1200	378	0.2468	0.152	1161
1200	177	0.2468	0.1926	1168
1200	487	0.493	0.2453	1121
1200	378	0.4938	0.2847	1111
1200	178	0.4931	0.36	1105
1200	487	0.9876	0.3464	933
1200	379	0.9905	0.4126	908
1200	175	0.988	0.5486	880
1200	487	1.9682	0.444	774
1200	378	1.9653	0.5766	751
1200	377	1.9683	0.5832	753
1200	486	3.937	0.6193	686
1200	377	3.953	0.8556	653
1200	172	3.951	1.3382	603

Tong and Tang (1997) reviewed the gap effect based on Lee and Little's (1962) dumbbell section results. It was found that the minimum gap size also had little effect on CHF at 6.6 MPa with a 10% mean exit quality and a vertical upflow of water. The gap between the rod surfaces was varied from 0.81 to 5.6 mm.

3.1.5 Fuel bundles

The fuel bundle shape is very complex and many characteristic geometry parameters are involved. These parameters include:

- Gap size (rod to rod spacing, and rod to pressure tube spacing)
- Curvature of the surfaces
- Subchannel shape
- Equivalent diameter (hydraulic diameter, heated diameter)

Tong and Tang (1997) summarized the rod spacing effect on CHF in rod bundles based on the work of Towell (1965). The rod spacing effect was found to be insignificant for water at a mass flux of about 1356 kg/m²s, a pressure of 6.9MPa, and an exit quality of 15-45% in the star test section - the central seven-rods of a 1.22 m long 19-rod bundle. Towell tested rod spacings of 0.45-1.27mm, and found no effect on CHF at constant exit enthalpy whether the adjoining surfaces were heated or not. Tong et al. (1967) investigated the CHF in both square and triangular rod arrays having the same water-to-fuel ratio. Their tests were conducted in subcooled and low-quality water flow at 13.8 MPa, and obtained similar results (i.e., the rod spacing had little effect.)

Groeneveld et al. (1992) reviewed the effect of element-element bowing and element spacing. They concluded that the effect of gap and rod bowing on CHF is uncertain and strongly dependent on flow conditions.

Nagino et al. (1978) conducted DNB-type CHF experiments in a sixteen-rod bundle. One of the central heater rods was bowed into contact with another rod. They found that the rod bowing effect on CHF depended strongly on pressure. At 10.5 and 12.7 MPa there was no

effect of rod bowing on CHF, while at high pressure (14.8 and 16.9 MPa) there was a considerable decrease in CHF.

Markowski et al. (1977) investigated CHF on a five-by-five square rod array with several bowing configurations. They reduced the gap size between two heated rods, and between a heated rod and an unheated rod. Three gap sizes, 3.55 mm, 1.65 mm and zero gap, were tested at pressures from 12.1 MPa to 16.5 MPa. In the case of a heated rod bowed towards another heated rod, the CHF decreased when the bowing was greater than 50%. In the case of an unheated rod bowed towards a heated rod, no effect on CHF was found, even when the unheated rod was bowed into contact with adjacent heated rods.

Macduff and Fighetti (1983) studied the effect of rod bowing in a five-by-five square rod array and got similar results. They reduced the gap between two centrally located high-powered rods by 71% from 3.46 mm to 1 mm. Their results indicated that at a pressure of 16.6 MPa, the gap effect on CHF was very small (only 5%) and had no effect at 13.8 MPa.

Nakajima et al. (1977) conducted CHF tests in a 36-rods bundle to determine the effect of rod to rod bowing. The minimum gap between the outer and inner rod was reduced from 2.1 mm (nominal value) to 0 mm. Their results showed that rod bowing only had a slight effect on CHF.

Nixon et al. (1975) tested a full-scale eight-by-eight rod bundle. They obtained similar results. The effect of reduced gap and rod bowing had little or no effect on CHF over the conditions tested.

3.2 Modeling Studies

This section reviews the existing literature on dryout-type CHF modeling. Dryout-type CHF is the most likely to occur in practical applications, where the channel lengths are typically several meters long. As a typical case and original dryout CHF model, the Whalley et al. (1974) model will be described in detail.

3.2.1 Whalley's Model

Whalley et al. (1974) presented a model of annular two-phase flow at the 5th International Heat Transfer Conference. The model was developed to predict dryout CHF in a vertical circular tube under steady state flow conditions. The essential functionality of the model is to analyze the phase distribution evolution along the uniformly heated tube based on mass balances. The detailed variation of droplet entrainment from the liquid film and droplet deposition to the liquid film is considered.

Whalley (1977) applied his dryout model to rod bundles by dividing the rod bundle into rod-centered subchannels. Whalley et al. (1978) modified and extended their model to cope with different boundary conditions, and time dependent situations.

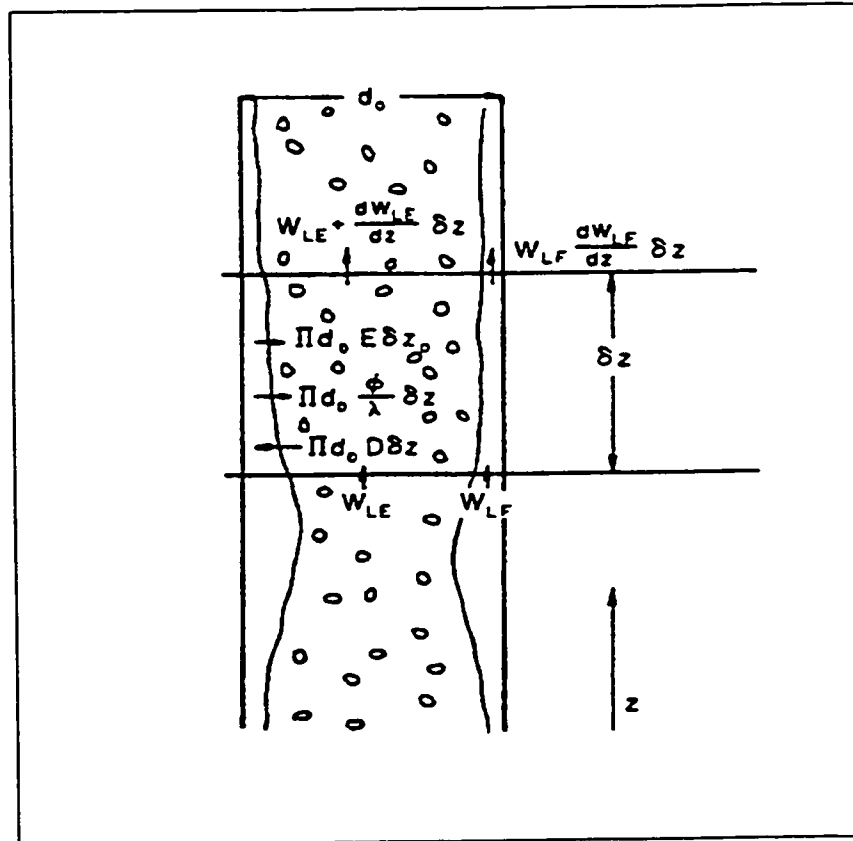


Figure 3.2 Annular flow (from Whalley, 1977).

As shown in Figure 3.2, the mass balance on the liquid film and on the entrained liquid in the core of an annular flow can be written as

$$\frac{dW_{LF}}{dz} = \pi d_o \left[D - E - \left(\frac{q''}{h_{fg}} \right) \right]$$

where W_{LF} is the liquid mass flux in liquid film along the wall at z . The integration of W_{LF} should be continued until the film flow rate becomes zero, when it is assumed that dryout has occurred. D is the rate of droplet deposition per unit area of the tube wall, E is the rate of entrainment per unit area of tube wall. q'' is the heat flux and h_{fg} is the latent heat of evaporation.

The deposition rate, D , is proportional to the concentration of droplets in the homogeneous core.

$$D = k C$$

k is the mass transfer coefficient and C is the concentration of entrained liquid droplets in the core region. The concentration of droplets C is obtained based on homogenous flow:

$$C = \frac{W_{LE}}{u_{GC} A_{GC}} = \frac{W_{LE}}{u_{GC} (A_G + A_{LE})} = \frac{W_{LE}}{u_{GC} A_G + u_{GC} A_{LE}} = \frac{W_{LE}}{u_G A_G + u_{LE} A_{LE}}$$

which reduces to

$$C = \frac{W_{LE}}{\frac{W_G}{\rho_G} + \frac{W_{LE}}{\rho_L}}$$

The deposition coefficient (k) is obtained from measurements. Figure 3.3 shows the experimental data for the deposition coefficient plotted against surface tension. The relationship is also shown in Table 3.6.

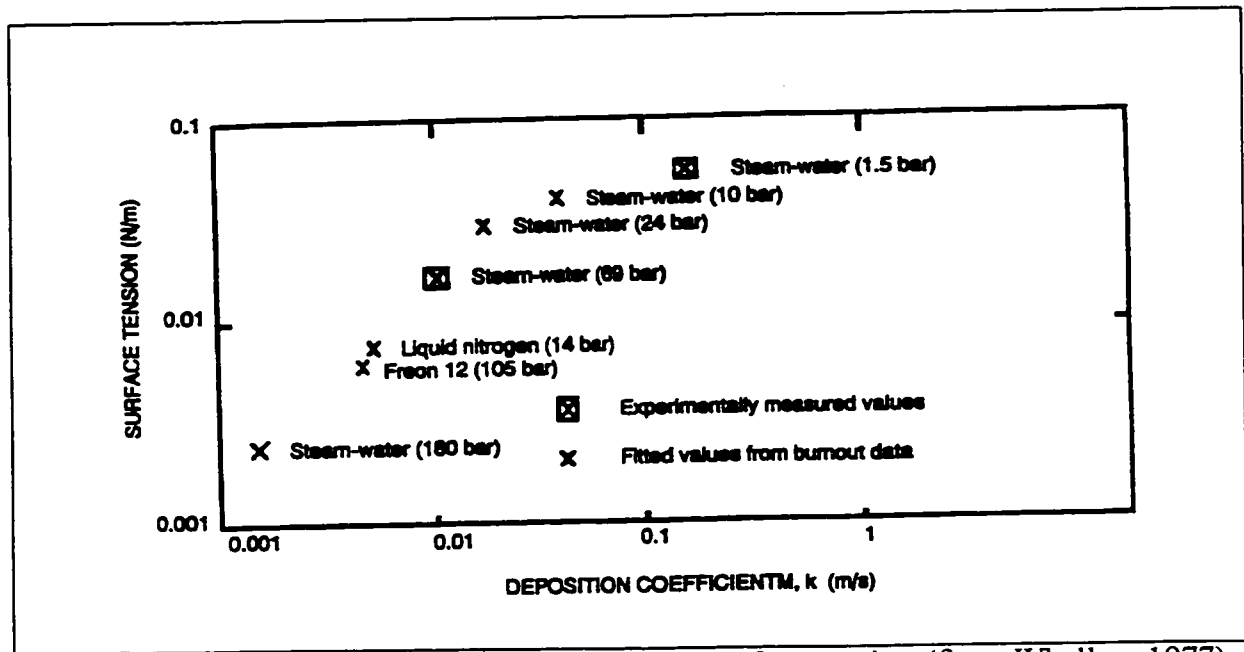


Figure 3.3 Variation of deposition coefficient with surface tension (from Whalley, 1977).

This relationship can be approximated by the formula:

$$k = f(\sigma)$$

σ is the surface tension. Vaitekunas (1996) fitted the following logarithmic polynomial to this experiment data:

$$\log_{10} k = 2.8163 + 4.0685 \log_{10} \sigma + 0.743 (\log_{10} \sigma)^2$$

Table 3.6 Variation of deposition coefficient with surface tension

Surface tension σ (N/m)	Deposition Coefficient k (m/s)	Surface tension σ (N/m)	Deposition Coefficient K (m/s)	Surface tension σ (N/m)	Deposition Coefficient k (m/s)
0.002	0.0011	0.008	0.005	0.45	0.04
0.003	0.002	0.009	0.0055	0.05	0.06
0.004	0.0024	0.01	0.006	0.055	0.085
0.005	0.003	0.02	0.011	0.06	0.14
0.006	0.0035	0.03	0.02	0.063	0.2
0.007	0.0043	0.04	0.027	0.07	0.5

The entrainment rate E is best measured using adiabatic hydrodynamic equilibrium flow. In the case of adiabatic flow in very long channels, hydrodynamic equilibrium can be established, and thus the deposition rate (equal to the entrainment rate) can be measured. E is proportional to C_E if k is a constant. The entrainment rate is therefore obtained using the following simple formula:

$$E = kC_E$$

It was found that the concentration of droplets in the gas core at hydrodynamic equilibrium is a function of the non-dimensional group $(\tau_i \delta / \sigma)$. Here, τ_i is the interfacial shear stress, σ is the surface tension, and δ is the liquid film thickness. Whalley et al. (1978) compiled the experimental data from nine sources, and the resultant C_E are presented in Figure 3.4.

Tahir et al. (1987) fitted the following correlation to the data:

$$C_E = \begin{cases} 0.036 - 4.22 \left(\frac{\tau_i \delta}{\sigma} \right) + 253 \left(\frac{\tau_i \delta}{\sigma} \right)^2 - 427 \left(\frac{\tau_i \delta}{\sigma} \right)^3, & \left(\frac{\tau_i \delta}{\sigma} \right) < 0.3 \\ 10^{0.33 + 2.22 \left(\frac{\tau_i \delta}{\sigma} \right)}, & \left(\frac{\tau_i \delta}{\sigma} \right) \geq 0.3 \end{cases}$$

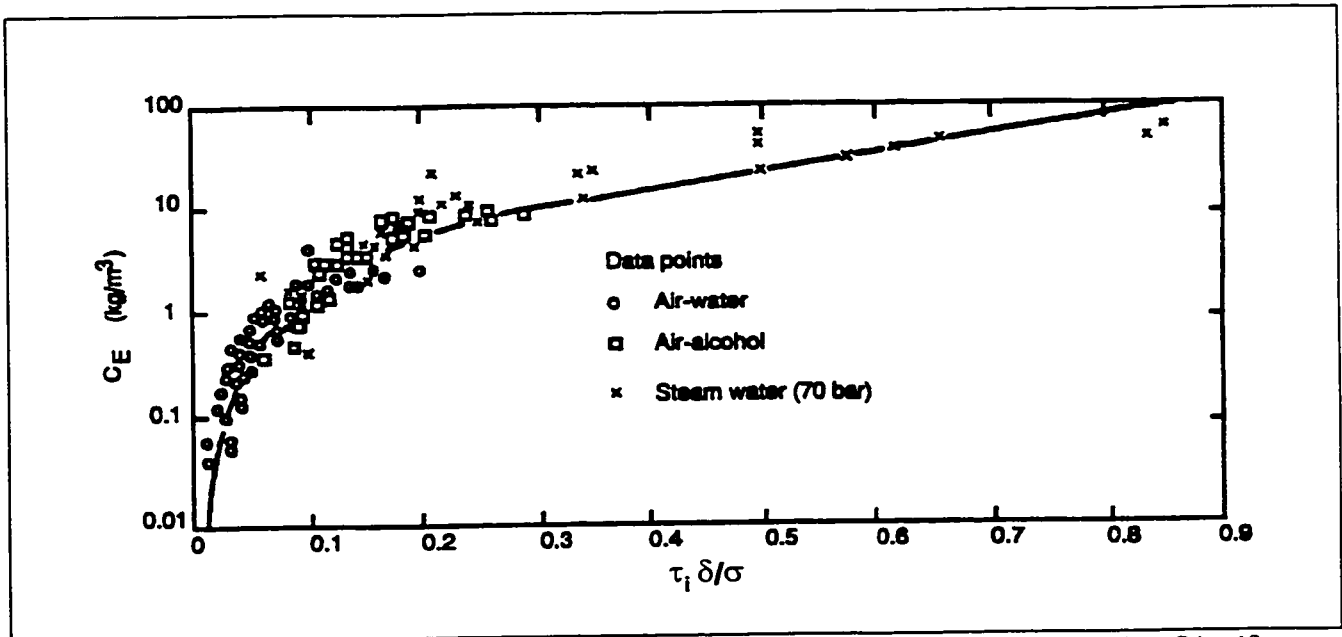


Figure 3.4 Variation of equilibrium concentration of entrained droplets with $\tau_i \delta / \sigma$ (from Whalley, 1977).

To obtain the entrainment rate, the interfacial shear stress and the film thickness are required. The three dependent variables being considered are W_{LF} , δ , and dP/dz , which can be related in such a way that any one of them can be calculated knowing the other two. This kind of 3-way relationship is termed a “triangular relationship”, and its use can be illustrated by considering the calculation of W_{LF} from δ and dP/dz .

This is based on the assumption that some known velocity profile is occurring in the liquid film, from which it follows that the interfacial shear stress and the film thickness can be related. The particular form of this triangular relationship is:

$$\frac{4\delta}{d_o} = \sqrt{\frac{(dP/dz)_{LF}}{dP/dz}}$$

where $(dP/dz)_{LF}$ is the pressure gradient which would occur in a single-phase liquid flow were the mass flow identical to that of the liquid film, and dP/dz is the pressure gradient in the two-phase flow. $(dP/dz)_{LF}$ was calculated from the liquid film fraction factor proposed by Hewitt and Hall Taylor (1970):

$$-\left(\frac{dP}{dz}\right)_{LF} = \frac{4f_{LF}}{d_o} \left(\frac{1}{2}\rho_L V_{LF}^2\right)$$

where superficial velocity is given by

$$V_{LF} = \frac{W_{LF}}{A\rho_L}$$

To calculate f_{LF} , the liquid film Reynolds number is required, thus:

$$\text{Re}_{LF} = \frac{W_{LF} d_o}{A\mu_L}$$

For turbulent flow, the friction factor could be calculated from the Blasius equation:

$$f_{LF} = 0.079 \text{Re}_{LF}^{-0.25}$$

For laminar flow,

$$f_{LF} = \frac{16}{\text{Re}_{LF}}$$

If the liquid film flow rate is known, to calculate the average film thickness and the shear stress (and hence the pressure gradient), one more equation is needed in addition to the triangular relationship. This is the interfacial roughness correlation. Wallis (1970) correlated a variety of data for annular flow using the following simple relation:

$$f_{gci} = f_{sgc} \left[1 + 360 \frac{\delta}{d_o} \right]$$

where f_{gci} is the interfacial friction factor and f_{sgc} is the friction factor for a smooth pipe.

$$f_{gci} = \frac{\tau_i}{\frac{1}{2} \rho_{gc} V_{gc}^2}$$

and ρ_{gc} is the homogeneous gas core density,

$$\rho_{gc} \approx \rho_G \left[\frac{W_G + W_{LE}}{W_G} \right]$$

$$V_{gc} = \frac{4W_G}{\pi \rho_G D^2}$$

The friction factor for a smooth pipe flow f_{sgc} can be calculated based on the gas-core Reynolds number in terms of the total mass flow in the core as follows:

$$Re_{sgc} = \frac{V_{gc} \rho_{gc} d_o}{\mu_G}$$

The calculation is started at the point in the tube where the quality reaches 1%, and it is generally found that the best results are produced when it is assumed that 99% of liquid phase is entrained at this starting point.

Whalley et al. (1974) did not give details on how (dP/dz) was calculated for a circular tube. Whalley (1977) gave a little more detail about how the pressure gradient (dP/dz) is calculated. He assumed that the interfacial shear stress was almost equal to the wall shear stress, claiming this assumption was good enough for thin liquid films. The friction pressure gradient can then be calculated from the hydraulic diameter and interfacial shear stress.

3.2.2 Saito's Model

3.2.2.1 Physical Model

Saito et al. (1978) developed multi-fluid models for annular two-phase flow and applied their models to predict dryout-type CHF in round tubes, annuli, and rod bundles. In a given channel, the liquid film(s) on the wall, the vapor core and the entrained liquid droplets are all considered a separate “fluid”. For the case of annular flow in a circular tube, a three-fluid (the liquid film, the vapor core, and entrained droplets) model was obtained. Annular flow in a concentric annulus required a four-fluid model (the two liquid films, vapor core, and entrained droplets).

To derive the multi-fluid physical model, Saito made following basic assumptions about annular flow:

- Steady-state;
- One-dimensional;
- Constant flow cross-sectional area;
- Gravity is the only body force;
- Vapor phase is always saturated;
- No liquid superheating;
- The entrained droplets do not contribute to vapor generation;
- Viscous dissipation is negligible.

In order to determine the flow characteristics from the subcooled inlet up to the annular flow regime transition, they used a conventional single-phase flow model and a homogenous two-phase flow model. Based on a review of several flow regime maps, the transition to annular flow was assumed to occur at a thermodynamic quality of 5%. From this location onward, the multi-fluid model is used. The initial conditions for the multi-fluid model are based on the following assumptions,

- The velocity ratio of entrained liquid and gas is 0.9;
- If there are more than two films, as in annuli, all film thickness are assumed to be equal and also all film velocities are assumed to be equal;

- The volumetric fraction of the entrained liquid α_E is given by

$$\alpha_E = (1.0 - \alpha_G) F_S$$

where F_S is a liquid flow split factor, specified to be 0.1.

3.2.2.2 Mechanisms Review of CHF in Annular Two-phase Regime

Saito et al. summarized the mechanisms of CHF in annular two-phase flow regime based on Tong and Hewitt's review (1973). The following are possible causes of CHF in the annular flow condition:

- a) Film boiling transition underneath the liquid film;
- b) Film disruption by hydrodynamic instability of the liquid film;
- c) Spontaneous breakdown of the liquid film;
- d) Breakdown due to thermal capillary effects;
- e) Equality between droplet deposition and evaporation;
- f) Dryout between waves;
- g) Liquid entrainment with no subsequent deposition;
- h) Film disruption due to nucleate boiling within the film;
- i) Integral entrainment-deposition effect leading to zero film flow rate at the point of onset of CHF.

Most of the investigation results have shown that CHF occurs when the liquid film becomes too thin and breaks down into rivulets around dry patches. In general, the condition of zero, or near zero, liquid film flow can be used as a criterion for CHF.

3.2.2.3 Formulation

The differential equations for continuity, momentum and energy conservation are then written for each fluid. In friction loss calculations, they assumed the entrained droplets to be spherical and of uniform size, and the liquid film-gas core interface to be smooth.

Saito et al. (1978) compiled information from deposition experiments. They found that the following factors may affect the deposition rate,

- Droplet concentration;

- Droplet diameter;
- Turbulence intensity in the vapor core;
- Evaporation rate of the liquid film;
- Total mixture flow rate;
- Gas core flow rate;
- Geometry of the channel wall.

Because the droplet concentration varies largely over the boiling length, the available data indicate that the deposition coefficient is a function of the concentration. All other effects are assumed negligible.

For round tubes,

$$k = 1.698 \times 10^{-2} - 2.117 \times 10^{-4} (C_H) + 2.896 \times 10^{-4} (C_H)^2 \\ - 2.192 \times 10^{-8} (C_H)^3 + 7.778 \times 10^{-11} (C_H)^4 - 1.027 \times 10^{-13} (C_H)^5$$

For annuli,

$$k = 1.1474 \times 10^{-2} - 1.2854 \times 10^{-4} (C_H) + 1.0150 \times 10^{-6} (C_H)^2 \\ - 4.2501 \times 10^{-9} (C_H)^3 + 6.8196 \times 10^{-12} (C_H)^4$$

C_H is the homogenous concentration of liquid in the core region.

Entrainment of liquid from the surface of the liquid film is related to the complex geometry and wave characteristics of the surface of the film. In boiling annular flow the liquid film thickness, gas velocity and heat flux vary along the channel, and the mechanisms of entrainment may also vary. The following phenomena are considered as possible mechanisms of entrainment,

- Break-up of small unstable waves;
- Break-up of large disturbance waves;
- Break-up of liquid slug at slug-annular transition;
- Emission of liquid due to bursting of bubbles at liquid surfaces;
- Emission of liquid due to nucleation at the wall-liquid interface.

It is now believed that the break-up of large disturbance waves is the most important mechanism over wide ranges of conditions. Saito et al. derived a series of expressions to correlate the equilibrium entrainment rate using the data from Hutchinson and Whalley (1973):

$$E = -2.695 \times 10^{-3} - 1.104 \left(\frac{\tau_i \delta}{\sigma} \right) + 4.444 \times 10 \left(\frac{\tau_i \delta}{\sigma} \right)^2 \\ + 1.241 \times 10^3 \left(\frac{\tau_i \delta}{\sigma} \right)^3 - 3.775 \times 10^3 \left(\frac{\tau_i \delta}{\sigma} \right)^4 \\ + 4.007 \times 10^3 \left(\frac{\tau_i \delta}{\sigma} \right)^5 - 1.510 \times 10^3 \left(\frac{\tau_i \delta}{\sigma} \right)^6 \quad \text{if} \quad \left(\frac{\tau_i \delta}{\sigma} \right) < 1$$

$$E = -4.504 \times 10 + 1.824 \times 10^2 \left(\frac{\tau_i \delta}{\sigma} \right) - 6.828 \times 10 \left(\frac{\tau_i \delta}{\sigma} \right)^2 \\ + 1.343 \times 10 \left(\frac{\tau_i \delta}{\sigma} \right)^3 \quad \text{if} \quad 1 < \left(\frac{\tau_i \delta}{\sigma} \right) \leq 5$$

$$E = 4.670 \times 10 + 4.725 \times 10 \left(\frac{\tau_i \delta}{\sigma} \right) - 4.534 \left(\frac{\tau_i \delta}{\sigma} \right)^2 \\ + 1.518 \times 10^{-1} \left(\frac{\tau_i \delta}{\sigma} \right)^3 \quad \text{if} \quad 5 < \left(\frac{\tau_i \delta}{\sigma} \right) \leq 10$$

$$E = 0.01 \rho_L \left\{ 1 - \exp \left[-4.605 \left(\frac{\tau_i \delta}{\sigma} - 10 \right) \right] \right\} \quad \text{if} \quad \left(\frac{\tau_i \delta}{\sigma} \right) > 10$$

In applying their model to rod arrays, Saito et al. used conventional (classical) subchannels and assumed zero shear surfaces to coincide with planes which were equidistant from the channel walls. They also assumed that there was no mixing between subchannels.

3.2.2.4 Modeling Results

In the case of circular tubes, the multi-fluid model predicts liquid film dryout generally within about 10%. A slight error trend with heat flux indicates that nucleation-induced disruption of the film may be present at higher heat flux. In the case of annuli, the model predicts CHF with an accuracy of 15%. The model tends to underestimate CHF for lower heat flux cases and overestimate CHF for high heat flux cases. When the model was applied

to the prediction of CHF in rod bundles, the model underestimated CHF (up to 50%) for all low mass fluxes tested ($G < 500 \text{ kg/m}^2\text{s}$). This inaccuracy may be due to an inaccurate entrainment correlation or the lack of subchannel cross-flow in the model.

3.2.3 Levy's Model

Levy, Healzer, and Abdollahian (1980) predicted the dryout flux in vertical pipes by a semi-empirical adiabatic model of liquid film flow and entrainment. It starts with a heat balance along the flow direction and a mass balance perpendicular to the flow direction:

$$G(\Delta h_{sub} + X h_{gr}) = \frac{2}{r} \int_0^z q'' dz$$

$$\frac{dG_{LF}}{dz} = 2\pi r \left[D - E - \left(\frac{q''}{h_{fg}} \right) \right]$$

Instead of using a constant value of the mass transfer coefficient k at each pressure given by Harwell (Whalley et al., 1973), Levy and Healzer (1980) developed an entrainment parameter β . G_{LF} and β are evaluated by solving the following two equations simultaneously:

$$G(1 - X_{eq}) - G_{LF} = G(1 - X_{eq})(\beta)^{-1/2}$$

G is the total mass flow rate in an adiabatic flow, X_{eq} is the equilibrium quality, and

$$\beta = 1 + \left\{ \left[\left(\frac{\rho_L}{\rho_G} \right)^{1/\beta} - 1 \right] \left(\frac{\sigma}{k_r} \right) \frac{\rho_L}{(GX_{eq})^2} \right\}^{1/2}$$

For a very thin liquid film, the value of β cannot be evaluated, and must be replaced by a new parameter β' , derived using the general turbulent boundary-layer profile in adiabatic flow.

3.2.4 Govan's Model

Govan et al. (1988) conducted a study relating to the Harwell annular flow code HANA. The code predicts the onset of the CHF in an evaporating liquid-liquid upflow in a vertical channel by calculating the point at which the liquid film flow rate reduces to zero, allowing for evaporation, droplet entrainment and droplet deposition. Then, based on new correlation

for entrainment and deposition, an improved version of the CHF modeling code was developed which showed good predictions not only for equilibrium entrainment in adiabatic flow but also for entrainment and CHF in heated tubes.

Hewitt and Govan (1990) used the following correlation for the deposition rate coefficient:

$$k = \begin{cases} 0.1815 \sqrt{\frac{\sigma}{\rho_G d}}, & \text{if } \frac{C}{\rho_G} \leq 0.3 \\ 0.1815 \sqrt{\frac{\sigma}{\rho_G d}} \left(\frac{C}{\rho_G} \right)^{-0.65}, & \text{if } \frac{C}{\rho_G} > 0.3 \end{cases}$$

The entrainment model departs from the original equilibrium flow approach to one relating the entrainment rate-to-vapor flow fraction to the liquid film flow rate:

$$\frac{E}{G_G} = \begin{cases} 5.75 \times 10^{-5} \left[(G_{LF} - G_{LFC})^2 \frac{d\rho}{\sigma \rho_g^2} \right]^{-.316}, & G_{LF} > G_{LFC} \\ 0, & G_{LF} \leq G_{LFC} \end{cases}$$

where the critical film mass flux for onset of entrainment is given by

$$\frac{G_{LFC} d}{\mu_L} = \text{Re}_{LFC} = \exp \left[5.8504 + 0.4149 \left(\frac{\mu_G}{\mu_L} \right) \sqrt{\frac{\rho_L}{\rho_G}} \right]$$

3.2.5 Lim's model

Lim and Weisman (1988) presented a model for rod bundle dryout that is similar to that used by Whalley (1977), but avoided the limitations of this earlier model. They found that the Whalley model made a very arbitrary assumption about the onset of annular flow. In contrast to Whalley's assumption that both droplets and vapor are redistributed, they assume that only vapor flows between the channels.

For the sake of simplicity, the effects of mixing and cross flow are considered for the various channels after the subchannel computations are completed. Since it is assumed that the

conditions are uniform in the droplet-droplet core of a channel, channel average (homogeneous) conditions are first determined by averaging the steam qualities and mass fluxes of the component subchannels. The interchannel droplet mixing is then treated in the same manner suggested by Whalley.

They followed Levy's approach to calculate the onset of annular flow. Initially, it was assumed that the annular flow began when any channel reached $\alpha = 0.8$. However, it was found that in some cases, that the method failed to produce dryout prediction with this initial condition. When the onset of annular flow was based on the lowest void channel having an $\alpha = 0.8$, prediction could be obtained for all cases.

3.2.6 Sugawara's Model

Sugawara (1990a) developed a dryout-type CHF prediction model for their Film Dryout Analysis Code in Subchannels (FIDAS). The model is based on three-fluid/three-field models, assuming that the fluid consists of a liquid film, a continuous vapor, and entrained liquid droplets, as shown in Figure 3.5. Sugawara presented a new formulation for the entrainment and deposition relationship, as well as thermal non-equilibrium effects and inter-subchannel mixing.

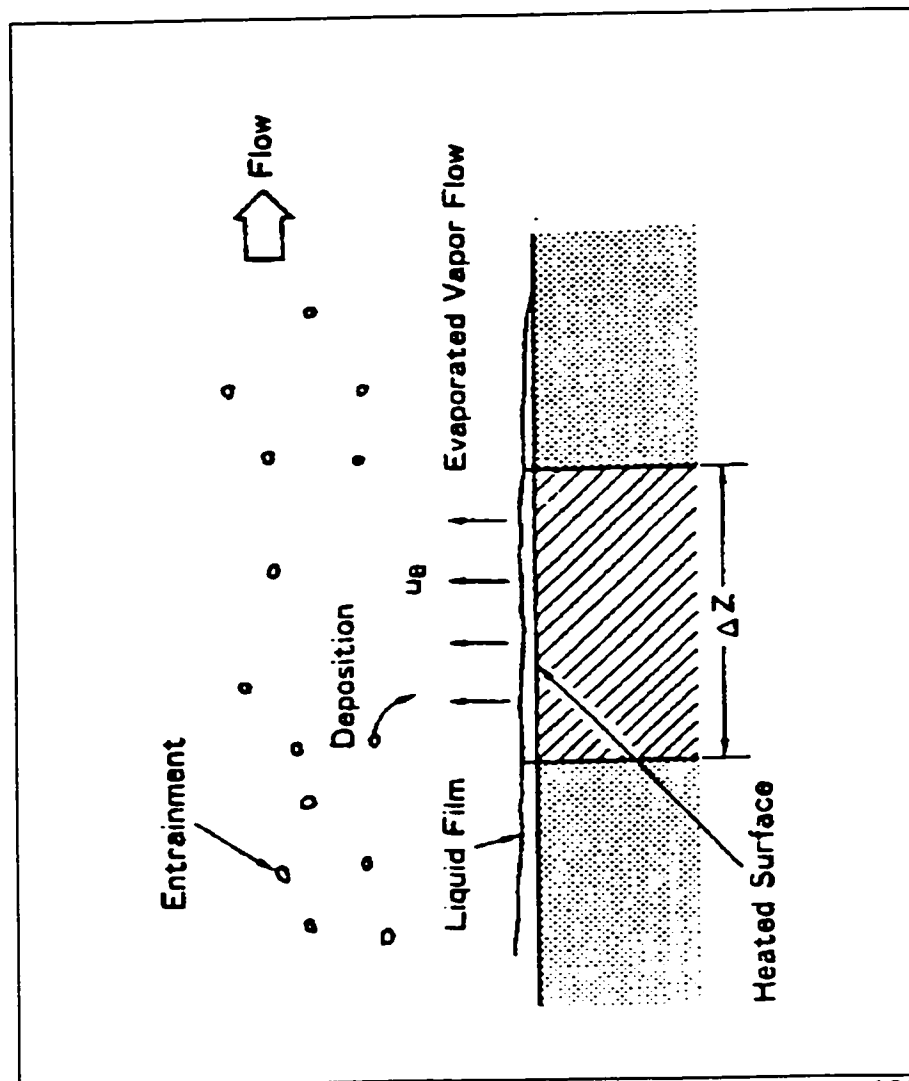


Figure 3.5 Film dryout model in FIDAS code (from Sugawara, 1990a).

When describing interfacial mass transfer, Sugawara (1990a) considered the suppression phenomenon of droplet deposition by evaporation. A new term, the suppression rate of droplet deposition by evaporation per unit area was added, which is a function of radial vapor velocity and liquid droplet concentration in the vapor core.

Sugawara used the analogy between heat and mass transfer and the Lewis Number to determine the deposition coefficient. The assumption is made that the thermal diffusivity and mass diffusivity of the fluid are equal. The temperature and concentration profiles are therefore similar so that the heat transfer relation can be applied to mass transfer. The deposition coefficient k is estimated using the following correlation:

$$\frac{k}{u} = 9.0 \times 10^{-3} \left(\frac{C}{\rho_G} \right)^{-0.5} \text{Re}_G^{-0.2} \text{Sc}^{-2/3}$$

where,

$$C = \frac{W_{LE}}{\frac{W_G}{\rho_G} \cdot \frac{u_{LE}}{u_G} + \frac{W_{LE}}{\rho_L}}$$

The entrainment rate E is estimated from the following correlations:

$$E = 1.07 S_R$$

$$S_R = \left(\frac{\tau_i \Delta l_{eq}}{\sigma} \right) \left(\frac{u_G \mu_{LE}}{\sigma} \right) \left(\frac{\rho_L}{\rho_G} \right)^{0.4}$$

$$\Delta l_{eq} = \begin{cases} k_s & \text{if } (\text{Re}_G \geq 1 \times 10^5) \\ k_s (2.13 \log_{10}(\text{Re}_G) - 9.8) & \text{if } (\text{Re}_G < 1 \times 10^5) \end{cases} \quad (\Delta l_{eq} \geq 0.0)$$

$$k_s = 0.57 \delta + 21.73 \times 10^3 \delta^2 - 38.8 \times 10^6 \delta^3 + 55.68 \times 10^9 \delta^4$$

where τ_i is the interfacial shear stress between liquid film and gas core, Δl_{eq} is the equivalent height of roll waves, δ is the mean film thickness, σ is the surface tension and k_s is the sand roughness of roll waves.

Annular flow is assumed to start when α reaches the value obtained from the following equation formulated from the slug/annular transition boundary in the well-known Baker flow regime diagram.

$$\alpha_{ONAF} = \left(P + \frac{0.9}{P_0} \right) e^{-\frac{G}{G_0}} + \left(3.1 - \frac{P}{P_0} \right)$$

where P_0 and G_0 are reference pressure and mass flux at 4.0 MPa and 729 kg/m²s respectively.

Chapter 4

EXPERIMENTAL STUDY

This research work is divided into two parts: experimental investigation and modeling study. The experimental investigation focuses on acquiring and analyzing the dumbbell-shaped tube CHF data. The modeling study focuses on developing the complex geometry CHF model and its application to the dumbbell-shaped flow channel. This chapter describes the experiments conducted.

4.1 General

The main purpose of this experiment is to systematically investigate the effect of gap size on CHF. The CHF experiments were conducted using dumbbell-shaped tubes. As shown in Figure 4.1, the dumbbell-shaped tube consists of two subchannels interconnected via a narrow gap. Its shape closely resembles two adjacent subchannels in a nuclear fuel-rod bundle. The narrow gap region simulates the gap between the two rods. Zero gap simulates the situation in which the fuel rods bow and come into contact with each other.

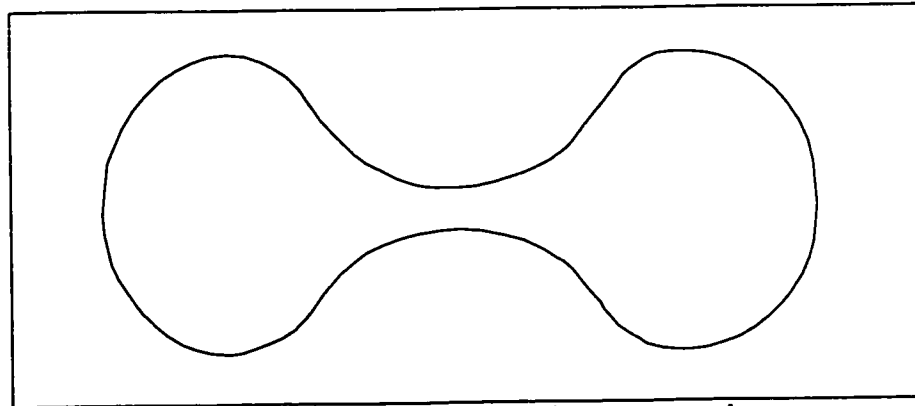


Figure 4.1 Dumbbell-shaped tube cross section.

The CHF measurements were conducted using various gap sizes at different flow conditions. Wide ranges of the narrow gap size (from 0 to 1 mm), pressure, mass flux and quality were

covered.

4.2 Dumbbell-Shaped Tube

4.2.1 Geometry Parameters

It is assumed that the cross section of the dumbbell-shaped tube is composed of three sets of tangential circles and two sets of tangential lines, as shown in Figure 4.2.

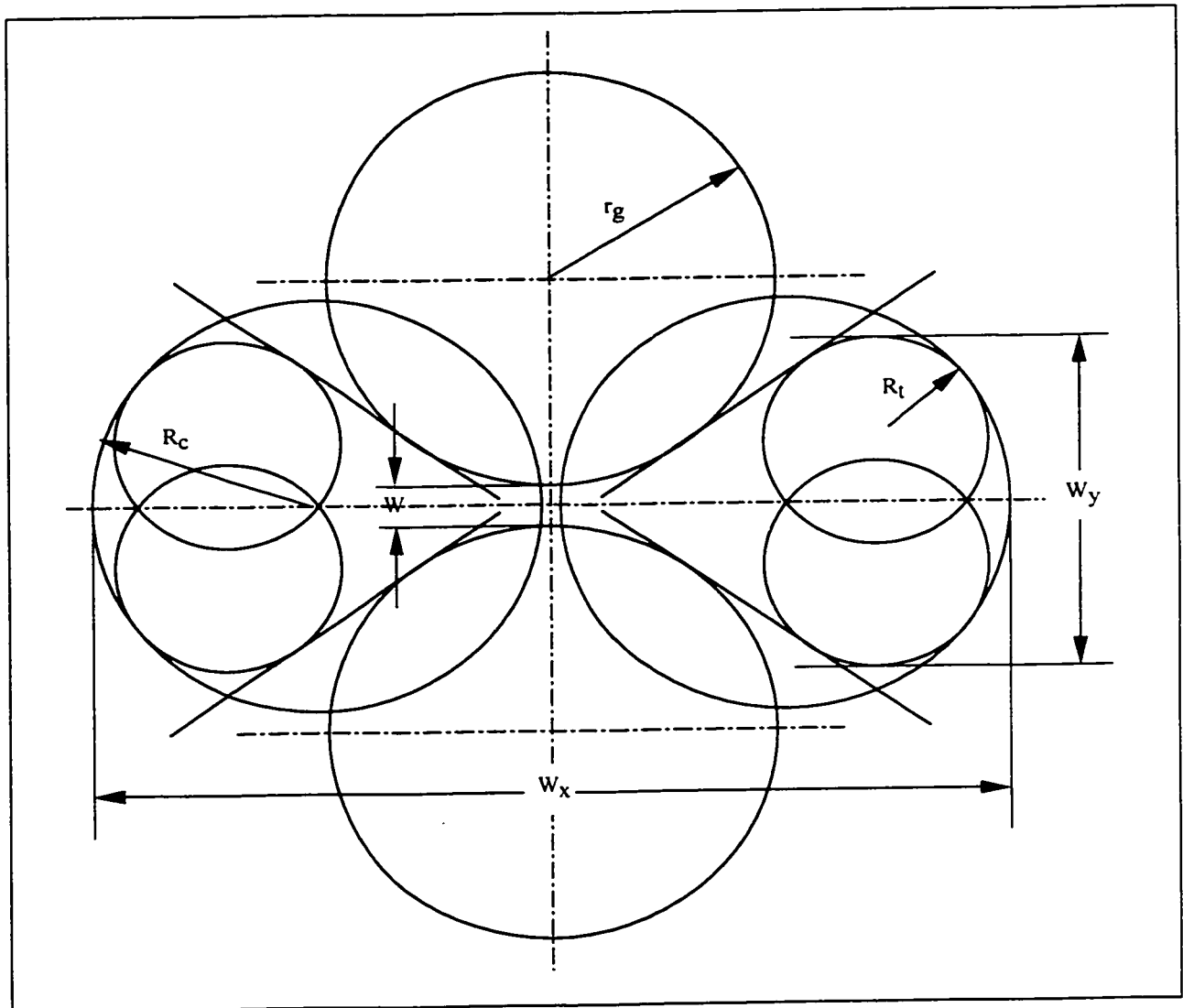


Figure 4.2 Dumbbell-shaped tube geometry parameters.

By analyzing the diagram, it is found that the geometric contour of the dumbbell-shaped tube is

determined uniquely by the following six measurable geometric parameters:

- the maximum width of the tube in x direction W_x . Note that for the inner surface, $w_x = W_x - 2\delta_{\text{wall}}$;
- the maximum width of the tube in y direction W_y . Note that for the inner surface, $w_y = W_y - 2\delta_{\text{wall}}$;
- the minimum gap width W_g , which is the minimum gap size of the tube plus the tube thickness. Note that for the inner surface, the gap size $\delta = W_g - 2\delta_{\text{wall}}$;
- the radius of gap region circle r_g (inner surface $R_g = r_g + \delta_{\text{wall}}$);
- the radius of main channel circle R_c (inner surface $r_c = R_c - \delta_{\text{wall}}$);
- the radius of transition circle R_t (inner surface $r_t = R_t - \delta_{\text{wall}}$).

For the dumbbell-shaped tube, the radius of the transition circle R_t is much smaller than the radius of the main channel circle R_c . It is noted that all these measurable parameters are for the outer surface of the dumbbell-shaped tube. The flow area and hydraulic diameter of the dumbbell tube are calculated based on the parameters of the inner surface, which can be derived from the above geometric parameters and the tube wall thickness. These parameters were measured using a micrometer and radius gauge at different gap sizes.

4.2.2 Flow Area

To compute the flow area of the dumbbell-shaped tube, only a quarter of the dumbbell-shaped tube needs to be considered due to the symmetry of the geometry. The flow area is the area enclosed by the inner surface of the dumbbell-shaped tube. The shape of the surface is determined by the six measurable parameters: w_x , w_y , w_g , R_g , r_c , and r_t .

The dumbbell-shaped tube is a complex geometry and there is no simple equation available to compute the area. As shown in Figure 4.3, the cross section can be broken up to four regions: quadrangle $OP_5P_3P_4$, pentagon $O_2O_3P_2P_3P_5$, sector $P_0O_2P_1$, sector $P_1O_3P_2$. Then the flow area of the dumbbell-shaped tube is given by

$$S = 4 \times (S_{OP_5P_3P_4} + S_{O_2O_3P_2P_3P_5} + S_{P_0O_2P_1} + S_{P_1O_3P_2})$$

where

$$S_{OP_5P_3P_4} = S_{O_1OP_5} - S_{P_3O_1P_4}$$

These areas can be obtained if the co-ordinates of the characteristic points O, O₁, O₂, O₃, P₁, P₂, P₃, P₄, P₅ are known. The co-ordinates can be derived by determining the angles (α_1 , α_2 , and β_1) and the six measurable parameters.

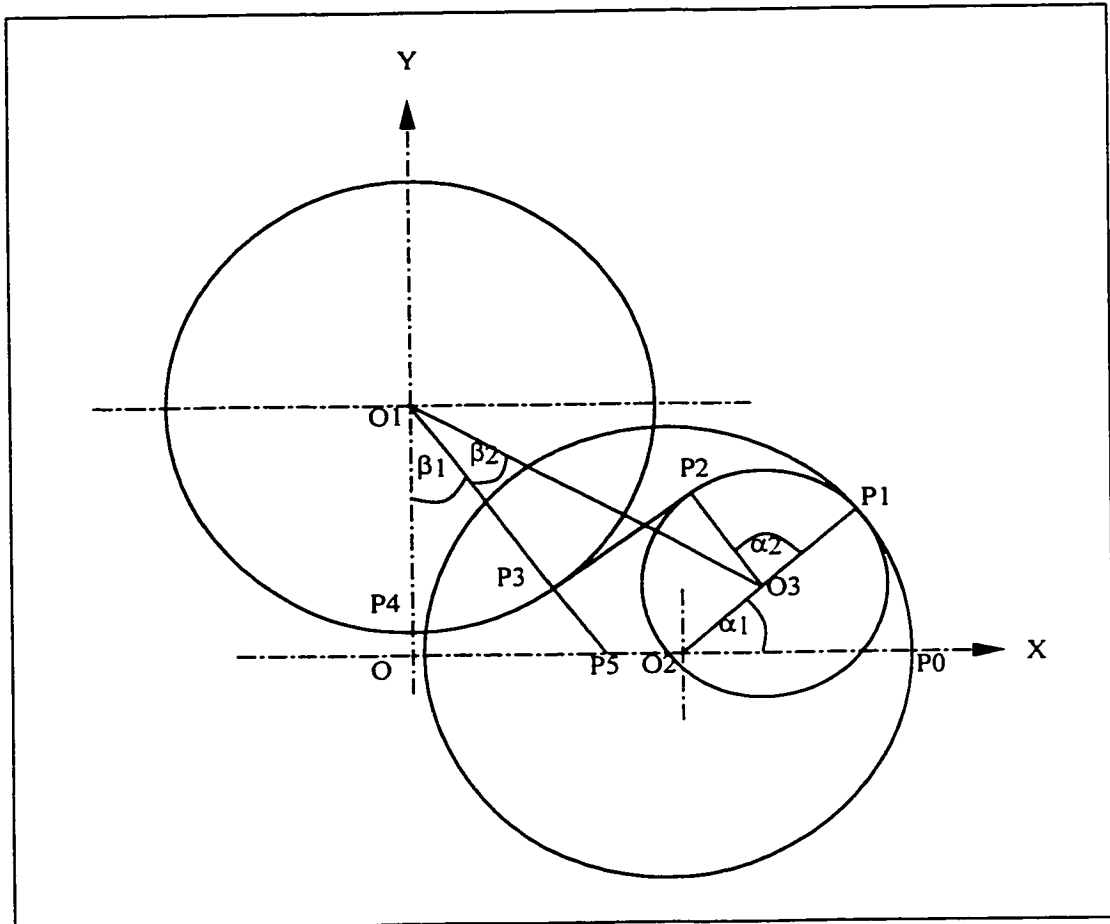


Figure 4.3 Dumbbell-shaped flow area calculation.

The flow area can also be calculated through numerical integration over the curve P₄P₃P₂P₁P₀. The mathematical functions for the section P₄P₃(circle O₁), P₃P₂(straight line), P₂P₁(circle O₃), P₁P₀(circle O₂) are obtained according to the co-ordinates of the characteristic points O₁, O₂, O₃, P₁, P₂, P₃, and P₄.

4.3 Working Fluid

R-134a (HFC-134a) was selected as a working fluid for this experimental investigation. As described in Chapter 1, heavy water is normally used as primary coolant for nuclear reactors.

However, modeling fluids (also called “scaling fluids”) are normally used as a working fluid for CHF experiments based on the CHF fluid-to-fluid scaling theories. Refrigerants, such as R-12 (CHC-12), have been widely used as modeling fluids.

There are numerous advantages of using modeling fluids to measure CHF over water. The CHF experiment will cost dramatically less because of the lower pressure, lower temperature and heat flux involved. For example, the CHF in R-134a at 1.7 MPa corresponds to CHF in water at 10 MPa. The system pressure is reduced considerably by using a modeling fluid. The corresponding saturation temperature is reduced from 310°C to 60°C. Furthermore, the latent heat of R-134a is only 5% of that for water at the corresponding pressure. This reduces the system power requirements by an order of magnitude. Tain (1994) concluded that the following advantages are derived from using scaling fluids:

- The lower heat flux and lower temperature result in much lower temperature excursions at the CHF condition. Thus the possibility of physical burnout is extremely low.
- The accuracy and reproducibility of CHF data are much better than in water.
- Instrumentation is less complex and experimental testing is safer, easier and quicker.

Commonly used refrigerants for CHF experiments are:

- CFCs;
CFCs stands for chlorofluorocarbons, which are molecules made up of chlorine, fluorine and carbon atoms. CFC-12 (usually known as R-12) used to be the most widely used fluid for CHF measurement.
- HCFCs;
HCFCs stands for hydrochlorofluorocarbons, which are molecules made up of hydrogen, chlorine, fluorine, and carbon atoms.
- HFCs;
HFCs stands for hydrofluorocarbons, which are complex molecules made up of hydrogen, fluorine and carbon atoms.

It has been found that CFCs deplete the ozone layer which filters out harmful ultraviolet

radiation produced by the sun. The ozone molecule is unstable. CFCs have the highest ozone depletion potential or ODP, and CFC-11 and CFC-12 are the most harmful of the commonly used refrigerants: they have ODP factors equal to 1.0. HCFCs have an ODP approximately 1/20 that of CFCs. HFCs have zero ODP and pose no danger to the ozone layer, but have a global warming potential (GWP) as greenhouse gases. HFC-134a (usually known as R-134a) has an ODP factor equal to 0.

To take advantage of refrigerants while avoiding the depletion of the ozone layer, R-134a is a good fluid for CHF experiments.

4.4 Fluid-to-Fluid Modeling of CHF

Fluid-to-fluid modeling techniques make it possible to derive the CHF in water from the experimental CHF data obtained in a modeling fluid at equivalent conditions. The theory underlying fluid-to-fluid modeling of CHF is based on the principle of similitude.

The first similitude principle gives the property of similarity. The third similitude principle defines the similarity conditions required for two phenomena. Based on the research of Tain et al. (1993), for CHF in a uniform heated steady-state up-flow channel, if the model and prototype flows have geometric similarity, hydrodynamic similarity and thermodynamic similarity, the two CHF phenomena are similar. Thus, the dimensionless CHF groups are equal.

Generally, by using the same L/D ratio in model and prototype, geometric similarity is achieved.

It can be expressed as

$$\left(\frac{L}{D}\right)_M = \left(\frac{L}{D}\right)_W$$

where, subscript 'M' denotes the modeling fluid and subscript 'W' indicates the equivalent value for water or the prototype system.

For hydrodynamic similarity, a similar non-dimensional mass flux in both systems is required, i.e.,

$$G_M^* = G_W^*$$

where G^* can be expressed in various ways. After examining seven different fluids, Groeneveld et al. (1992) found that the most accurate CHF mass flow modeling parameter is given by

$$G^* = \frac{GD^{0.5}}{(\sigma\rho_f)^{0.5}}$$

A similar density ratio in both systems is required, i.e.,

$$\left(\frac{\rho_f}{\rho_g}\right)_M = \left(\frac{\rho_f}{\rho_g}\right)_W$$

Thermodynamic similarity requires the equivalent thermodynamic properties of the fluid. It is achieved when qualities in both systems are the same at any axial location (z/D) along the length, i.e.,

$$X(z)_M = X(z)_W$$

From the heat balance equation

$$X(z) = \left(\frac{\dot{q}_{CHF}}{h_{fg}G}\right) \left(\frac{A}{zl_p}\right) - \left(\frac{\Delta h_i}{h_{fg}}\right)$$

where, A and l_p are area and perimeter of the channel cross section.

If the above dimensionless parameters are matched, the two CHF phenomena are similar. According to the first principle of similitude, all dependent non-dimensional groups are equal. Thus, the non-dimensional CHF groups for the two fluids are equal:

$$\left(\frac{\dot{q}_{CHF}}{h_{fg}G}\right)_M = \left(\frac{\dot{q}_{CHF}}{h_{fg}G}\right)_W$$

The fluid-to-fluid modeling technique has been successfully applied to a wide range of test section geometries and flow conditions.

4.5 Flow Loop

The CHF experiment investigation was carried out in the Multi-Fluid Boiling Loop at the University of Ottawa. This loop was designed specifically to perform CHF tests on non-aqueous

fluids, such as CFC-12 and HFC-134a.

4.5.1 General

As shown schematically in Figure 4.4, the two gear pumps circulate the fluid through the control valves, to the flow meter (Micro Motion, Sensor Model D12, Transmitters Model RFT9712), preheater, test section, heat exchanger (Model Coax-2201-J-08-128), and pressurizer. The loop is built entirely of stainless steel except for the brass heat exchanger and some minor connections. It is capable of supplying fluid at pressures up to 3.45 MPa (500 psi), mass fluxes up to $4000 \text{ kg m}^{-2} \text{ s}^{-1}$, powers up to 5 kW, and test section lengths up to 1.8 m.

4.5.2 System Parameters and Measurements

4.5.2.1 Pressure

The pressures are adjusted using a pressurizer. A tubular heater with a maximum power of 500 W is located at the bottom of the pressurizer to boil the liquid, while a cooling coil is mounted at the top to condense the vapor. The pressure is controlled by adjusting the power supplied to the heater and the water flow rate in the cooling coil. The absolute system pressures are measured using an absolute pressure transducer located at the outlet of the test section.

4.5.2.2 Mass Flux

Valves at upstream of the test section and within the by-pass sections control the flow rate and provide a stable operating condition during the tests. A Coriolis type mass flow meter at the upstream of the test section measures the mass flow rate. The mass flux is derived by dividing the mass flow rate by the cross section area.

4.5.2.3 Inlet Subcooling

The inlet conditions to the test section are adjusted by the power input and the flow rate of coolant to the preheater. The power to the preheater is finely regulated with a variable transformer. The inlet temperature is monitored by the thermocouples installed at the inlet of the test section.

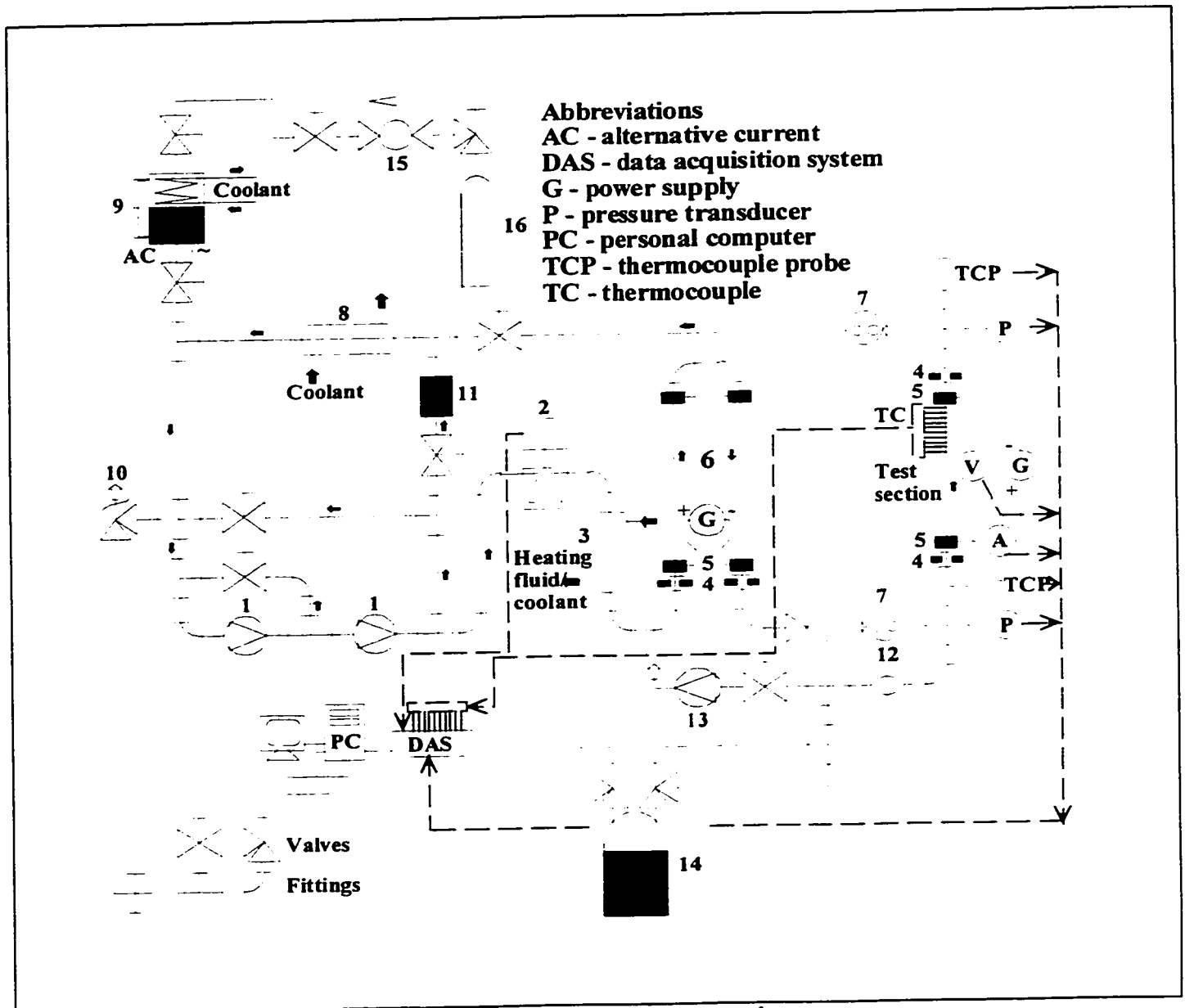


Figure 4.4 Schematic of the test loop.

1. Gear pump	2. Coriolis type mass flow meter	3. Preheater
4. Dielectric fittings	5. Electric power terminal	6. Electrical preheater
7. Sight glass	8. Condenser	9. Pressurizer
10. Pressure relief valve	11. Refrigerant filter-drier	12. Ball valve
13. Vacuum pump	14. Refrigerant storage tank	15. Pressure reducer
16. High pressure nitrogen		

4.5.2.4 Heat Flux

Low-voltage, high-current DC power is supplied directly to the test section. The power supply (Sorenson Model 40-125A) is connected to the test section and has a capacity of up to 5 kW. The voltage drop across the test section is measured using a multi-meter. The DC current is derived by measuring the voltage drop through a shunt resistor, where the total heat supplied to the test section is calculated from the product of the DC current and voltage drop across the test section. The average heat flux is obtained by dividing the total power by the area of the inner surface.

4.5.3 Test section

4.5.3.1 General

The test section is basically a heated dumbbell-shaped tube mounted vertically with subcooled liquid entering at the bottom. The test section with the dumbbell shape is made of Inconel 600 circular tube (9.8 mm I.D., 10.5 mm O.D.). Since test sections are directly heated by DC power, the heated length can be adjusted simply by changing the location of the upper power clamp. The test section is connected to the flow loop via two dielectric fittings (Swagelok, SS-4-DE-6) so that the test section is electrically insulated from the other parts of the loop. The test section is insulated with fiber glass wool, and a schematic of the test section is shown in Figure 4.5.

4.5.3.2 Minimum Gap Size Control and Measurement

The minimum gap size is a key parameter in the CHF experimental investigation. Due to the properties of the tube material, the minimum gap size can vary with temperature and pressure. Therefore, before using for the CHF test, the dumbbell test sections were systematically measured. The wall effect and the system pressure effect on the minimum gap size were also tested.

It was found that wall temperature effect on the minimum gap size is negligible within the temperature range from 0°C to 100°C (the maximum saturation temperature is less than 60°C). However, the minimum gap size changes with system pressure.

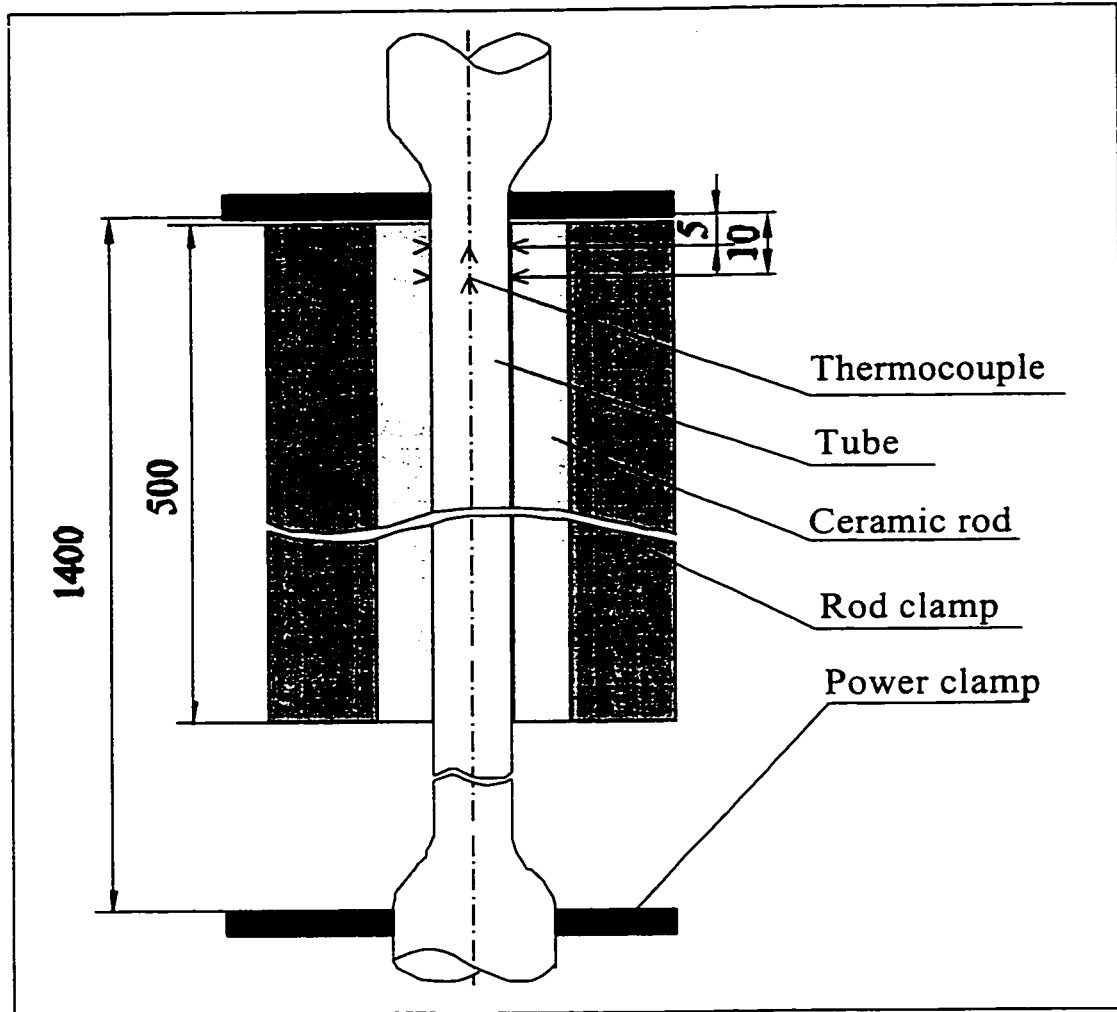


Figure 4.5 Sketch of the dumbbell-shaped test section.

To restrain the minimum gap size expansion with pressure, a pair of special clamps were designed and applied to the test section. As illustrated in Figure 4.5, two ceramic rods are placed between the dumbbell-shaped tube and act as both electrical and thermal insulator. The ceramic material was chosen because of its hardness and low heat and electricity conductivity as well. The rod radius is a little bit smaller than that of the narrow gap region, which ensures that the rods are well in contact with the tube wall in the minimum gap region. By tightening the screws, the clamps push the rods against the tube wall in the narrow gap region. The gap size is therefore restrained and controlled by adjusting the screws.

The ceramic rod and the clamps make it difficult to measure the minimum gap size. It is not possible to measure the size directly. Therefore, special pins with fixed length were made and installed to the test section for the purpose of gap size measurement. The pins, ceramic rods, tube walls, and gap form a chain.

As shown in Figure 4.6, the total length of the chain was measured using a micrometer. It is noted that at the zero gap size case, the total length of the chain is equal to the sum of the pin lengths, rod diameters and wall thickness. For non-zero gap, the minimum gap size was obtained by deducting the pin lengths, rod diameters and wall thickness (the sum equals the length for zero gap) from the total length of the chain.

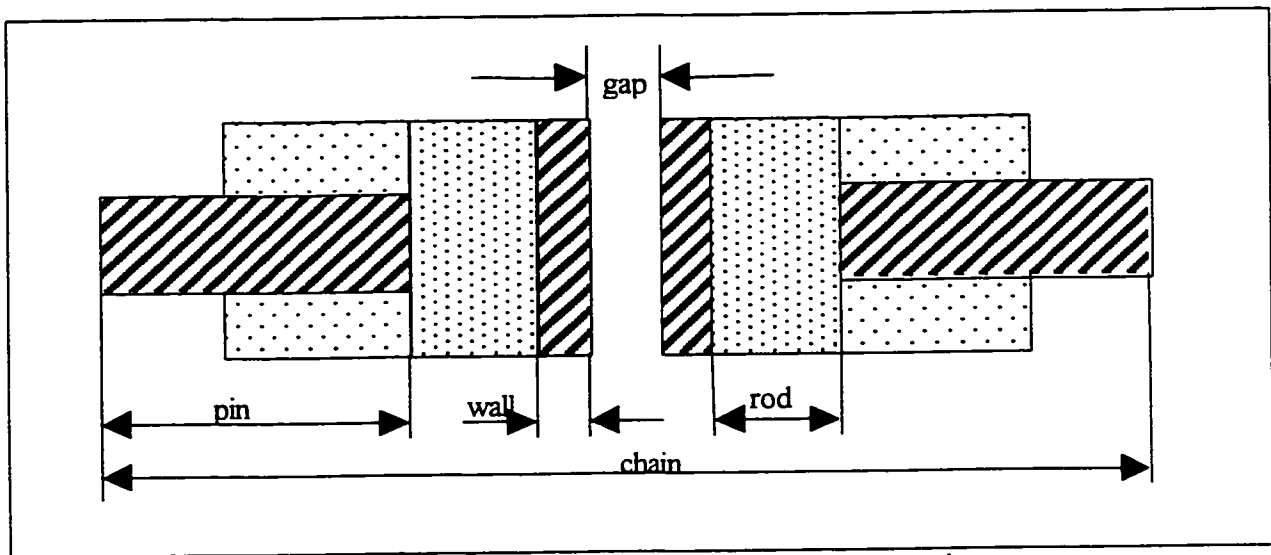


Figure 4.6 The chain for measuring minimum gap size.

4.5.3.3 Monitoring of CHF

CHF normally occurs first at the exit of the test section. In this experimental study, thermocouples were mounted in the vicinity of the flow exit of the test section to monitor the temperature jump which is characteristic of CHF. It is noted that for monitoring the CHF an accurate measurement of wall temperature value is not required. Because of the geometry complexity, flow in the dumbbell tube is not

uniform and the CHF may occur at different locations of the dumbbell cross section. Based on the dumbbell shape analysis in section 4.2, eight ungrounded K-type thermocouples were installed at characteristic points on the outside tube surface around the periphery of the dumbbell-shaped tube cross section. Ungrounded thermocouples were used to prevent electrical conduction.

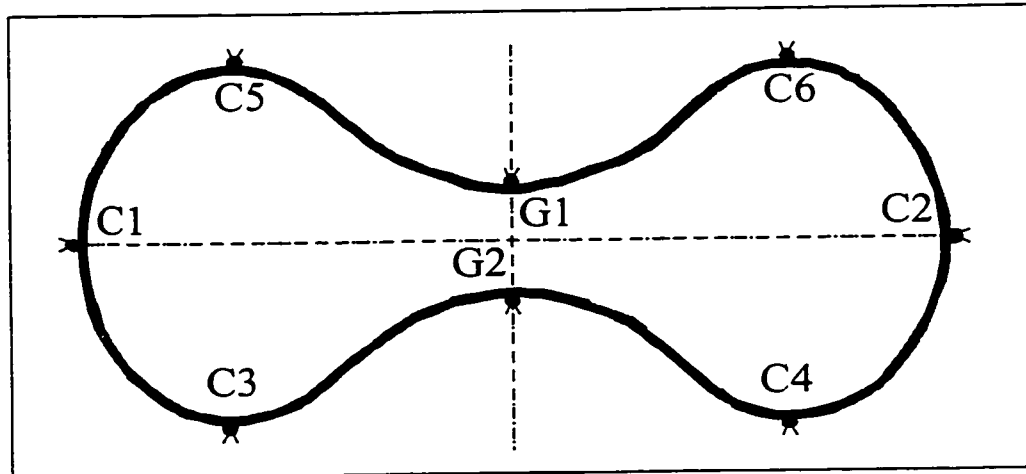


Figure 4.7 Thermocouples for monitoring CHF.

As shown in Figure 4.7, thermocouples labeled G1 and G2 are placed at the center of the narrow gap region of the test section to monitor the temperature at the minimum gap size points. The thermocouples C1 and C2 are located at two ends of the center line. These thermocouples are used to monitor the CHF at different flow conditions.

As described in the previous section, ceramic rods and clamps are applied to the narrow gap region of the test section to restrain the gap size change. This makes it hard to install the thermocouples in the gap region. A special thermocouple holder was therefore designed and added to the rod clamps for measuring the tube wall temperature in the minimum gap region. Small holes were drilled in the rods, which allows the thermocouples to pass through the rods and keep a tight contact with the dumbbell-shaped tube wall.

As shown in Figure 4.8, the thermocouples (stainless steel sheathed, ungrounded) go through the rod and clamp, and keep a tight contact with the tube wall with the aid of a compressed spring. The spring

is installed between the screw and the peg. The peg is fastened with thermocouples, while the screw is fastened in the clamp. By turning the screw, the compressed spring pushes the peg and keeps the thermocouples in good contact with the tube wall.

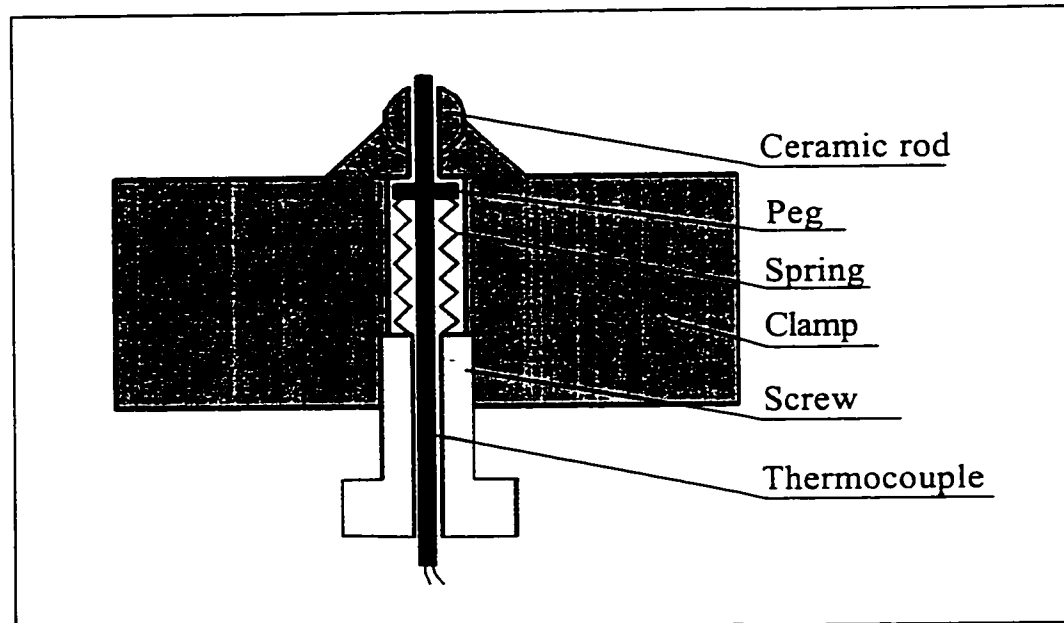


Figure 4.8 Assembly sketch of rod, clamps and thermocouples.

4.5.4 Contamination Control

As shown in Figure 4.4, a filter is installed in one of the by-pass paths in order to minimize the influence of contamination. Furthermore, evacuation is performed using a vacuum pump before charging the working fluid. This removes unwanted moisture and non-condensable gases from the system. By lowering the system pressure, the boiling point of water is lowered to a point much less than the room temperature. Thus, evaporation occurs easily. This vaporized moisture can then be drawn out by the vacuum pump.

4.5.5 System Leak Testing

After the test section was connected to the loop, a leak check was performed using high pressure nitrogen (around 2 MPa) before charging refrigerant. Note that the maximum operating pressure for this CHF investigation is 1.66 MPa. The leaks were traced using a simple bubble test liquid made of soap and water.

4.5.6 System Charging and Discharging

Before charging R-134a to the loop the gases inside the loop was evacuated using the pump. This removes the non-condensable gases and water vapor in the system. The pump operation lasted about 15 minutes. When the system was fully evacuated, the vacuum control valve was closed and the pump was shut off. The system pressure was monitored to ensure that the vacuum level held for at least 5 minutes. Then R-134a was charged to the loop. The charging procedure can be accomplished faster by placing the refrigerant storage tank in hot water bath. The amount of working fluid used was controlled by monitoring the liquid level in the pressurizer.

Whenever it is necessary to replace a test section, the system must be discharged. To return the refrigerant in the loop back to the storage tank, the tank was submerged in ice water. The refrigerant will condense at low temperature, which lowers the tank pressure. Thus, the refrigerant in the loop at a higher pressure is sucked back to the storage tank.

4.5.7 Heat Balance Testing

Heat balance tests using single phase R-134a were performed before CHF experiments. It was found that the heat loss was less than 1%.

4.6 Data Acquisition System

The flow and the heat transfer parameters of the main loop and test section were measured by thermocouples, absolute pressure transducers and flow rate transducer. A data acquisition system (Sciometric Data Acquisition and Control System 200) was used to collect, process and record these parameters. In the experiments, the following parameters are of primary interest:

- wall temperatures in the vicinity of the exit of the test section,
- inlet and outlet temperatures of the test section,
- mass flux,
- pressure at the outlet of the test section.

The parameters were also displayed using real time graphs. The CHF occurrence was detected

by monitoring the wall temperature jump in the graph. The system parameters (pressure, mass flux, and inlet temperature) and power supplied (voltage drop and DC current) were recorded at CHF conditions.

4.7 Test Procedure

The CHF condition is approached by increasing test section power at constant mass flux, inlet temperature, and pressure. The experimental procedure consists of the following:

1. Fill the system with the test fluid; starting the circulating pumps; adjusting the mass flux with the regulating valves;
2. Regulate the system inlet temperature and pressure by adjusting the desired power to the preheater and the heater inside the pressurizer;
3. Supply power to the test section at about 80% of its predicted CHF value;
4. Slowly increase the test section power until CHF has occurred. At each step, maintain the test section power, and carefully monitor the mass flux and system pressure to ensure both are at steady state.
5. Record current and voltage to the test section, mass flux, system pressure, fluid inlet and outlet temperatures.
6. Reduce the test section power to 80-90% of the CHF value and adjust the flow condition for the next CHF test.

4.8 Test Conditions

The objective of this experiment is to obtain data for the geometric effect on CHF, and the influence of flow parameters on this effect. The experiments covered a wide range of flow conditions and the overall test matrix is summarized in Table 4.1.

Table 4.1 Test matrix

Parameter	Test range
Gap size	0, 0.15, 0.3, 0.5, 0.7, 1 mm
Pressure	1130 kPa (7 MPa water equivalent) 1660 kPa (10 MPa water equivalent)
Mass flux	1000, 2000, 3000 (kg/m ² s)
Mass flow rate	1, 3, 5 (kg/min)
Quality	0.1 to 0.7
Inlet temperature	14 to 57 °C
Heated length	0.7 to 1.4 m

4.9 Flow Visualization

A simple adiabatic two-phase flow visualization test has been performed for mechanistic analysis and CHF modeling. Transparent dumbbell-shaped test sections were constructed by squeezing a plastic circular tube in between two glass rods.

Tap water and high-pressure nitrogen were mixed upstream of the dumbbell-shaped test section to form two-phase flow. Water and gas valves in the upstream of the test section controlled the mass flow and quality. Annular flow in the dumbbell-shaped tube was observed.

4.10 Data Reduction

For a given geometry, the CHF is a function of P, G, X at local condition:

$$CHF = f(P, G, X)$$

The pressure P can be measured directly, while the mass flux G and quality X are derived from

$$G = \frac{W}{A}$$

$$X = \frac{h_{exit} - h_f}{h_{fg}}$$

where W is the mass flow rate, and A is the flow area.

$$h_{exit} = h_m + \frac{Q}{W}$$

where Q is the total power supplied to the test section. The heat flux is calculated from

$$CHF = \dot{q}_{CHF} = \frac{Q}{A}$$

where A is the inner surface area of the dumbbell-shaped tube.

The CHF can also be expressed as a function of inlet condition:

$$CHF = f(P, G, \Delta h_{sub}, L)$$

where L is the heated length which is measurable. The subcooling Δh_{sub} is derived from the inlet temperature and the pressure.

4.11 Error Analysis

Please refer to Appendix II for details.

Chapter 5

MODELING STUDY

This chapter describes a new theoretical model developed to study dryout CHF in complex flow geometries. The model is called a CHF partition model since it partitions a complex flow channel into two interactive pseudo-channels. Using the partition concept, a deposition factor is introduced to account for the effect of irregular geometry on deposition and its distribution to different parts of the subchannel.

This modeling study was carried out in three stages. First, simplifying assumptions were made to approximate the real two-phase annular flow and heat transfer system with a physical model. Some of these assumptions refer to specific annular flow processes, while others pertain to the partitioning of flow channels. Second, analytical formulation of annular flow heat transfer was performed based on these physical assumptions and the conservation equations. The final stage (computation) focused on how to solve the equations obtained in the formulation step. It involved the design and implementation of a computer model for annular two-phase flow and heat transfer.

Sections 5.1, 5.2, and 5.3 of this chapter describe the aforementioned three stages of model development.

5.1 Partition Model

5.1.1 Introduction

The CHF in complex flow geometry is a three-dimensional two-phase flow and heat transfer problem. However, due to the complexity of the CHF phenomenon, CHF models are normally one-dimensional. To deal with the CHF in complex flow channels under the one-dimensional flow limitation, the partition model was created in this study. Instead of explicitly expressing the

three-dimensional flow and heat transfer in a complex geometry, the partition model divides the flow channel into two interactive pseudo-channels. Each pseudo-channel is viewed as a quasi-simple geometry. Two sets of one-dimensional conservation equations are then applied to the two pseudo-channels. The CHF in the complex geometry is computed by solving equations of the two pseudo-channels and overall conservation equations of the entire channel.

The partition model created in this study is a new approach to the CHF modeling. However, the concept of partition is not new. As early as the 1930s, Prandtl developed his well-known boundary layer theory. The theory is essentially a partition approach which divides the flow over a surface into two regions, boundary layer and main flow region. Similarly, for modeling the CHF in a complex geometry, the partition model divides the flow inside the channel into two regions: the narrow gap region, and the main flow region. The geometry effects on CHF are examined by analyzing the flow and heat transfer difference between the two channels.

Most complex geometries of interest can be partitioned reasonably. For example, rectangular and triangular geometries can be divided into a corner region and a main flow region; concentric annuli can be divided into the inner region and outer region; geometries like the bundle and the dumbbell, can be divided into a gap region and a main flow region.

5.1.2 Basic Assumptions

The partition model was derived from Whalley's "annular flow model" (Whalley et al., 1974) with the concept of partitioning a complex geometry. It is limited to annular two-phase flow and dryout-type CHF. This subsection describes the basic assumptions specific to annular flow. They are general to most of annular two-phase flow model like Whalley's model (Whalley et al., 1974) and Govan's model (Hewitt and Govan, 1990). The subsequent sections will depict the assumptions specific to the partition model and the dumbbell-shaped tube.

Two-phase flow and heat transfer in a complex flow channel is simplified with the following assumptions:

- Steady state

No effect of transition will be taken into account. The model cannot be applied to time-dependent situations.

- Pseudo one-dimensional

The flow area of the cross-section is assumed constant. The flow inside the channel is a forced flow. The flow is thus assumed 1-D in the flow direction. However, for the annular two-phase flow, the evaporation and droplet exchange processes in lateral direction have to be taken into account. Thus, to be precise, the flow is assumed quasi one-dimensional.

- Vertical upward flow

As this study is concentrating on the effect of geometry on CHF, the model is restricted to vertical upward flow. The effect of flow direction will not be taken into account.

- Three flow regions in flow direction

In the case of subcooled inlet conditions, flow inside the channel is divided into three regions: single-phase flow, homogenous flow, and annular flow. In the single flow region, the liquid is heated up to saturation condition. From saturation to onset of annular flow, the flow is assumed to be homogenous. For annular flow, a liquid film flows along the tube wall. Vapor and entrained liquid droplets comprise a homogeneous core surrounded by the liquid film.

- The onset of annular flow (ONAF) condition is identified according to the assumption made by Whalley's model: the initial condition for annular flow is a thermodynamic quality of 0.01 where 99% of the liquid phase is entrained. Earlier modeling works have shown the predicted CHF to be relatively insensitive to the assumption of the ONAF condition for long tubes (Whalley, 1977).

- Thermodynamic equilibrium in two-phase flow regions

All fluid properties involved are constant and the vapor phase is always saturated. The saturation properties are determined by the exit pressure.

- Homogenous flow of the liquid droplets and vapor in the core.

- Gravity is the only body force involved.

- Viscous energy dissipation is neglected

No viscous energy dissipation is taken into account. For boiling heat transfer, it is a

minor quantity.

5.1.3 Partition Assumptions

In addition to the basic annular flow assumptions, the assumptions specific to the partition model are as follows:

- Any flow channel can be divided into two pseudo-channels
Partitioning a flow channel is simply a different view of the channel, which will not affect the flow and heat transfer inside the channel. Overall conservation is preserved.
- Flow inside each pseudo-channel is assumed to be an annular flow.
- The flows inside the two pseudo-channels interact with each other by sharing the common gas core.
- The mass transfer of the liquid films between the two pseudo-channels is negligible
For each liquid film of a pseudo-channel, the mass transfer in the lateral direction is primarily due to evaporation, deposition and entrainment. The mass transfer across the liquid film boundary between the two pseudo-channels is minor since the liquid film is very thin and the flow in the peripheral direction is secondary.
- The deposition rates in the two channels may be different due to the shape effect.
Section 5.2.4.5 describes the geometry effect on deposition.

5.1.4 Partitioning Flow Channels

The partition model isolates the particular region from the main flow region to take into account local flow and heat transfer characteristics. Therefore, partitioning a flow channel becomes a key part of the model. Like the boundary layer theory, the definition of boundary of the boundary layer is a key part of the theory. However, unlike boundary layer theory, when partitioning a channel (defining the boundary of the two pseudo -channels), the geometry characteristics of the channel needs to be taken into account. Assumptions are made for partitioning each flow channel.

5.1.4.1 Partitioning a Circular Tube

The circular tube is a special case. It can be divided arbitrarily into two sectors as shown in

Figure 5.1 due to its symmetry. In this study, the partition model is applied to the circular tube geometry not for the purpose of predicting new CHF data but to verify the model.

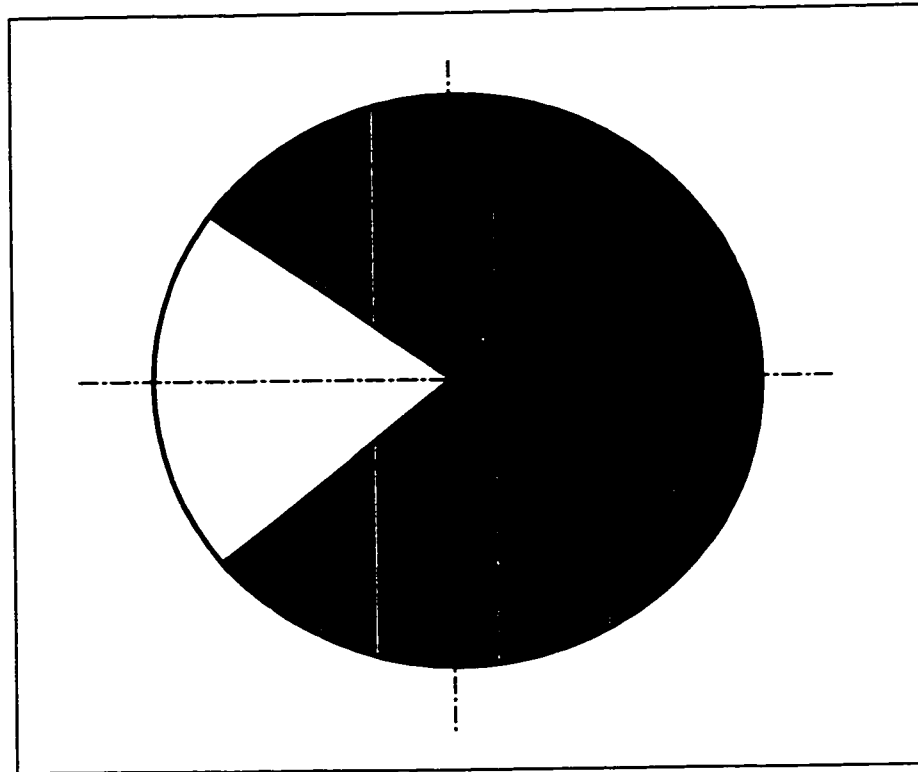


Figure 5.1 Partition of circular tube.

5.1.4.2 Partitioning a Dumbbell-Shaped Tube

The partition model was initiated for predicting the CHF in dumbbell-shaped tubes. In theory, the same methodology can be used to predict CHF in other flow geometries. Due to symmetry, only half of the dumbbell shape needs to be considered, further simplifying the analytical treatment of the flow geometry.

As shown in Figure 5.2, the dumbbell-shaped tube flow channel is divided into two flow regions (two shaded areas in the figure), a narrow gap flow channel and a main flow channel. It is noted that the curvature of the dumbbell-shaped tube changes. The boundary (shown as a double line in the figure) between the two flow regions is set at the point of transition from concave to convex. By this partition, the surface of the main channel is concave while the surface of the gap region is convex (from interior view).

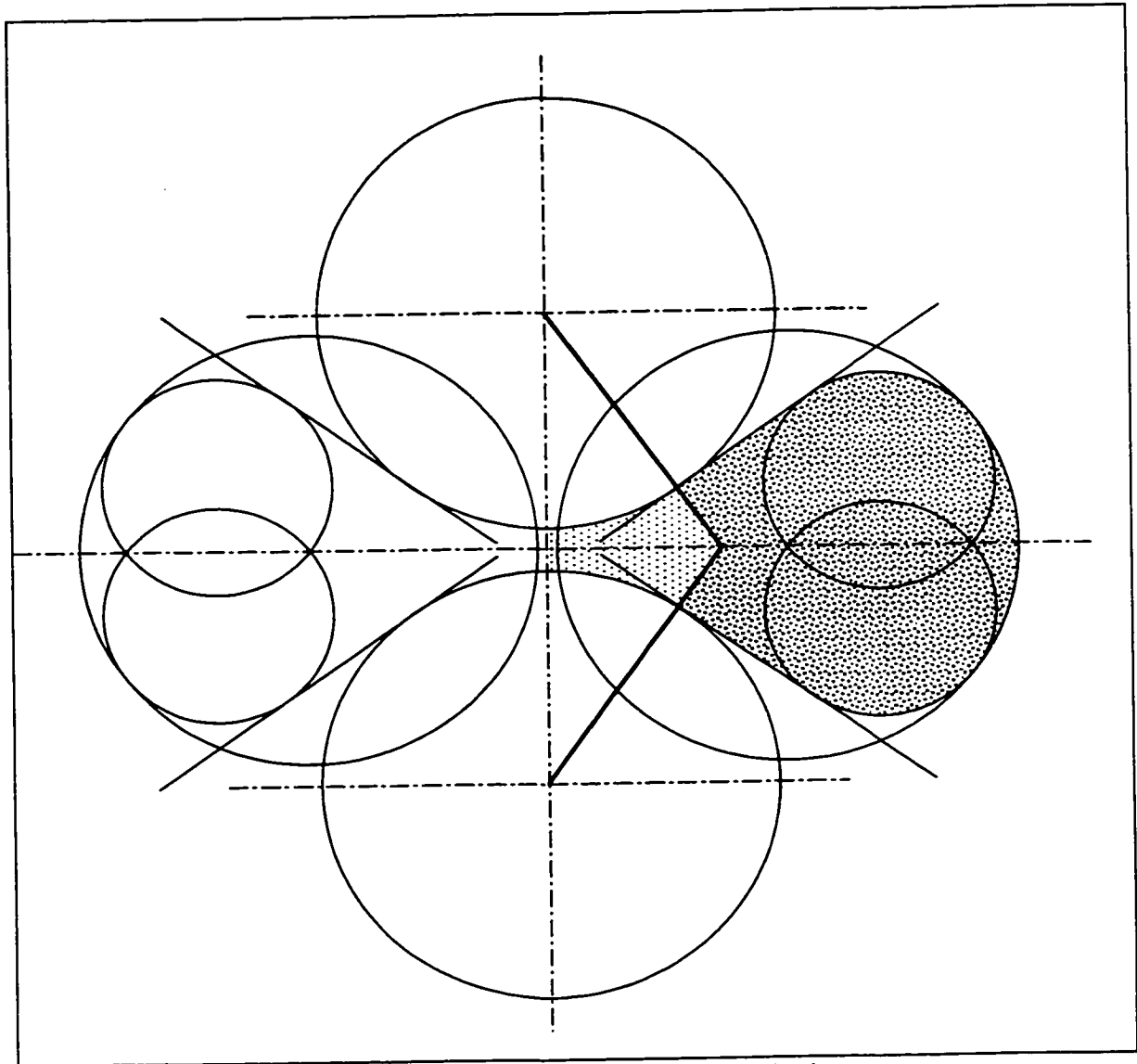


Figure 5.2 Partition of dumbbell-shaped tube.

5.1.5 Dumbbell Flow Channel Assumptions

This section describes the detailed assumptions made specifically about the dumbbell-shaped tube.

5.1.5.1 Two-phase Flow Regimes in Dumbbell-Shaped Tube

To determine the flow patterns and heat transfer regimes in the dumbbell-shaped tubes,

sophisticated diabatic flow visualization and measurement are required. However, no such experimental investigation has been found currently. This subsection simply postulates the flow in a dumbbell-shaped tube based on researches on other geometries, including small triangular and rectangular channels.

Triplett et al. (1999a) conducted an experimental investigation on two-phase flow patterns in micro-channels. Circular tubes with diameters of 1.1 and 1.45 mm and semi-triangular channels with hydraulic diameters of 1.09 and 1.49 mm were tested. The semi-triangular is a triangular with one of the three round corners. Five major flow patterns, bubbly, slug, churn, slug-annular and annular, were observed in the experiments. It was found that the flow patterns in the semi-triangular channels were similar to their corresponding flow patterns in the circular tube. Xu et al. (1999) investigated flow regimes of air and water flow in rectangular channels with narrow gaps of 0.3, 0.6 and 1.0 mm. It was found that the flow regimes for gaps of 1.0 and 0.6 mm were similar to those having a larger gap, which can be classified into bubbly flow, slug flow, churn-turbulent flow and annular flow. For the rectangular channel having a mini size gap of 0.3 mm, cap-bubbly flow, slug-bubbly flow, churn flow and annular-droplet flow were observed.

Based on these results in the small semi-triangular and rectangular channels, it is assumed that the flow patterns in the dumbbell-shaped tube should be similar to those in the circular tube. As the subcooled working fluid enters the dumbbell-shaped tube, it is in the liquid phase. Like the flow in a circular tube, as the fluid proceeds farther downstream, the flow patterns change from liquid single-phase flow, bubbly flow, slug flow, to annular flow.

With the flow patterns changing, the regions of heat transfer change from single-phase convective heat transfer, subcooled boiling, saturated nucleate boiling, to two-phase forced convective heat transfer. In the boiling regions, the bubbles will eventually depart from the wall surface. It is postulated that no bubble will bridge the two convex surfaces of the tube wall in the narrow gap region.

At the ONAF point, two vapor cores are formed at the centers of the two subchannels of the

dumbbell-shaped tube. At this state, the liquid bridges the gap completely and separates the two vapor cores. In the region of annular flow, no bubble is generated at the surface of the tube wall.

The void is generated at the surface of liquid film, which merges into the turbulent vapor core immediately after generation. As the quality increases, the liquid film around the tube wall becomes thinner and thinner, as does the liquid bridge in the narrow gap region. For most of the flows, a point may eventually be reached where the liquid bridge breaks. The liquid bridge disappears and becomes two liquid films on the two convex surfaces of the tube in the narrow gap region. Upon the complete evaporation of the liquid films, dryout-type CHF takes place.

In summary, the flow patterns and heat transfer regimes in the dumbbell-shaped tubes are assumed to be similar to those in circular tubes.

5.1.5.2 Circumferential Distribution of Liquid Film Thickness

The partition model is restricted to annular flow. The CHF is predicted by monitoring the first occurrence of liquid film dryout. Different from the circular tube, the circumferential distribution of liquid film thickness along the periphery of the dumbbell-shaped tube may not be uniform due to the geometry effect. This subsection makes further simplifications on the circumferential distribution of liquid film thickness in the dumbbell-shaped tube.

A measurement of the circumferential distribution of liquid film thickness in the dumbbell-shaped tube is needed as a basis for prediction. However, this is beyond the scope of this study. In this subsection, assumptions of the liquid film distribution are made by examining the mechanisms of liquid film spreading instead of conducting an experimental investigation.

Butterworth (1973) investigated the circumferential distribution of liquid film thickness and summarized the mechanisms of liquid film spreading. He proposed the following four mechanisms or factors of liquid film spreading:

1. Spreading of the film by a wave action
2. Transfer of liquid by entrainment and deposition of droplets
3. Spreading by circumferential shear force due to secondary gas flow

4. Spreading by surface tension force.

The partition model is based on data for circular tubes. For the annular flow inside a complex geometry like the dumbbell-shaped tube, the problem is converted to the effect of geometry on these four factors. Currently, no literature has been found on the geometry effect on the circumferential distribution of liquid film thickness. For the effect of geometry on entrainment and deposition, an approach is introduced in this study, which will be discussed in section 5.2.4 of this chapter in detail.

For the other three factors, the effect of geometry in the vertical dumbbell-shaped tube is analyzed based on the results of previous researches on horizontal tubes. The three mechanisms of liquid spreading are similar for all the annular flows, including vertical flow in the dumbbell-shaped tube and horizontal flow in the circular tube. In a horizontal or near-horizontal circular tube, the effect of gravity is the only source that may lead to a non-uniform circumferential distribution of liquid film thickness. For the vertical flow in the dumbbell-shaped tube, the effect of geometry is the only source that causes a non-uniform distribution. The assumptions will be made for the geometry effect on the three factors of liquid spreading in the vertical dumbbell-shaped tube according to the gravity effect in the horizontal tube.

In a horizontal two-phase flow, liquid film climbs up towards the top of the channel cross section against gravity if the flow rate of gas is high enough. This phenomenon clearly shows the existence of some kind of force that is strong enough to transfer the liquid film upwards against the gravity. Laurinat et al. (1985) proposed a theoretical model to calculate the circumferential distribution of film thickness in horizontal pipes. Among the four mechanisms of liquid film spreading, the secondary gas flow was considered to be the most important factor that affects the distribution. The role of surface tension was considered negligible. Against gravity, the circumferential shear force caused by the secondary gas flow transfers the liquid towards the top of the horizontal tube. Fores et al. (1995) investigated the secondary flow in horizontal annular two-phase flow. They also concluded that the secondary flow in the vapor core plays the principle role in full development of the liquid annulus.

Fukano and Ousaka (1989) conducted a theoretical research on the circumferential distribution of liquid film thickness in horizontal and near horizontal annular flows. They obtained different results. It was found that the disturbance waves play an important role in maintaining a liquid film near the top of the tube. The pumping action of disturbance waves was thought to be the mechanism of the liquid moving upwards to the top of the cross section of a pipe. According to this flow model, the liquid is pumped up from the bottom of the channel through a disturbance wave and any parts of a disturbance wave move with the same axial velocity. Jayanti and Hewitt (1993) also found that the secondary flow mechanism for liquid film distribution is not applicable in the case of annular flow. They concluded that the disturbance wave, and entrainment and deposition are probably responsible for the film distribution.

Fukano et al. (1997) indicated that “the mechanism of transporting liquid toward the top of the pipe in horizontal annular flow is still in question.” However, it can be concluded from the previous investigations that the effect of surface tension is negligible if compared to the effect of the disturbance wave or the circumferential shear force caused by the secondary gas flow. The effect of the disturbance wave or the circumferential shear force or both is so strong that the liquid film is formed on the top of horizontal channels, which leads to a more uniform distribution of liquid film thickness in the circumferential direction.

According to the results of previous researches, the partition model will not need to take the effect of surface tension into account for the vertical dumbbell-shaped tube. Among the four mechanisms of liquid spreading, the surface tension plays little role on the circumferential distribution in the case of annular flow. The disturbance wave and the circumferential shear stress are strong and will result in a relatively uniform distribution like flow in a circular tube. As described in previous section, the partition model assumed that the liquid film in each flow region was uniform. This uniform distribution is the results of the combined effect of the two mechanisms (the disturbance wave and the circumferential shear force). In other words, the partition model takes the effects of these two mechanisms into account by assuming that the distribution of liquid film in each pseudo-channel is uniform. It is noted that in this study, the

mass transfer of liquid through the liquid film boundary between the two pseudo-channels is negligible.

5.1.5.3 Zero Gap Case

In the case of zero gap, the two convex surfaces of the dumbbell-shaped tube contact each other in the narrow gap region. This is an extreme case. It is noted that this may not mean that a severe adverse effect on CHF will occur. The vapor phase is able to migrate into the main flow region.

Flow channels can be grouped into two categories: uniform spacing like concentric annuli or parallel plates, and non-uniform spacing like eccentric annuli. For the flow channel with uniform spacing, if the spacing is very small, the bubble may be deformed and bridge the surface due to the constraint of the flow channel wall, which will more likely lead to premature CHF. On the other hand, the non-uniform spacing channel, like the dumbbell-shaped tube, has an open flow area – the main flow channel. The evaporated vapor may move to the vapor core from the narrow gap region instead of stagnating in the narrow gap region.

In theory, only one point of the surface comes into contact in the case of zero gap. Different from the non-zero gap size, the liquid bridge in the narrow gap region will not break until dry out. The dumbbell-shaped tube with zero gap is closer to a triangular flow channel. Figure 5.3 illustrates the analogy between the two flow channels.

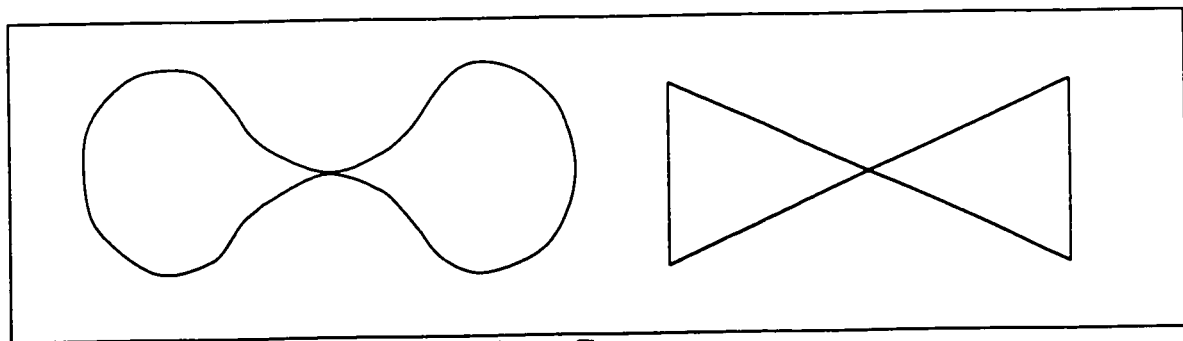


Figure 5.3 Zero gap case.

A lower velocity in the narrow gap region may lead to the liquid film being superheated. This may lead to a lower CHF. The partition model takes account of such an effect by considering the

difference between the narrow gap region and the main flow region.

5.2 Formulation

This section presents the mathematical description of the partition model.

5.2.1 Conservation of Mass

The equations of mass conservation are obtained from mass balances for the liquid films and vapor core for the control volume shown in Figure 5.4. The figure shows a typical finite control volume, which can be either the main flow or the gap flow region.

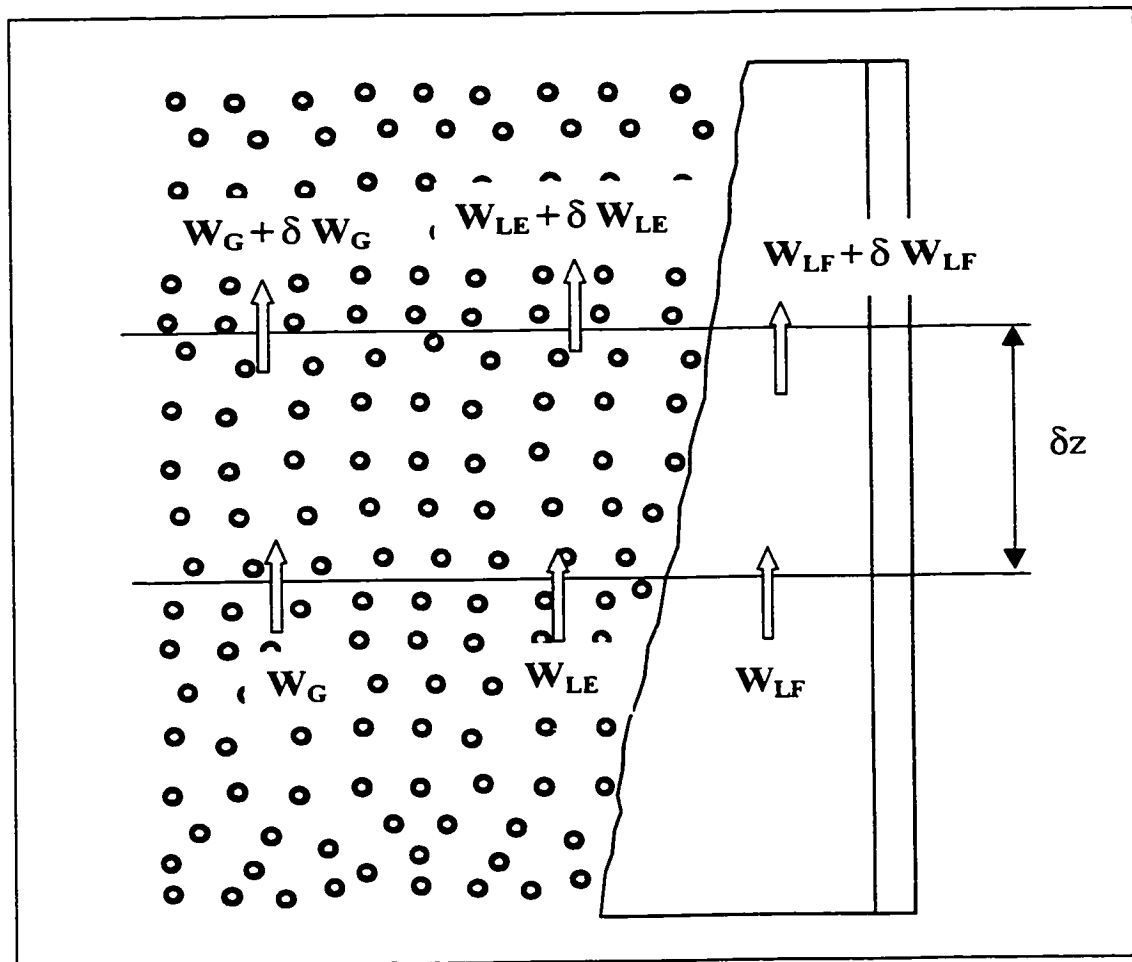


Figure 5.4 Mass balance on a control volume of a pseudo-channel.

Applying a mass balance to the control volume for the liquid film of pseudo-channel 1 gives

$$\frac{dW_{LF1}}{dz} = l_{p1} \left(D_1 - E_1 - \frac{q_1''}{h_{fg}} \right)$$

Applying mass conservation to the control volume for the liquid film of pseudo-channel 2 gives

$$\frac{dW_{LF2}}{dz} = l_{p2} \left(D_2 - E_2 - \frac{q_2''}{h_{fg}} \right)$$

Applying mass balance to the entrained liquid droplets in the vapor core gives

$$\frac{dW_{LE}}{dz} = l_{p1} (E_1 - D_1) + l_{p2} (E_2 - D_2)$$

where,

- W_{LF1} is the mass flow rate of entrained liquid droplets in pseudo-channel 1;
- W_{LF2} is the mass flow rate of entrained liquid droplets in pseudo-channel 2;
- Z is the axial co-ordinate in the flow direction;
- l_{p1} is the perimeter of pseudo-channel 1;
- l_{p2} is the perimeter of pseudo-channel 2;
- D_1 is the deposition rate of droplets onto the liquid film in pseudo-channel 1;
- D_2 is the deposition rate of droplets onto the liquid film in pseudo-channel 2;
- E_1 is the entrainment rate of droplets from the liquid film in pseudo-channel 1;
- E_2 is the entrainment rate of droplets from the liquid film in pseudo-channel 2;
- q_1'' is the heat flux applied to pseudo-channel 1.
- q_2'' is the heat flux applied to pseudo-channel 2.
- h_{fg} is the latent heat of vaporization of the liquid.

For the mass flow rate, many other notations are also used:

- W is the total mass flow rate in the entire channel;
- W_{LF} is the total mass flow rate of the liquid film in the entire channel;
- W_{LE} is the total mass flow rate of the entrained liquid droplets in the entire channel;
- W_G is the total mass flow rate of the gas in the entire channel;
- W_1 is the total mass flow rate in the pseudo-channel 1;
- W_2 is the total mass flow rate in the pseudo-channel 2;
- W_{E1} is the total mass flow rate of entrained liquid droplets in the pseudo-channel 1;

- W_{E2} is the total mass flow rate of entrained liquid droplets in the pseudo-channel 2;
 W_{G1} is the total mass flow rate of gas in the pseudo-channel 1;
 W_{G2} is the total mass flow rate of gas in the pseudo-channel 2.

These mass flow rates have following relationships:

$$W = W_{LF} + W_{LE} + W_G$$

$$W_1 = W_{LF1} + W_{LE1} + W_{G1}$$

$$W_2 = W_{LF2} + W_{LE2} + W_{G2}$$

$$W_{LF} = W_{LF1} + W_{LF2}$$

$$W_G = W_{G1} + W_{G2}$$

$$W_{LE} = W_{LE1} + W_{LE2}$$

5.2.2 Conservation of momentum

The pressure drop of a two-phase flow generally consists of three components: friction loss, momentum change, and elevation pressure drop arising from the effect of the gravitational force field.

Applying the Newton's second law of motion to the elemental control volume for pseudo-channel 1 and 2, respectively, yields

$$\begin{aligned}
 -\left(\frac{dp}{dz}\right)_1 &= \tau_{w1} \frac{l_{p1}}{A_1} + \frac{d}{dz} \left(\frac{W_1}{A_1} u_1 \right) + \rho_1 g \\
 -\left(\frac{dp}{dz}\right)_2 &= \tau_{w2} \frac{l_{p2}}{A_2} + \frac{d}{dz} \left(\frac{W_2}{A_2} u_2 \right) + \rho_2 g
 \end{aligned}$$

where,

- P is the pressure of the system;
 τ_1 is the wall shear stress in pseudo-channel 1;
 τ_2 is the wall shear stress in pseudo-channel 2;
 ρ_1 is the average density for pseudo-channel 1;
 ρ_2 is the average density for pseudo-channel 2;

- g is the acceleration of gravity;
 A_1 is the total area of the cross section for pseudo-channel 1;
 A_2 is the total area of the cross section for pseudo-channel 2;
 u_1 is the average actual velocity for pseudo-channel 1;
 u_2 is the average actual velocity for pseudo-channel 2;

It is assumed that the pressure drops in cross section are small compared to the axial pressure drops, then

$$\left(\frac{dp}{dz}\right)_1 = \left(\frac{dp}{dz}\right)_2$$

The average density is determined based on the homogeneous assumption. This approach is reasonable since the liquid films are very thin and the homogeneous vapor core occupies the flow channel. Therefore, the density is expressed as a function of liquid and gas phase density and the quality as follows:

$$\rho_1 = \frac{\rho_G \rho_L}{\rho_L \frac{W_{G1}}{W_1} + \rho_G \left(1 - \frac{W_{G1}}{W_1}\right)}$$

$$\rho_2 = \frac{\rho_G \rho_L}{\rho_L \frac{W_{G2}}{W_2} + \rho_G \left(1 - \frac{W_{G2}}{W_2}\right)}$$

The mean velocities are calculated from:

$$u_1 = \frac{W_1}{\rho_1 A_1}$$

$$u_2 = \frac{W_2}{\rho_2 A_2}$$

5.2.3 Conservation of Energy

In a boiling heat transfer system, work interaction and kinetic and potential energy changes are negligible. Thus, the overall conservation of energy is written as

$$h = h_m + \frac{l_p z q''}{W}$$

where q is the average heat flux over the entire tube.

$$q = \frac{l_{p1} \bar{q}_1 + l_{p2} \bar{q}_2}{l_p}$$

In the case of uniform heating, q_1 and q_2 are equal to q . In the case of non-uniform heating, like bilaterally heated annuli, q_1 and q_2 are normally known. For the dumbbell-shaped tube, the distribution of the heat flux can be obtained by using simplifying assumptions.

For a given location z , the quality is calculated as

$$x = \frac{h - h_f}{h_{fg}}$$

The vapor phase mass flow rate is calculated from

$$W_G = W x$$

Base on the assumption of homogeneous vapor core, W_{G1} and W_{G2} are simply the part of W_G in the partition.

5.2.4 Entrainment and Deposition Models

Deposition and entrainment are distinctive phenomena of annular flow. The deposition of entrained liquid droplets onto the liquid film and entrainment of liquid from the waves on the liquid film are among the most important aspects of annular two-phase flow. Accurate modeling and prediction of these phenomena are required for accurate prediction of the characteristics of the liquid film. This section reviews the detailed processes of entrainment and deposition and describes the new partition model approach to deal with deposition and entrainment in complex geometries.

5.2.4.1 Entrainment Process in Simple Geometry

The entrainment process is associated with the presence of waves on the surface of the liquid film. According to Whalley (1987), the disturbance waves occur in an annular flow if

$$Re_{lf} > 200$$

where Re_{lf} is the liquid film Reynolds number which is given by:

$$Re_{lf} = \frac{G_{lf} d}{\mu_l}$$

The entrainment mechanism can be considered as that part of the wave is undercut by the vapor flow. The major contribution to entrainment is attributed to the break-up of large disturbance waves. Figures 5.5 and 5.6 show the entrainment process and mechanism in a side view and an axial view respectively. The most striking feature seen is the presence of filaments of liquid which eventually shatter, leaving drops behind. This process appears to be the main source of entrainment of drops in the flow.

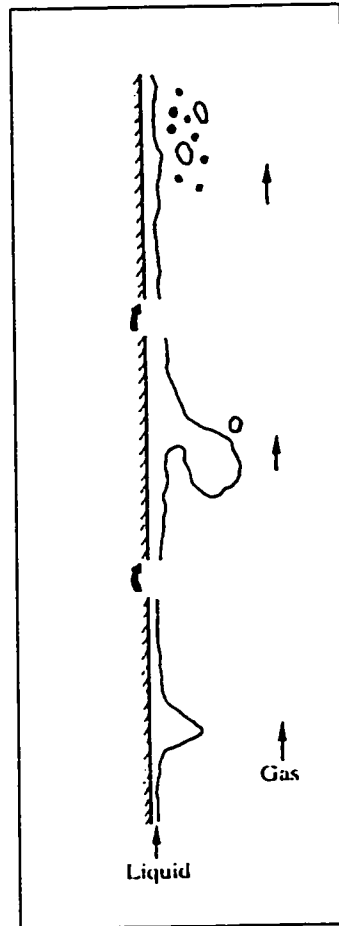


Figure 5.5 Entrainment process in a side view (from Whalley, 1987).

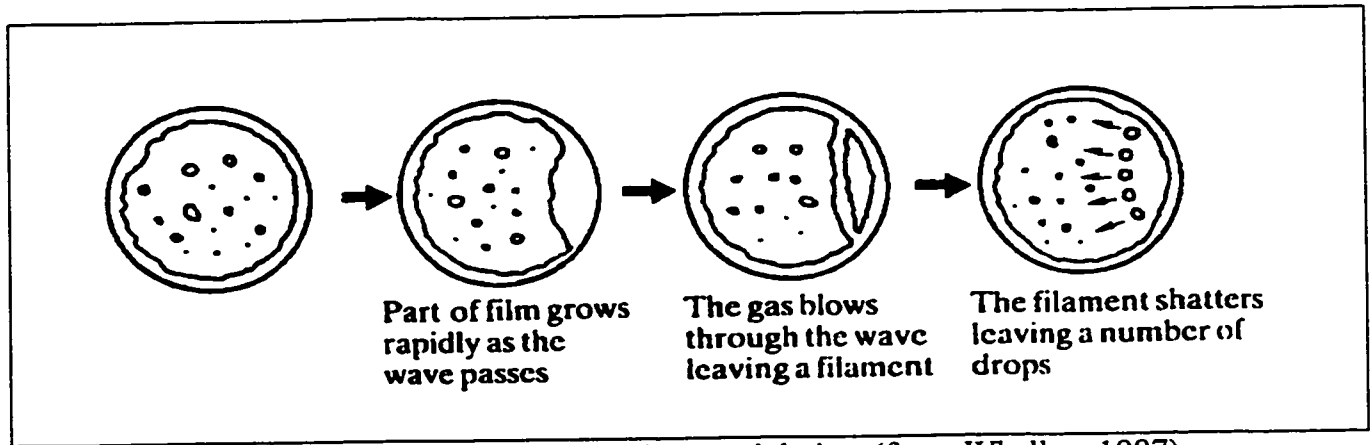


Figure 5.6 Entrainment process in an axial view (from Whalley, 1987).

The entrainment rate, E ($\text{kg/m}^2\text{s}$), is used to express the mass flow rate of liquid entrained into the gas core per unit area of tube wall. One way to determine entrainment rate is based on adiabatic hydrodynamic equilibrium flow. For an equilibrium flow, the entrainment rate is equal to the deposition rate, or

$$E = D$$

At the equilibrium state,

$$D = k C_E$$

where k is the deposition mass transfer coefficient (m/s) and C_E is the droplet concentration (kg/m^3) in the vapor core.

Hence,

$$E = k C_E$$

Note that C_E is not the actual droplet concentration of the state, but the concentration that would be in equilibrium with the liquid film.

There are two forces dominating the liquid film, surface tension and interfacial shear stress. Surface tension σ holds the liquid film together while interfacial shear stress τ_i disrupts it. According to Whalley et al. (1978), interfacial shear stress and surface tension can be made into a dimensionless group s if the average film thickness δ is used

$$s = \frac{\tau_i \delta}{\sigma}$$

The equilibrium droplet concentration C_E is a function of s .

$$C_E = f\left(\frac{\tau_i \delta}{\sigma}\right)$$

The correlation is obtained by experiments. At equilibrium states, the entrainment and deposition rates are equal, so that the concentration of liquid drops is the equilibrium concentration. Whalley et al. (1978) compiled the experimental data from nine sources. The data are illustrated in Figure 3.4.

5.2.4.2 Deposition Process in Simple Geometry

Many researchers carried out photographic investigations to observe the deposition process. As far as their non-axial movement is concerned, the liquid drops move in a straight line from the point where they are entrained to the point where they are deposited. Figure 5.7 shows typical tracks of liquid droplet migration.

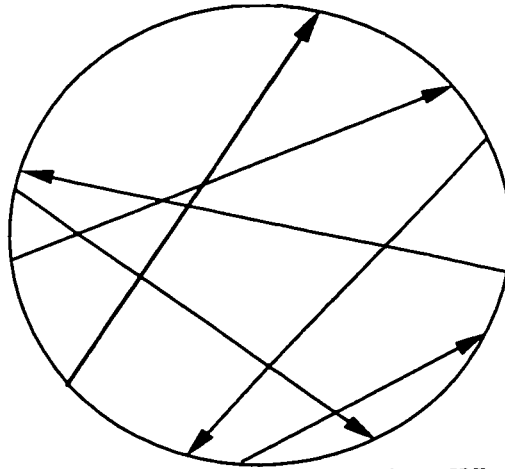


Figure 5.7 Droplet trajectories (redraw based on Whalley, 1987).

The deposition rate, D ($\text{kg/m}^2\text{s}$), is used to express the mass flow rate per unit area of tube wall. D is given by

$$D = k C$$

where C is the concentration of liquid drops in the gas core (kg/m^3). Note that here it is implicitly assumed that the effective concentration of droplets at the tube wall is zero. This is because the tube wall is assumed to remove all the droplets hitting it. Here k is the mass transfer

coefficient (m/s).

5.2.4.3 Deposition Model for Complex Geometry

Unlike the simple geometry, the deposition in complex geometry may not be uniform. The deposition rate may vary along the tube wall due to the geometry effect. Instead of using the average deposition rate over the entire channel, the partition model takes account of such variation by dealing with the two pseudo-channels.

It is noted that the two pseudo-channels are not independent but interactive. Based on the partition assumptions, the two pseudo-channels share a common vapor core. In most applications, the real flow of the vapor core is turbulent. The concentration of liquid droplets can be assumed uniform over the vapor core. The two pseudo-channels share such a uniform vapor core and have the same concentration of entrained liquid droplets.

The concentration in the vapor core, C , is determined by

$$C = \frac{W_{LE}}{\frac{W_G}{\rho_G} + \frac{W_{LE}}{\rho_L}}$$

According to the mechanism of deposition, the liquid droplets move in straight lines. If the liquid droplets inside the vapor core are distributed uniformly, the deposition rates of the two pseudo-channels may be different due to the geometry shape. The deposition coefficients k_1 and k_2 take account of the difference of the deposition rates in the two pseudo-channels. Thus, the deposition rates of liquid droplets in the two pseudo-channels are given by

$$D_1 = k_1 C$$

$$D_2 = k_2 C$$

The deposition coefficients k_1 and k_2 can be obtained from measurement. However, the cost of the experiments to measure the deposition coefficients for all geometries of interest is prohibitive. Also, it will be beyond the scope of this study to measure the deposition coefficients.

Instead of measuring the deposition coefficients k_1 and k_2 , a new approach is developed in this study. Since the correlation for the deposition coefficient in circular tube is available, a new concept of deposition factor ϕ is introduced in this study to denote the geometry effect on the deposition. Then the deposition coefficients for the two flow regions are given by

$$k_1 = \phi_1 k$$

$$k_2 = \phi_2 k$$

where ϕ_1 and ϕ_2 are deposition factors for the two pseudo-channels. k is the deposition mass transfer coefficient for a circular tube. For a circular tube, k is determined by the surface tension of the working fluid. The correlation is shown in Figure 3.3.

The problem of the geometry effect is thus converted to that of determining the deposition factor. Section 5.2.5 will give the details of deriving the deposition factor.

5.2.4.4 Entrainment Model for Complex Geometry

In this study, a complex geometry is partitioned into two pseudo-channels. The entrainment rates for the two pseudo-channels are obtained from the following relationship:

$$E_1 = k C_{E1}$$

$$E_2 = k C_{E2}$$

where, C_{E1} and C_{E2} are equilibrium droplet concentrations for the two pseudo-channels respectively.

k is the deposition coefficient for a circular tube. It is noted that the deposition coefficient for a circular tube instead of the local deposition coefficient k_1 or k_2 is used. For a pseudo-channel, the entrainment rate is determined based on an adiabatic hydraulic equilibrium flow in an equivalent circular tube. Therefore, the entrainment rate is a function of the equilibrium droplet concentration and the deposition coefficient for a circular tube.

Due to geometry effect, the liquid film thickness and interfacial shear stress on the film surface are different in the two pseudo-channels. These in turn affect the entrainment. The equilibrium droplet concentration takes account for the geometry effect on the entrainment. C_{E1} and C_{E2} are

given by

$$C_{E1} = f\left(\frac{\tau_{i1} \delta_1}{\sigma}\right)$$

$$C_{E2} = f\left(\frac{\tau_{i2} \delta_2}{\sigma}\right)$$

Like the simple geometry, the experimental data compiled by Whalley et al. (1978) are used for this analysis.

5.2.4.5 Deposition Factor and Shape Factor

To deal with the geometry effect on the deposition, a new approach has been created in this study. A new concept of deposition factor is defined as follows to account for the geometry effect:

$$\varphi_1 = \frac{k_1}{k}$$

$$\varphi_2 = \frac{k_2}{k}$$

where k is the equivalent deposition coefficient for the entire tube, which represents the average deposition rate over the cross section of the tube. k_1 and k_2 are deposition coefficients for the two pseudo-channels.

The problem becomes to determine the amount of entrained droplets in the vapor core that may deposit onto a particular pseudo-channel surface. In order to evaluate the amount, a concept of “shape factor” is introduced to this study. The concept comes from the radiation shape factor (view factor, or angle factor) for radiation heat transfer analysis. Then, the shape factors for the two pseudo-channels are defined as:

- F_1 = fraction of entrained droplets in the gas core which may deposit potentially on the surface of pseudo-channel 1.
- F_2 = fraction of entrained droplets in the gas core which may deposit potentially on the surface of pseudo-channel 2.

To determine a general relation for F_1 or F_2 , consider a droplet in the gas core as shown in Figure 5.8. The angles α_1 and α_2 represent the angles subtended by pseudo-channel surface 1 and surface 2 respectively.

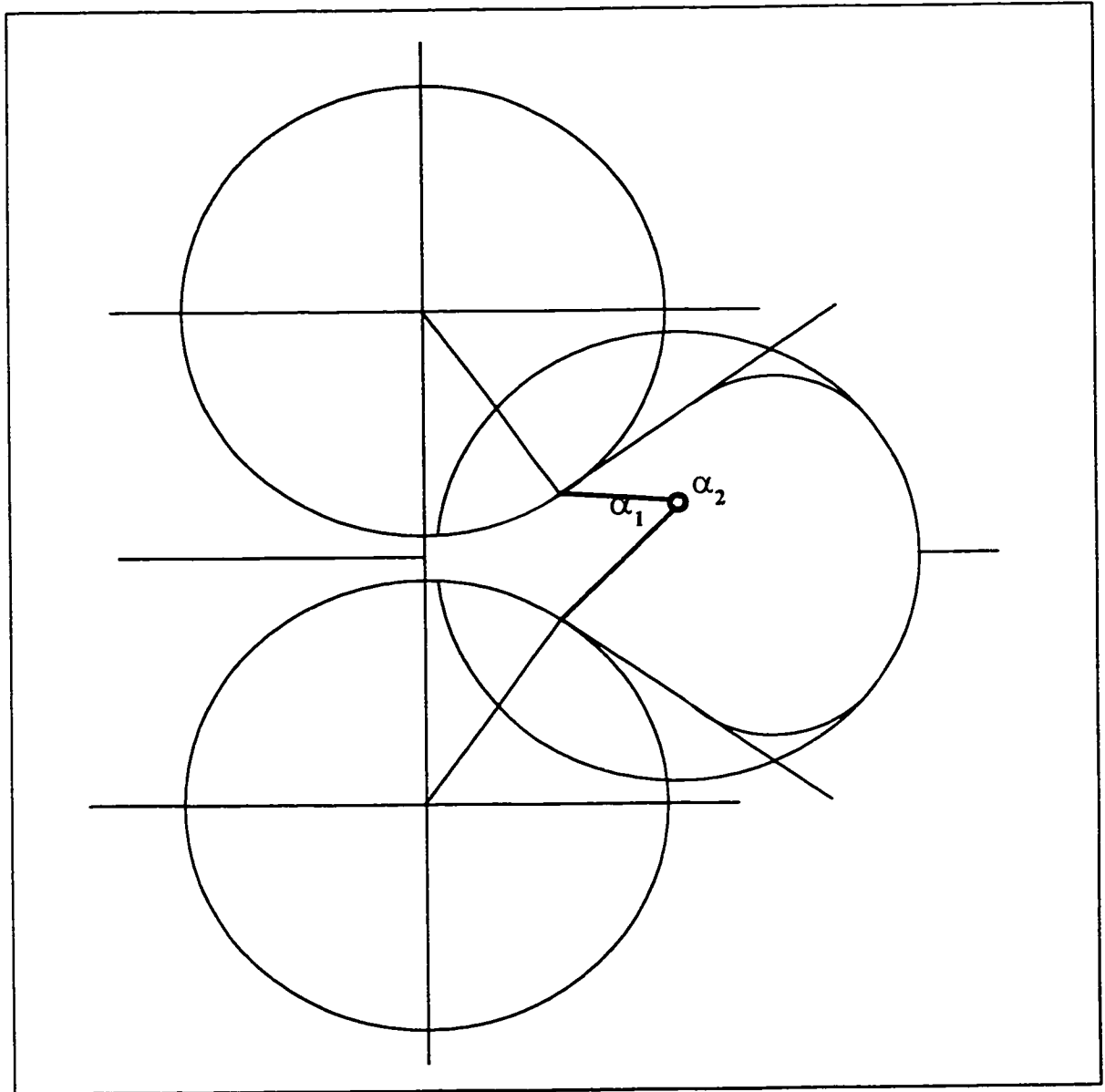


Figure 5.8 Calculation of shape factor.

The probability of the entrained droplet being deposited on the surface of pseudo-channel 1 is:

$$\alpha_1 / 2\pi$$

The probability of the entrained droplet being deposited on the surface of pseudo-channel 2 is:

$$\alpha_2/2\pi$$

where

$$\alpha_1 + \alpha_2 = 2\pi$$

Since the liquid film is very thin, shape factors F_1 and F_2 can be obtained by integrating over the entire cross section:

$$F_1 = \frac{1}{A} \iint_A \frac{\alpha_1}{2\pi} dA$$

$$F_2 = \frac{1}{A} \iint_A \frac{\alpha_2}{2\pi} dA$$

where

$$F_1 + F_2 = 1$$

Similar to the calculation of the flow area for the dumbbell shaped tube as shown in Figure 4.3, F_1 and F_2 can be determined through numerical integration.

The deposition factors are calculated based on the shape factors. Considering the entrained droplet mass balance,

$$D_1 l_{p1} dz = F_1 D l_p dz \Rightarrow D_1 l_{p1} = D l_p F_1$$

$$D_2 l_{p2} dz = F_2 D l_p dz \Rightarrow D_2 l_{p2} = D l_p F_2$$

where l_p , l_{p1} and l_{p2} are perimeters for the real channel, pseudo-channel 1 and pseudo-channel 2 respectively, and dz is the unit length of the tube in the flow direction. $D l_p dz$, $D_1 l_{p1} dz$, and $D_2 l_{p2} dz$ are liquid deposition volumes to the unit tube surfaces of the entire tube, gap region and main channel region, respectively.

The two pseudo-channels share a common vapor core,

$$D = kC \quad D_1 = k_1 C \quad D_2 = k_2 C$$

Therefore,

$$k_1 C l_{p1} = k C l_p F_1 \Rightarrow \frac{k_1}{k} = F_1 \frac{l_p}{l_{p1}}$$

$$k_2 C l_{p2} = k C l_p F_2 \Rightarrow \frac{k_2}{k} = F_2 \frac{l_p}{l_{p2}}$$

The deposition factors are then given by

$$\varphi_1 = F_1 \frac{l_p}{l_{p1}}$$

$$\varphi_2 = F_2 \frac{l_p}{l_{p2}}$$

5.2.5 Interfacial Shear Stress

The interfacial shear stresses are given by

$$\tau_{i1} = \frac{1}{2} f_{gci,1} \rho_{gc} V_{gc}^2$$

$$\tau_{i2} = \frac{1}{2} f_{gci,2} \rho_{gc} V_{gc}^2$$

where interfacial friction factors $f_{gci,1}$ and $f_{gci,2}$ are calculated from

$$f_{gci,1} = f_{sgc} \left[1 + 24 \left(\frac{\rho_L}{\rho_G} \right)^{1/3} \frac{\delta_1}{d_1} \right]$$

$$f_{gci,2} = f_{sgc} \left[1 + 24 \left(\frac{\rho_L}{\rho_G} \right)^{1/3} \frac{\delta_2}{d_2} \right]$$

and the homogeneous gas core density is given by

$$\rho_{gc} = \frac{\frac{W_G}{\rho_G} + \frac{W_{LE}}{\rho_L}}{\frac{W_G}{\rho_G} + \frac{W_{LE}}{\rho_L}}$$

The friction factor for a smooth tube flow can be calculated based on the gas core Reynolds number in terms of the total mass flow in the core as follows:

$$f_{sgc} = 0.079 \text{Re}_{sgc}^{-0.25}$$

$$\text{Re}_{sgc} = \frac{V_{gc} \rho_{gc} d}{\mu_G}$$

where the superficial velocity is

$$V_{gc} = \frac{W_G}{\rho_G A}$$

5.2.6 Triangular relationship

As described in Chapter 3, the triangular relationship is required to determine the entrainment rate. For the two pseudo-channels, the triangular relationships are given by

$$\frac{4\delta_1}{d_1} = \sqrt{\frac{(dP/dz)_{LF,1}}{dP/dz}}$$

$$\frac{4\delta_2}{d_2} = \sqrt{\frac{(dP/dz)_{LF,2}}{dP/dz}}$$

where $(dP/dz)_{LF,1}$ and $(dP/dz)_{LF,2}$ are the pressure gradients which would occur in a single phase liquid flow where the mass flow was identical with that in the film, and dP/dz is the pressure gradient in the two-phase flow. d_1 and d_2 are hydraulic diameters for the two pseudo-channels. $(dP/dz)_{LF,1}$ and $(dP/dz)_{LF,2}$ are calculated from the liquid film friction factor due to Hewitt and Hall Taylor (1970)

$$-\left(\frac{dP}{dz}\right)_{LF,1} = \frac{4f_{LF,1}}{d_1} \left(\frac{1}{2}\rho_L V_{LF,1}^2\right)$$

$$-\left(\frac{dP}{dz}\right)_{LF,2} = \frac{4f_{LF,2}}{d_2} \left(\frac{1}{2}\rho_L V_{LF,2}^2\right)$$

where $V_{LF,1}$ and $V_{LF,2}$ are superficial velocities which are calculated from

$$V_{LF,1} = \frac{W_{LF,1}}{A_1 \rho_L}$$

$$V_{LF,2} = \frac{W_{LF,2}}{A_2 \rho_L}$$

To calculate friction factors $f_{LF,1}$ and $f_{LF,2}$, the liquid film Reynolds number is required, thus

$$\text{Re}_{LF,1} = \frac{W_{LF,1} d_1}{A_1 \mu_L}$$

$$\text{Re}_{LF,2} = \frac{W_{LF,2} d_2}{A_2 \mu_L}$$

For turbulent flow, the friction factors could be calculated from the Blasius equation:

$$f_{LF,1,turb} = 0.079 \text{Re}_{LF,1}^{-0.25}$$

$$f_{LF,2,turb} = 0.079 \text{Re}_{LF,2}^{-0.25}$$

For laminar flow, the friction factors could be calculated from the equation:

$$f_{LF,1,lam} = \frac{16}{Re_{LF,1}}$$

$$f_{LF,2,lam} = \frac{16}{Re_{LF,2}}$$

For a smooth transition from laminar to turbulent flow, the friction factors could be calculated using the equation Hewitt and Hall-Taylor (1970) suggested.

$$f_{LF,1} = \sqrt{f_{LF,1,turb}^2 + f_{LF,1,lam}^2}$$

$$f_{LF,2} = \sqrt{f_{LF,2,turb}^2 + f_{LF,2,lam}^2}$$

5.2.7 Frictional Pressure Gradient

To calculate the frictional pressure gradient, the wall shear stress must be known. This study follows Whalley's method, which assumes that the wall shear stress is almost equal to the interfacial shear stress. The assumption is reasonable as the liquid films are thin.

5.3 Computing Model

5.3.1 General

This section describes the computing model based on the mathematical model created in the previous section. The complexity of the partition model formulation requires a numerical solution with the aid of computer. The channel is converted into discrete nodal points in flow direction. A software system CHFA++ (CHF Analysis Plus Plus) has been designed and implemented based on the partition model.

The computing model was designed using Rumbaugh's OMT (Object Modeling Technique) methodology (Rumbaugh et al., 1995) and was implemented using C++.

5.3.2 Two-Phase Flow and Heat Transfer System

Figure 5.9 shows the architecture of the CHFA++ software system. The partition model is

initialized by specifying the geometry shape and fluid type. The system stops running and records the outlet condition when CHF occurs at the outlet of the tube.

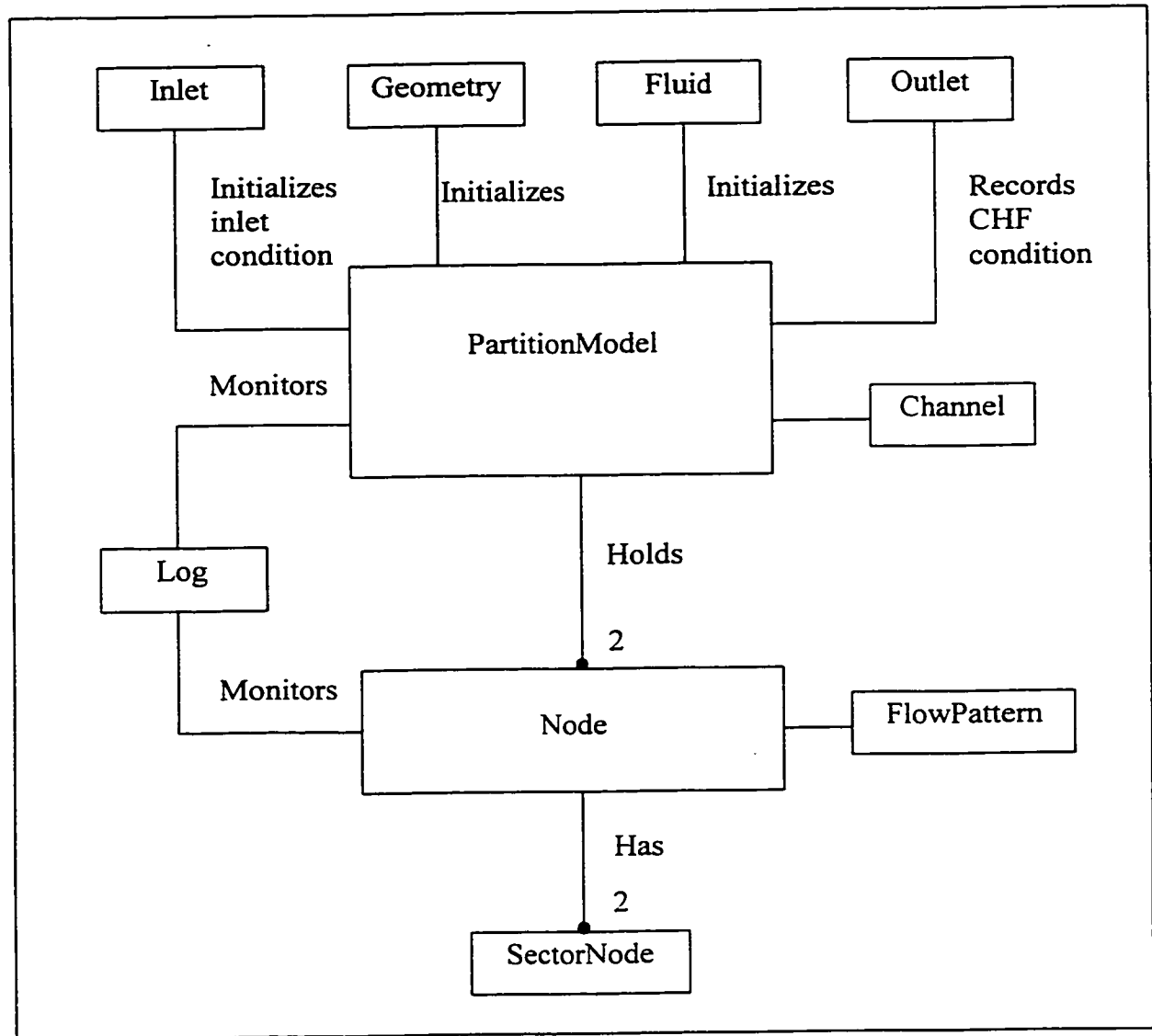


Figure 5.9 Object diagram for two-phase flow and heat transfer.

5.3.3 Subsystems

As shown in Figure 5.10, the CHFA++ system is divided into two subsystems: kernel and application. The kernel provides conservation equations, deposition/entrainment model, interfacial shear stress and triangular relationship, etc. The kernel is flow channel geometry

shape and fluid type independent. It only deals with three channels (the entire physical channel and the two pseudo-channels partitioned). The partition of the flow channel is transparent to the kernel. Outside the kernel, are the surrounding application modules, which include geometry specific modules and fluid modules. The geometry specific modules describe the peculiarities of each geometry shape, such as the deposition factor. These modules are geometry shape specific. The fluid modules provide properties of each working fluid. Currently, the CHFA++ package supports water and R-134A.

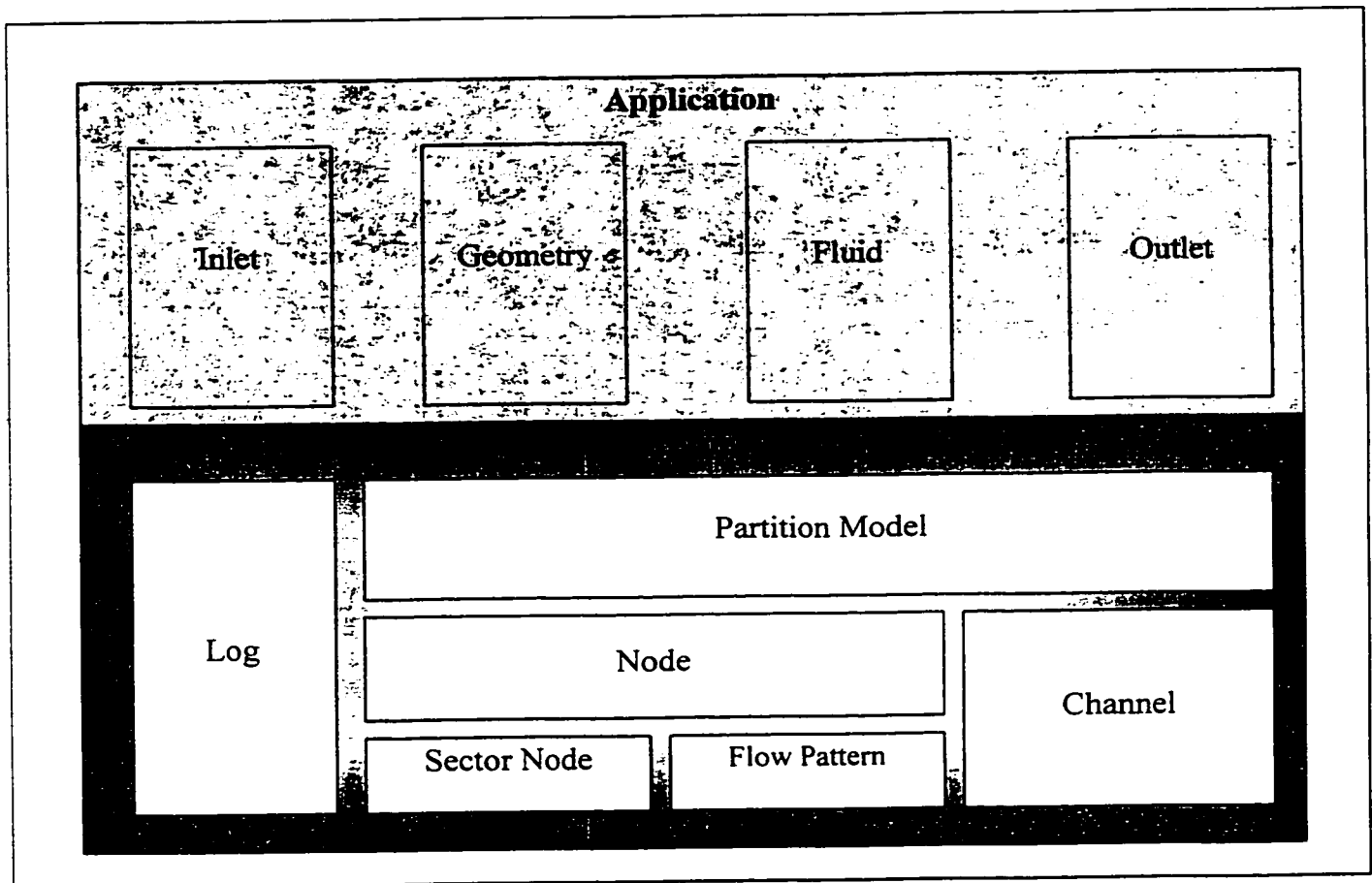


Figure 5.10 Partition model subsystem hierarchy.

The software system is designed in such a way that it is very easy to extend to other geometry and fluid in the future. For the further research for modeling other geometries, no code change for the kernel is needed.

Another advantage of the design makes it easy to verify the software system. The kernel is the

same for simple geometry and complex geometry. The kernel can be verified utilizing the most reliable CHF data, like Groeneveld's CHF lookup table or real test data for circular tube. Then the verified kernel can be used to predict the CHF in complex geometry for which not many CHF measurement data are available.

5.3.3.1 Kernel

The kernel handles the conservation equations, deposition and entrainment models, triangle relationship, etc. It is fundamental of the partition model.

5.3.3.1.1 Partition Model

Partition Model module is the umbrella of the system and plays the role of controller for the entire system. It provides the interface to applications and controls the process for CHF computing. Like the CHF experiment, at the given inlet conditions, heat flux is adjusted until the CHF phenomenon occurs at the exit of the channel.

5.3.3.1.2 Node

The control equations that describe the two-phase flow inside the channels can only be solved using numerical methods. Therefore, the continuous flow channel is divided into a series of discrete nodes and the flow and the heat transfer inside the channel are calculated incrementally.

As shown in Figure 5.11, while the working fluid is heated up inside the channel, the node state is changed from subcooled to CHF condition.

As illustrated in the section 2.1, the states of node changes from inlet condition, subcooled, saturation, homogeneous, onset of annular flow, annular flow, to CHF condition. For this study, the system simulates the real CHF test process. Inlet state is usually subcooled, and the inlet state can be considered as a special case of subcooled state. However, inlet condition is not limited to the subcooled condition. In case of saturation inlet condition, subcooled state is skipped.

5.3.3.1.3 Log

Log provides a facility to record and monitor the process of computation when it is turned on.

Error messages will be logged during the computing. In the case of events such as unreasonable results and infinite loops, the program will terminate. All the computing parameters will be logged for the purpose of debugging.

5.3.3.1.4 *Flow Pattern*

In current study, the model is limited to dryout-type CHF and the flow is restricted to annular. A flow pattern facility is added to the system to check if the flow type is annular.

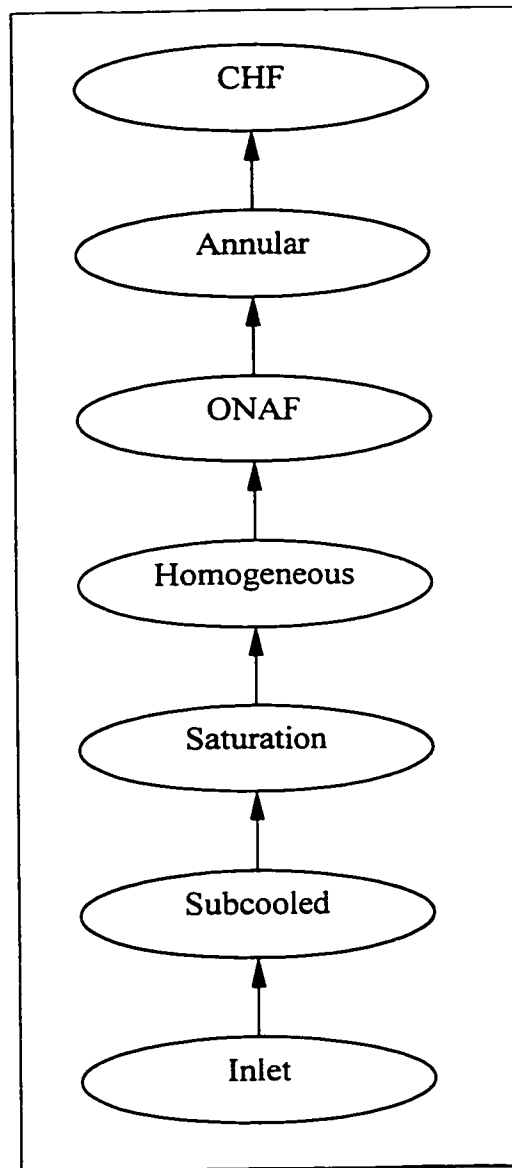


Figure 5.11 Node states.

5.3.3.2 Application

While the kernel provides the fundamentals of the partition model, the application provides the geometric or fluid specific modules as well as input and output facilities.

5.3.3.2.1 Geometry

The geometry module provides the geometric specific parameters. It computes the flow areas, perimeters, the shape factors, and deposition factors for both pseudo-channels. In the current study, two geometries, circular and dumbbell-shaped tubes, are implemented. In case of the dumbbell-shaped tube, the computation of these parameters is complicated.

The partition model divides a flow channel into two pseudo-channels. Therefore, three channels are involved in the CHF computing, one physical channel and two pseudo-channels. It is worthwhile to distinguish the geometry parameters with following categories:

- Entire physical channel specific geometry parameters;
- Per pseudo-channel based geometry parameters;
- Node specific geometry parameters.

5.3.3.2.2 Fluid

This module provides the properties of fluids. In this study, the deposition coefficient, k , is considered to be a function of surface tension σ . Therefore, based on Whalley's approach (Whalley, 1977), the deposition coefficient is given in this module as a property.

Figure 5.12 shows the hierarchy of classes of fluid. Currently, only two fluids are implemented. Class Fluid is an abstract class that works as a base for concrete classes Water and R134a. Much more fluids can be added to this fluid class hierarchy based on the UO property code.

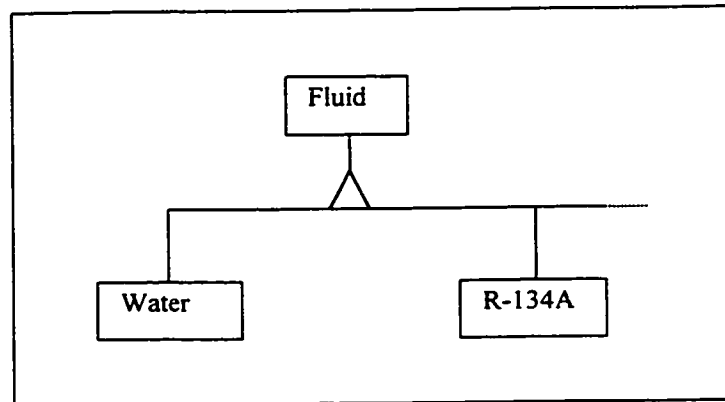


Figure 5.12 Fluids.

5.3.4 Scheme of CHF Computing

5.3.4.1 General

The main scheme of CHF computation is illustrated in Figure 5.13. The model is initialized based on the system parameters. Before the computing start, an initial heat flux value is guessed based on empirical correlation. In fact, this initial heat flux value will not affect final results, but a good initial value may reduce the iteration time. The node is then set to the inlet state. The transition from the inlet state to ONAF can be calculated by simply using the heat balance. The annular flow node state is calculated until the CHF condition is reached. Because the heat flux value used is a guessed value, the CHF may not occur at the exit of the channel. In this case, a new heat flux is assumed by multiplying with a factor of previous CHF location. Similar to the real CHF experiments, the heat flux is fine tuned until CHF occurs at the exit of the channel. Unlike the real CHF experiments, the previous CHF location is known, so that the process of finding CHF value is much faster.

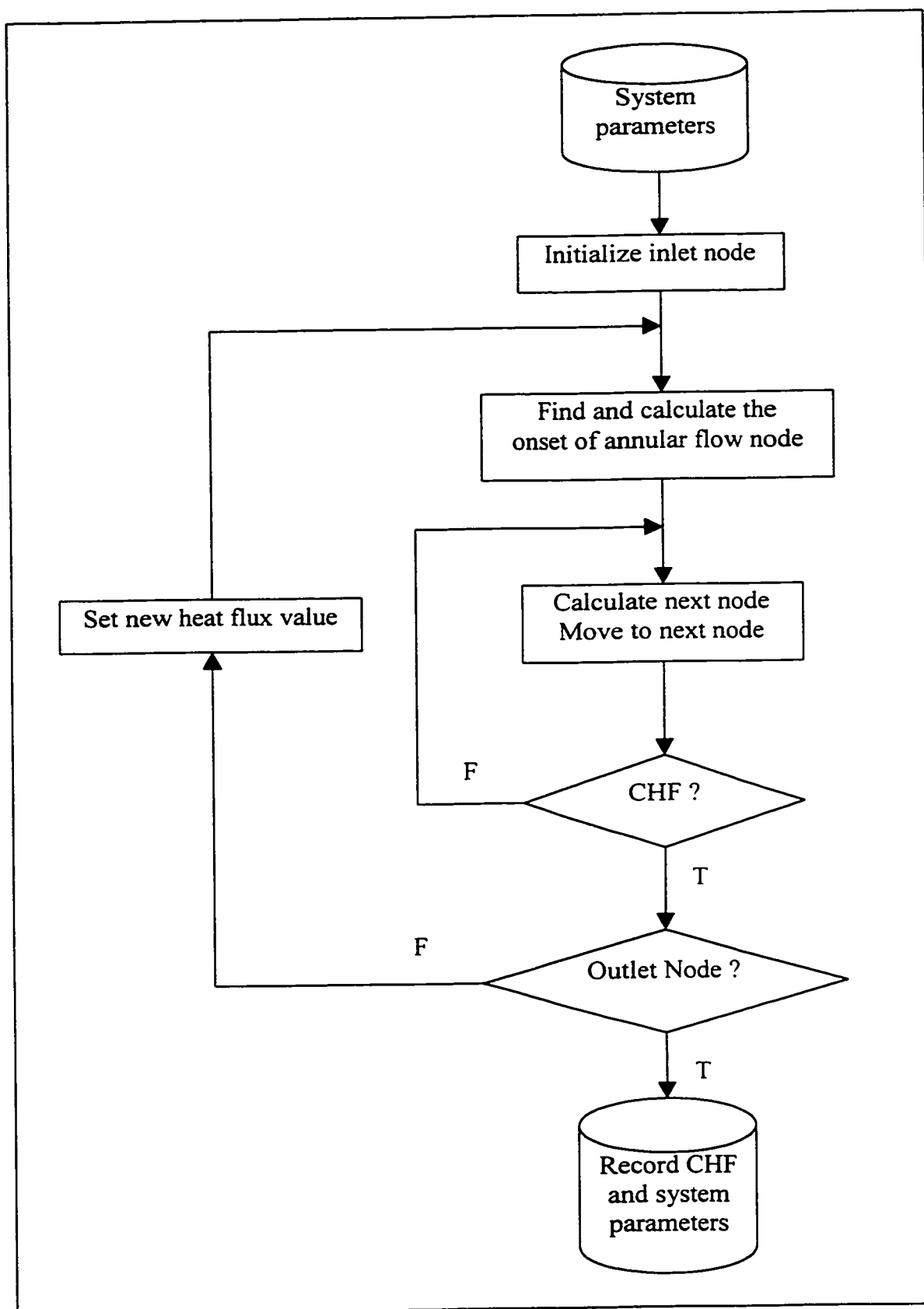


Figure 5.13 General scheme of CHF computing.

5.3.4.2 Increments

The computation of next node is started by increasing axial location from z to $z+dz$. Using the triangle relationship and interfacial roughness correlation, the interfacial shear stress and film thickness are calculated for each pseudo-channel. With these data, the equilibrium concentrations of liquid droplets are obtained. Mass flow rates are then calculated based on mass conservation equations.

Chapter 6

EXPERIMENTAL RESULTS AND ANALYSIS

This chapter presents the experimental results and analysis. A total of 624 CHF data points were obtained in the dumbbell-shaped tube during this experimental study. Appendix III tabulates all the test data, including the location at which CHF first occurred.

Section 6.1 gives an overview of the overall CHF test results and the general parametric trends observed. The experimental CHF data are analyzed with respect to various flow parameters, including pressure P , mass flux G , quality X and inlet enthalpy h_{in} . Section 6.2 analyzes the geometry effect in detail. The influences of quality, mass flux, pressure, and inlet enthalpy on the gap effect are discussed. In section 6.3, a comparison is made between the CHF data for dumbbell-shaped tubes and a circular tube of equivalent hydraulic diameter. Section 6.4 presents the observation results of the flow visualization test in a dumbbell-shaped tube. The results of this experimental study are summarized in section 6.5.

6.1 General Trend

As discussed in Chapter 2, for a given geometry, there are four main flow parameters that affect CHF:

- inlet enthalpy (subcooling), $h_{in}(\Delta h_{in})$;
- local quality, X ;
- mass flux, G ;
- system pressure, P .

The effects of these four parameters are analyzed in this section. It is noted that these effects are not all independent but are mutually related.

Normally, the CHF is discussed based on either fixed inlet condition or local condition. For a given geometry at fixed inlet condition, the CHF is a function of pressure, mass flux, inlet

enthalpy/subcooling and heated length:

$$CHF = f(p, G, h_m, L)$$

At fixed local condition, the CHF is expressed as

$$CHF = f(p, G, x)$$

In this section, the experiment results for the CHF in a dumbbell-shaped tube are discussed based on both conditions. Note that this section only presents the CHF trends against flow parameters.

6.1.1 Inlet Condition

To discuss the CHF based on inlet condition, the heated length has to be fixed. For all the runs conducted in this experimental study, the fluid entering the tube was subcooled. Due to the restriction of the cooling method, the inlet temperature is limited, and therefore the degree of subcooling was small in all cases. To extend the test range, the CHF was measured at different heated lengths. As described in Chapter 4, the heated length of the test section was varied from 0.7 m to 1.4 m.

For the purpose of CHF data comparison, all the test data are converted to a fixed pseudo heated length l_{pseudo} in this section. The pseudo inlet enthalpy based on the pseudo heated length $(h_{in})_{pseudo}$ is then derived according to the energy balance:

$$(h_{in})_{pseudo} = (h_{in})_{test} + (l_{he} - l_{pseudo}) \frac{q l_p}{G A}$$

where, l_{he} is the actual heated length; l_p and A are perimeter and flow area of the dumbbell-shaped tube, respectively; $(h_{in})_{test}$ is the real inlet enthalpy that is determined by inlet temperature and system pressure. The converted inlet enthalpy data values are tabulated in Appendix IV.

6.1.1.1 Inlet Enthalpy/Subcooling Effect

The effect of inlet enthalpy on CHF can be considered as the effect of inlet subcooling at a fixed system pressure. The inlet subcooling is the liquid saturation enthalpy minus inlet enthalpy.

$$(\Delta h_{sub})_{in} = h_f - h_{in}$$

Figure 6.1 and Figure 6.2 illustrate the variation of the CHF with inlet enthalpy. For each curve in the figures, the gap size, pressure, mass flux or mass flow rate are fixed. The heated length is

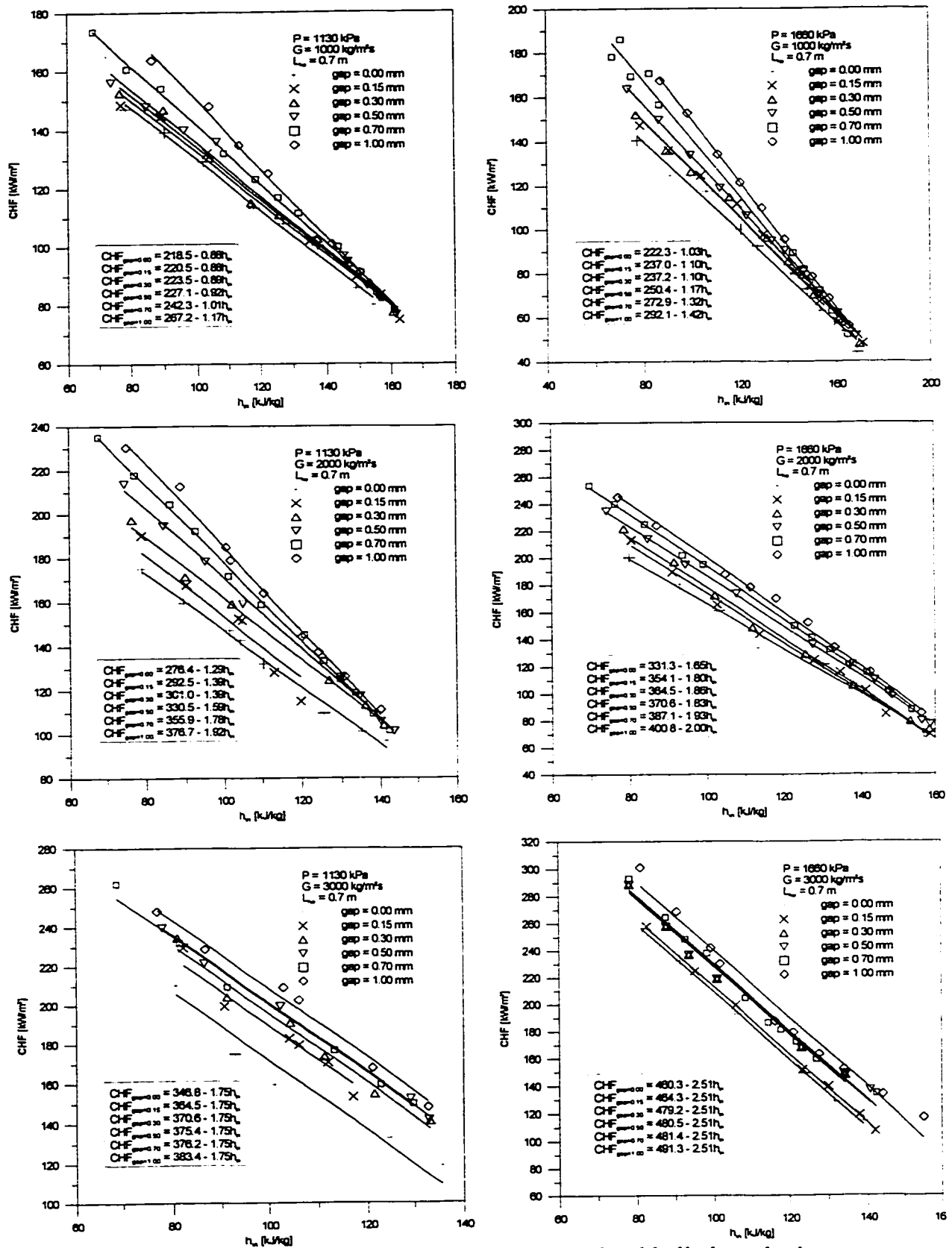


Figure 6.1 Inlet enthalpy effect on CHF in dumbbell-shaped tube.

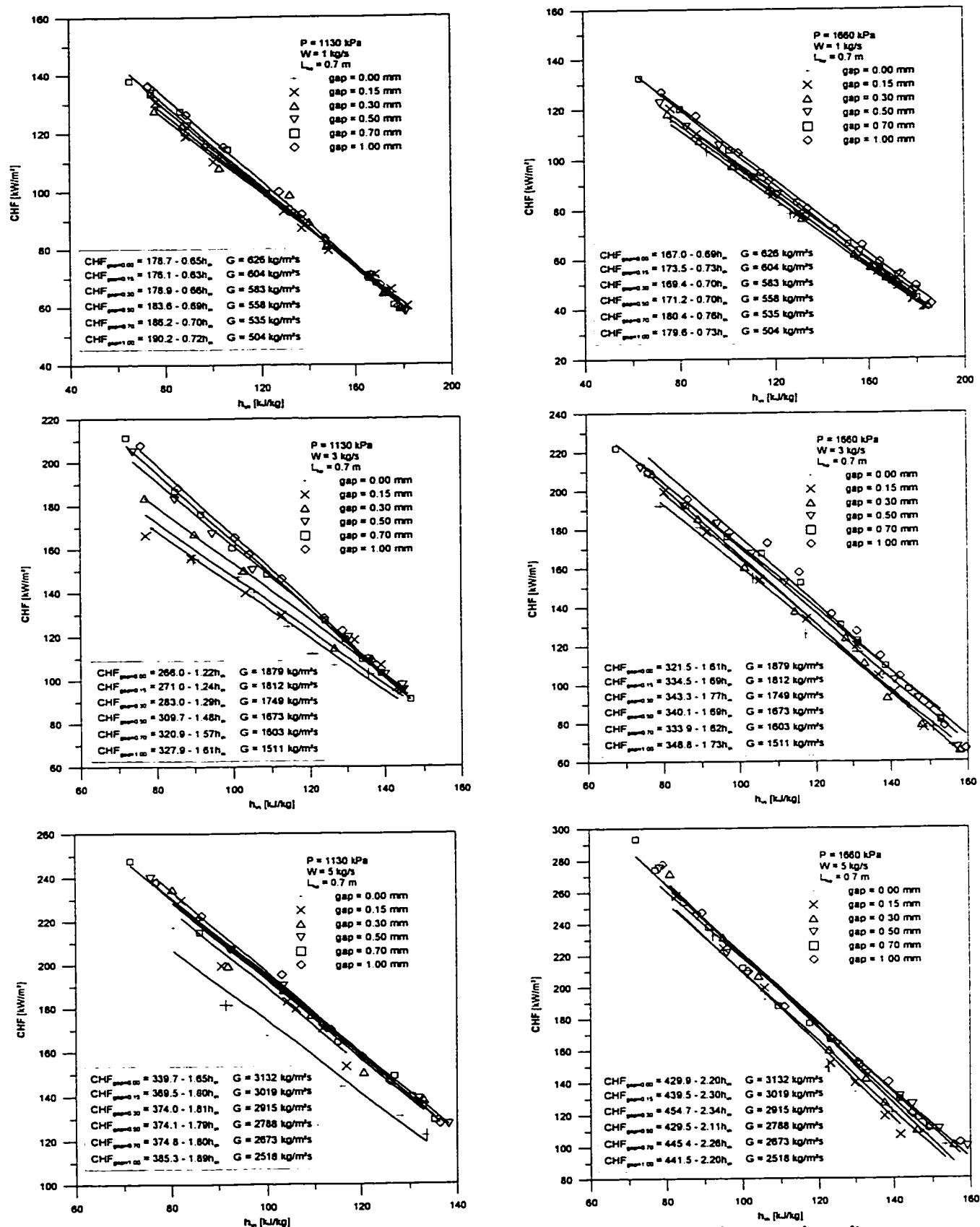


Figure 6.2 Inlet enthalpy effect on CHF in dumbbell-shaped tube (continued).

set to 0.7 m for the purpose of comparison.

The figures show that for a given pressure, mass flux (or flow rate) and tube heated length, the critical heat flux in the dumbbell-shaped tube decreases almost linearly with increased inlet enthalpy (or decreased inlet subcooling). This trend can be explained in that more energy is required for lower inlet enthalpy flow. In general, the effect of inlet subcooling or inlet enthalpy on CHF in the dumbbell-shaped tube has the same trend as that in a circular tube. No severe effect is found even for the zero gap case based on fixed inlet condition. The gap effect is analyzed in detail in section 6.2.

6.1.1.2 Mass Flux Effect at Fixed Inlet Condition

Figure 6.3 and Figure 6.4 present the effect of mass flux on CHF in the dumbbell-shaped tubes at fixed inlet condition. It is found from the figures that the CHF increases with mass flux at fixed inlet enthalpy (subcooling), pressure, and geometric parameters. For a given geometry at fixed inlet condition, the higher the mass flux, the more energy is needed to reach the CHF condition.

For the three mass fluxes tested, the CHF rises more rapidly at low flow (from 1000 kg/m²s to 2000 kg/m²s) than at high flow (from 2000 kg/m²s to 3000 kg/m²s). This trend is also the same as that of the circular tube.

It is also found from the figures that the mass flux effect on CHF depends on inlet condition. The CHF at different mass flux converges with an increase of inlet enthalpy. The higher the inlet enthalpy is, the less energy is required.

In summary, the trend of mass flux effect on CHF in the dumbbell-shaped tube is the same as that in a circular tube. At a fixed inlet condition, all the runs with different gap sizes have a similar trend with mass flux.

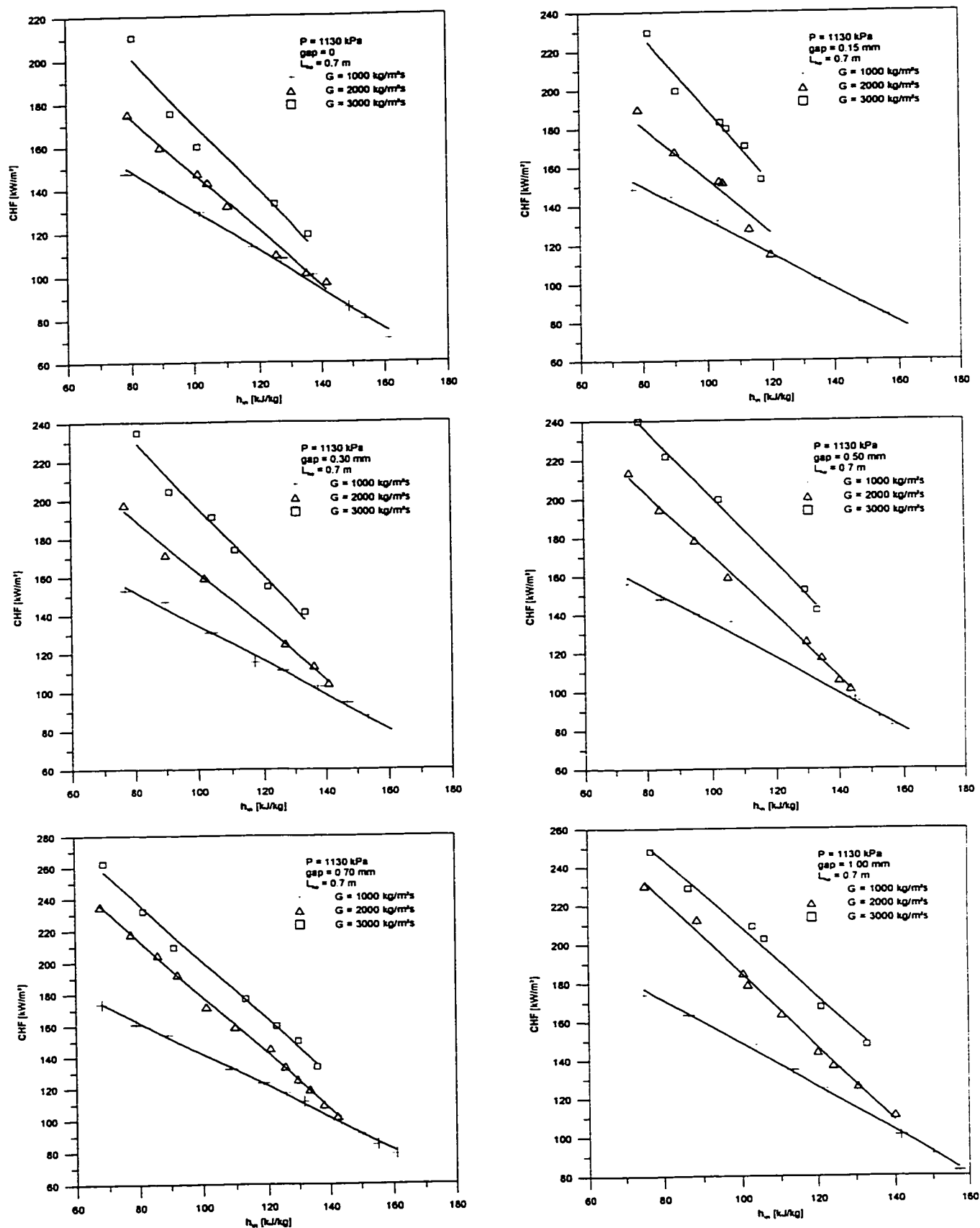


Figure 6.3 Mass flux effect on CHF in dumbbell-shaped tube (inlet condition, 1130kPa).

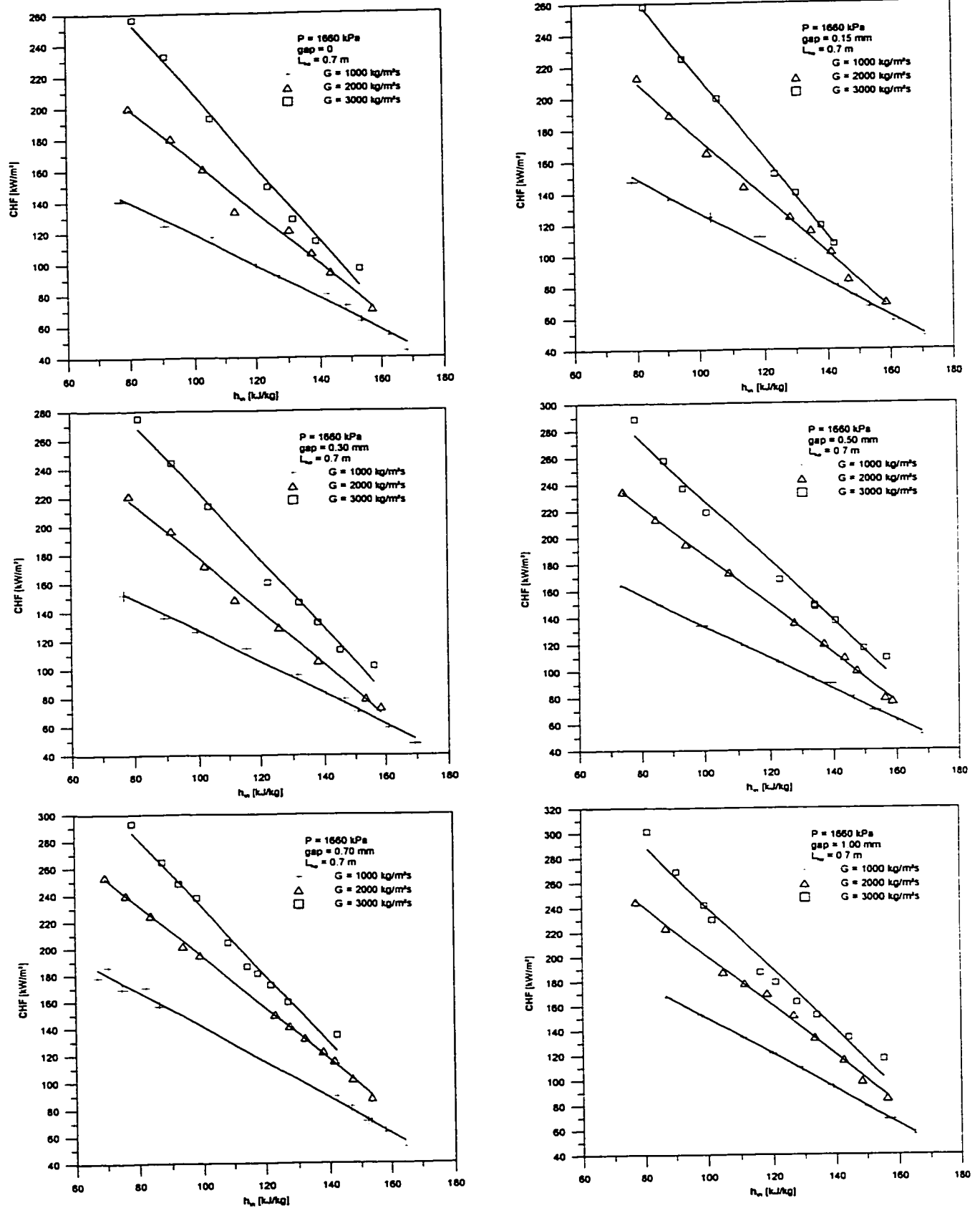


Figure 6.4 Mass flux effect on CHF in dumbbell-shaped tube (inlet condition, 1660kPa).

6.1.1.3 Pressure Effect at Fixed Inlet Condition

Figure 6.5 illustrates the pressure effect on CHF at fixed inlet subcooling. When comparing CHF values at different pressures based on the fixed inlet condition concept, inlet subcooling is usually used. It is noted that the inlet subcooling is different at different pressures for the same inlet enthalpy.

Only the results for a 0.7 mm gap tube are plotted because the pressure effect shows a similar trend for all other gap sizes tested. In this experimental study, CHF was measured at two different pressures, 1130 kPa and 1660 kPa, which are equivalent to 7 MPa and 10 MPa in water.

As discussed in Chapter 2, pressure has a strong effect on CHF because of its effect on thermodynamic properties. For circular tubes, the critical heat flux initially increases with increasing pressure and then decreases with increasing pressure. As described by Whalley (1987), for steam-water flows, the CHF reaches a maximum value at a pressure about 7 MPa. According to fluid-to fluid modeling technique, this transition pressure is equivalent to 1130 kPa for R-134a.

The dumbbell tube test results show that the CHF at 1130 kPa is higher than the CHF at 1660 kPa. This implies that the pressure effect on CHF in the dumbbell-shaped tube appears to show a similar trend as that in a circular tube at fixed inlet condition (subcooling).

Therefore, it can be concluded that at fixed inlet condition, the effect of the flow parameter on CHF in the dumbbell-shaped tube has a similar trend to that in a circular tube.

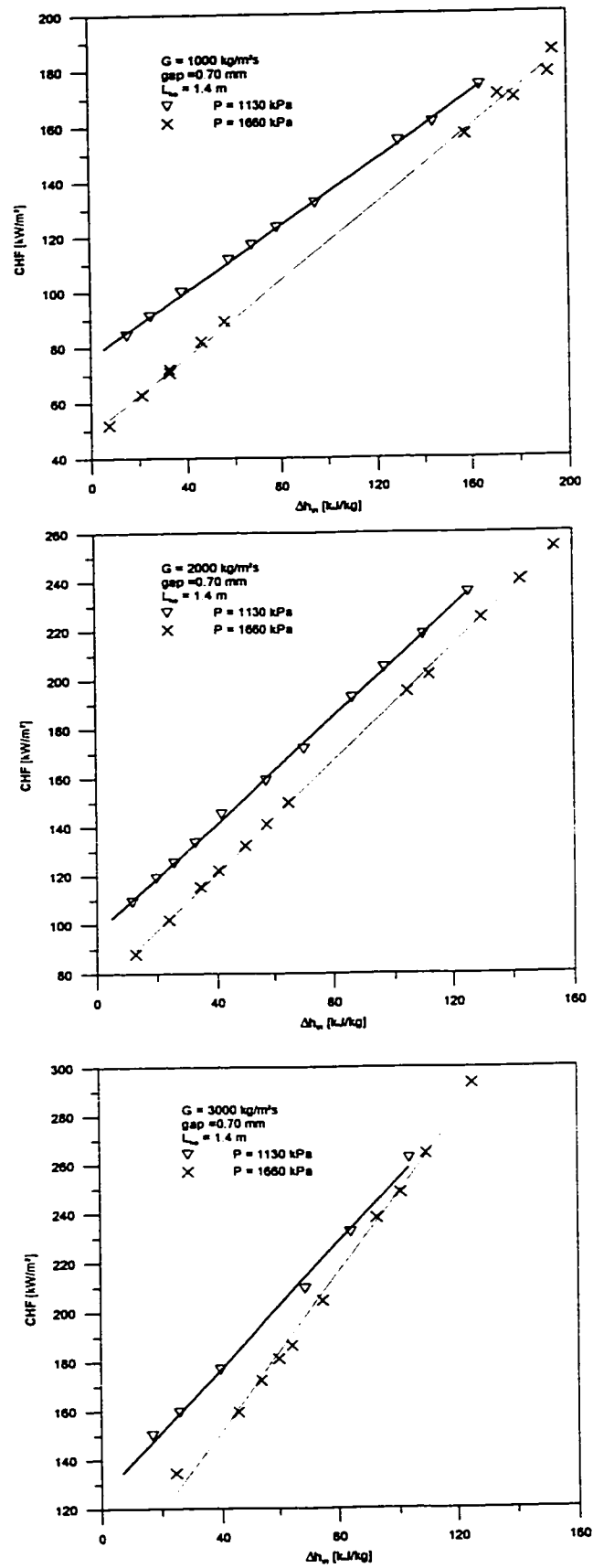


Figure 6.5 Pressure effect on CHF in dumbbell-shaped tube (inlet condition).

6.1.2 Local Condition

6.1.2.1 Quality Effect at Fixed Local Condition

Figure 6.6 to Figure 6.11 show the CHF in the dumbbell-shaped tube plotted against quality at a fixed pressure and mass flux. The effect of critical quality X on CHF in the dumbbell-shaped tube shows a similar trend to that in circular tubes. The CHF decreases with increased quality. The effect of quality on CHF is due primarily to the local wall voidage, which impairs the critical heat flux, and secondarily to the bulk voidage, which affects the flow pattern. At 100% quality the CHF is zero. No severe adverse effect (decreasing CHF more than 50%) is observed in the dumbbell-shaped tubes at the flow conditions tested.

The limiting critical quality phenomenon is detected at low flow ($G = 1000 \text{ kg/m}^2\text{s}$) in the dumbbell-shaped tube as shown in Figure 6.6 and Figure 6.9. The phenomenon of “the limiting critical quality” is characterized by a sharp drop in CHF at the “limiting quality”. The phenomenon occurs at higher quality (around 0.6-0.68 at $P = 1130 \text{ kPa}$; 0.46-0.51 at $P = 1660 \text{ kPa}$) for all gap sizes.

A comprehensive literature survey for the limiting quality phenomenon is given by Kitto (1980). Whalley (1987) found that many of the data exhibiting such a limiting quality were obtained in experiments for which the fluid at the tube inlet was not liquid but a mixture of gas and liquid. The method of mixing the liquid and gas at the inlet affects the initial fraction of liquid entrained, and this affects the critical heat flux results. One of the greatest difficulties with the limiting quality hypothesis is to describe mechanistically the apparent sudden instability of the liquid film once a certain quality has been reached.

It is also found that the limiting quality moves to a lower quality with an increase in gap (around 0.68 at zero gap vs. around 0.6 at 1 mm gap at $P = 1130 \text{ kPa}$). It is noted that the hydraulic diameter increases with increased gap size. For a circular tube, the limiting quality switches to a lower quality with an increase in diameter. Further research is needed to investigate the effect of geometry on the limiting critical quality phenomenon.

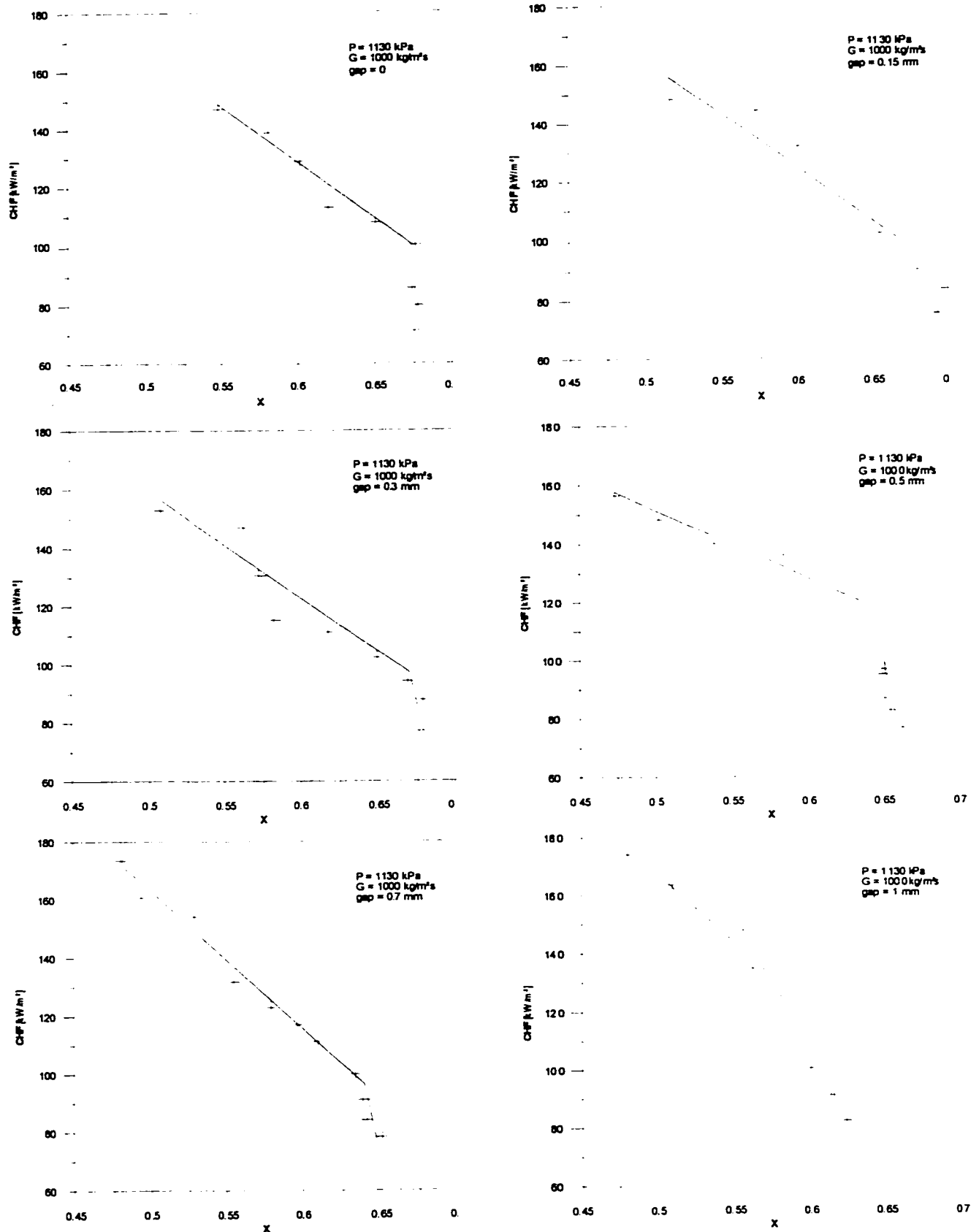


Figure 6.6 Exit quality effect on CHF in dumbbell-shaped tube (1130kPa, 1000kg/m²s).

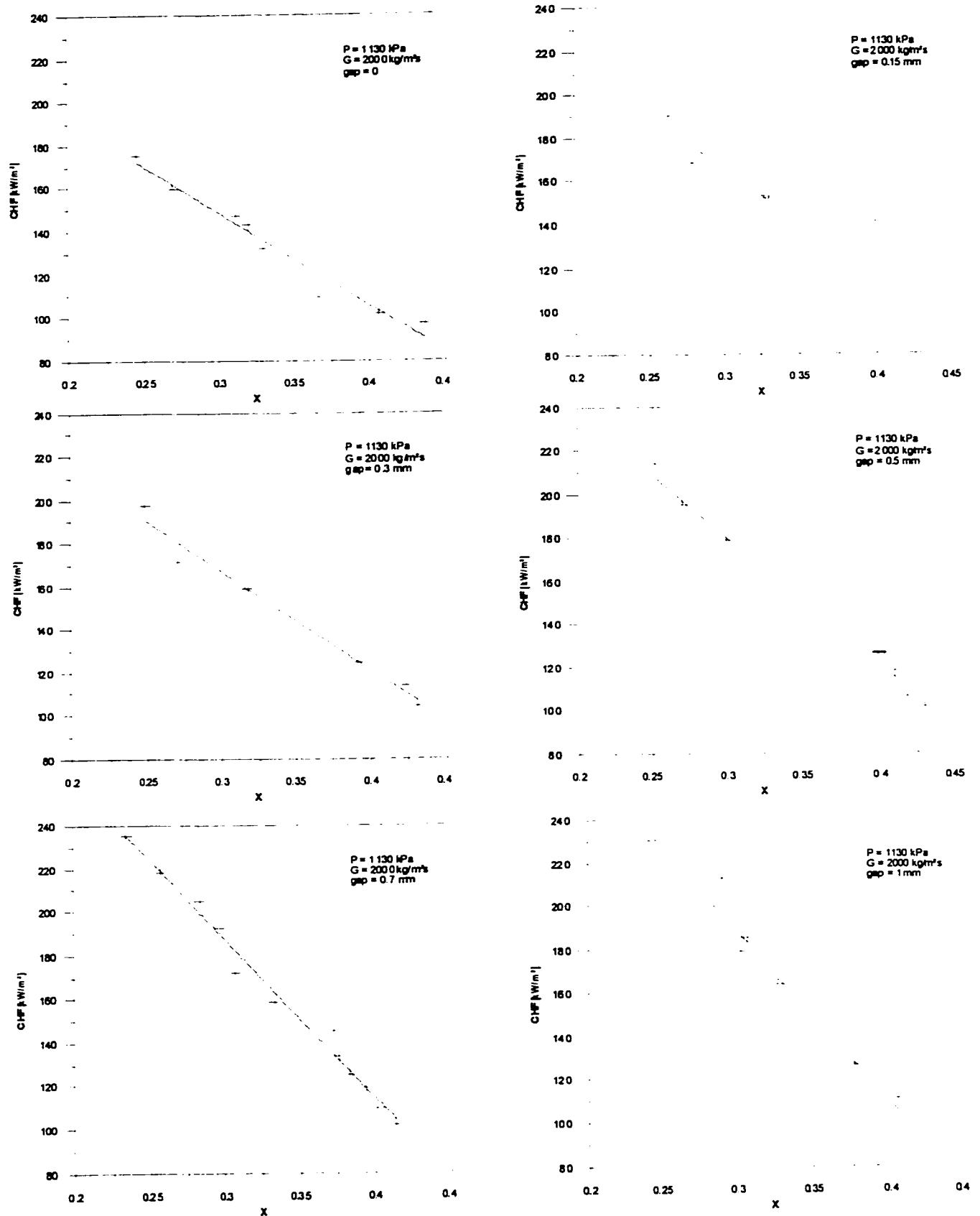


Figure 6.7 Exit quality effect on CHF in dumbbell-shaped tube (1130kPa, 2000kg/m²s).

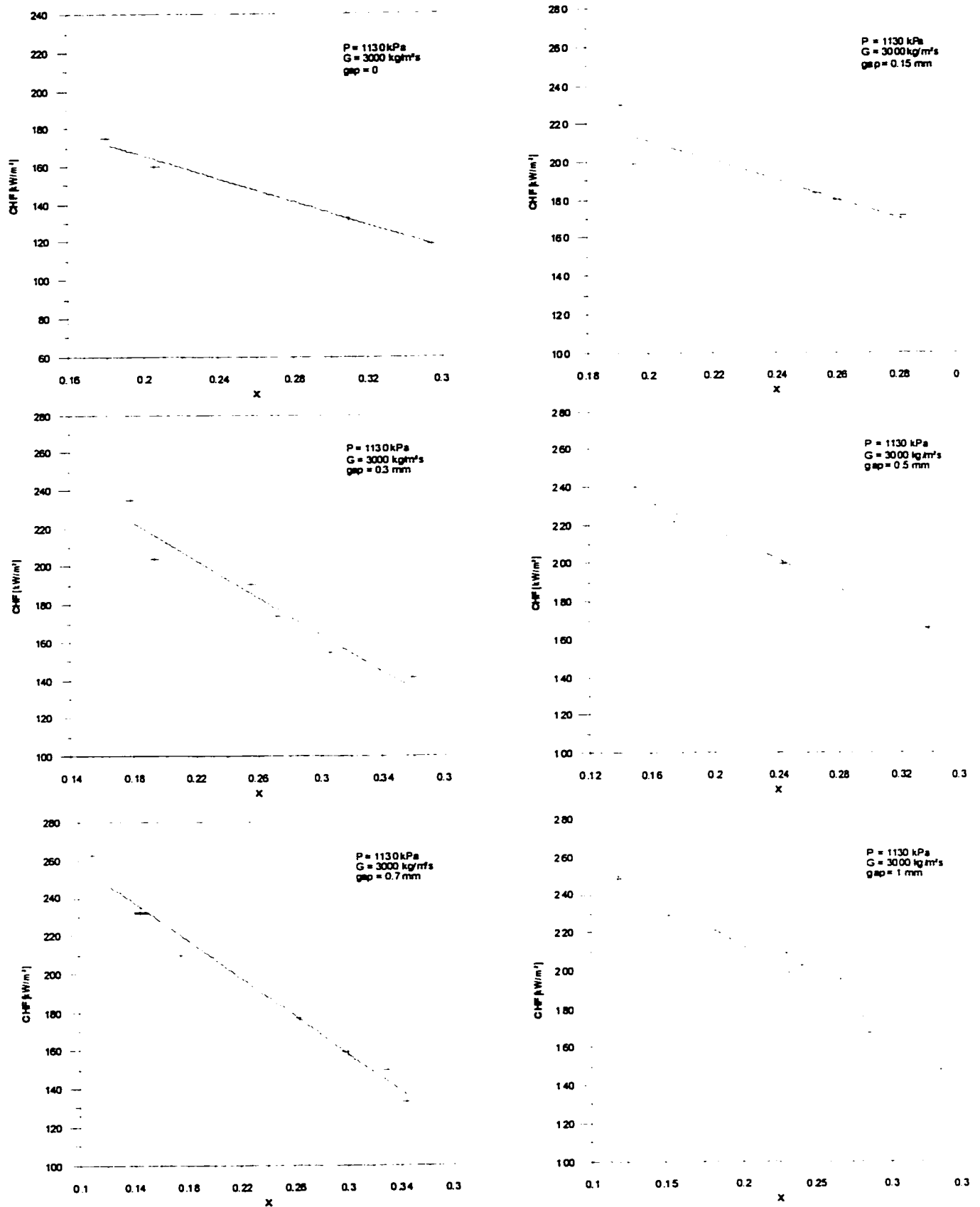


Figure 6.8 Exit quality effect on CHF in dumbbell-shaped tube (1130 kPa , $3000 \text{ kg/m}^2\text{s}$).

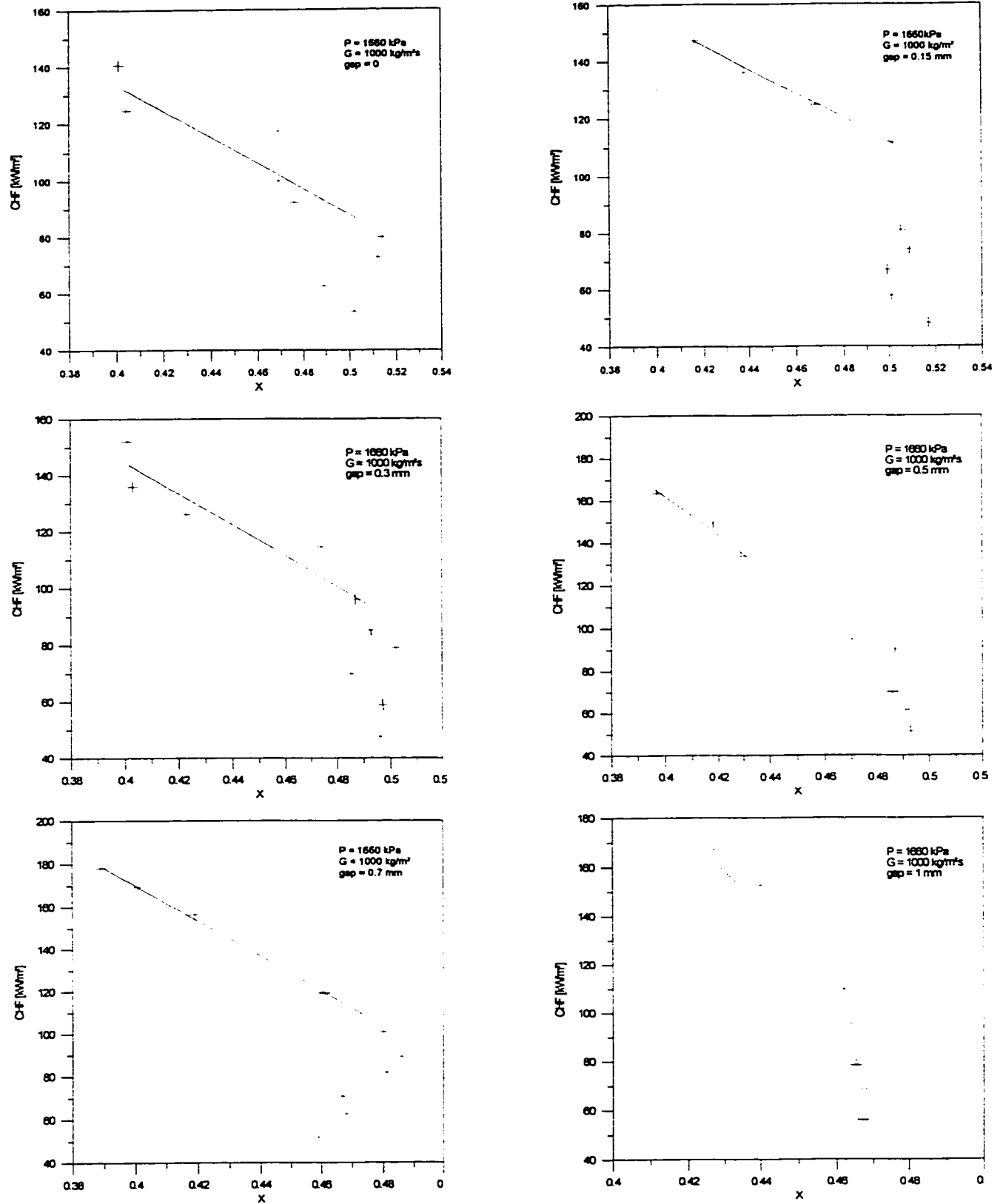


Figure 6.9 Exit quality effect on CHF in dumbbell-shaped tube (1660kPa, 1000kg/m²s).

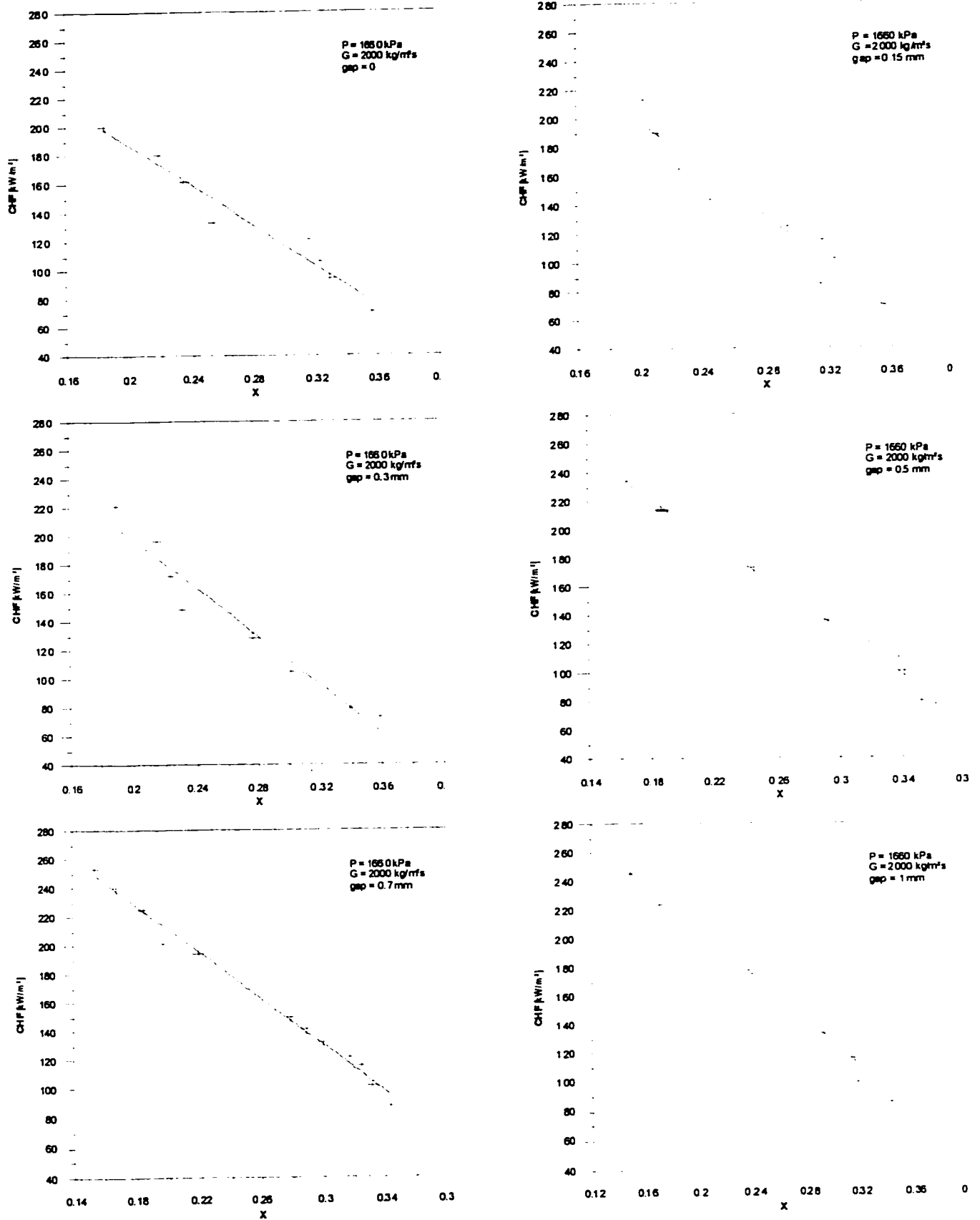


Figure 6.10 Exit quality effect on CHF in dumbbell-shaped tube (1660kPa, 2000kg/m²s).

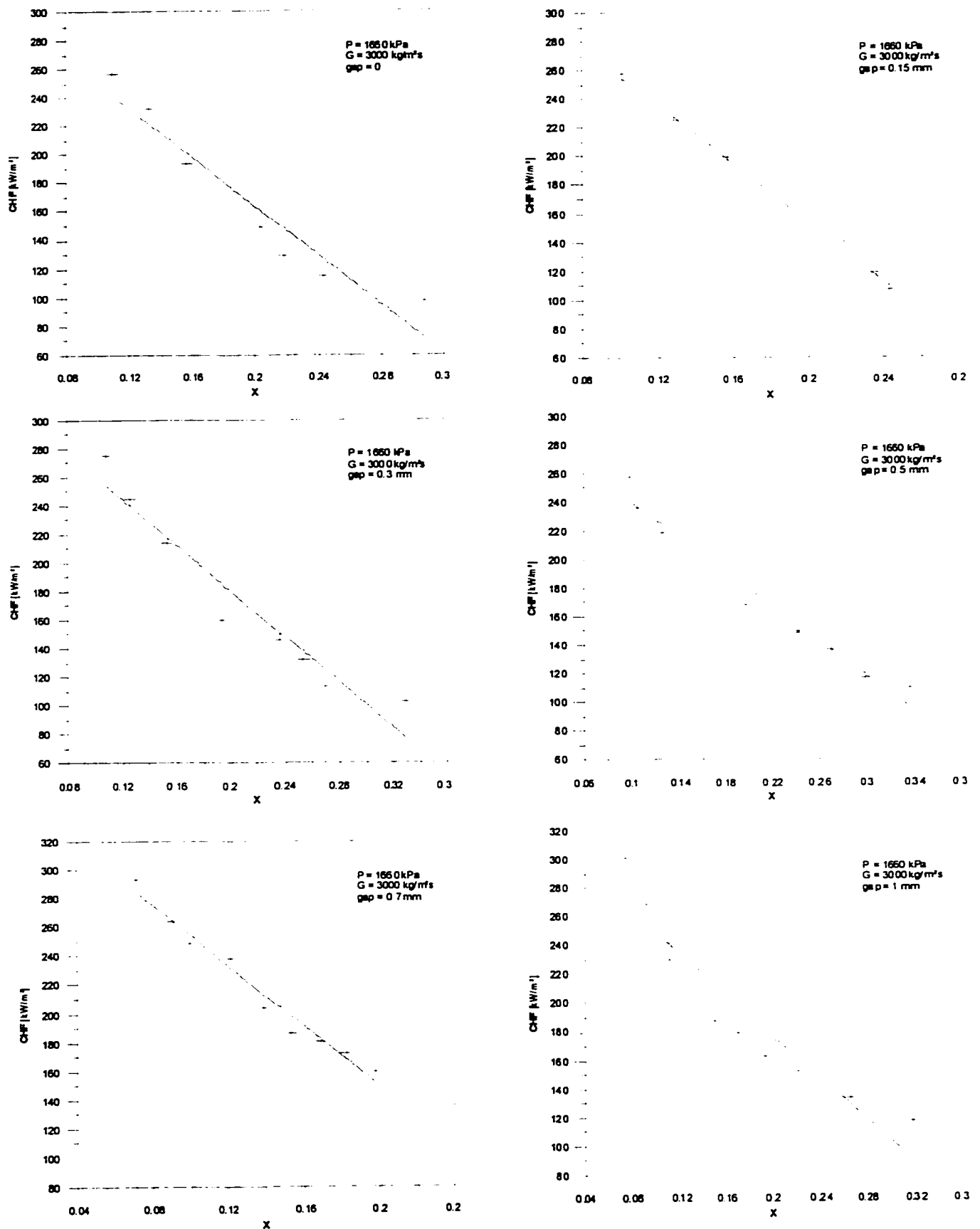


Figure 6.11 Exit quality effect on CHF in dumbbell-shaped tube (1660kPa, 3000kg/m²s).

6.1.2.2 Mass Flux Effect at Fixed Local Condition

Figure 6.12 and Figure 6.13 illustrate the effect of mass flux on CHF at fixed local condition. The CHF has little overlap for all the three mass fluxes tested, but the curves clearly demonstrate the effect of mass flux on CHF. The CHF decreases with increased mass flux. The CHF drops much more rapidly at lower mass flux than that at higher mass flux. This trend is similar to that in a circular tube.

As described in Chapter 2, at higher qualities, dryout-type CHF occurs due to liquid film depletion. Higher mass flux results in a wavier liquid film and thus increases entrainment, which leads to an early liquid film break down and lower CHF.

6.1.2.3 Pressure Effect at Fixed Local Condition

Figure 6.14 shows the pressure effect on CHF in the dumbbell-shaped tube at fixed local condition. Similar to section 6.1.1.3 for fixed inlet condition, the CHF results for a 0.7 mm gap size are used to illustrate the pressure effect since all other gap sizes tested show a similar trend.

At fixed local condition, the CHF at the 1660 kPa is lower than the CHF at 1130 kPa, i.e., the CHF decreases with increased pressure. As described in Chapter 2, CHF decreases with increasing pressure if the pressure is higher than a transition value. The pressure effect on CHF in the dumbbell-shaped tube has a similar trend as that in circular tube at fixed local condition.

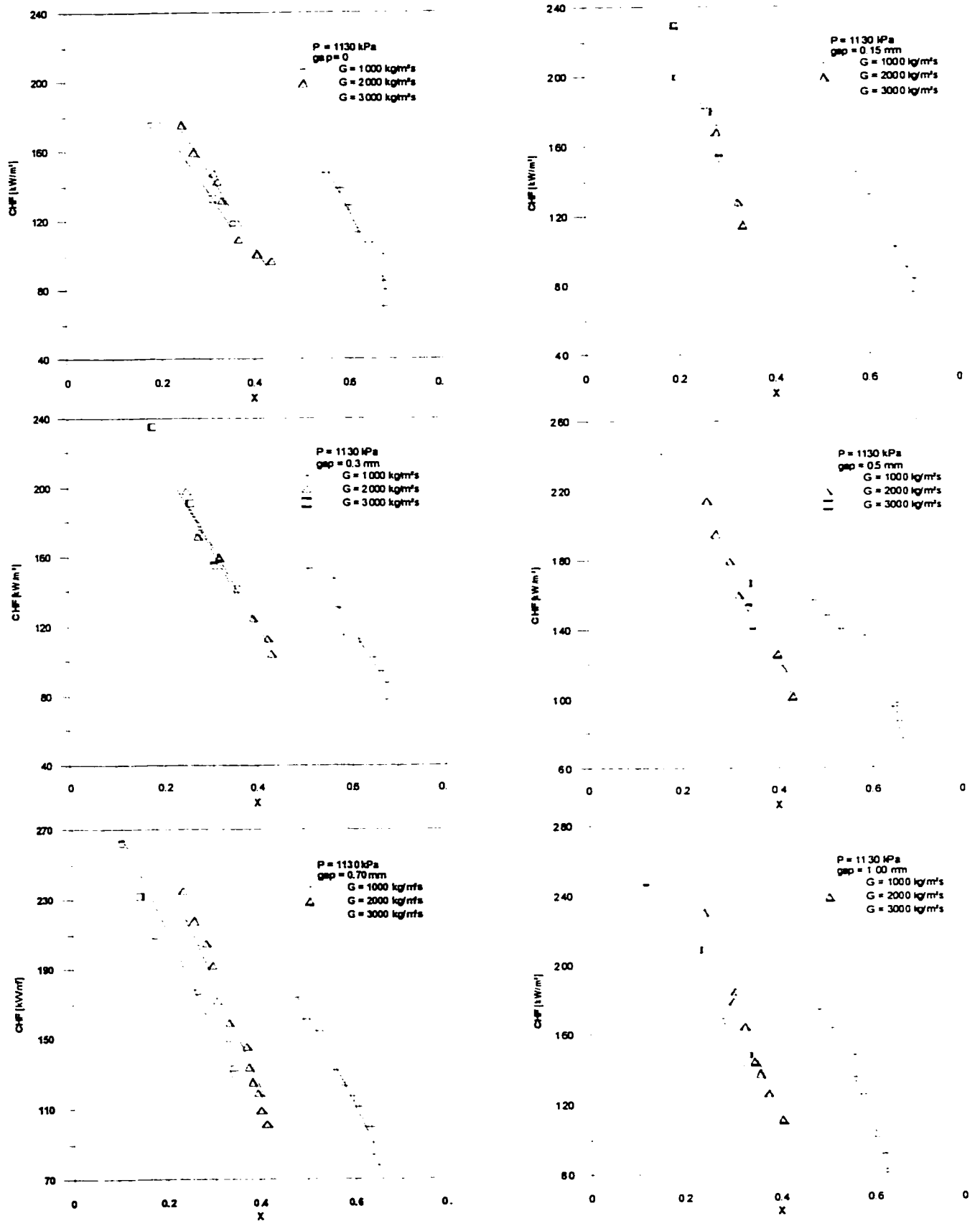


Figure 6.12 Mass flux effect on CHF in dumbbell-shaped tube (local condition, 1130kPa).

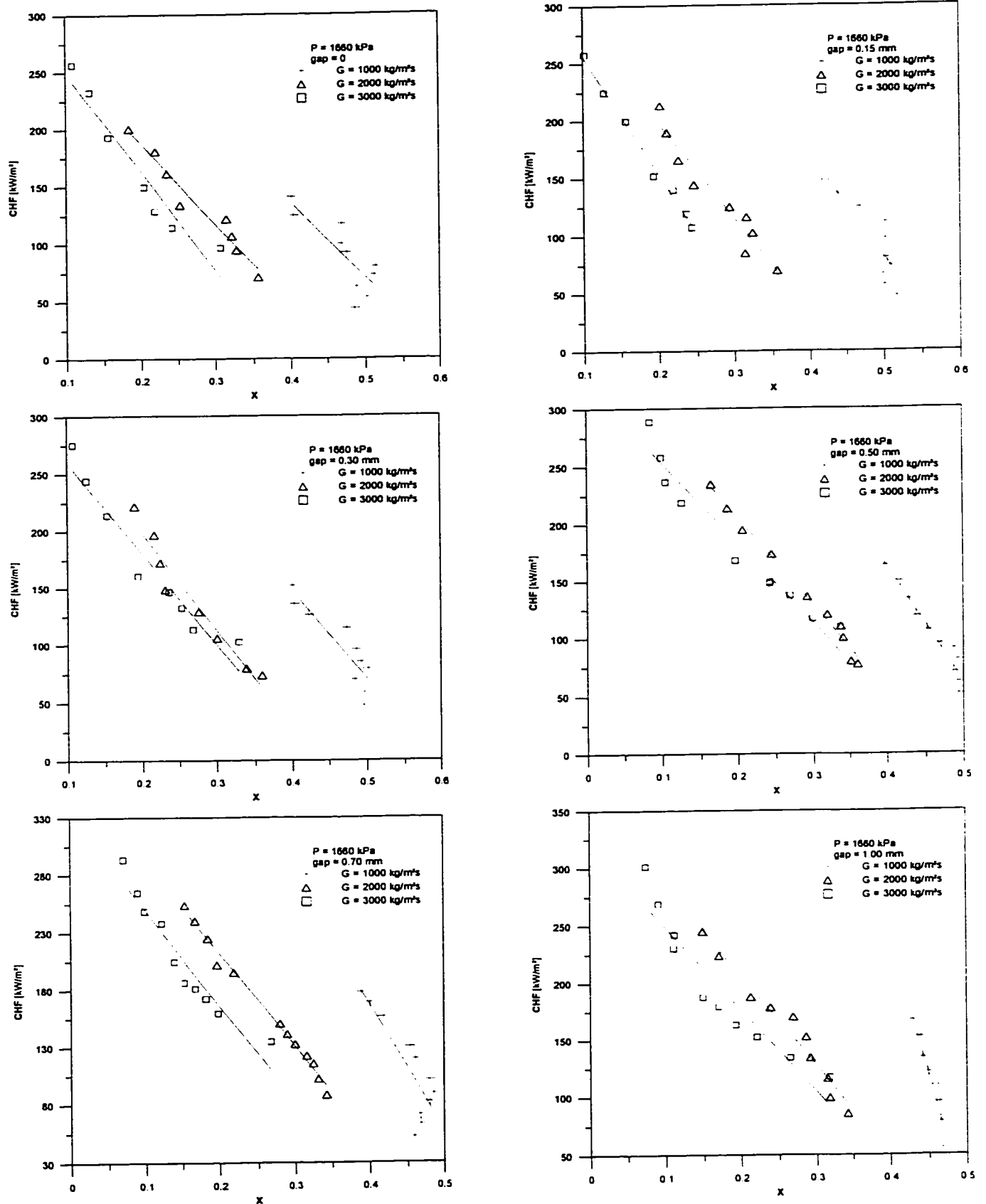


Figure 6.13 Mass flux effect on CHF in dumbbell-shaped tube (local condition, 1660kPa).

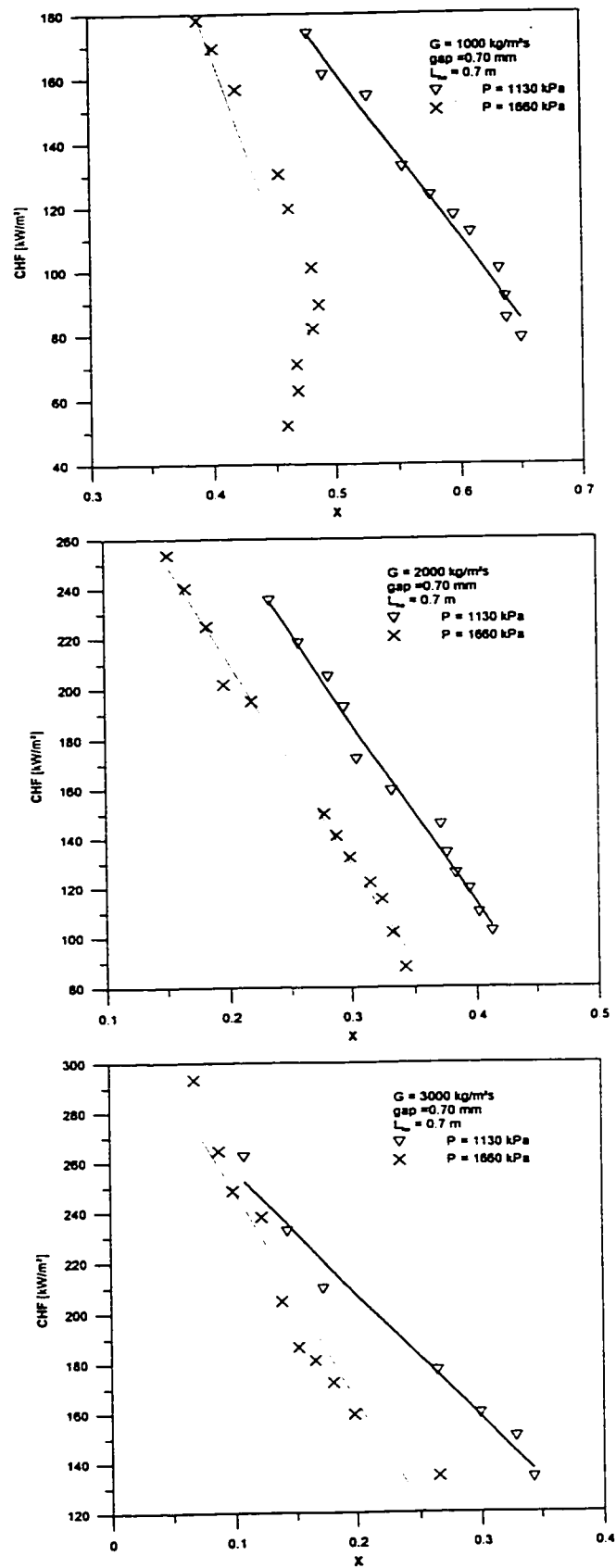


Figure 6.14 Pressure effect on CHF in dumbbell-shaped tube at fixed local condition.

6.2 Gap Effect

The main purpose of this experiment is to investigate the gap effect on CHF. It is noted that in this experiment the flow area increases with increased gap size. With the gap size increasing from 0 to 1 mm, the hydraulic diameter of the dumbbell-shaped tube increases from 3.43 mm to 4.30 mm due to an increase in flow area. This indicates that the measured CHF is a result of the combined effect from all geometric parameters. The overall geometry effect on CHF is presented in subsection 6.2.1 of this section. Subsection 6.2.2 focuses on isolating the gap effect from the overall geometry effect.

Similar to the previous section, the geometry effect is analyzed based on fixed inlet condition and local condition. It is noted that in this study, a CHF ratio ($\text{CHF}/\text{CHF}_{\text{base}}$) is used to represent the gap effect. The dumbbell-shaped tube with 1 mm gap size is chosen as the base. If the CHF decreases with the gap size, the ratio is less than 1. The smaller the value of the CHF ratio, the stronger the gap effect.

To discuss the geometry effect, the flow parameters should be fixed. However, the real experimental data are limited set of data points which are not continuous. In this section empirical equations were generated based on the experimental data using a least squares fit. For a given flow condition, CHF was calculated based on the equations.

6.2.1 Overall Geometry Effect on CHF

6.2.1.1 Overall Geometry Effect at Fixed Inlet Condition

This subsection describes the general CHF trend against the dumbbell-shaped tube geometry. The gap size is used as the characteristic geometric parameter. Figure 6.15 and Figure 6.16 show the effect of geometry based on fixed inlet condition. For each curve, the inlet enthalpy, mass flux, pressure as well as heated length are fixed. The figures at the left-hand side show the CHF data versus gap size, while the figures at the right-hand side present CHF ratios versus gap size.

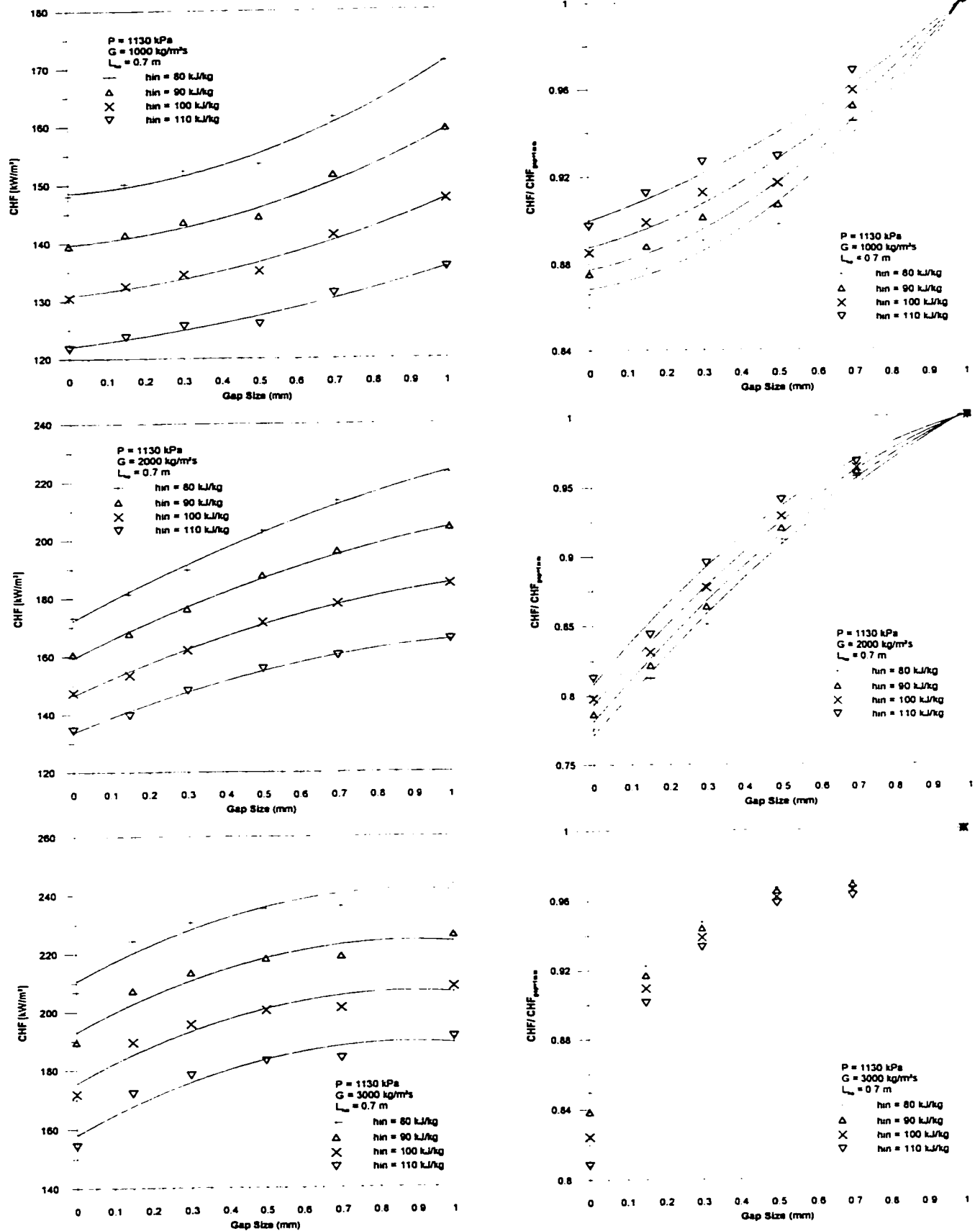


Figure 6.15 Geometry effect on CHF at fixed inlet condition (1130 kPa).

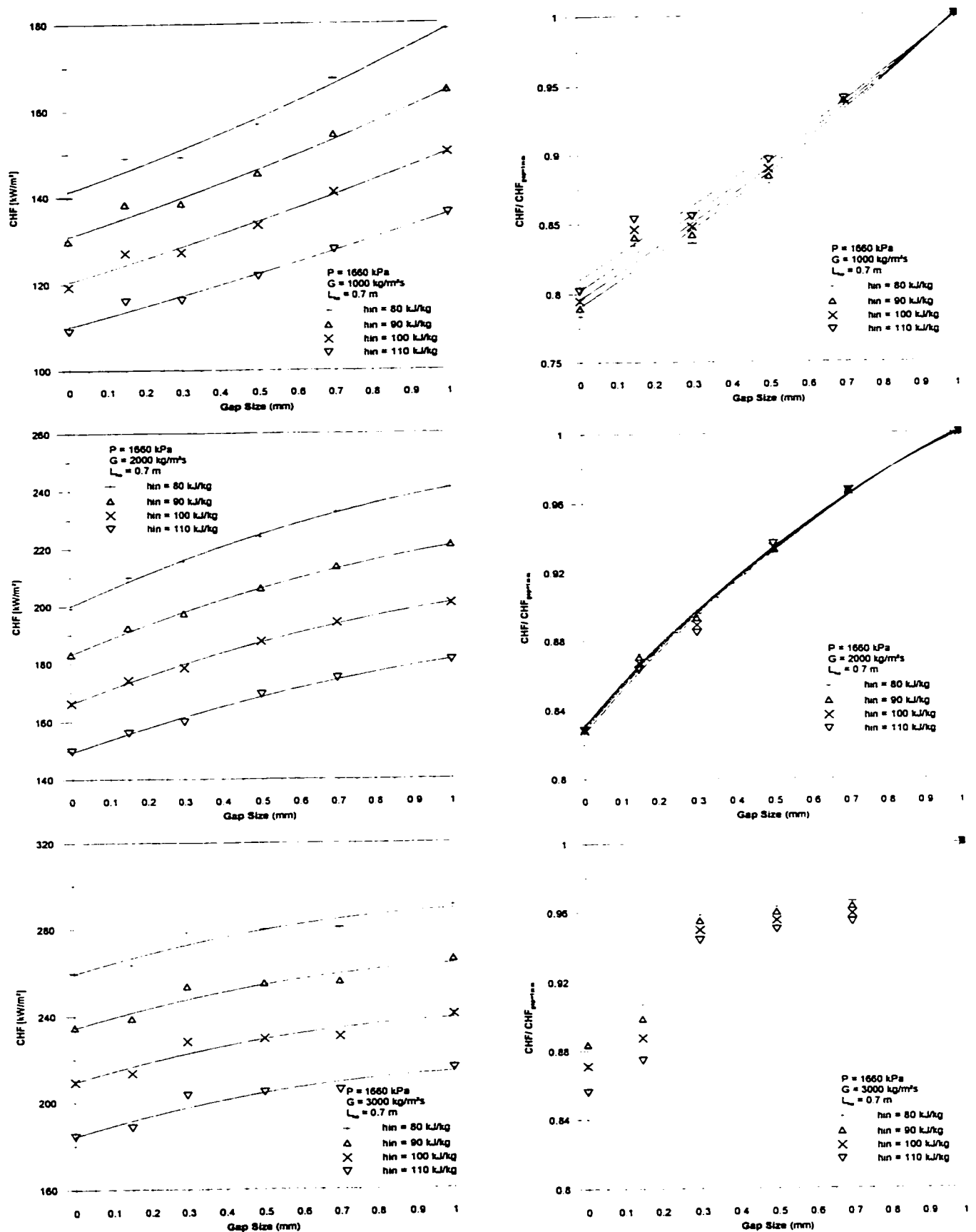


Figure 6.16 Geometry effect on CHF at fixed inlet condition (1660 kPa).

In general, the CHF decreases with decreased gap size. Over the test range, the CHF decreases around 17.2% on average when the gap size is decreased from 1 mm to 0. Table 6.1 and Table 6.2 list some of the detailed data for the CHF ratio.

Table 6.1 $\text{CHF}_{\text{gap}=0}/\text{CHF}_{\text{gap}=1\text{mm}}$ at pressure of 1130 kPa

$\begin{matrix} \text{G} \\ h_{\text{in}} \end{matrix}$	1000 (kg/m ² s)	2000 (kg/m ² s)	3000 (kg/m ² s)	Average
80 (kJ/kg)	0.866	0.776	0.850	0.831
90 (kJ/kg)	0.875	0.786	0.838	0.833
100 (kJ/kg)	0.885	0.798	0.824	0.836
110 (kJ/kg)	0.897	0.813	0.808	0.839
Average	0.881	0.793	0.830	0.835

Table 6.2 $\text{CHF}_{\text{gap}=0}/\text{CHF}_{\text{gap}=1\text{mm}}$ at pressure of 1660 kPa

$\begin{matrix} \text{G} \\ h_{\text{in}} \end{matrix}$	1000 (kg/m ² s)	2000 (kg/m ² s)	3000 (kg/m ² s)	Average
80 (kJ/kg)	0.783	0.828	0.893	0.835
90 (kJ/kg)	0.789	0.828	0.883	0.833
100 (kJ/kg)	0.795	0.828	0.871	0.831
110 (kJ/kg)	0.802	0.829	0.856	0.829
Average	0.792	0.828	0.876	0.832

6.2.1.2 Geometry Effect on CHF at Fixed Mass Flow Rate

Normally, the CHF is analyzed under a fixed mass flux, which eliminates the influence of flow area. In this experimental study, the CHF was measured at fixed mass flow rate also. The purpose of this test is to find the gap effect on CHF for a fixed mass flow. This effect can be viewed as an aggregated effect of flow area and gap size at fixed mass flux condition.

Figure 6.17 shows the geometry effect on CHF at fixed mass flow rate. It is found from the figure that the CHF decreases with decreased gap size. On the average, when the gap size decreases from 1 mm to zero, the CHF decreases by about 8.4%.

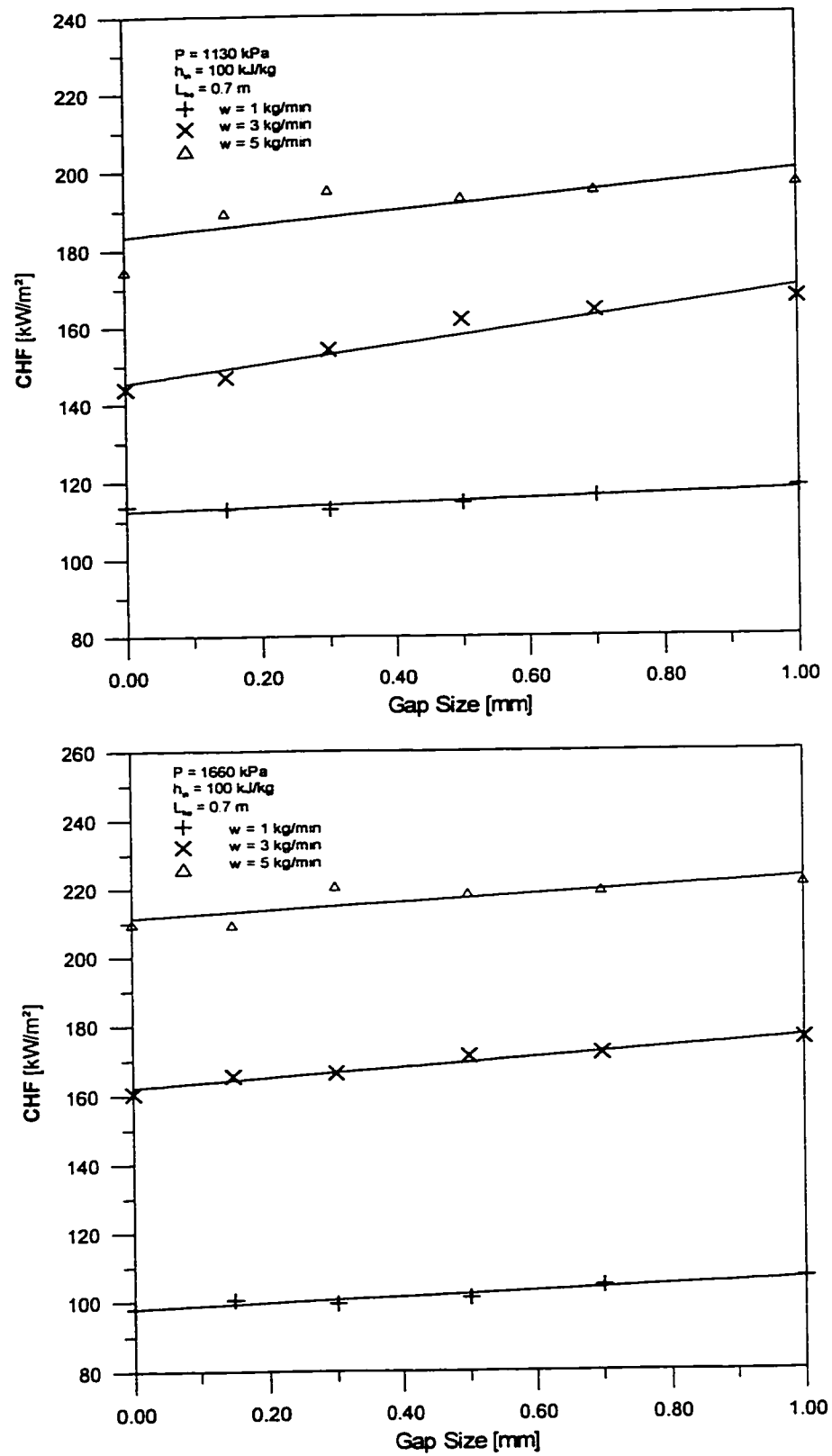


Figure 6.17 Geometry effect on CHF in dumbbell-shaped tube at fixed mass flow rate.

6.2.1.3 Overall Geometry Effect at Fixed Local Condition

Figure 6.18 shows the overall geometry effect on CHF at fixed local condition. In general, the dumbbell-shaped tube CHF decreases with decreased gap size due to the overall geometry effect.

When the gap size decreases from 1 mm to zero, the CHF decreases by up to about 22% depending on flow conditions. On the average, the CHF drops by about 11.5%.

There is an exception for the case of low mass flux and low pressure. This may be because of the limiting quality phenomenon. Due to the limitation of the range of experiment, the overlap of experimental data is small and the qualities of 0.55 and 0.6 were selected to show the geometry effect. These qualities are away from the limiting quality for small gaps but close to the limiting quality for large gaps. The fact that CHF drops at the limiting quality may result in a relatively low CHF for large gap. Thus, the CHF does not decrease much with decreased gap size at the low mass flow rate and pressure as it does at other flow conditions.

The locations of the first CHF are also shown in Figure 6.18. In the figure, G represents the CHF occurs at the gap region first, C represents the CHF occurs at the main channel region first, and GC represents the CHF occurs at the gap region and main channel region simultaneously. The first CHF location will be discussed in section 6.2.3.

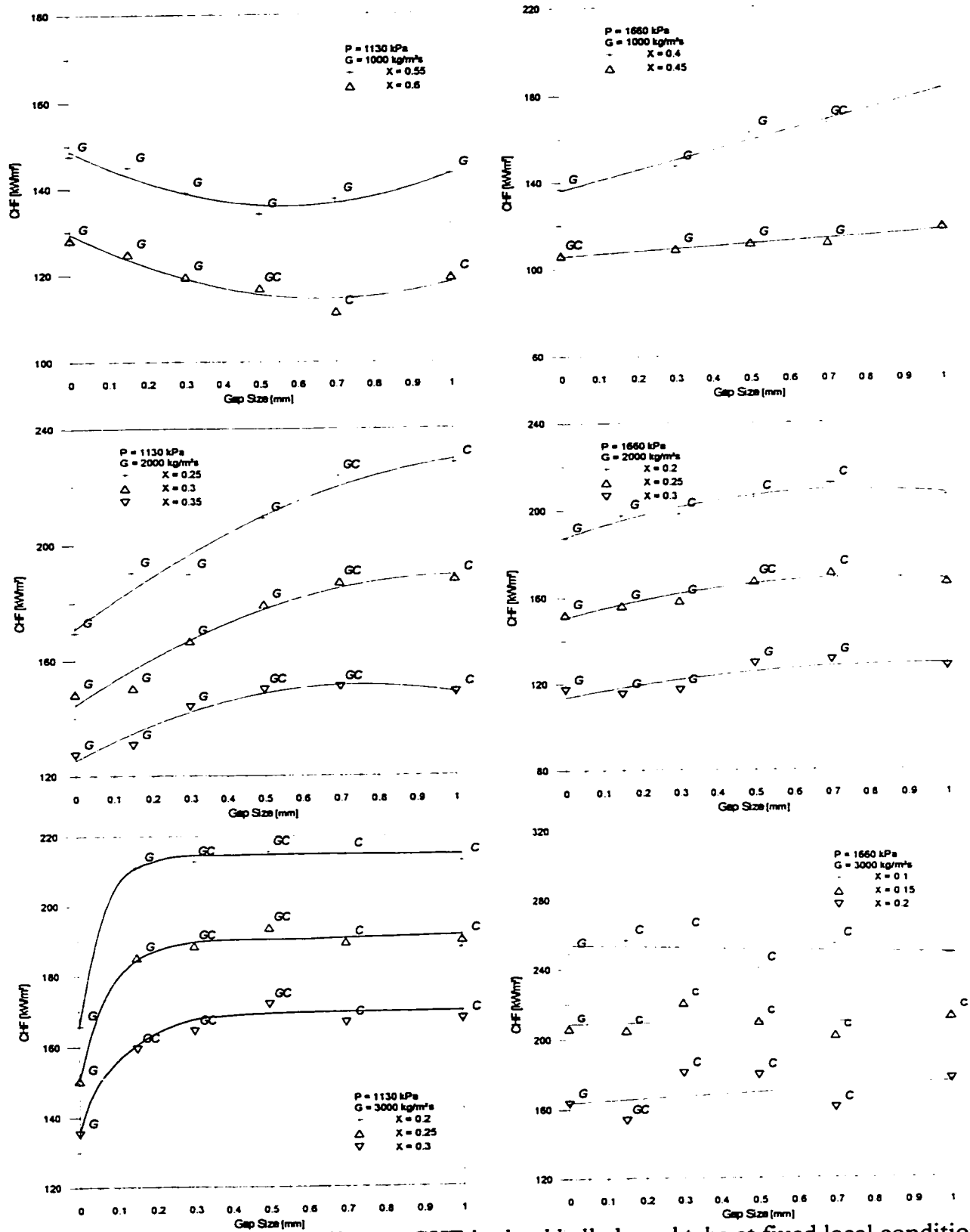


Figure 6.18 Geometry effect on CHF in dumbbell-shaped tube at fixed local condition.

6.2.2 Gap Effect on CHF

6.2.2.1 Approach to Isolate Gap Effect

To find out the gap effect on CHF, a method is needed to isolate gap effect from the overall geometry effect. For the dumbbell-shaped tube, there are three main geometric parameters that control the shape of the tube as follows:

- Gap size
- Curvatures
- Flow area

As described in Chapter 4, when the gap size decreases from 1 mm to zero the change of the curvatures is very small and can be neglected. Thus, the overall geometry effect can be considered as a combination of gap effect and flow area effect.

The problem is that the gap effect and the flow area effect may not be independent but coupled. For the dumbbell-shaped tube, the gap effect mainly comes from the local narrow gap region, while the flow area effect is due to the entire tube. In this study, it is assumed that the gap effect could be isolated from the overall geometry effect by simply deducting the effect of flow area. This assumption needs to be verified in future research.

6.2.2.2 Flow Area Effect

The hydraulic diameter effect accounts for the flow area change. The effect of flow area is thus determined using the results of diameter effect on CHF for a circular tube.

The lookup table (Groeneveld et al., 1996) was used to determine the diameter effect on CHF for fixed local condition. It is one of the most accurate methods to predict CHF in a circular tube. The CHF data of the lookup table are based on water flow in an 8 mm circular tube. To account for the diameter effect on CHF, the lookup table introduced a simple correction equation to account for the diameter effect at a given local condition:

$$\frac{CHF_D}{CHF_{D=8mm}} = \left(\frac{D}{8}\right)^n$$

A value of $-1/2$ for the exponent n in the equation is recommended.

The hydraulic diameter effect on CHF in the dumbbell-shaped tube is then calculated according to the following correlation,

$$\frac{CHF}{CHF_{base}} = \left(\frac{D}{D_{base}} \right)^{-0.5}$$

In this study, the tube with a gap size of 1 mm was used as a base for data comparison. The tube has a hydraulic diameter of 4.30 mm.

To apply the data to this experiment, flow parameters of R-134a are converted into water equivalent flow parameters according to the scaling laws proposed by Tain et al. (1993). Table 6.3 shows the equivalent flow parameters between R134a and water.

Table 6.3 Water equivalent flow parameters

Fluid	R-134a	Water	R-134a	Water
Pressure (kPa)	1130	7000	1660	10000
Mass Flux (kg/m ² s)	1000	1421	1000	1402
	2000	2843	2000	2803
	3000	4264	3000	4205

6.2.2.2.1 Hydraulic Diameter Effect on CHF at Fixed Local Condition

As shown in Figure 6.19, the CHF increases by about 12% as the hydraulic diameter decreases from 4.30 to 3.43 mm (corresponding to a decrease of the gap size from 1 mm to zero).

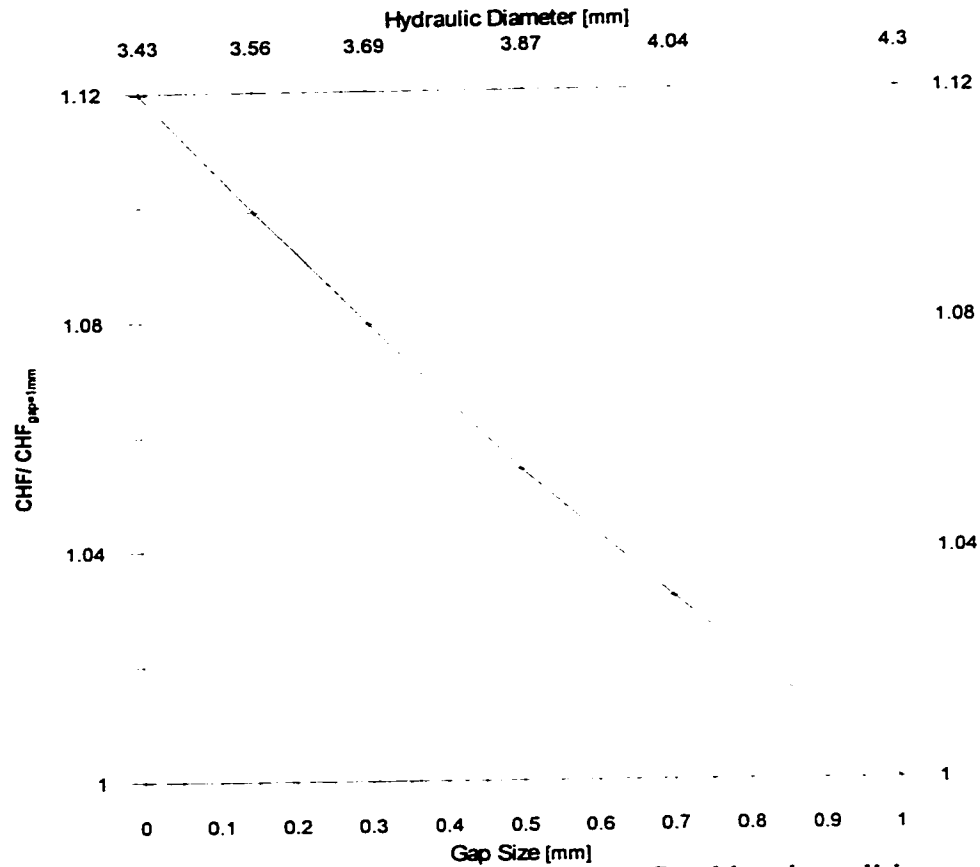


Figure 6.19 Hydraulic diameter effect on CHF at fixed local condition.

6.2.2.2.2 Hydraulic Diameter Effect on CHF at Fixed Inlet Condition

It is noted that the lookup table is based on fixed local condition. To extend the lookup table to fixed inlet condition, a computer program was developed using iteration. Figure 6.20 shows the flow chart for computing the CHF at a given inlet condition using the lookup table.

Table 6.4 and Table 6.5 summarize CHF changes when the diameter is decreased from 4.30 to 3.43 mm (the gap size changes from 1 mm to zero) at various inlet conditions. The diameter effect on CHF is thus obtained numerically for a given inlet condition. On the average over the test range, the CHF decreased by about 12.0% when the hydraulic diameter was decreased from 4.3 mm to 3.43 mm.

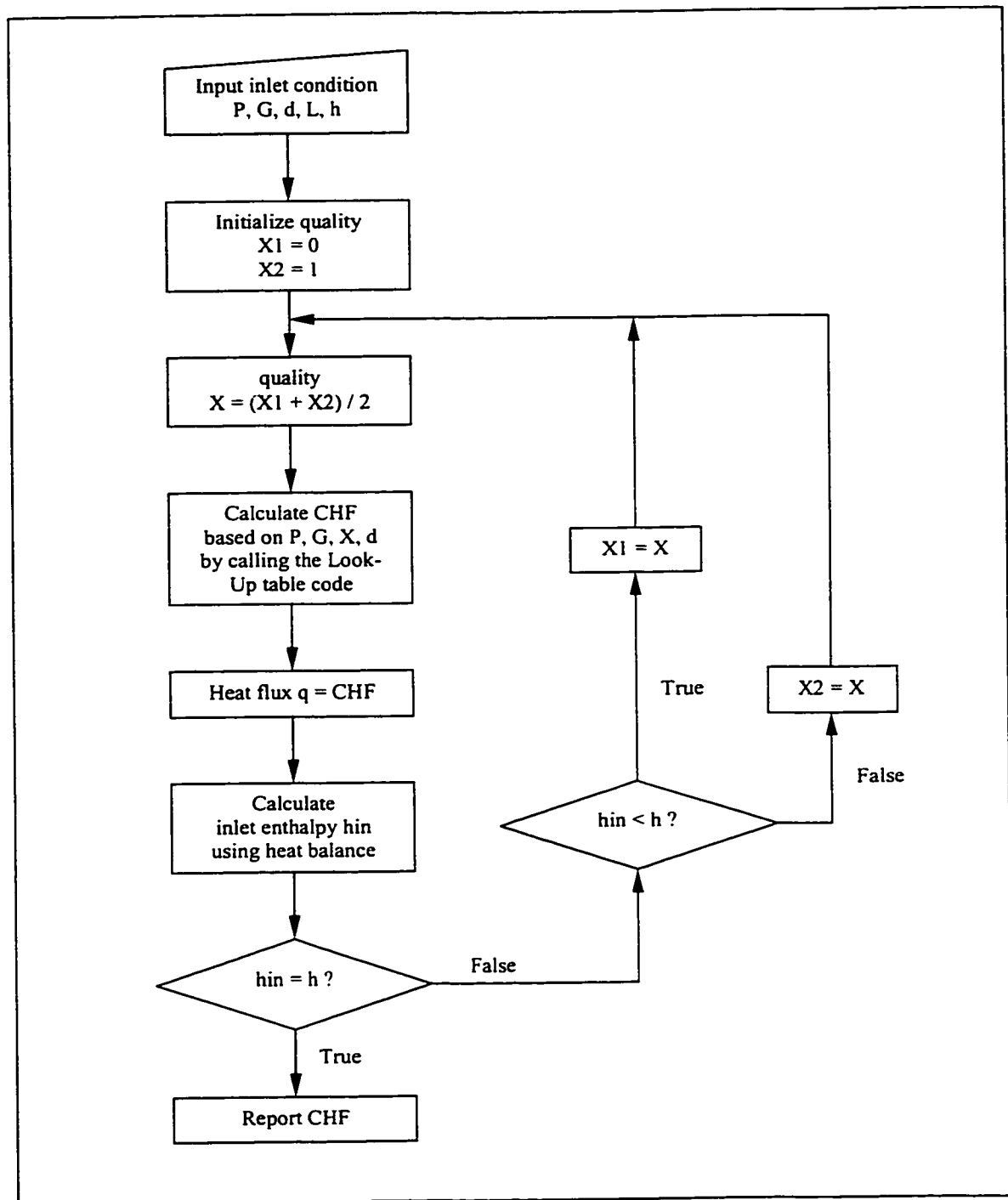


Figure 6.20 Flow chart for computing the CHF at a given inlet condition.

Table 6.4 $\text{CHF}_{d=3.43\text{mm}}/\text{CHF}_{d=4.30\text{mm}}$ at pressure of 1130 kPa

$\begin{matrix} \text{G} \\ h_{\text{in}} \end{matrix}$	1000 (kg/m ² s)	2000 (kg/m ² s)	3000 (kg/m ² s)	Average
80 (kJ/kg)	0.847	0.884	0.982	0.904
90 (kJ/kg)	0.847	0.876	0.941	0.888
100 (kJ/kg)	0.847	0.873	0.919	0.880
110 (kJ/kg)	0.847	0.869	0.910	0.875
Average	0.847	0.876	0.938	0.887

Table 6.5 $\text{CHF}_{d=3.43\text{mm}}/\text{CHF}_{d=4.30\text{mm}}$ at pressure of 1660 kPa

$\begin{matrix} \text{G} \\ h_{\text{in}} \end{matrix}$	1000 (kg/m ² s)	2000 (kg/m ² s)	3000 (kg/m ² s)	Average
80 (kJ/kg)	0.827	0.893	0.870	0.863
90 (kJ/kg)	0.827	0.899	0.880	0.865
100 (kJ/kg)	0.831	0.904	0.895	0.877
110 (kJ/kg)	0.843	0.893	0.921	0.886
Average	0.832	0.897	0.892	0.873

6.2.2.3 Gap Effect

6.2.2.3.1 Gap Effect on CHF at Fixed Inlet Condition

At fixed inlet condition, the CHF decreases with decreased diameter. The pure gap effect is then obtained by the direct deduction of the hydraulic diameter effect from the overall geometry effect. By comparing Table 6.4 and Table 6.5 with Table 6.1 and Table 6.2, it is found that on the average the CHF decreased by about 5.2% as the gap size decreased from 1 mm to zero. This result indicates that the narrow gap size has a small effect on CHF under a fixed inlet condition.

For the range of data tested, annular flow and dryout-type CHF most likely occur inside the dumbbell-shaped test section. Instead of bubble nucleation on the tube wall, the liquid phase evaporates on the liquid film surface and merges to the vapor core. The liquid film depletion results in dryout-type CHF. The gap effect on the dryout-type CHF based on the inlet condition

concept can be considered as a combination of the following effects.

- Effect on evaporation

Under the uniform heating assumption, the evaporation rate distribution along the heating surface is uniform. However, with a decrease in gap size, the shape of the narrow gap region becomes sharper. The same heating surface encapsulates less fluid. The fluid per cubic meter in the narrow gap region absorbs more heat. The intensity of evaporation in the narrow gap region increases with decreased gap size. This tends to enhance the liquid depletion in the narrow gap region and results in premature CHF in the region. This is one of the main reasons why decreasing gap size decreases CHF.

This effect can be illustrated by considering a flow in a uniformly heated circular tube. The mass flow rate G , tube diameter d , and the heat flux q are given. At a time period $\delta\tau$, a fluid unit $\frac{1}{4}\pi d^2 G \delta\tau$ flows over a unit tube surface of δz and absorbs heat $q\pi d \delta z \delta\tau$.

The heat absorbed by unit of fluid is given by

$$\frac{q\pi d \delta z \delta\tau}{\frac{1}{4}\pi d^2 G \delta\tau} \propto \frac{q}{d}$$

For a given G and q , the smaller the tube, the more heat a unit of fluid obtains. In the narrow gap region, the hydraulic diameter is smaller and the heating is more intense.

- Effect on deposition

Because of the dumbbell tube geometry shape, the deposition rate distribution along the dumbbell tube cross section is not uniform. Non-uniform deposition results in a non-uniform liquid film distribution. Decreasing the gap size increases the unevenness of the deposition and thus reduces CHF.

- Effect on entrainment

Gap size may affect velocity and liquid film distribution, which will eventually affect the entrainment. But this gap effect on entrainment may be minor because the entrainment mainly depends on the main flow.

- Non-uniform heating

The narrow gap region of the dumbbell-shaped tube is mainly convex while the main flow channel region is mainly concave. Since the dumbbell-shaped test section was

heated by direct current during the experiments, the narrow gap region has a lower heat flux than the main flow channel region. This compensates the adverse gap effect. Section 7.2.4.1 will discuss the boundary condition in detail.

It is found from Tables 6.1 to 6.4 that the flow parameters have little influence on the gap effect. The results also show that the influence of flow parameters on gap effect is mutually related.

6.2.2.3.2 Gap Effect on CHF at Fixed Local Condition

At fixed local condition, the CHF increases with increased diameter. The gap effect is thus given by

$$\left(\frac{CHF_{gap=1mm} - CHF}{CHF_{gap=1mm}} \right)_{gap\ effect} = \left(\frac{CHF_{gap=1mm} - CHF}{CHF_{gap=1mm}} \right)_{geometry\ effect} - \left[1 - \left(\frac{D}{D_{D=4.30mm}} \right)^{-0.5} \right]_{diameter\ effect}$$

Figure 6.21 shows the gap effect on CHF based on a fixed local condition. It is found that the gap size has a noticeable effect on CHF. In general, the CHF decreases with decreasing gap size.

As the gap size decreases from 1 mm to 0, the CHF decreases by as much as 34% due to pure gap effect. On the average over the test range, the CHF decreases about 23.5% when the gap size decreases from 1 mm to zero. Similar to the discussion in previous section for the fixed inlet condition, at fixed local condition, the gap effect on CHF can be considered as the integral of the gap effects on evaporation, deposition, and entrainment.

It is also found from Figure 6.21 that the gap effect on CHF may vary with flow condition. The local quality has influence on gap effect but the effect is coupled with mass flux. In general, the

CHF ratio $\left(\frac{CHF}{CHF_{gap=1mm}} \right)$ increases with increased mass flux. In other words, the gap effect

$\left(\frac{CHF_{gap=1mm} - CHF}{CHF_{gap=1mm}} \right)$ becomes stronger with decreasing mass flux. There is an exception for the

case of low pressure and low mass flux. As described in the previous section, the influence of limiting quality may lead to such an exception.

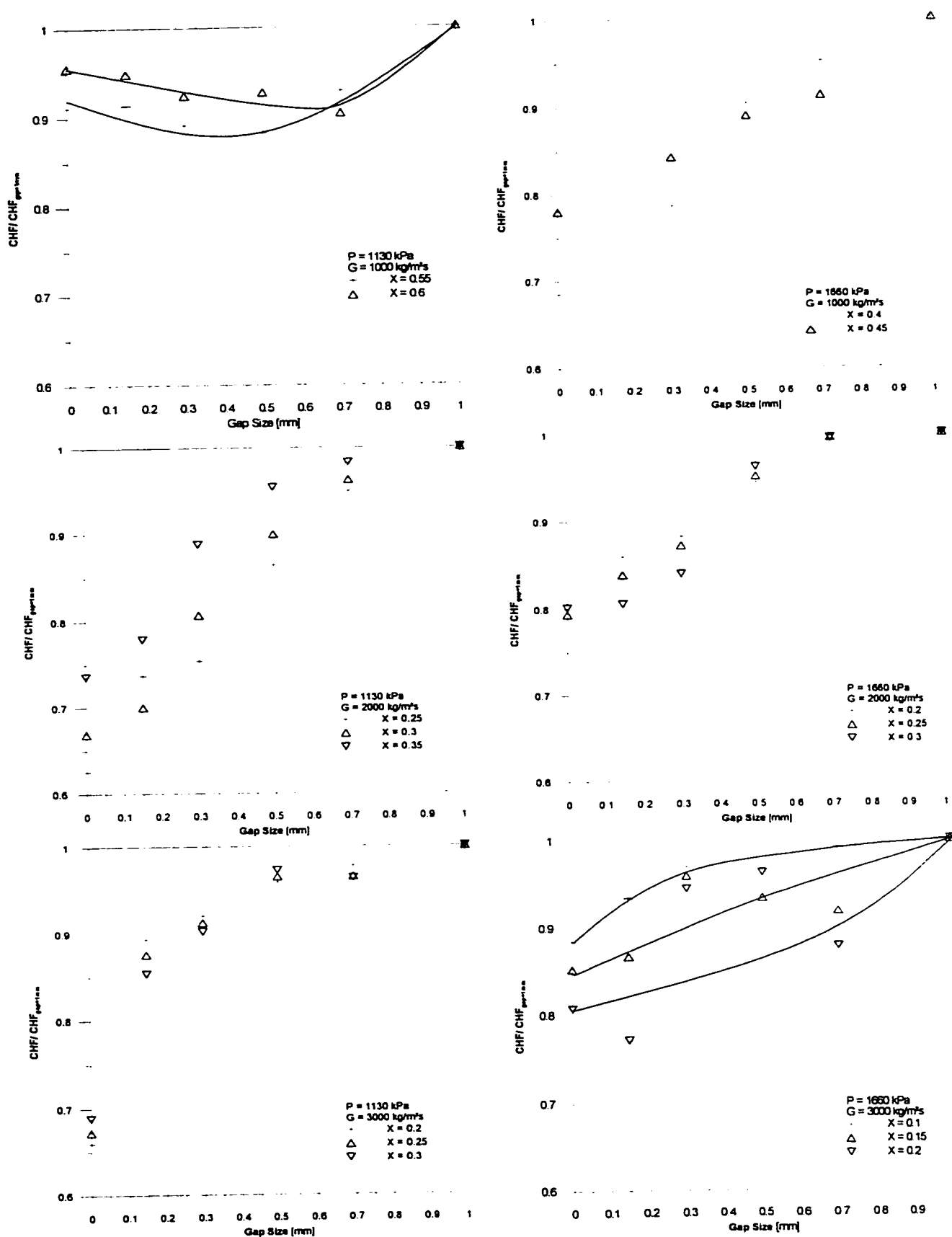


Figure 6.21 Gap effect on CHF in dumbbell-shaped tube at fixed local condition.

It is hard to see the pressure influence on the gap effect from the figure because there is not much overlap of the test data. By comparing the CHF data for the flow at $X = 0.3$ and $G = 2000$ kg/m²s, it is found that the gap effect is stronger at lower pressure.

6.2.3 CHF Location

The location where the first CHF occurs is recorded during the test as shown in appendix III. Figure 6.18 presents the location where the CHF first occurs. It is found from the figure that CHF does not always occur in the narrow gap region first. For the dumbbell-shaped tube with large gap size, CHF may occur first in the main flow region. As described in the previous section, the test section was not uniformly heated. The main flow channel region has a higher heat flux than the gap region. High heat flux in the main flow region compensates the adverse gap effect.

The CHF tends to occur in the narrow gap region where the gap size is decreased. In the case of high mass flux and high quality, the CHF occurs in the main flow channel region. At a higher pressure, the CHF tends to occur in the main flow channel region.

6.3 Comparison Against Circular Tube Data

In this section, the CHF test results of dumbbell-shaped tube are compared directly against the CHF data of a circular tube with the same hydraulic diameter.

6.3.1 Inlet Condition

Figure 6.22 shows the comparison of the experimental CHF data of the dumbbell-shaped tube against the lookup table CHF data using equivalent hydraulic diameter approach and fluid-to-fluid scaling technique. Within the test range, the relative error of the CHF the dumbbell-shaped tube based on the lookup table falls into the range from -20% to +10%.

The result indicates that the lookup table could be used for predicting the CHF in the dumbbell-shaped tube at fixed inlet condition if a correction could be made to take account for the gap effect.

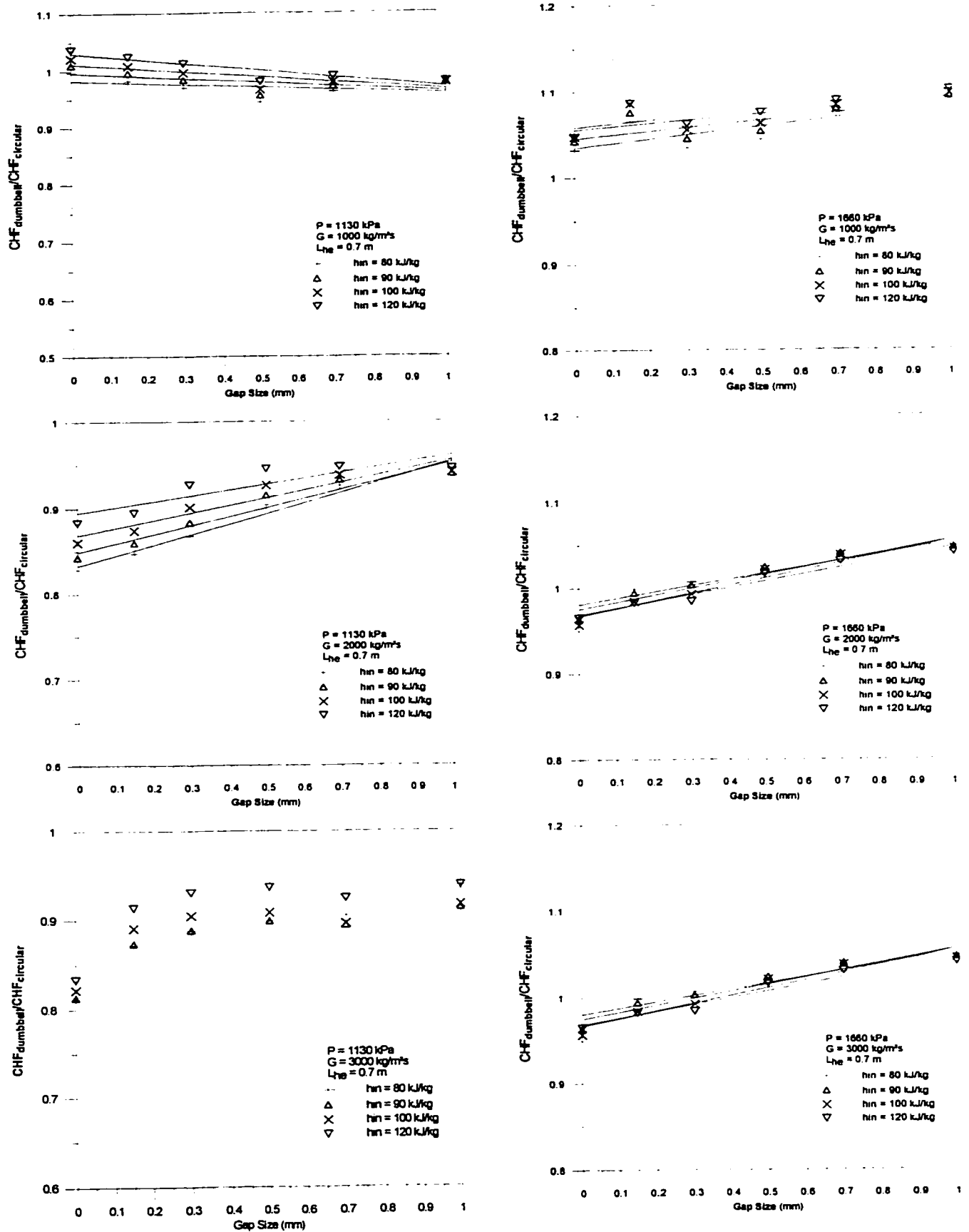


Figure 6.22 Variation of CHF ratio with gap size at fixed inlet condition.

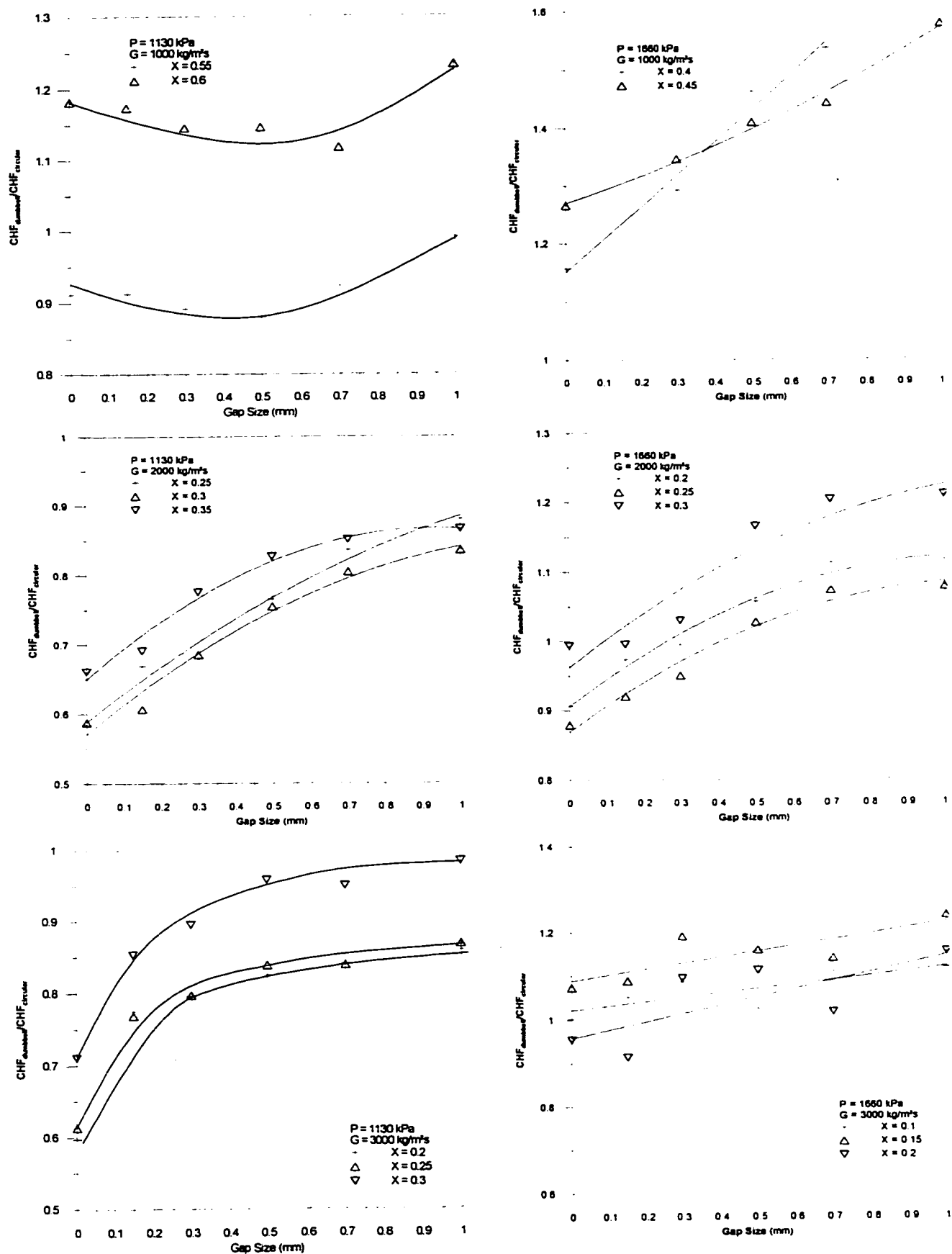


Figure 6.23 Variation of CHF ratio with gap size at fixed local condition.

6.3.2 Local Condition

Figure 6.23 shows the comparison of CHF data between the dumbbell-shaped tube and circular tube. The CHF data for circular tube were predicted using the lookup table. Within the test range, the relative error ranges from -40% to 60% .

This result implies that the hydraulic diameter approach may lead to a large error when it is used to predict the CHF in the dumbbell-shaped tube based on a given local condition. This is because the hydraulic diameter approach does not account for the geometry effect.

Figure 6.23 also shows the gap effect. In general, the CHF ratio based on the circular tube decreases with decreased gap size.

6.4 Flow Visualization

A qualitative adiabatic two-phase flow visualization test has been carried out in this study, which is used to assist the mechanistic analysis and modeling of CHF in the dumbbell-shaped tube. Based on the CHF test results, the local quality is high. For the range of test condition, annular flow most likely occurs in a dumbbell-shaped tube within the test range. Therefore, the flow visualization focused on observing annular flow patterns.

Normally, in annular flow, an annular liquid film forms around the tube wall while the gas phase with the entrained liquid droplets flows in the center of the tube as a vapor core. For the dumbbell-shaped tube, there are two subchannels interconnected via a narrow gap. The flow behavior near the narrow gap region has a great importance for the CHF mechanistic analysis.

It was observed that in the annular flow two gas cores formed at the centers of both subchannels of the dumbbell-shaped tube, which were separated by the liquid in the narrow gap region. The flow looks like two independent annular flows in parallel. No bubble or gas was found in the narrow gap region.

This observation suggests that in annular flow no gas or bubble can stagnate around the narrow gap region. Therefore, only the dryout-type CHF should occur in the dumbbell-shaped tube in annular flow. It can be assumed that under heating condition the evaporated vapor near the narrow gap region will move and merge to one of the two vapor cores.

6.5 Summary

The trend of flow parameter (pressure, mass flux/mass flow rate, quality, and inlet enthalpy/subcooling) on CHF in the dumbbell-shaped tube is the same as that in the circular tubes. No severe adverse effect was found in the dumbbell-shaped tube under the test range of this study.

The experimental results in summarized in Table 6.6. The results shown in the table is the average of the CHF data tested.

Table 6.6 Summary of experimental results

	Overall Geometry Effect	Pure Gap Effect
At fixed inlet condition	17.2%	5.2%
At fixed local condition	11.5%	23.5%

At a fixed inlet condition, the CHF decreases with decreasing gap size. However, the narrow gap size has only a small effect on CHF. As gap size decreases from 1 mm to zero CHF drops by about 17.2% on the average. If deducting the flow area effect, the CHF drop reduces to about 5.2% due to the pure gap effect.

At a fixed local condition, the gap size has a noticeable effect on CHF. In general, the CHF decreases with decreasing gap size. As the gap size decreases from 1 mm to 0, the CHF decreases by 11.5% depending on flow conditions on the average, considering the overall geometry effect. If one takes the flow area changes into account, the CHF drops about 23.5% considering the pure gap effect. The gap effect is influenced by flow conditions.

Flow visualization test results show that for annular flows, two vapor cores form, one at the center of each subchannel. The liquid phase in the narrow gap region separates these two gas cores, and no bubble stagnation occurs in the narrow gap region.

Chapter 7

MODELING RESULTS AND ANALYSIS

This chapter describes the assessment of the partition model. Section 7.1 presents the results of the partition model for a circular tube as well as the comparison of predicted CHF with data obtained from other sources. Sections 7.2 and 7.3 describe the detailed assessment of the model against data from dumbbell-shaped tube experiments. Both the general trend and the gap effect are presented and discussed. Section 7.2 is dedicated to uniform heating while section 7.3 discusses non-uniform heating. The detailed partition model results for the dumbbell-shaped tube are tabulated in Appendix IV and Appendix V. The modeling study is summarized in section 7.4.

7.1 Partition Model Application to Circular Tube

The partition model was initiated for predicting the CHF in the dumbbell-shaped tube. However, as indicated in Chapter 5, the partition model and the software package (CHFA++) were designed in such a way that they are not restricted to the dumbbell-shaped channel. The software can be extended to simple geometry (circular tube) as well as other complex geometries. The main purpose of applying the partition model to a circular tube is to verify the kernel of the model and the software package. The kernel describes the conservation equations, deposition/entrainment model, interfacial shear stress and triangular relationship. It is independent to flow channel geometry shape and fluid.

This section is divided into two subsections. The first subsection describes the verification of the kernel in-depth, which not only checks the final results (quality and CHF) but also examines the details of flow and heat transfer behavior. A steam-water flow case was used. The other subsection describes tests of the partition model works over a wide range of flow conditions. The CHF results of R-134a flows were examined.

7.1.1 Steam-Water Flow Case

The partition model is based on the Whalley model (1974). Whalley used a steam-water flow case to present his CHF model. In this case, saturated water at 30 bar enters a 15-mm-diameter circular tube with a mass flux of 1000 kg/m²s and dryout-type CHF occurs at the exit of the tube. Vaitekunas (1995) used this flow case to evaluate annular flow models. Due to the availability of detailed data, the case is also used in this study for an in-depth verification of the partition model.

7.1.1.1 Whalley's Annular Flow Model Results

As a first phase in the model development, Whalley's annular flow CHF model was re-implemented using the state-of-the-art Object-Oriented software design methodology and programming language. The CHF prediction data for the water-steam flow test case is shown in Table 7.1. The results from other resources (see Vaitekunas, 1995) are also listed in the table for comparison.

Table 7.1 Comparison of CHF prediction data at fixed inlet condition

Prediction Method		X	CHF [kW/m ²]	Deviation %
Whalley's model	This Study	0.628	847.1	-12.4%
	Whalley [1974]	0.55	766	-20.8%
	Vaitekunas [1995]	0.624	841.4	-13.0%
Empirical correlation	Bowring (1972)	0.615	828.7	-14.3
	Katto (1984)	0.658	886.5	-8.4%
Lookup table	Groeneveld (1995)	0.718	967.3	-

where, the relative deviation based on lookup table is given by

$$Deviation\% = \frac{CHF - CHF_{Lookup\ table}}{CHF_{Lookup\ table}}$$

It is noted that Whalley's model results vary with different implementations. The basic annular flow assumptions are identical. However, the detailed approach for deposition and entrainment, method for solving differential equations, accuracy control, etc., all of these factors can lead to

the deviation of the final results. The result obtained from this study is almost the same as Vaitekunas' result ($< 1\%$).

Figures 7.1 to 7.5 present the details of the variations of the mass flux of liquid film, liquid film thickness, interfacial shear stress, deposition rate, and entrainment rate. Note that with the fluid flowing forward the quality increases. The X axis of the figure is represented using the quality instead of tube length. The data show a similar trend as Whalley results (1974) and the values are within the same order of magnitude.

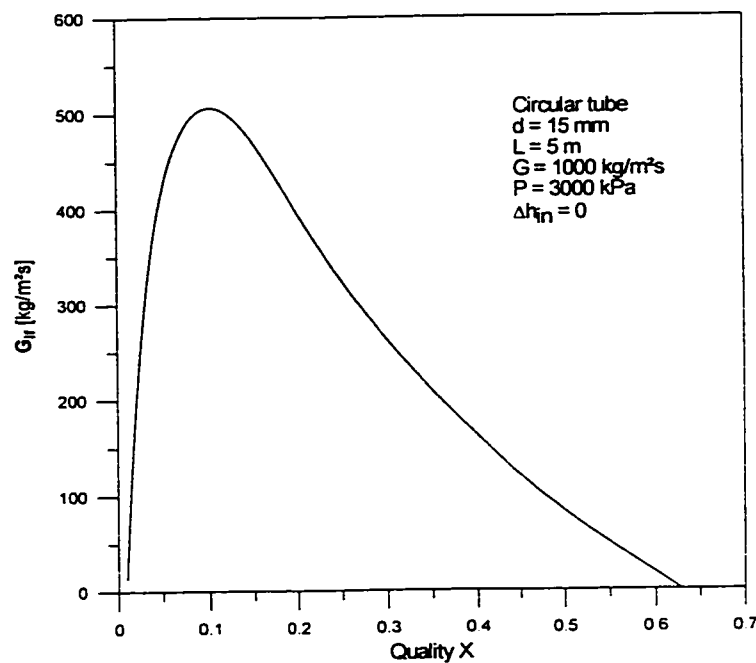


Figure 7.1 Variation of liquid film flow rate along the tube axis.

As described in Chapter 5, the onset of annular flow (ONAF) was assumed to start at a thermodynamic quality of 1% and 99% of the liquid phase is entrained. From Figures 7.1 and 7.2, it is found that the liquid film mass rate increases rapidly after the ONAF. This implies that the assumption of ONAF may under-estimate liquid film mass rate and liquid film thickness at ONAF. However, the assumption has very little effect on the final CHF result. According to mass conservation, a lower mass flow rate of liquid film leads to a higher droplet concentration and thus a higher deposition rate. It is noted that both the liquid film thickness and flow rate

initially increase very rapidly from the ONAF point due to the higher deposition rate. In a very short section, the simplified assumption for onset of annular flow condition is corrected automatically by the internal balance mechanism.

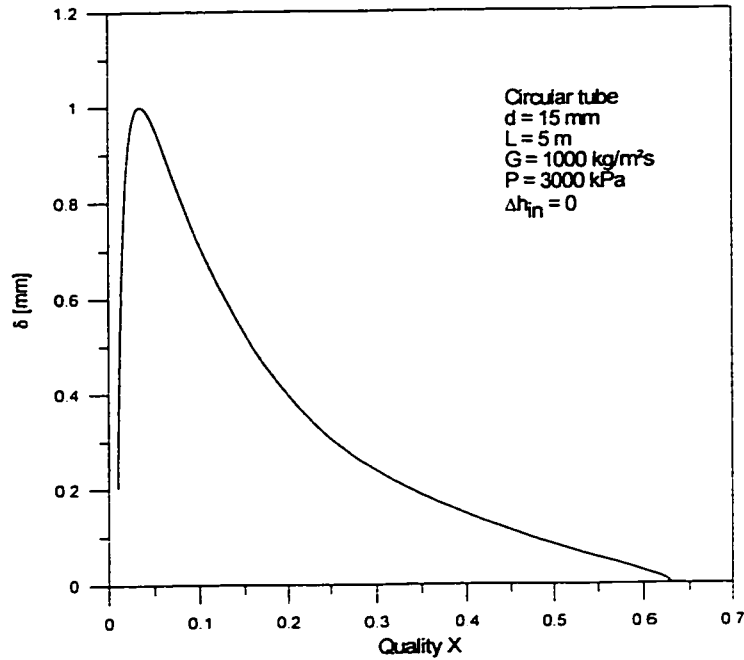


Figure 7.2 Variation of liquid film thickness along the tube axis.

For evaporation rate, deposition rate and entrainment rate, the newly implemented model generates the similar results as Whalley model (1974). The evaporation rate is given by:

$$\frac{q''}{h_{fg}} = \frac{847.1 [kW / m^2]}{1796.6 [kJ / kg]} = 0.47 [kg / m^2s]$$

It is found from Figure 7.4 and Figure 7.5 that both the deposition rate and entrainment rate have the same order of magnitude (10^{-1}) as the evaporation rate for the flow in the tube except the deposition right after ONAF and the entrainment near the exit.

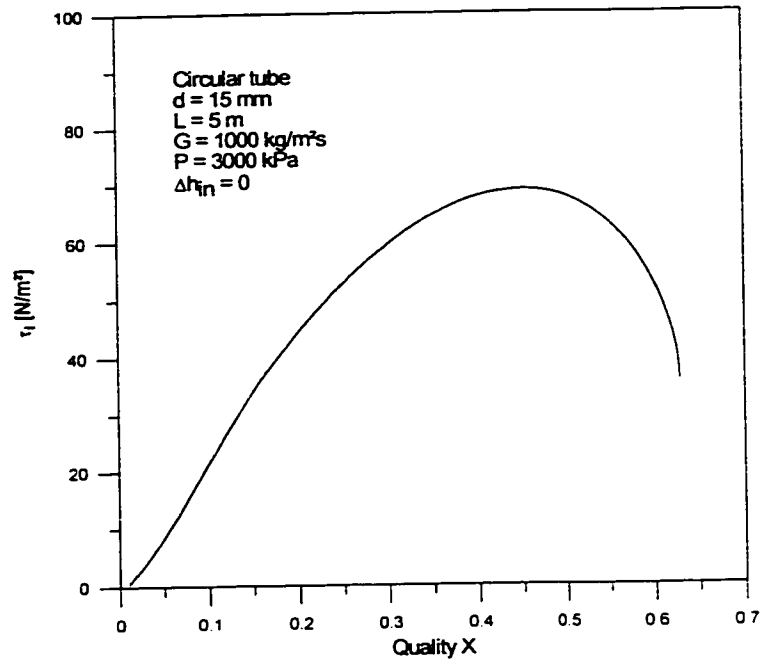


Figure 7.3 Variation of interfacial shear stress along the tube axis.

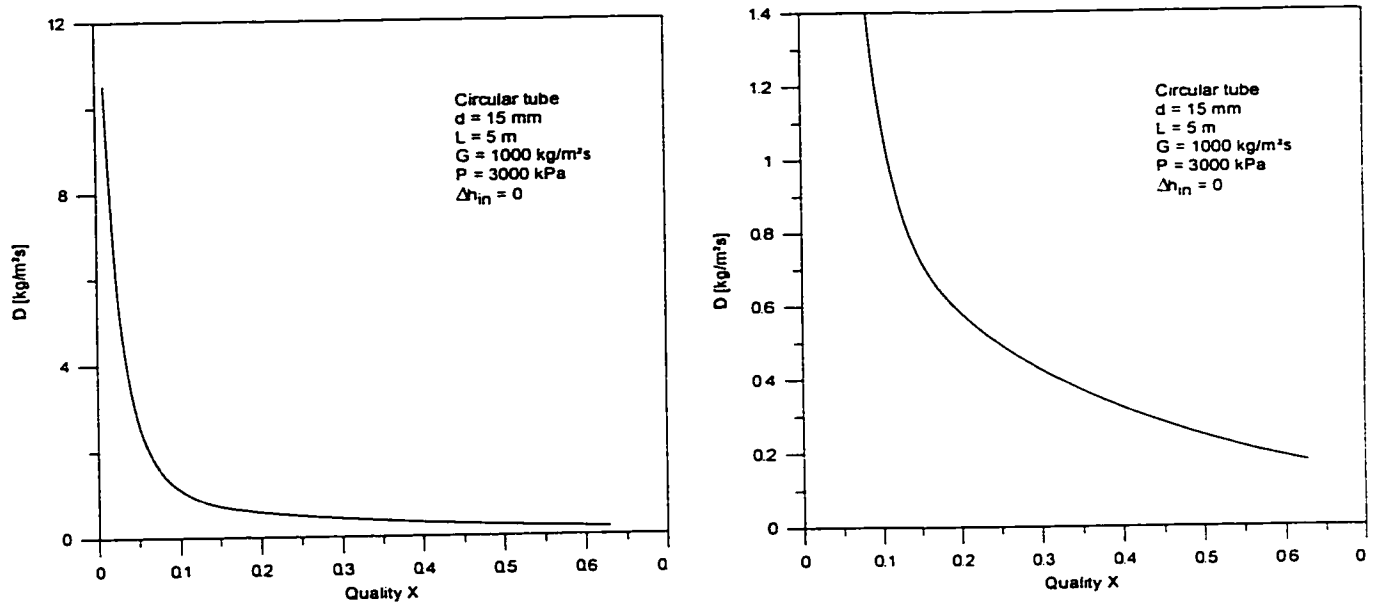


Figure 7.4 Variation of deposition rate along the tube axis.

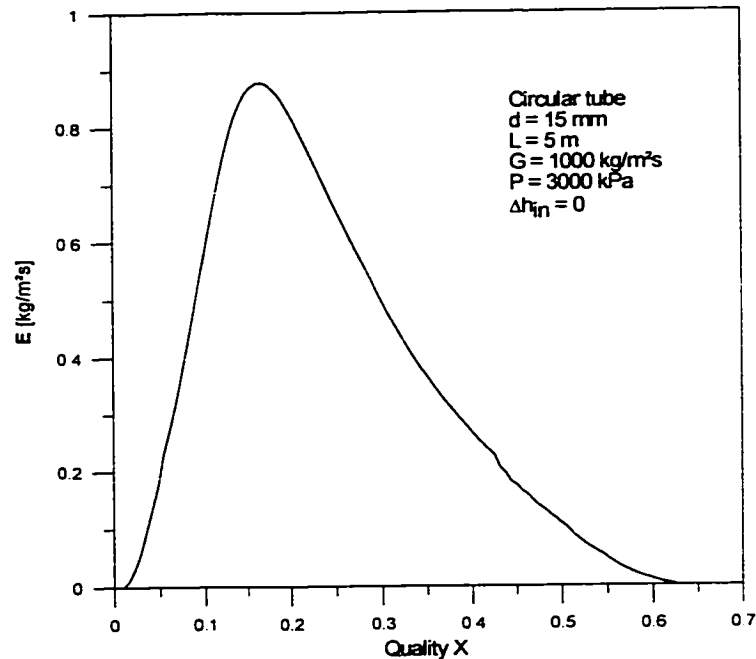


Figure 7.5 Variation of entrainment rate along the tube axis.

7.1.1.2 Partition Model Results

The partition model was applied to circular tubes for the purpose of model verification. As described in Chapter 5, a circular geometry can be partitioned arbitrarily into two sectors. For simplicity, the circular tube described in the previous section was divided into two pseudo-channels, $\frac{1}{4}$ and $\frac{3}{4}$ each, and the partition model was then used to predict CHF.

When applied to the circular tube, the partition model produces identical results as Whalley's model (new implementation). The partition model is derived from Whalley's model to handle the complex geometry. For the simple geometry of a circular tube, no matter how a circular tube is partitioned, the values of all the parameters those describe the flow and heat transfer in the two sectors or the two pseudo-channels are identical to that for the physical flow channel. Deposition factors for both sectors are the same as a circular tube because of the radial symmetry. Therefore, Figures 7.1 to 7.5 also represent the results of the partition model for the water-steam flow.

In conclusion, the partition model predicts the CHF in circular tube with the same results as Whalley's model.

7.1.2 R-134a Flow Cases

The previous sub-section verified the partition model in depth by examining a steam-water flow case. This sub-section assesses the model in wide range of flow conditions using a set of R-134a flows. Piro (1998) conducted a systematic test of CHF in circular tube with R-134a using the same loop as this study (refers to Chapter 4 for detailed description of the loop). Piro's R-134a CHF test data comprised the main set of data used to verify the partition model. The reasons to use these data are as follows:

- Reliability

Piro compared his test results with the lookup table based on the fluid scaling techniques. The test data and lookup table results match very well.

- Similarity

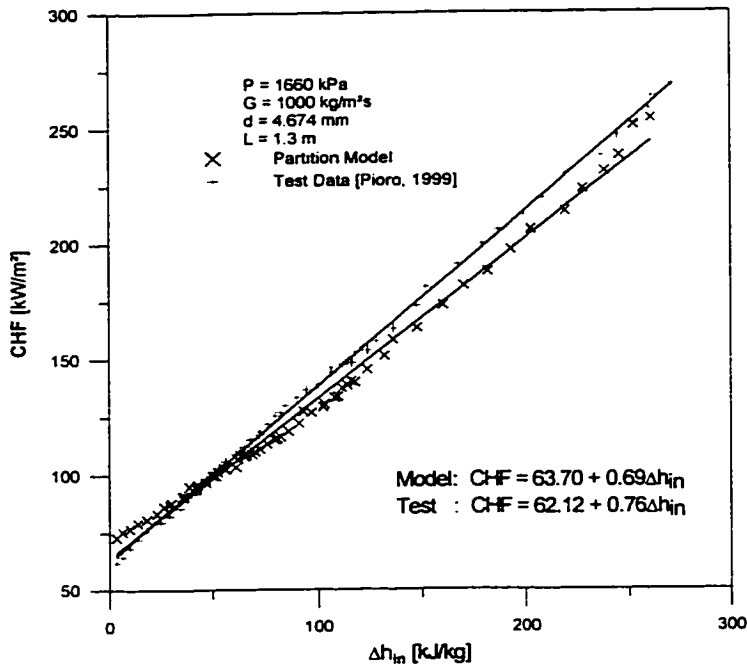
The data were obtained from the same test loop with the same working fluid and the same measurement equipment.

The predicted CHF results for the circular tube are tabulated in Appendix VI. Piro's CHF data are also listed for the purpose of comparison.

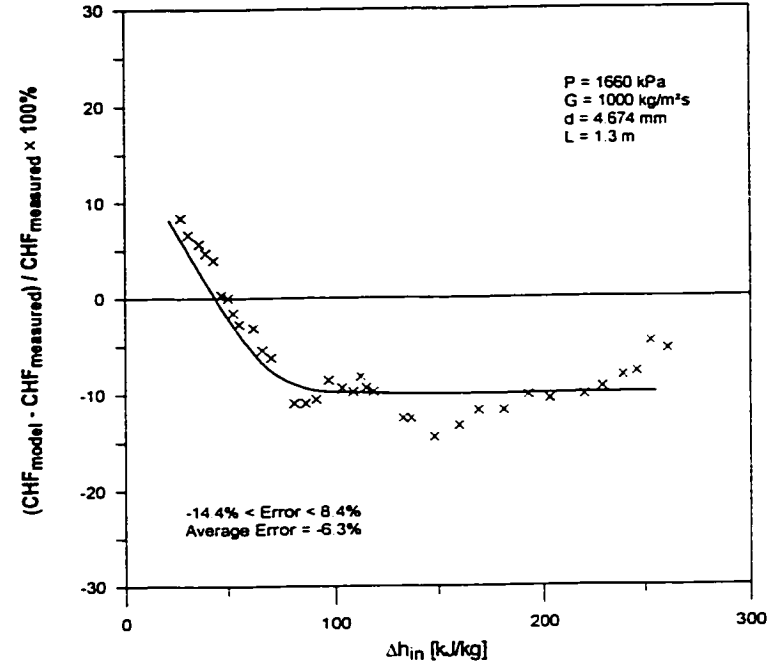
7.1.2.1 General Trend

It is noted that the partition model results are identical to Whalley's model when applied to a circular tube. The predicted CHF trend against inlet subcooling is shown in Figures 7.6 to 7.8. The test data (Piro, 1999) are also presented in the figures for the purpose of data comparison. The correlation for the CHF and inlet subcooling is given in the figures also.

The figures show that the partition model predicts the correct trend in CHF. Generally speaking, the partition model predicts the CHF in a circular tube for R-134a flow with good accuracy. For most flow conditions, the relative error is within a range of $\pm 10\%$. The worst case occurs at high mass flow and very low inlet subcooling, but the relative error is still less than 30%.

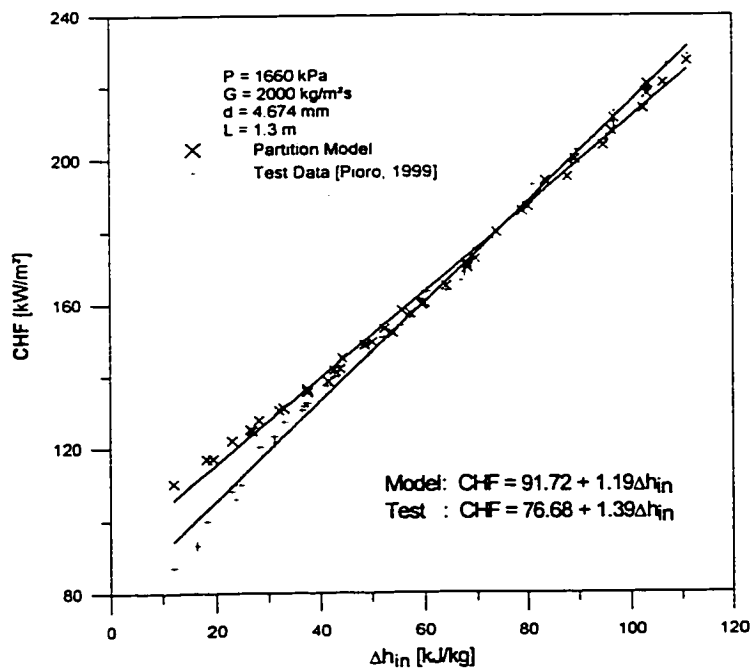


(a)

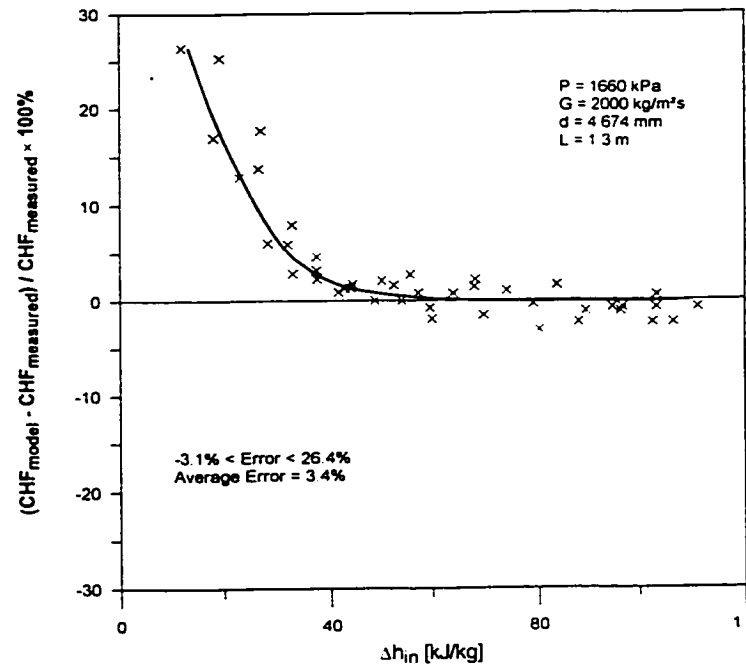


(b)

Figure 7.6 Predicted CHF for circular tube ($G = 1000 \text{ kg/m}^2\text{s}$).



(a)



(b)

Figure 7.7 Predicted CHF for circular tube ($G = 2000 \text{ kg/m}^2\text{s}$).

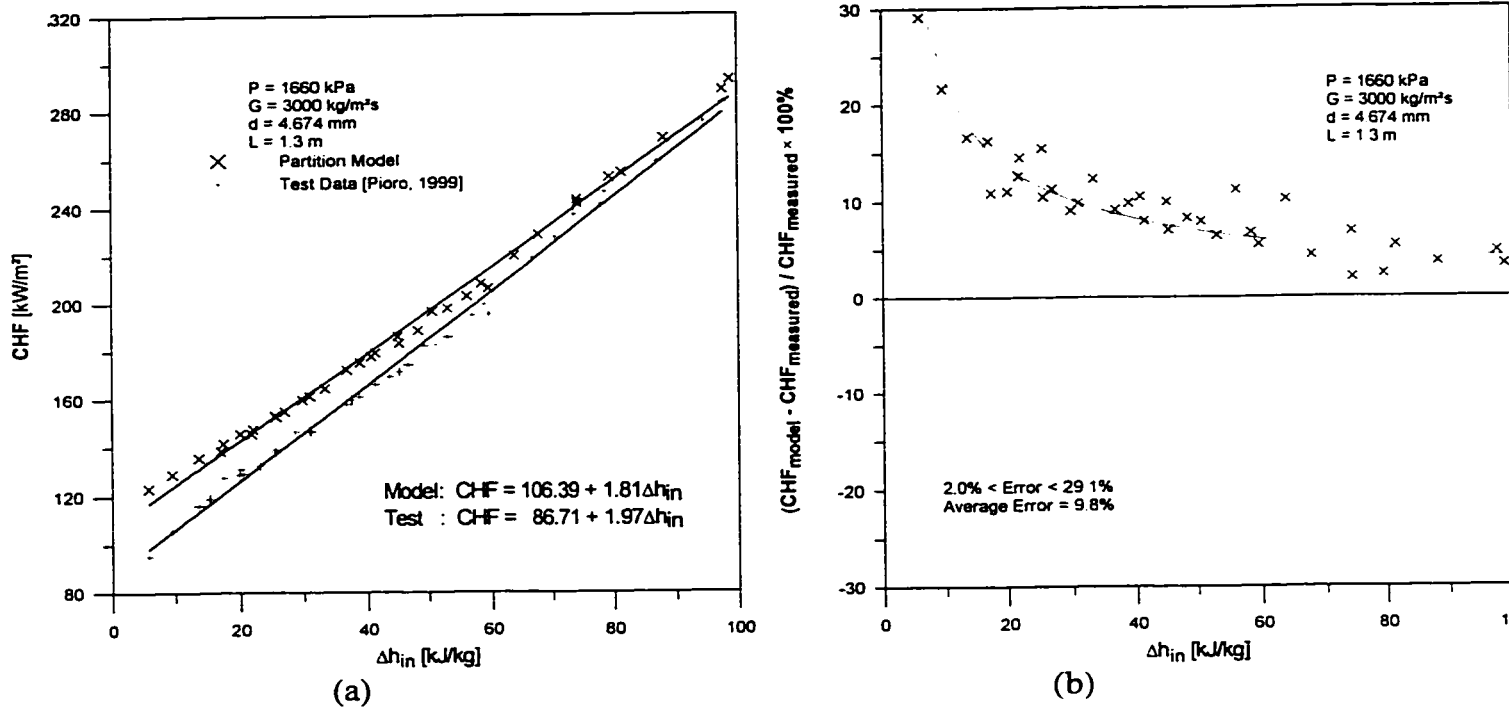


Figure 7.8 Predicted CHF for circular tube ($G = 3000 \text{ kg/m}^2\text{s}$).

7.1.2.2 Mass Flux Influence on Model Accuracy

From Figures 7.6 to 7.8, it is found that the accuracy of the model varies with the flow conditions. In general, at low flow rate ($1000 \text{ kg/m}^2\text{s}$), the model under-predicts the CHF by about -10%. At moderate flow rate ($2000 \text{ kg/m}^2\text{s}$), the partition model predicts the CHF very accurately (within $\pm 5\%$). At high flow rate ($3000 \text{ kg/m}^2\text{s}$), the model over-predicts the CHF by about 10% for most of the data.

The variation of accuracy with mass flux results from the deposition and entrainment models used. The results indicate that the model may under-predict the effect of mass flux on the entrainment rate. At fixed inlet condition, a higher mass flux requires more energy to dryout and thus leads to a higher CHF. This is the overall effect of the mass flux. However, there is an opposite effect underneath. A higher mass flux will result in a higher entrainment rate, which in turn will lower the CHF. However, it is not enough to offset the increase in CHF due to the increase in mass flux. The model results imply that the predicted entrainment rate may not

increase enough with the increase of the mass flux.

On the other hand, the entrainment rate for R-134a flow is a relatively small quantity if compared to the rates of evaporation and deposition. The entrainment does not dominate the CHF results. Since the predicted entrainment rate is small, the negative effect of mass flow rate on CHF is not strong. Therefore, for the higher mass flow rate, the model CHF value is a little higher than the test data due to low entrainment. For the lower mass flow rate, the model CHF value is a little lower than the test data, as shown in Figures 7.6 to 7.8. Overall, Whalley's deposition and entrainment models work well for R-134a flow.

It is noted that in Whalley's model the mass transfer coefficient is a function of surface tension. In reality, the mass transfer coefficient may vary with flow conditions. Vaitekunas (2000) evaluated four entrainment and deposition models, including Whalley's model. Whalley's model predicts CHF with good accuracy in overall, and is the most accurate model for certain flow conditions. The partition model could be enhanced with further research on deposition and entrainment models.

7.1.2.3 Inlet Subcooling Influence on Model Accuracy

It is also found from Figures 7.6 to 7.8 that the partition model predicts a relatively higher CHF value for small inlet subcooling (high inlet enthalpy). When the inlet subcooling is larger than a certain value, the relative error of the model becomes very stable. This transition seems to occur at a CHF value around 125 kW/m² for all three mass flow rates. In this case, the evaporation rate has a value of 0.9 kg/m²s.

As described in Chapter 5, the CHF is controlled by three terms: deposition rate, entrainment rate, and evaporation rate. For annular flow at fixed inlet condition, the higher the inlet subcooling, the more energy is required to reach CHF. In other words, the evaporation rate is low for small inlet subcooling. Based on the deposition model assumption, the deposition will have a relatively stronger effect for the lower evaporation case. According to the analysis of the order of magnitude, the evaporation rate is of the same order of magnitude as the deposition rate

at low inlet subcooling conditions. Therefore, the model predicts a relative higher CHF for lower inlet subcooling conditions due to a stronger deposition effect.

If the heat flux exceeds a certain value, the evaporation becomes the main term among the three terms in the control equation (evaporation, deposition and entrainment). The value of the deposition rate is relatively low and the effect of deposition is relatively small. Thus, the accuracy of the prediction model becomes stable at high subcooling conditions.

7.1.2.4 Variations of Flow Parameters

This subsection presents the order of magnitude for each related parameter. The following case of R-134a flow is analyzed in detail:

$$P = 1660 \text{ kPa}, G = 1000 \text{ kg/m}^2\text{s}, \Delta h_{\text{in}} = 3.5 \text{ kJ/kg}$$

In this case, the evaporation rate is:

$$\frac{q''}{h_{fg}} = \frac{66.1 [kW / m^2]}{140.1 [kJ / kg]} = 0.47 [kg / m^2s]$$

For simplification, the water flow case described in section 7.1.1 is used as a reference to illustrate the order of magnitude of the R-134a flow. It is noted that the steam-water flow is not thermodynamically equivalent to the R-134a flow. For the two cases, the evaporation rate is almost the same.

For R-134a at a pressure of 1660 kPa, the surface tension is about $3.89 \times 10^{-3} \text{ N/m}$. For water at 3000 kPa, the surface tension is about $3.01 \times 10^{-2} \text{ N/m}$. This means that the surface tension of R-134a flow case is about one order of magnitude smaller than that of water flow case. According to Whalley's deposition and entrainment models, the mass transfer coefficient, k , is a single valued function of surface tension. The mass transfer coefficient of R-134a is about one order smaller than that of water.

The mass transfer coefficient affects the deposition and entrainment rates. At the onset of annular flow, it is assumed that 99% of liquid is entrained into the vapor core. As described in Chapter 5, the deposition rate D and the entrainment rate E are given by,

$$D = k C$$

$$E = k C_E$$

where k is the mass transfer coefficient, and C_E is the equilibrium concentration of entrained droplets.

The initial deposition rate for the R-134a flow ($2.2 \text{ kg/m}^2\text{s}$) is about one order smaller than that of the water flow ($10.5 \text{ kg/m}^2\text{s}$). As described above, the mass transfer coefficient of R-134a flow is about one order smaller too. This implies that the initial concentrations (C) for the two flows are of same order. With the fluid flows upward, a lower deposition rate leads to a higher value of the concentration of droplets. Thus, the overall deposition rate for R-134a has the same order as that for water.

Figure 7.9 shows the variation of deposition rate along the tube at the critical heat flux for R-134a flow. Comparing the figure with Figure 7.4 for water, it is found that deposition rates for both fluids are very close although the initial values are of one order difference.

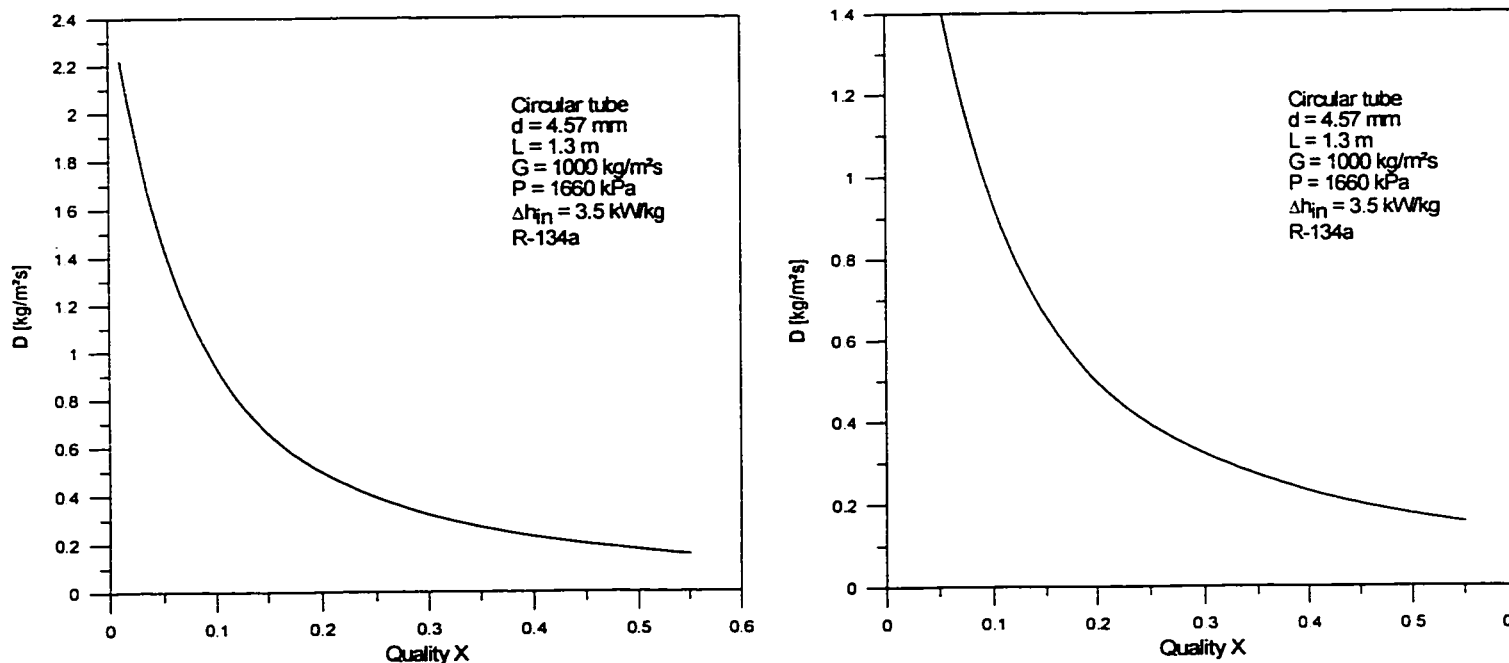


Figure 7.9 Variation of deposition rate along the tube axis (R-134a).

Unlike the deposition rate, the entrainment rate is determined by the equilibrium concentration of

entrained droplets, C_E , which is a single value function of the group $(\tau_i \delta / \sigma)$. The equilibrium concentration of the R-134a flow is of same order of magnitude as that for water. Therefore, as shown in Figure 7.10, the entrainment rate of the R-134a flow is much smaller than that of water due to the small mass transfer coefficient of the R-134a flow. The results imply that the entrainment effect in R-134a flow is not as important as that in water flow.

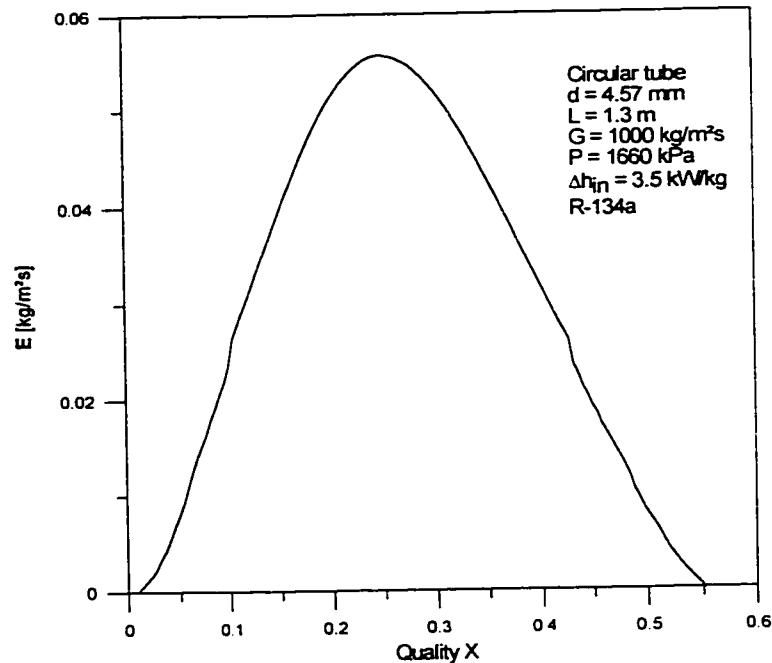


Figure 7.10 Variation of entrainment rate along the tube axis (R-134a).

In summary, for the R-134a flow, the deposition rate has the same order of magnitude as the evaporation rate (10^{-1}) while the entrainment rate is one order smaller.

Because of the lower initial deposition rate, the increase of film thickness and liquid film flow rate for R-134a flow is not as rapid as that for water, as shown in Figure 7.11 and Figure 7.12. Also, because of lower deposition rate, the liquid film thickness of R-134a flow is much smaller than that of water (maximum of 0.16 mm for R-134a vs. 1.00 mm for water).

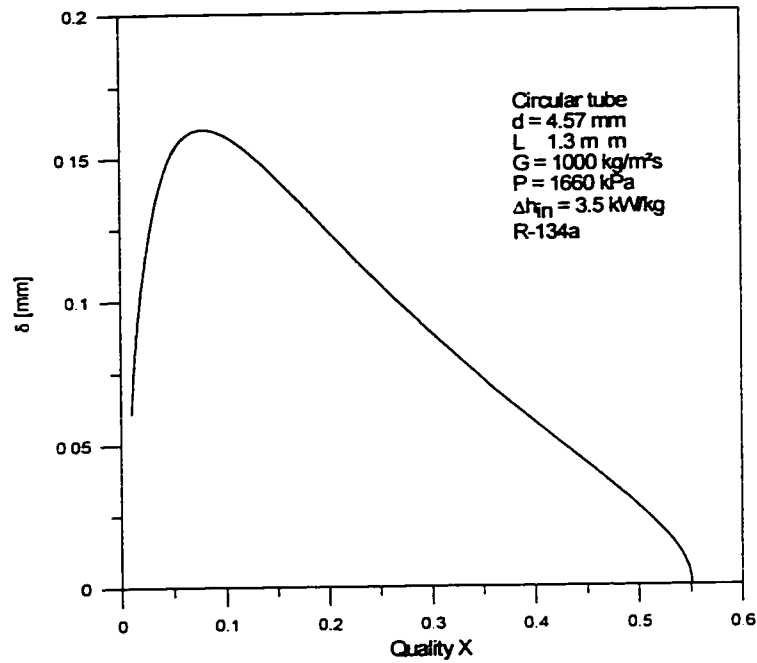


Figure 7.11 Variation of liquid film thickness along the tube axis (R-134a).

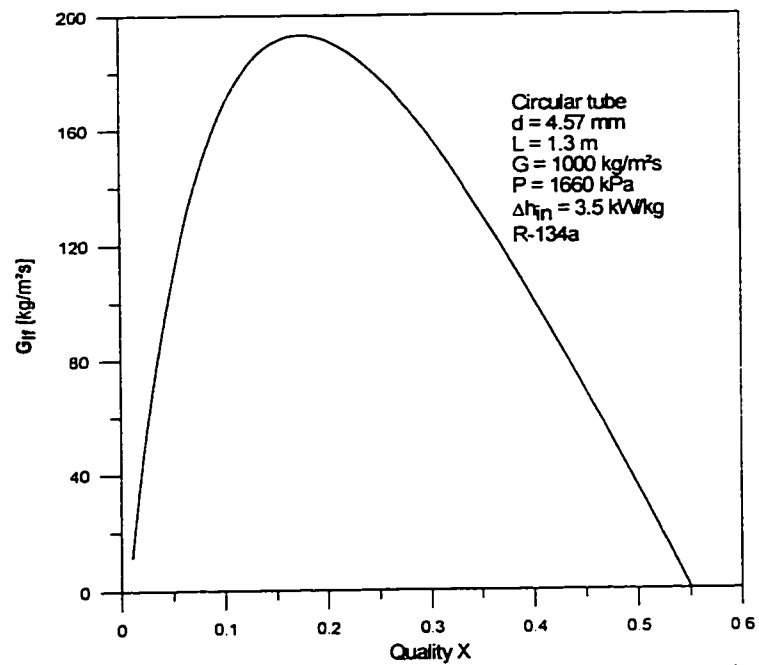


Figure 7.12 Variation of liquid film mass flow rate along the tube axis (R-134a).

7.2 Partition Model Application to Dumbbell-Shaped Tube

This section presents a detailed assessment of the partition model application to the dumbbell-shaped tube. Theoretically, the partition model can be applied to any dumbbell-shaped tube under the annular flow conditions. This section is limited to the dumbbell-shaped tube with the same nominal geometry and thermalhydraulic parameters as the experiments. As described in Chapter 6, the gap size of the dumbbell-shaped tube varies from 1.0 mm, 0.7 mm, 0.5 mm, 0.3 mm, 0.15 mm to zero.

In this section, subsection 7.2.1 presents the results for deposition factors. Subsections 7.2.2 and 7.2.3 describe the general CHF trend and gap effect under uniform heating assumptions. The comparison of the predicted data against the experimental results is presented in subsection 7.2.4.

7.2.1 Deposition Factors for Dumbbell-Shaped Tube

In this study, one of the key approaches used to deal with the geometry effect on CHF is to consider the geometry effect on deposition. Deposition factors (ϕ_1 and ϕ_2) were therefore introduced. As shown in Figure 5.2, the dumbbell-shaped tube (physical flow channel) was partitioned into two pseudo flow channels: narrow gap and main flow. The main flow region is of a concave shape while the narrow gap region is mainly convex. The complex contour leads to a non-uniform distribution of deposition rate along the periphery of the channel. As a part of the partition model, two deposition factors were computed to account for the difference in deposition between the two regions.

Table 7.2 Deposition factors for the dumbbell-shaped tube

Gap size (mm)	Deposition factor	
	ϕ_1	ϕ_2
0	0.792	1.136
0.15	0.830	1.105
0.3	0.864	1.079
0.5	0.908	1.049

0.7	0.950	1.025
1	1.008	0.996

Table 7.2 shows the deposition factors for the dumbbell-shaped tube at different gap sizes based on the computing method described in section 5.2.4.5. ϕ_1 and ϕ_2 denote deposition factors for the narrow gap region and main flow region respectively. For the partition model, it is assumed that every liquid droplet entrained in the gas core will eventually deposit onto one of the two surfaces of the pseudo-channels. The possibility of a droplet deposit onto which surface depends on the view factors.

The deposition factors for both pseudo-channels change with the gap size. For a liquid droplet inside the gas core, the view angle extended to the narrow gap region increases with increasing gap size. As a result, the deposition factor for the narrow gap region increases with increased gap size.

7.2.2 General Trend

This section presents the predicted CHF general trends for the dumbbell-shaped tube. As in Chapter 6, the trends are discussed from two different views: fixed inlet condition and fixed local condition.

7.2.2.1 Fixed Inlet Condition

7.2.2.1.1 Inlet Enthalpy (Subcooling) Effect

Figures 7.13 to 7.15 illustrates the variation of the CHF with inlet enthalpy/subcooling at a fixed heated length at various gap sizes, pressures, mass flow rates. As shown in the figures, the CHF decreases with increasing inlet enthalpy. If compared with the experimental results shown in Figure 6.1 and Figure 6.2, it is found that the partition model predicts a correct CHF trend against inlet enthalpy (subcooling).

The figures also present the empirical correlation for the relationship between the CHF and inlet

subcooling. The correlation is obtained by fitting the modeling data using the least squares method. The purpose to provide the empirical correlation is for interpolating the CHF value at a given inlet condition.

7.2.2.1.2 *Mass Flux Effect at Fixed Inlet Condition*

The mass flux effect on the CHF in the dumbbell-shaped tube at fixed inlet condition is shown in Figure 7.16. Only 0.7mm gap size results are shown here since the results for other gap sizes have a similar trend.

By comparing Figure 7.16 with Figure 6.3 and Figure 6.4, it is found that the partition model predicts the correct trend of mass flux effect on CHF at fixed inlet condition. The CHF increases with an increase of mass flux at a fixed inlet enthalpy or subcooling. The CHF rises more rapidly at a lower flow than that at a higher flow.

7.2.2.1.3 *Pressure Effect at Fixed Inlet Condition*

Figure 7.17 illustrates the pressure effect on CHF at fixed inlet condition. The figure only shows the results for the gap size of 0.7mm. The results of other gap sizes are not shown due to the similarity. If one compares the figure with Figure 6.5, it is found that the partition model predicts a correct trend of the pressure effect on CHF at fixed inlet condition. The CHF increases with decreasing pressure.

In summary, at fixed inlet condition the partition model predicts similar trends of the CHF against inlet subcooling, mass flux, and pressure, as measured. Both the measured and predicted CHF trends in the dumbbell-shaped tube are also appeared in the circular tube.

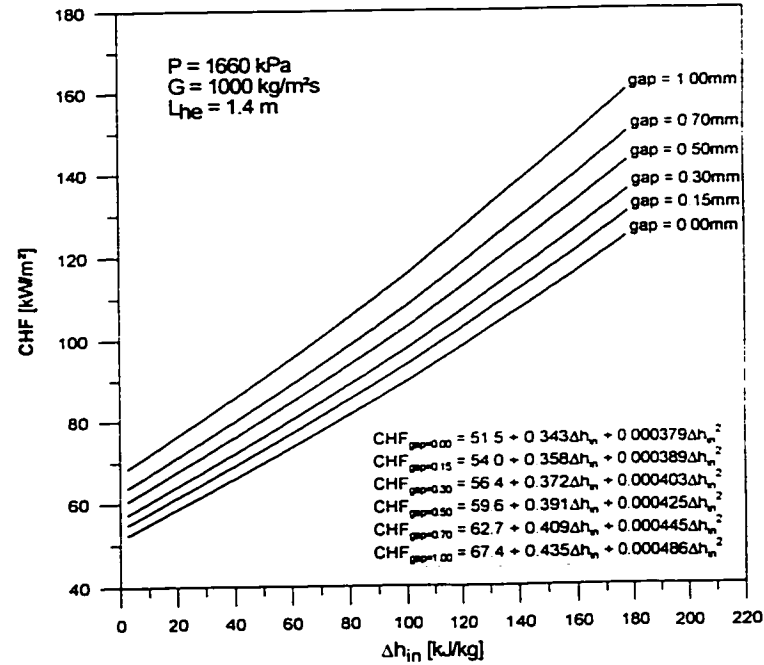
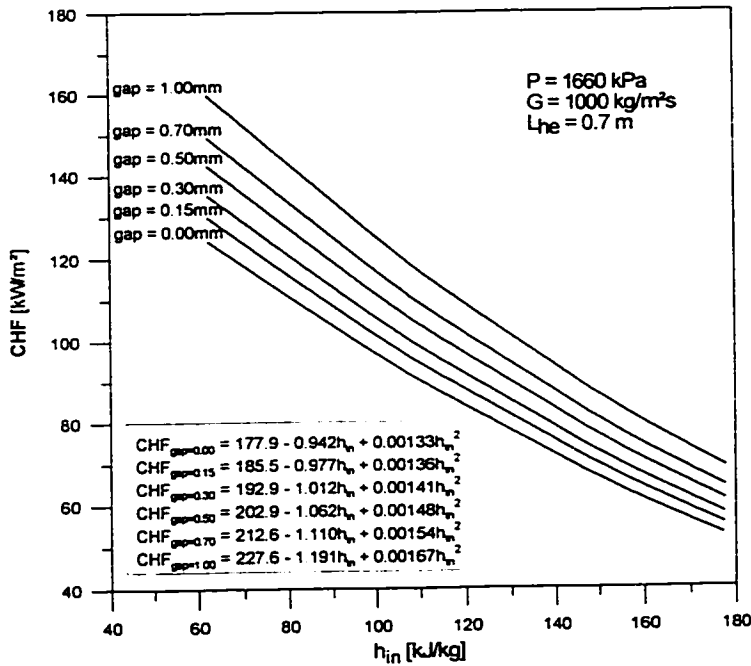
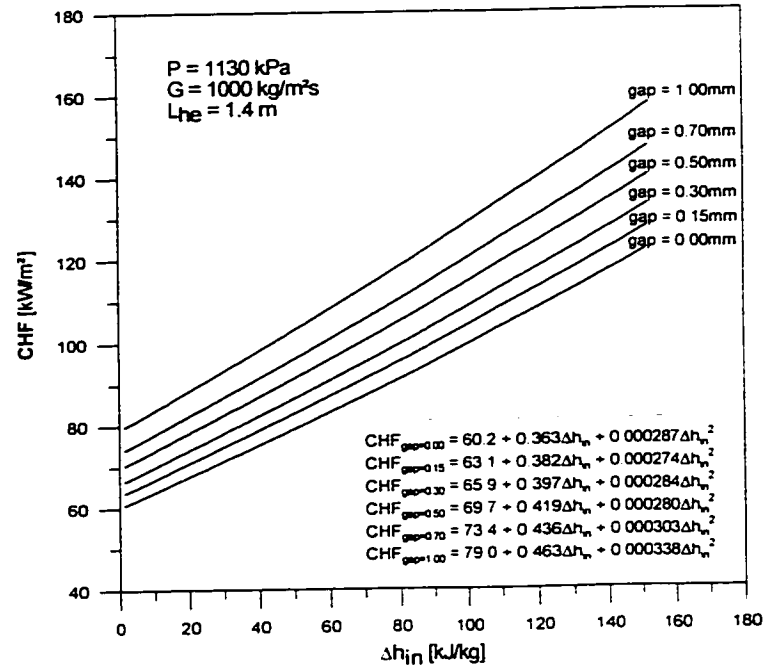
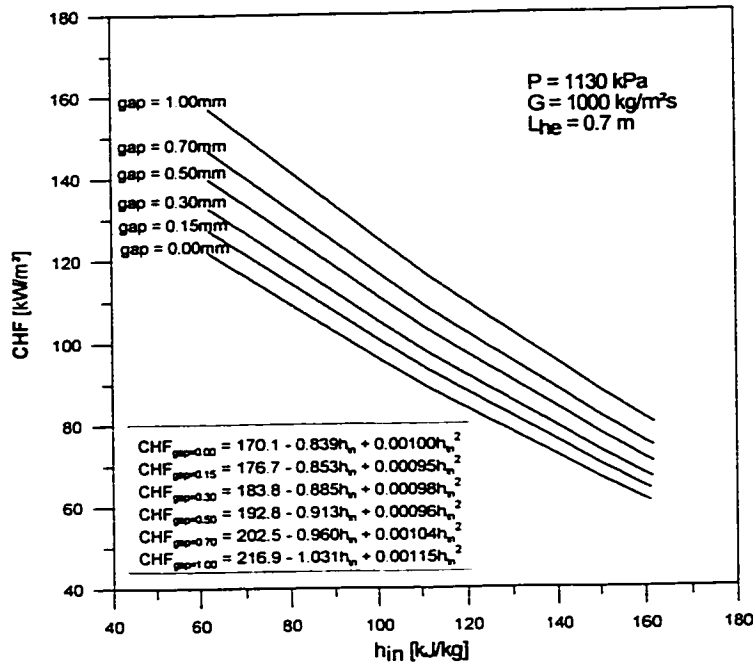


Figure 7.13 Inlet enthalpy/subcooling effect on the CHF in dumbbell-shaped tube ($G = 1000 \text{ kg/m}^2\text{s}$).

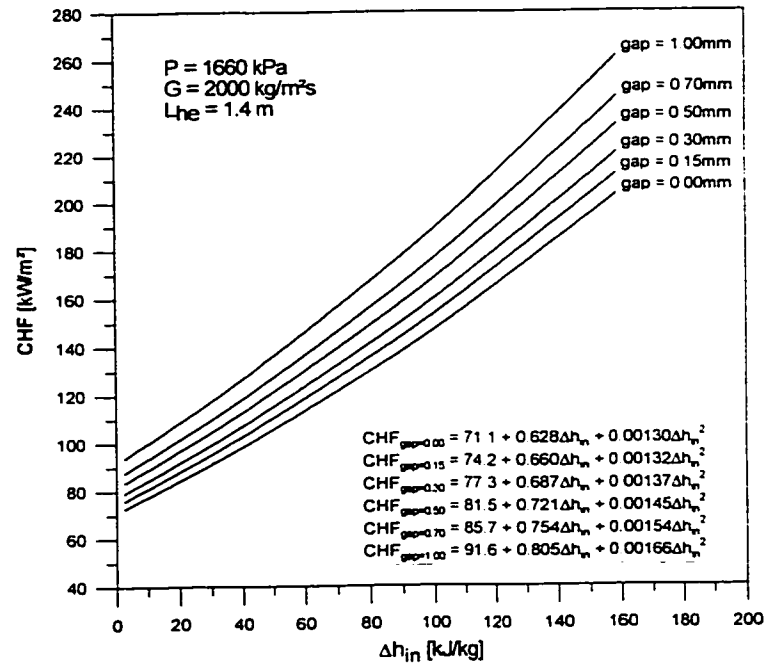
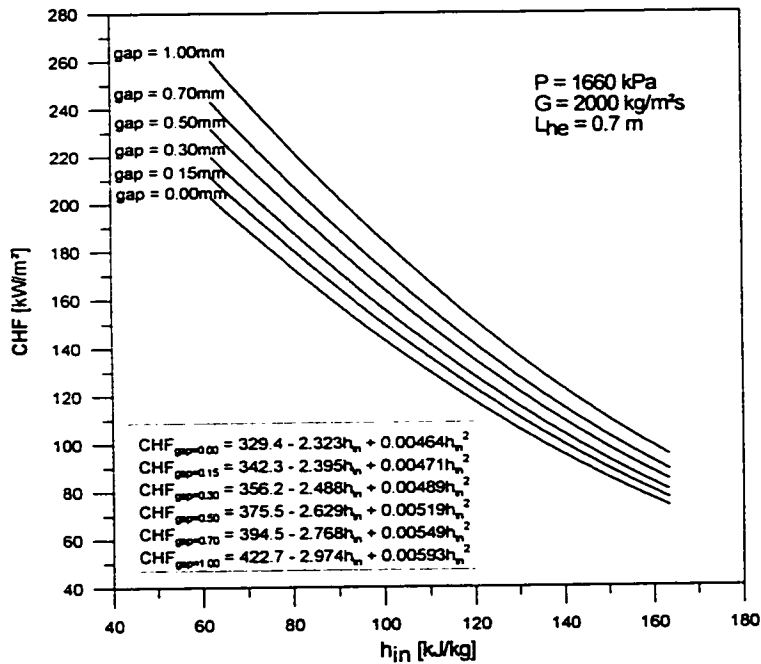
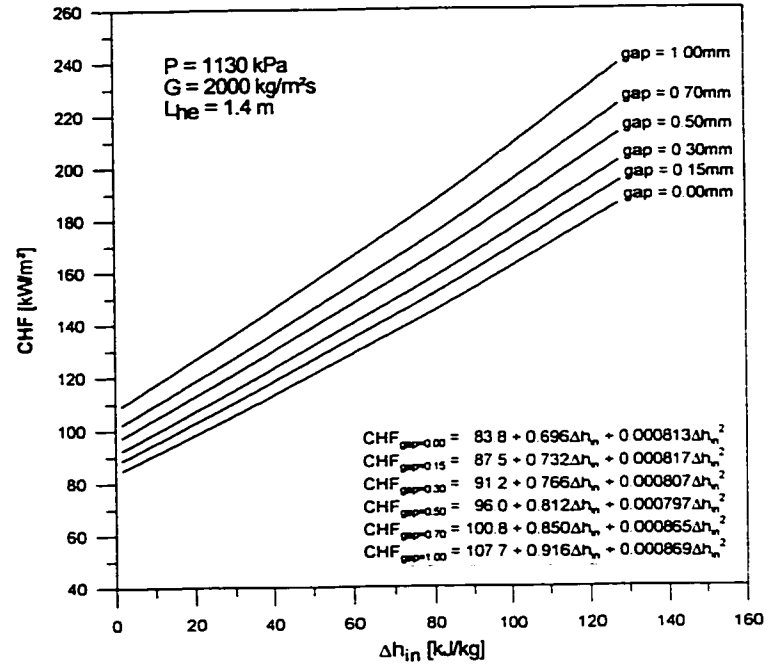
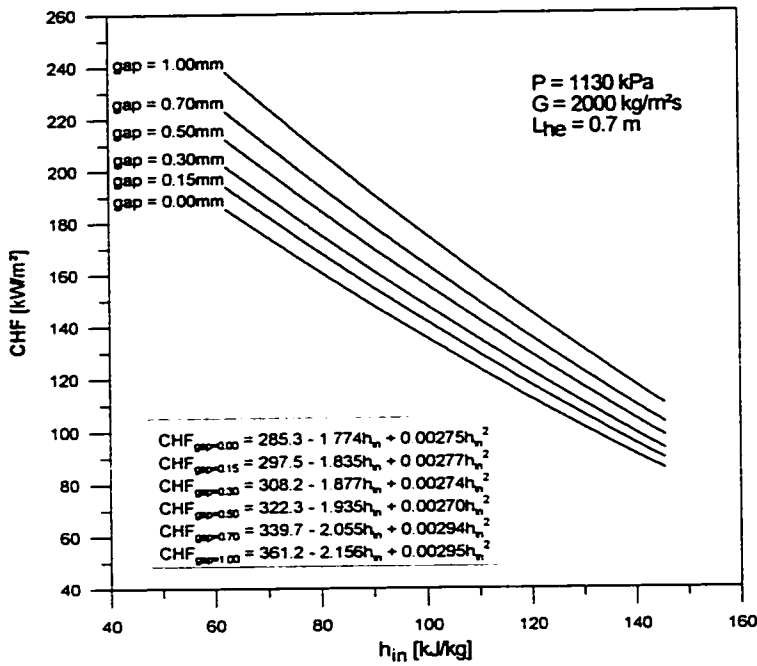


Figure 7.14 Inlet enthalpy/subcooling effect on the CHF in dumbbell-shaped tube ($G = 2000 \text{ kg/m}^2\text{s}$).

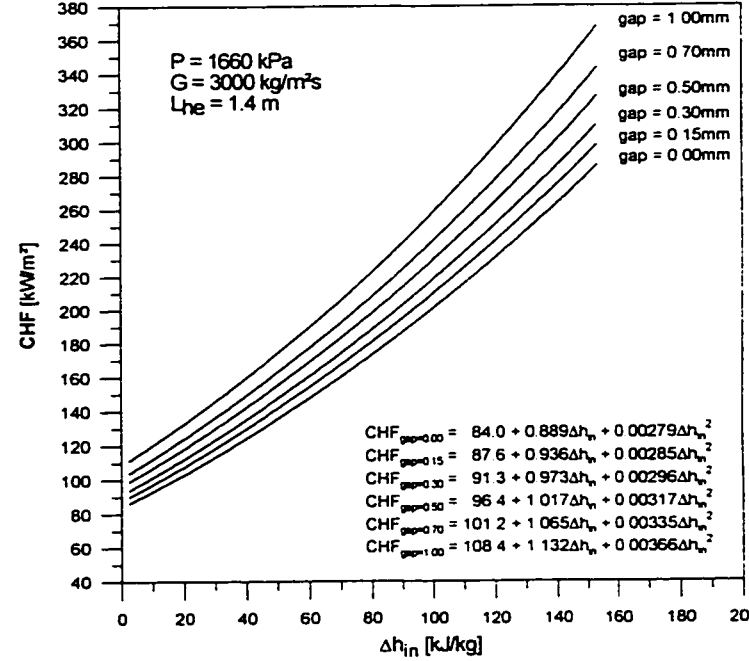
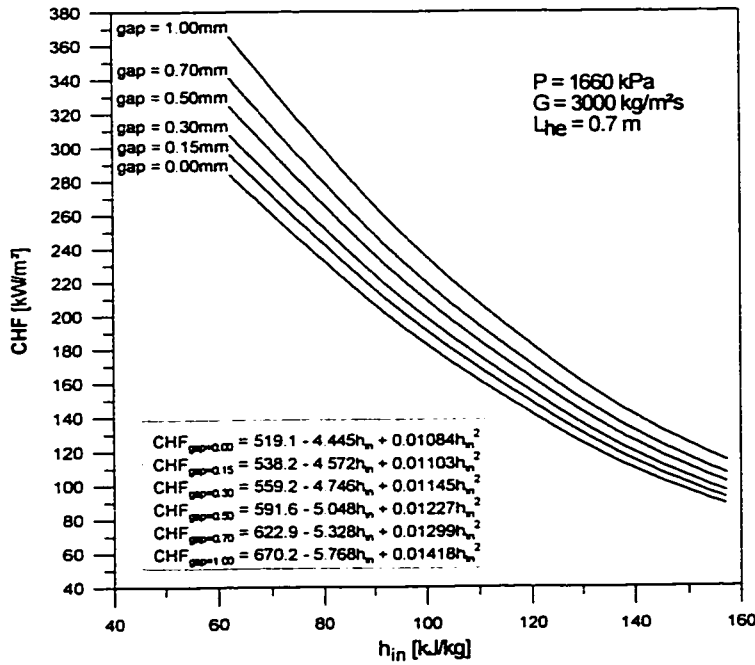
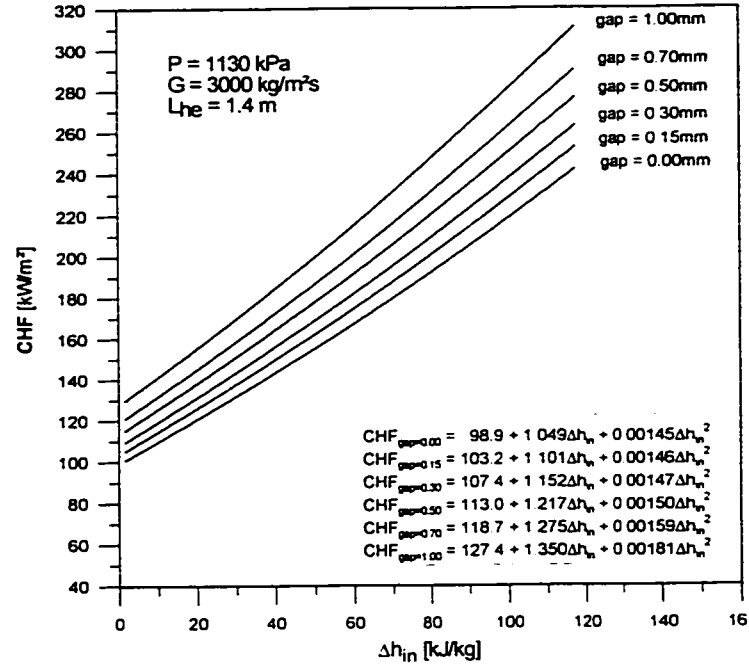
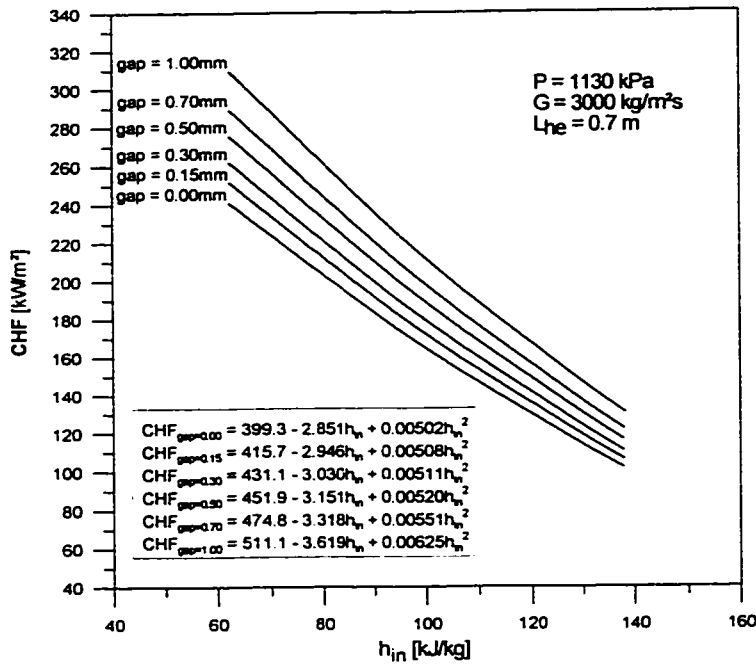


Figure 7.15 Inlet enthalpy/subcooling effect on the CHF in dumbbell-shaped tube ($G = 3000 \text{ kg/m}^2\text{s}$).

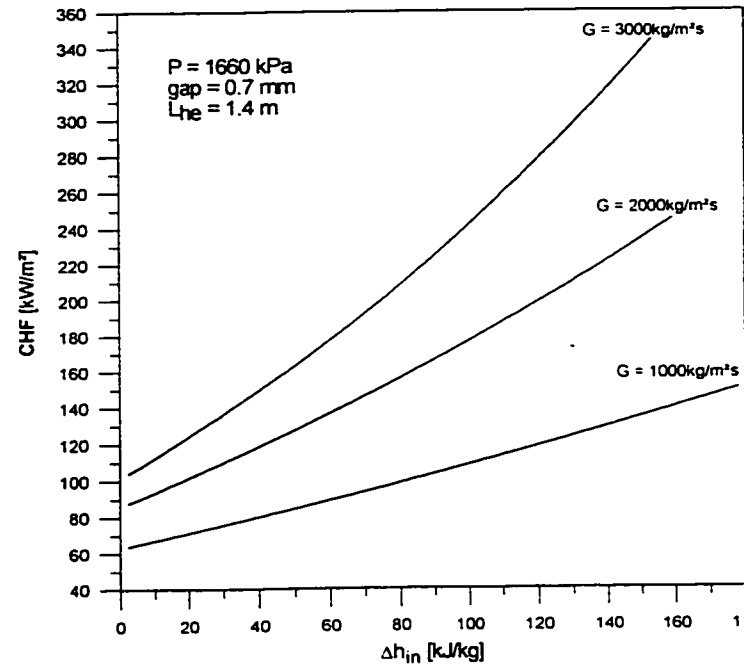
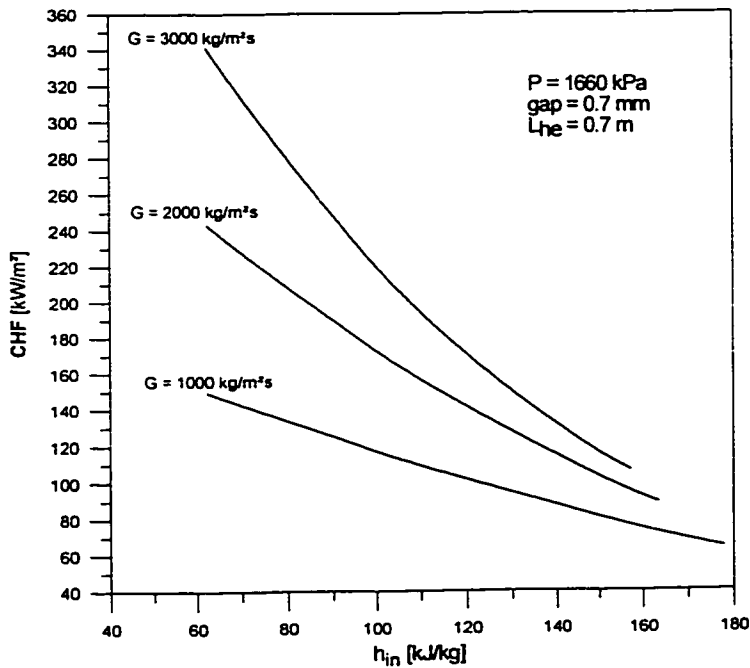
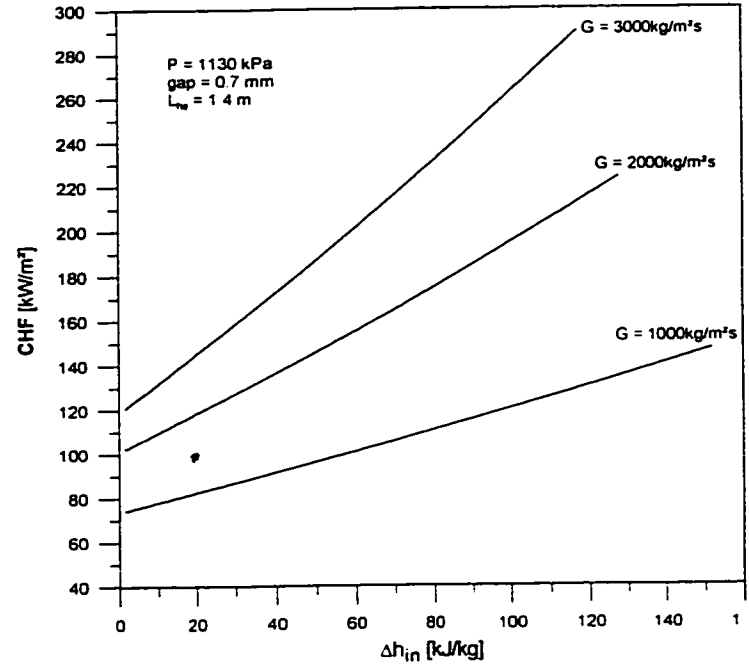
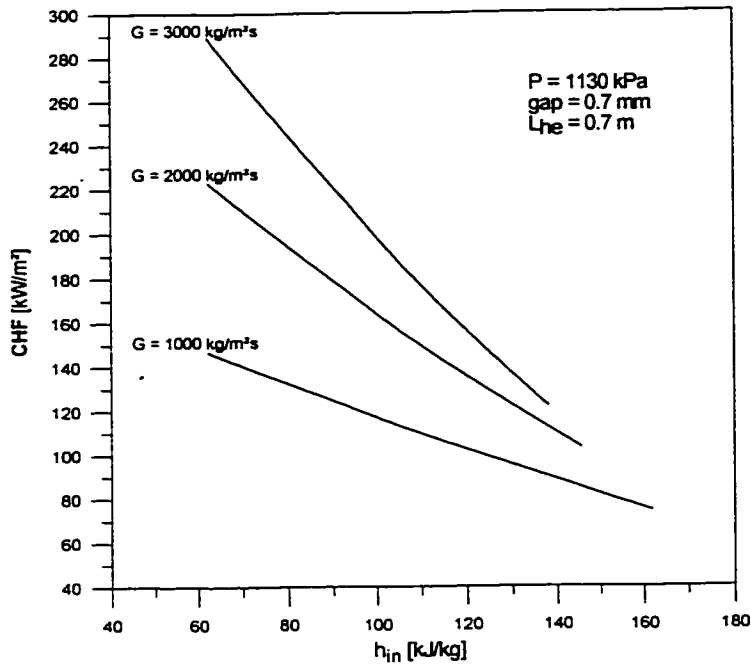


Figure 7.16 Mass flux effect on the CHF in dumbbell-shaped tube at fixed inlet condition.

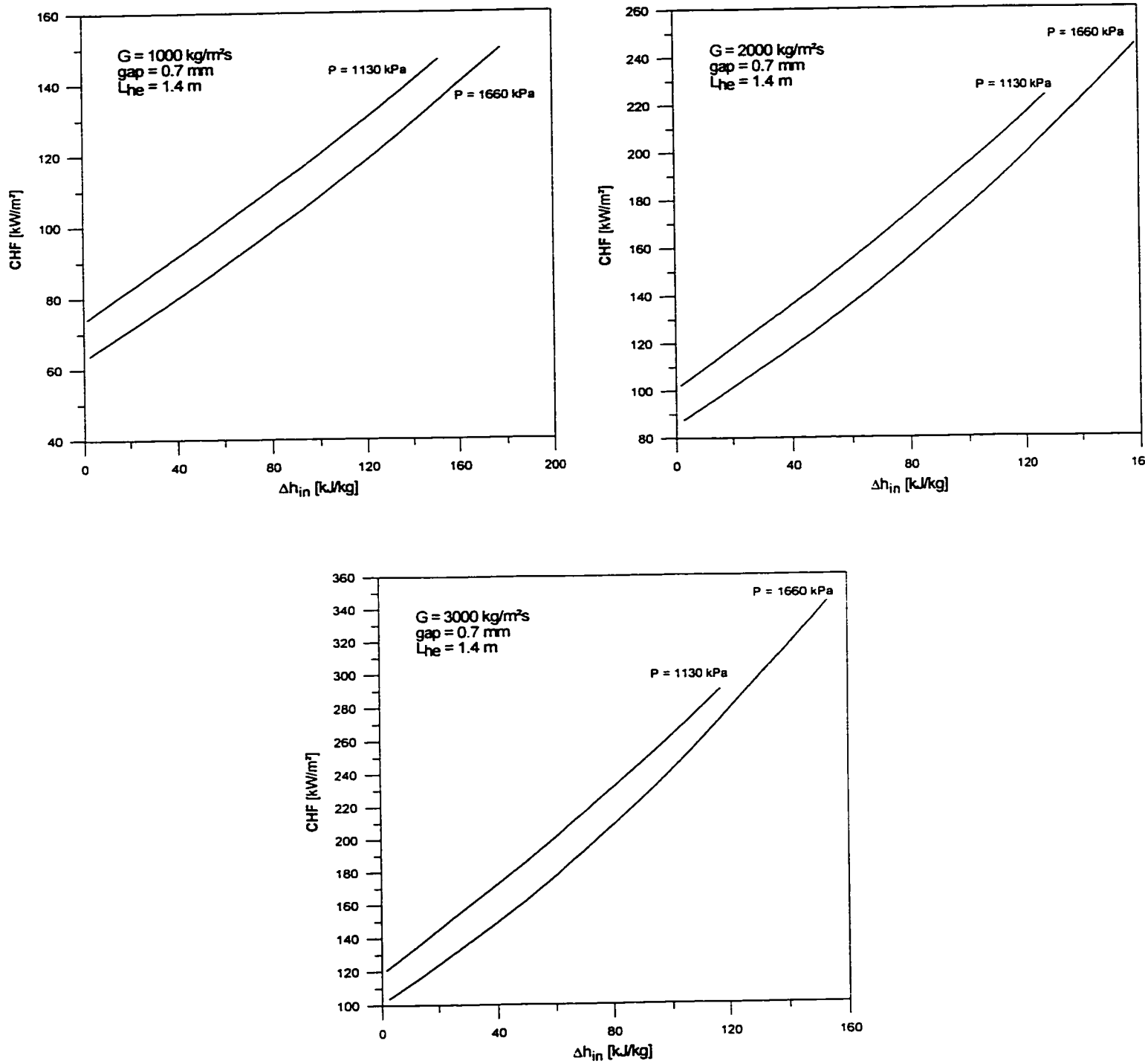


Figure 7.17 Pressure effect on the CHF in dumbbell-shaped tube at fixed inlet condition.

7.2.2.2 Fixed Local Condition

7.2.2.2.1 *Quality Effect*

Figure 7.18 shows the dumbbell-shaped tube CHF plotted against quality at a fixed pressure and mass flux. In general, the partition model predicts a similar CHF trend against quality if one compares the figure with Figures 6.6 to 6.11. The CHF decreases with an increase of critical quality. However, different from the experimental results, the model does not predict the “limiting quality” phenomenon. This is because the partition model does not take the “limiting quality” phenomenon into account. Further research on the mechanism of the “limiting quality” phenomenon and modeling is required. Currently, no literature on modeling the “limiting quality” has been found, even for the simple geometry.

The empirical correlation for the relationship between the CHF and local quality is also shown in Figure 7.18. The correlation is obtained by fitting the modeling data using the least squares method. The purpose in providing the empirical correction is for interpolating the CHF value at a given local condition.

7.2.2.2.2 *Mass Flux Effect at Fixed Local Condition*

Figure 7.19 illustrates the effect of mass flux on CHF in the dumbbell-shaped tube at a fixed local condition. Only 0.7mm gap size results are shown here since the results for other gap sizes have a similar trend. By comparing the figure with Figure 6.12 and Figure 6.13, it is found that the partition model predicts a similar trend in mass flux effect on CHF at fixed local condition. The CHF decreases with increased mass flux for a fixed local condition in a higher quality region. For annular flow, the higher flow results in a higher entrainment and hence a lower CHF.

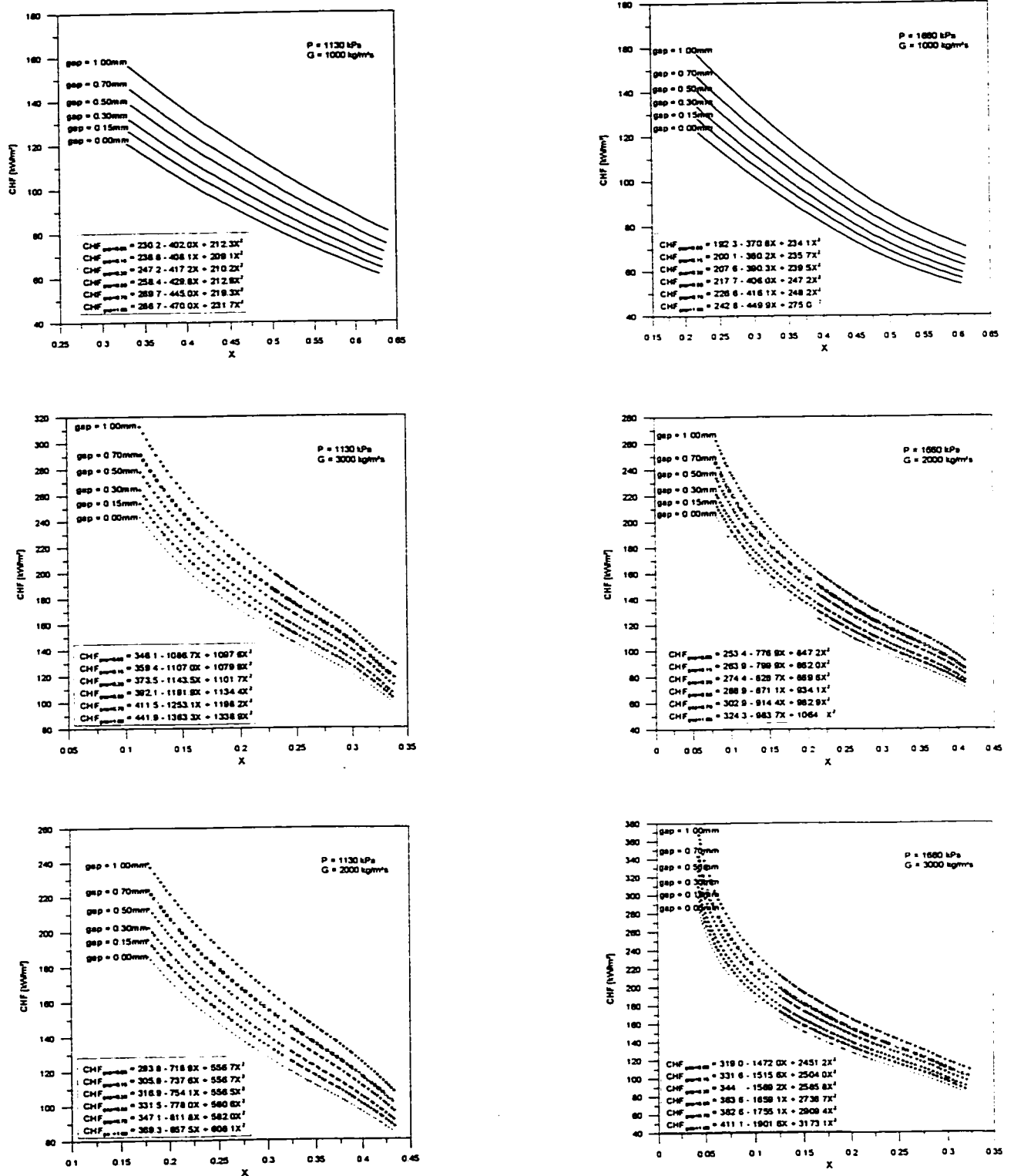


Figure 7.18 Exit quality effect on the CHF in dumbbell-shaped tube.

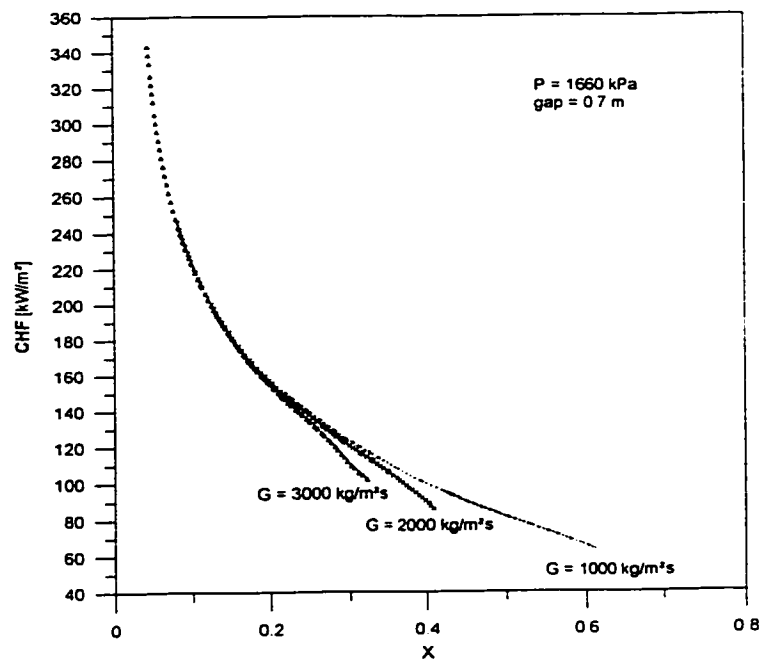
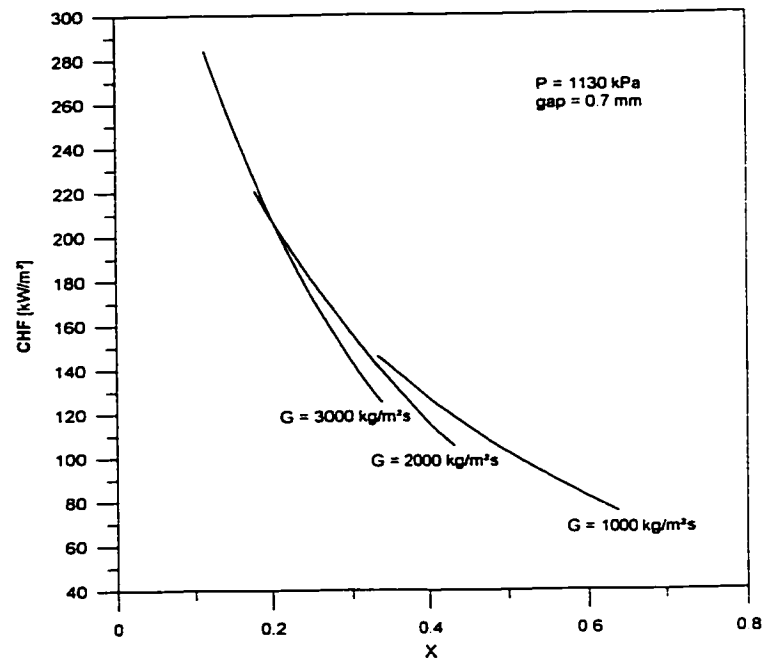


Figure 7.19 Mass flux effect on CHF in dumbbell-shaped tube at fixed local condition.

7.2.2.2.3 Pressure Effect at Fixed Local Condition

In this study, only two pressures were tested and computed. Figure 7.20 illustrates the pressure effect on CHF in a dumbbell-shaped tube at fixed local condition. Only the results of the dumbbell-shaped tube with a gap size of 0.7 mm are presented here since the trends of all gap sizes are similar. When comparing the figure with Figure 6.14, it is found that the partition model predicts a correct trend of the pressure effect on CHF at fixed inlet condition. At fixed local condition, the CHF decreases when the pressure increases from 1130 kPa to 1660 kPa.

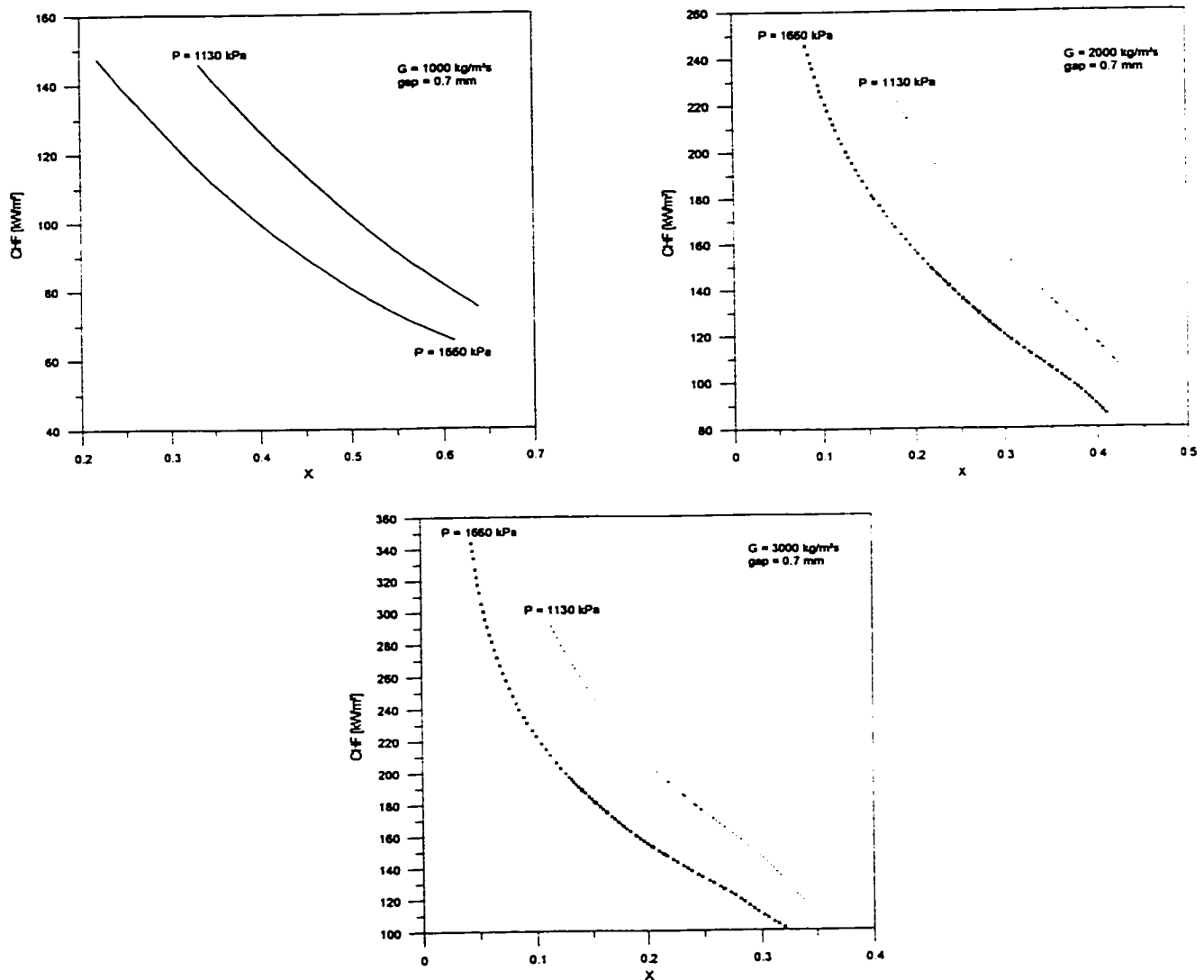


Figure 7.20 Pressure effect on CHF in dumbbell-shaped tube at fixed local condition.

In summary, the partition model predicts correct CHF trends at fixed local condition for the dumbbell-shaped tube.

7.2.3 Gap Effect

7.2.3.1 Inlet Condition

Figures 7.21 to 7.23 illustrate the overall geometry effect on CHF in dumbbell-shaped tube. It is noted that the data shown in here are combined effects of all geometric parameters. With a decrease of the gap size from 1 mm to zero, the flow area and hydraulic diameter also decrease. In this study, the gap size is used to characterize the entire geometry. Both the absolute CHF data and the ratio of the CHF based on the tube with 1 mm gap are plotted against the gap size. The CHF ratio curves represent the gap effect for each flow condition.

If comparing the figures with Figure 6.15 and Figure 6.16, it is found that the partition model predicts a correct trend of gap effect at fixed inlet condition. The CHF decreases with decreased gap size. When the gap size reduces from 1 mm to zero, the CHF drops about 23% due to the overall geometry effect.

The predicted results also show that the thermalhydraulic parameters have little influence on gap effect. From the figures of the CHF ratio, the four curves of the CHF ratio at different inlet subcooling almost converge to one. At same mass flux, the curves for the two different pressures are almost identical. For the three mass fluxes tested, the gap effect changes very little as shown in the figures of the CHF ratio.

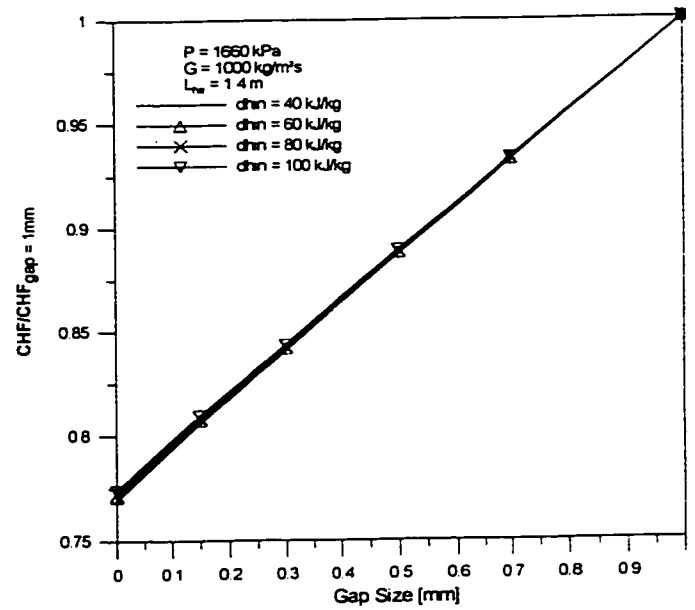
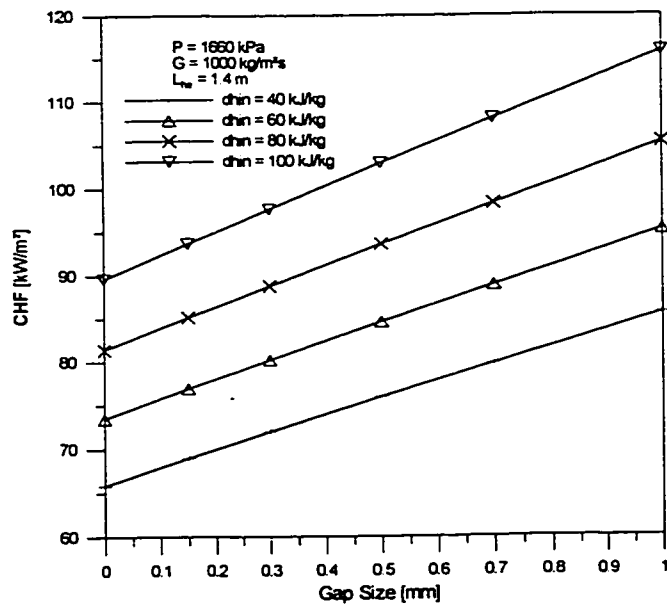
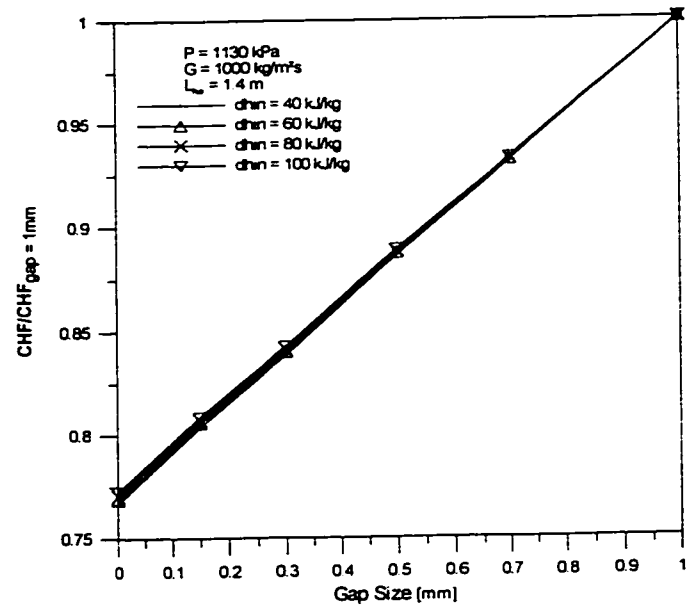
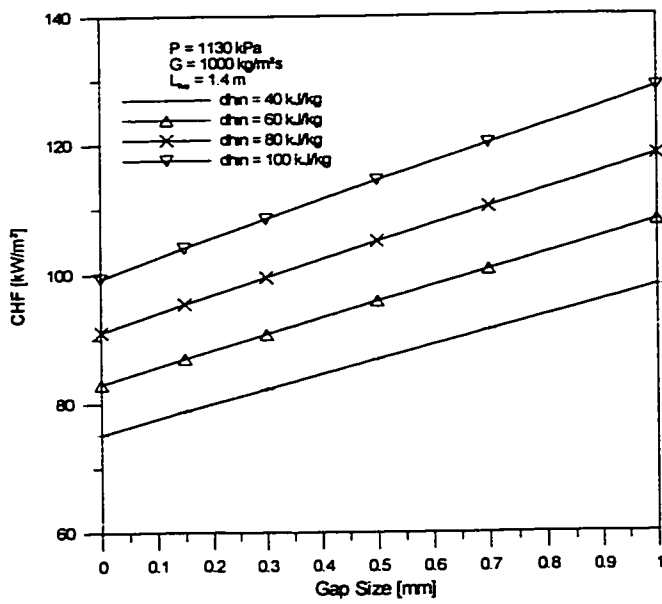


Figure 7.21 Geometry effect on CHF in dumbbell-shaped tube (inlet condition, $1000 \text{ kg/m}^2\text{s}$).

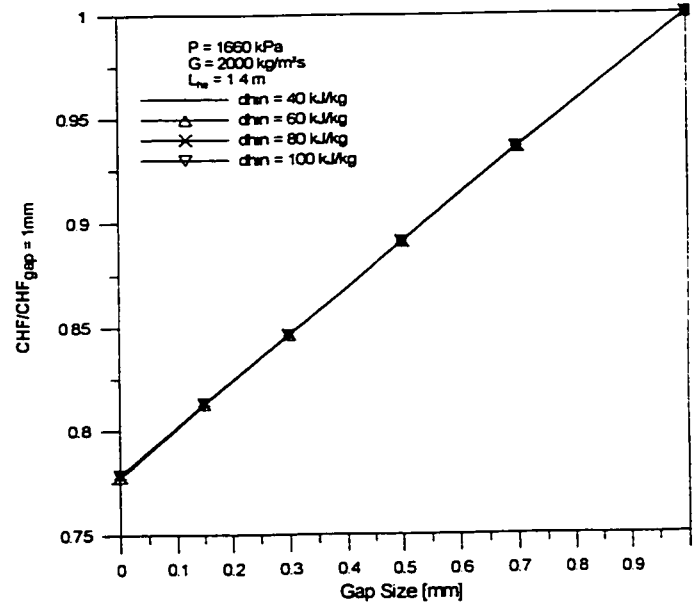
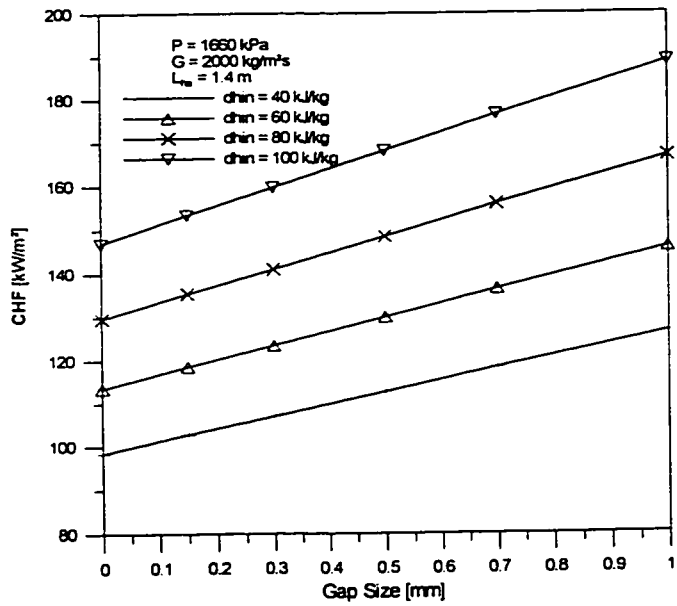
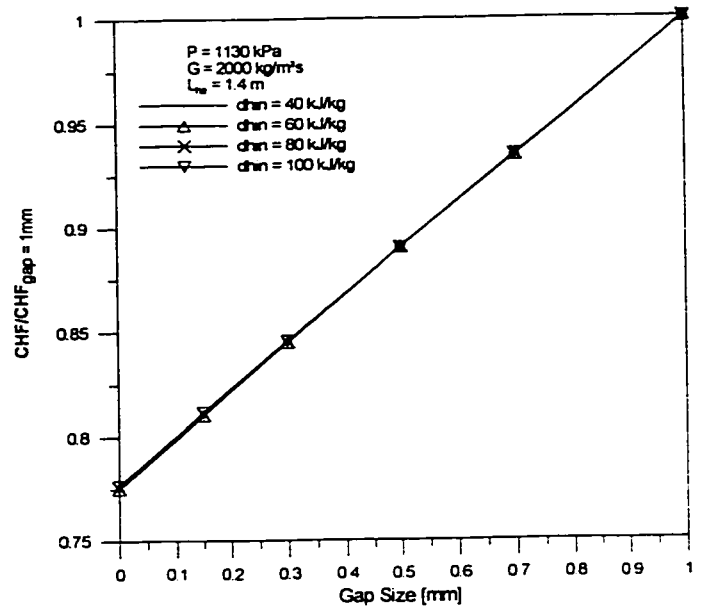
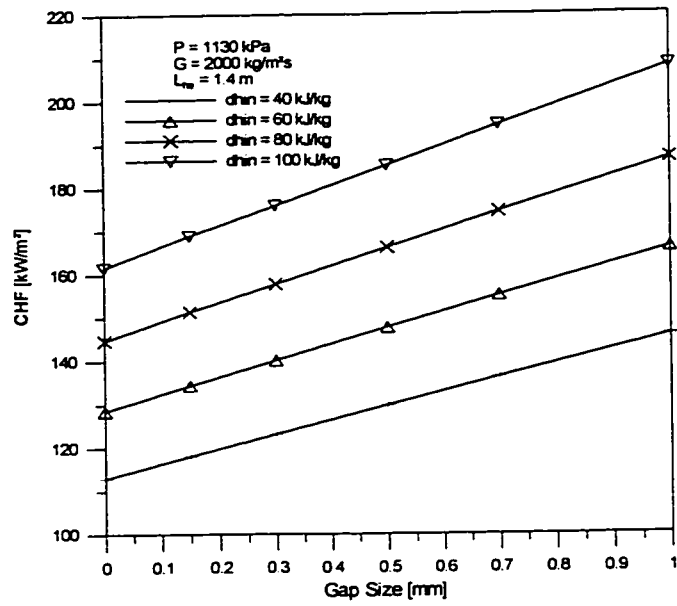


Figure 7.22 Geometry effect on CHF in dumbbell-shaped tube(inlet condition, 2000 kg/m²s).

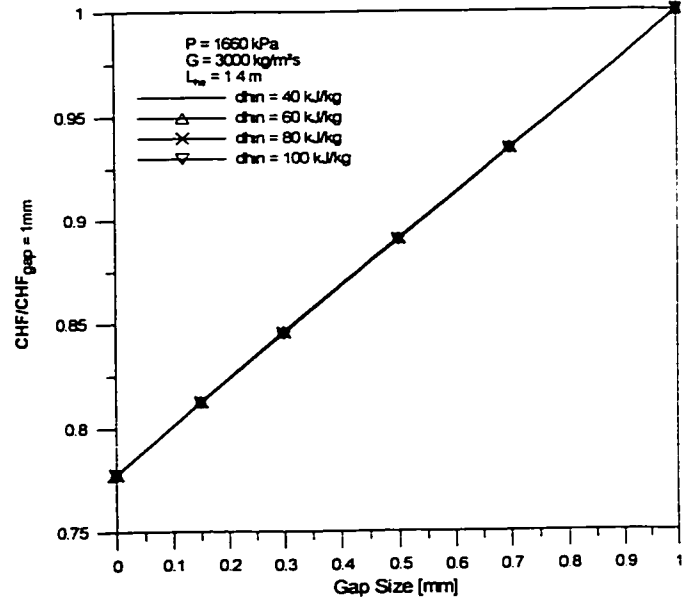
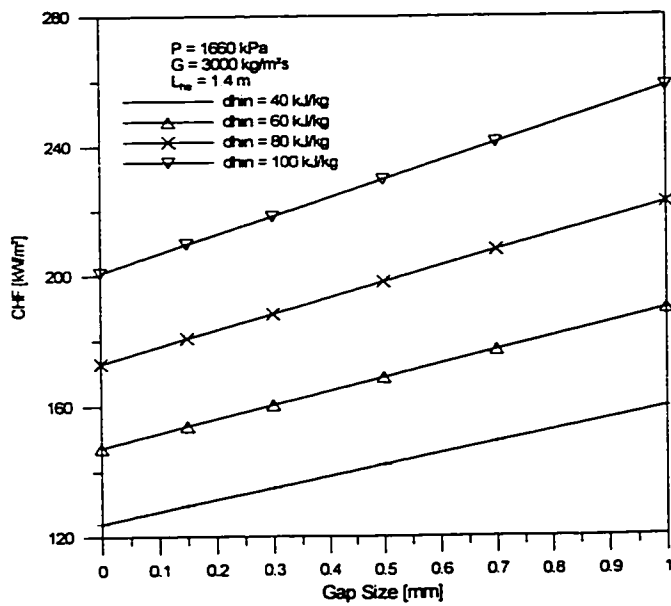
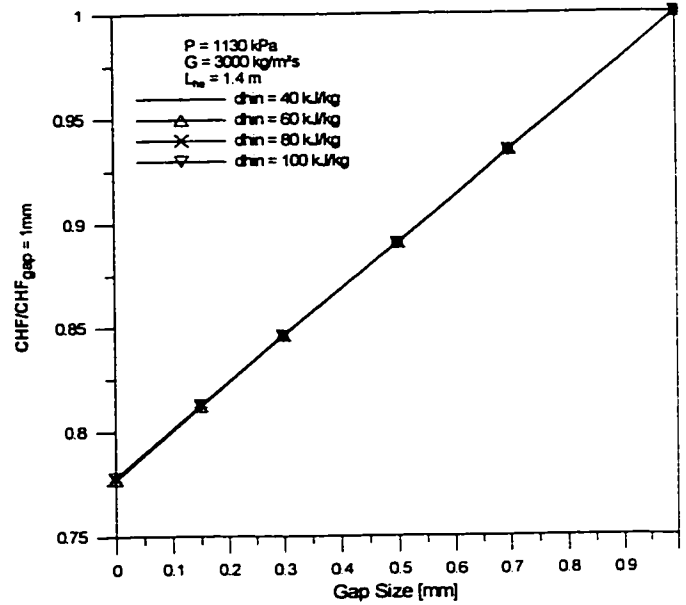
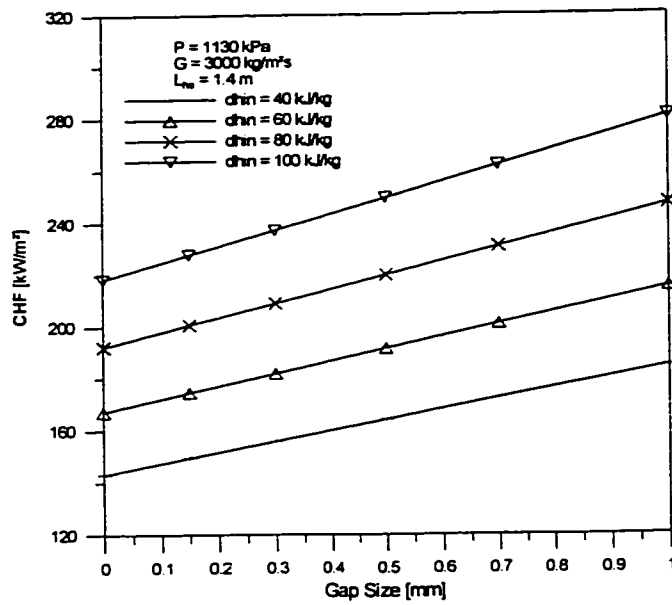


Figure 7.23 Geometry effect on CHF in dumbbell-shaped tube (inlet condition, 3000 kg/m²s).

7.2.3.2 Local Condition

Figures 7.24 and 7.25 present the overall geometry effect on CHF in a dumbbell-shaped tube at fixed local condition. Both the absolute CHF value and the ratio of the CHF based on the tube of 1 mm gap are shown in the figures. The CHF decreases about 23.5% on average as the gap size decreases from 1 mm to zero, depending on the flow conditions.

As described in Chapter 6, the overall geometry effect can be considered as the integrated effects of gap effect and hydraulic diameter effect. With the gap size decreasing from 1 to 0 mm, the hydraulic diameter of the dumbbell-shaped tube decreases from 4.30 mm to 3.43 mm due to the decrease of flow area. According to Groeneveld's CHF lookup table (1996), at fixed local condition the CHF increases about 12% when the hydraulic diameter decreases from 4.30 mm to 3.43 mm. Therefore, excluding the hydraulic diameter effect, the CHF in the dumbbell-shaped tube drops about 35.5% as the gap size decreases from 1 mm to zero at fixed local condition.

It is also found from the figures that the thermalhydraulic parameters have little influence on the gap effect at fixed local condition. The curves of the CHF ratio do not change much with changes of the thermalhydraulic parameters (p , G , and x).

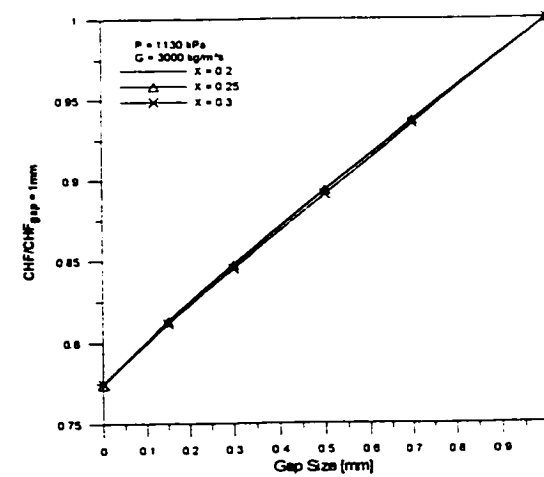
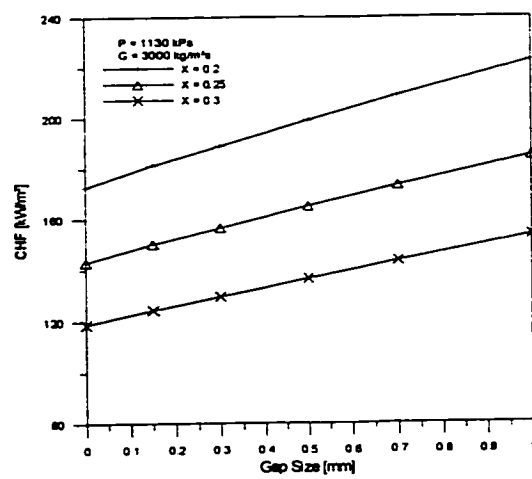
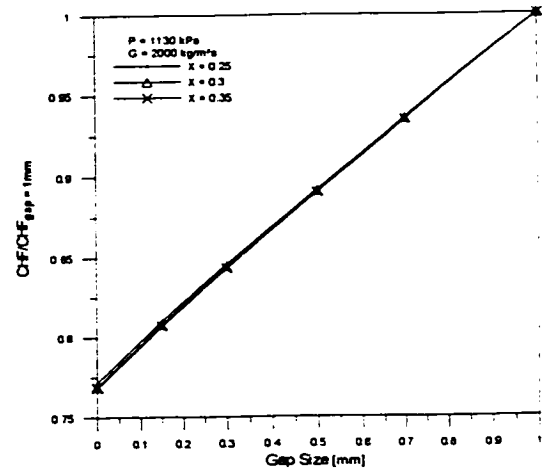
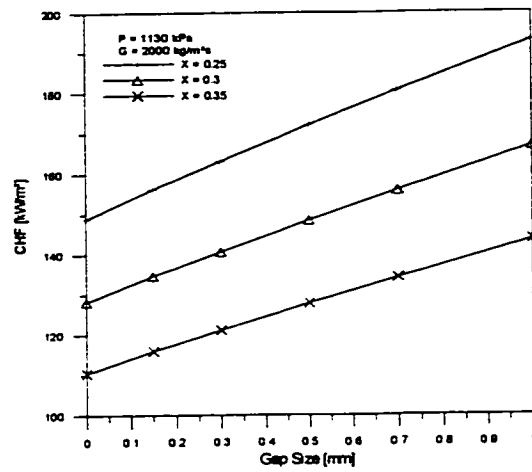
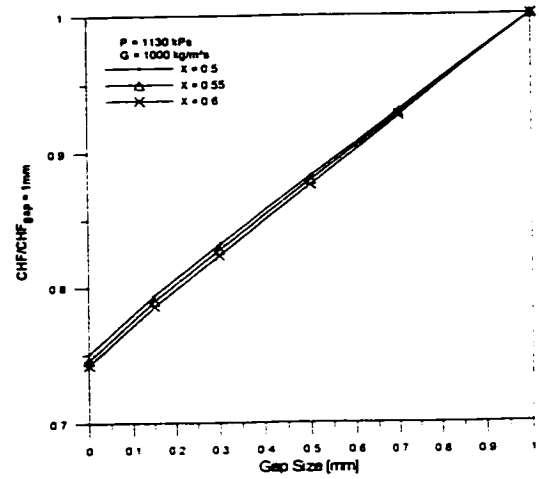
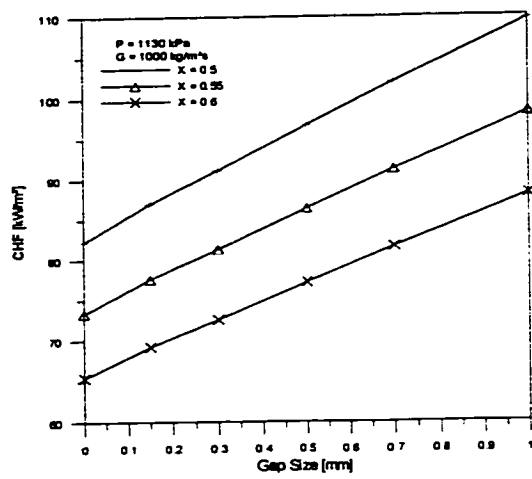


Figure 7.24 Geometry effect on CHF in dumbbell-shaped tube (local condition, 1130 kPa).

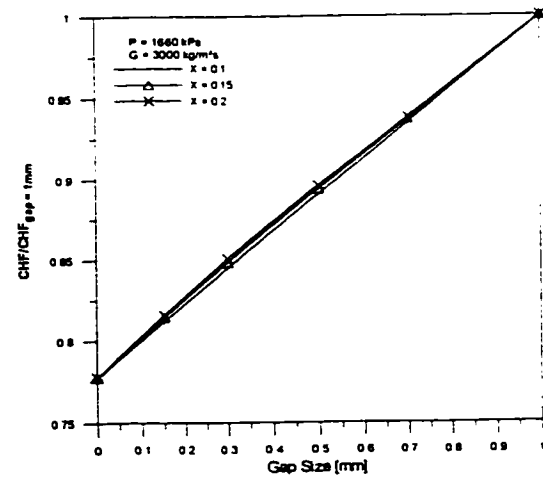
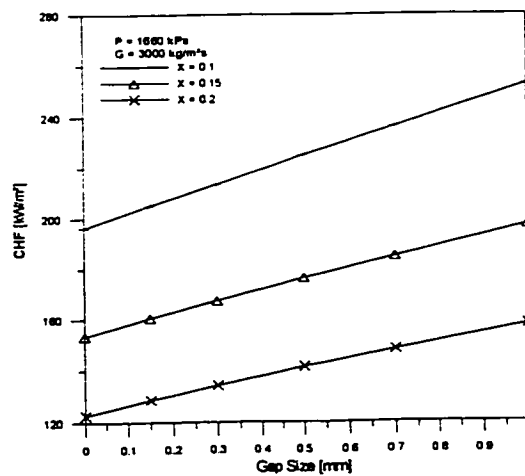
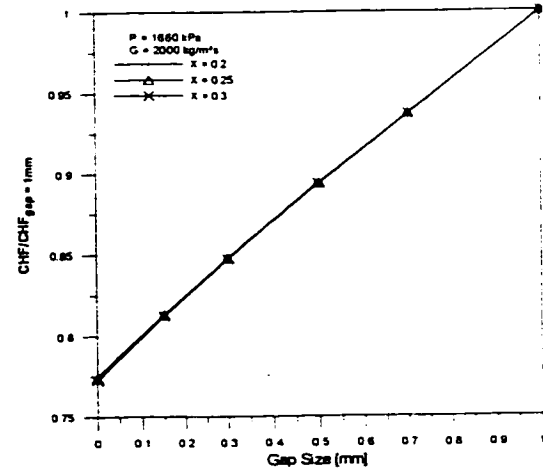
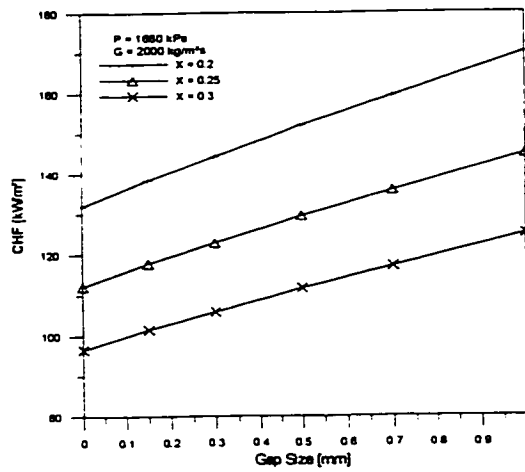
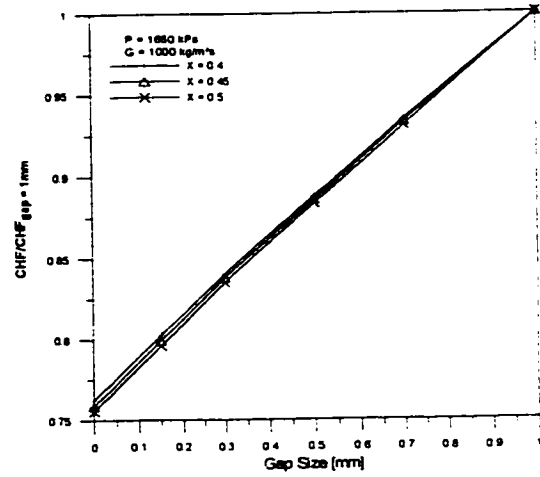
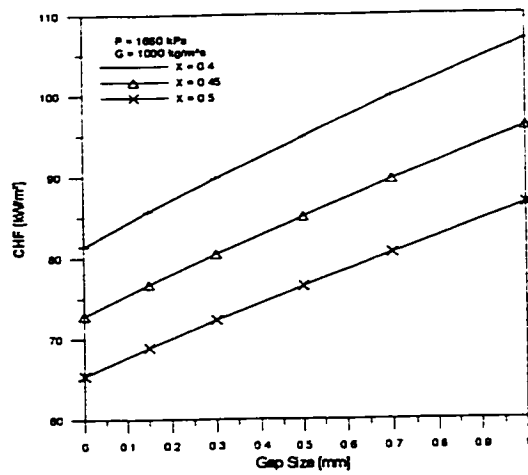


Figure 7.25 Geometry effect on CHF in dumbbell-shaped tube (local condition, 1660 kPa).

7.2.3.3 CHF Location

Under the uniform heating conditions, for all the flows computed using partition model, the CHF always occurred first in the narrow gap region. This is due to the relatively more intensive heating in the narrow gap region. The heat transferred to the working fluid per unit volume in the narrow gap region is higher than that in the main flow region. This leads to earlier dryout in the narrow gap region. Furthermore, the deposition factor for the narrow gap region is normally smaller than that for main flow region.

Figure 7.26 shows a typical curve of the variation of liquid film flow rate along the axis of the dumbbell-shaped tube. The mass flux of the liquid film in the main flow region is higher than that in the narrow gap region. When the CHF occurs in the narrow gap region at the exit of the tube in the narrow gap region, the liquid film in the main flow region does not dry out.

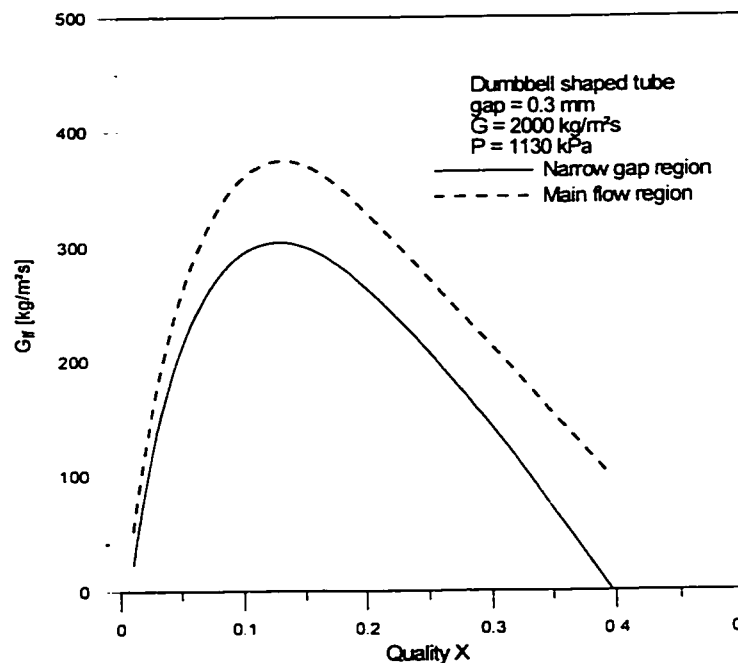


Figure 7.26 Variation of liquid film flow rate along the tube axis.

7.2.4 Comparison Against Test Results

This subsection presents the comparison between the predicted results and the experimental results for the dumbbell-shaped tube under the uniform heating assumption.

7.2.4.1 Boundary Condition

Theoretically, for each CHF point compared, the inlet and heating conditions should be the same for both cases. By design intention for the experimental investigation, the dumbbell-shaped tube was uniformly heated. However, due to the limitation of the heating method, the heat flux distribution along the periphery of the dumbbell-shaped tube is not uniform.

During the experiments, the dumbbell-shaped test section was heated by direct current. If the wall thickness of the dumbbell-shaped tube is assumed uniform, the heat generated by a unit volume of the tube wall is the same. The narrow gap region of the dumbbell-shaped tube is of convex shape while the main flow channel region is of concave shape. The curvature change makes the heat flux distribution non-uniform. The heat flux in the narrow gap region will be lower than that in the main flow channel region. This can be illustrated by considering a cylinder as shown in Figure 7.27. The cylinder generates a certain amount of heat. If all the heat generated is transferred to the inner surface, the heat flux will be higher than that transferred to the outer surface due to a smaller surface. The concave shape has a smaller heating surface than the convex shape.

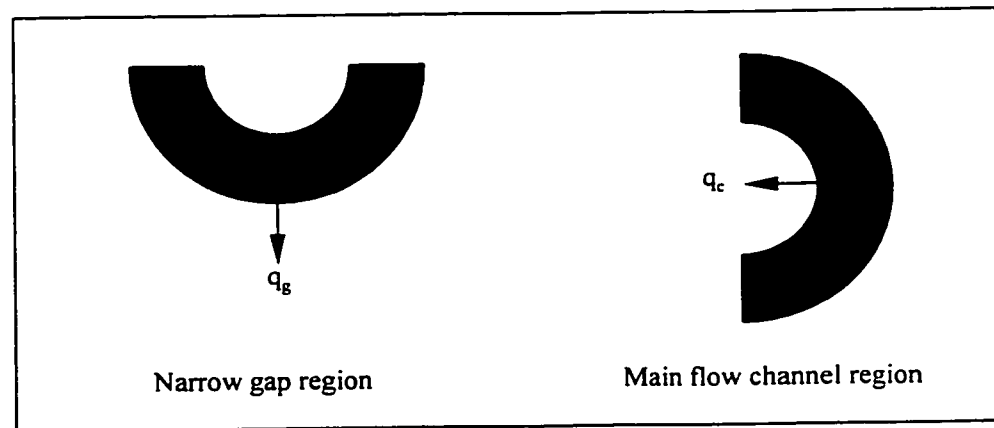


Figure 7.27 Difference of heat flux in the dumbbell-shaped tube.

The ratio of the heat flux at the narrow gap and the main flow channel region is given by

$$\frac{q_g}{q_c} = \frac{\frac{\rho dA_g dz}{dz dl_g}}{\frac{\rho dA_c dz}{dz dl_c}} = \frac{\frac{dA_g}{dl_g}}{\frac{dA_c}{dl_c}} = \frac{\frac{(R_g^2 - r_g^2)d\theta}{R_g}}{\frac{(R_c^2 - r_c^2)d\theta}{r_c}} = \frac{(R_g^2 - r_g^2)r_c}{(R_c^2 - r_c^2)R_g}$$

where ρ is the power generated per volume. R_g , r_g , R_c , and r_c are radii specified in Figure 4.2. Note that as described in Chapter 4, with the gap size decreases from 1 mm to 0, the changes of these radii are negligible. For the dumbbell-shaped tube tested,

$$\frac{q_g}{q_c} = \frac{(3.35^2 - 3^2) \times 2.65}{(3^2 - 2.65^2) \times 3.35} = 0.89$$

This means that the heat flux in the narrow gap region is about 11% less than that in the main flow channel region.

For the dumbbell-shaped test section, there are three heat fluxes: the average heat flux over the entire cross section of the dumbbell-shaped tube q , the average heat flux in the narrow gap region q_g , and the average heat flux in the main flow channel region q_c . Table 7.3 illustrates the difference of the heat fluxes.

Table 7.3 Heat fluxes of the dumbbell-shaped tube

Gap size (mm)	q_c/q	q_g/q
0	1.0456	0.9304
0.15	1.0434	0.9287
0.3	1.0421	0.9272
0.5	1.0401	0.9254
0.7	1.0382	0.9238
1	1.0356	0.9215

It is noted that the experimental data are presented using the average heat flux over the entire cross section. This means the reported experimental CHF value will be higher than the local CHF if it occurs first in the narrow gap region. It will be lower than the local CHF if it occurs first in the main flow channel region.

7.2.4.2 CHF Location

The modeling results show that the CHF always occurs first in the narrow gap region due to the gap effect. However, as described in Chapter 6, in the experiment, the CHF in the dumbbell-shaped tube may occur first in the main flow channel region. This different result between the prediction and measurement in CHF location is due to the difference in heating conditions of the two cases.

In the case of the modeling study, the dumbbell-shaped tube is uniformly heated. But, for the experimental investigation, it is not. The heat flux in narrow gap region is less than that in the main flow region. For the larger gap cases, when the gap effect is not so strong, the CHF may even occur at the main flow channel first due to higher heat flux.

7.2.4.3 General Trend

The comparison of the modeling results against the measured data is shown in Figure 7.28. Generally speaking, the partition model well predicts the CHF in the dumbbell-shaped tube. For most of data, the relative error is within $\pm 15\%$. For the worst case within the test range, the relative error is less than $\pm 27\%$.

For all the data tested, the average error is -6.6% . This means that the partition model under-predicts the CHF in the dumbbell-shaped tube. Under-prediction of the CHF in the dumbbell-shaped tube is mainly due to the different heating conditions. As indicated in the previous section, the test section for the experiment was not uniformly heated in reality. The reported experimental CHF is based on the average heat flux over the entire dumbbell-shaped tube. In the case that the CHF occurs first in the narrow gap region, the reported experimental CHF value is higher than the actual heat flux in the narrow gap region. In the case that the CHF occurs first in the main flow channel region, the reported experimental CHF value is lower than the actual heat flux in the main flow channel region. Since the CHF more likely occurs in the narrow gap region, the reported CHF data are higher than the actual heat flux in general.

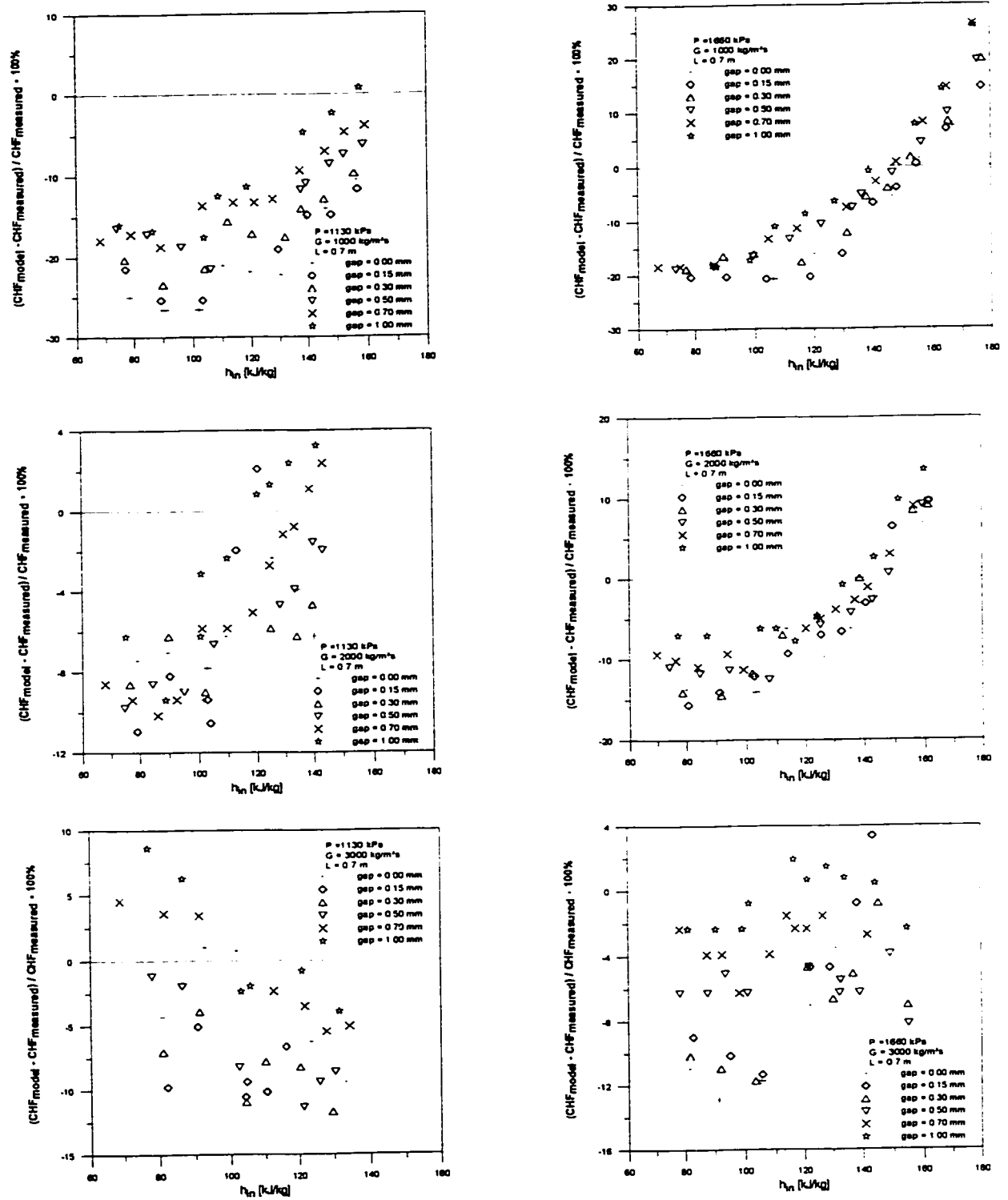


Figure 7.28 Comparison against experimental data at fixed inlet condition.

The model results are based on uniform heating. The predicted CHF always occurs in the narrow gap region and the reported modeling data represent the actual heat flux values at the CHF occurrence locations. Thus, the model data are normally lower than the reported experimental data.

It is also found from the figure that the prediction accuracy of the partition model varies with flow conditions. Since the model is derived from Whalley's annular flow model, all the characters of Whalley's model are inherited. Similar to the R-134a flow in circular tube, the partition model predicts a higher CHF value at higher flow rate than that at lower flow rate. For the moderate flow rate ($G = 2000 \text{ kg/m}^2\text{s}$), the model obtains the most accurate value. At high inlet enthalpy (low inlet subcooling), the predicted CHF value is high.

7.2.4.4 Gap Effect

The comparison between the predicted and measured gap effect (related to 1 mm) is shown in Figure 7.29 and Figure 7.30. The partition model predicts a similar trend in gap effect to that measured. The CHF decreases with decreased gap size. In general, the partition model accurately predicts the gap effect on CHF.

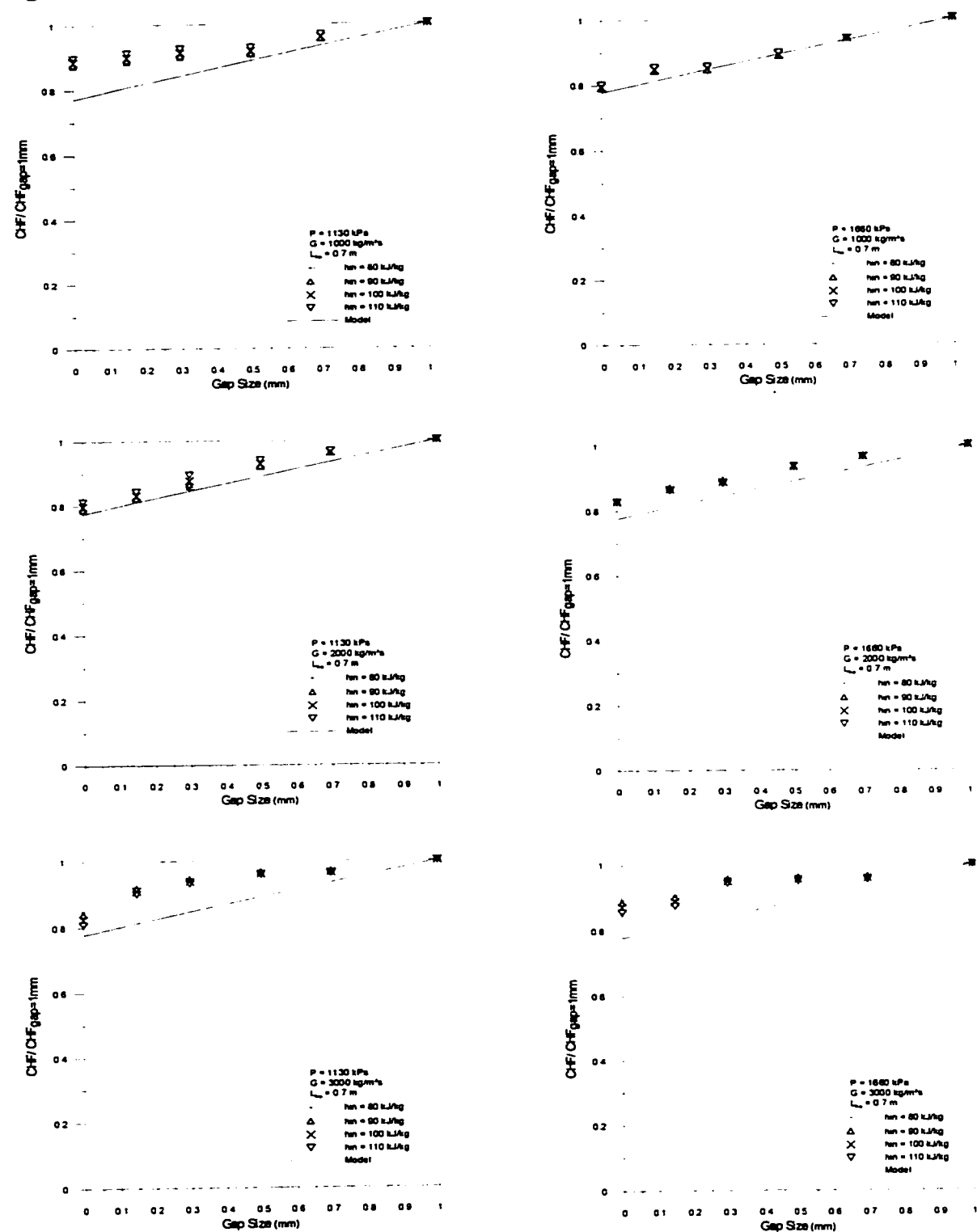
The partition model predicts a stronger gap effect than measured. Under fixed inlet condition, as the gap size decreases from 1 mm to zero, the CHF decreases about 22% as per the partition model due to the overall geometry effect. As described in chapter 6, the measured CHF drops about 17.2% on the average. Thus, the model over-predicts the gap effect about 5% based on fixed inlet condition.

Under the fixed local condition, as the gap size decreases from 1 mm to zero, the partition model predicts an average CHF drop of 37% (excluding the effect of the hydraulic diameter). This value is higher than the measured gap effect (the average CHF drop of 23.5% excluding the effect of the hydraulic diameter). The exception of the low flow and pressure case, which is assumed due to the limiting quality, affects the accuracy of the prediction.

Over-predicting the gap effect is due to the influence of the difference of heating conditions between the model and real experiment condition. The model is based on the assumption of uniform heating while in the experiment the test section is not uniformly heated. For the uniform heating model, CHF always occurred in the narrow gap region first due to the gap effect. For the experiments, as described in Chapter 6, the heat flux in the narrow gap region is lower than that in the main flow channel region. The measured CHF normally occurs first in the main flow channel region for 1 mm gap and in the narrow gap region for zero gap. The measured gap effect is relatively smaller due to a low heat flux in the narrow gap region.

The predicted results from the uniform heating model suggest that the partition model may predict even more accurate gap effects if the heating condition is closer to the actual case. The model results for the non-uniform heating conditions will be discussed in the subsequent section.

Figure 7.29 Comparison of predicted and measured geometry effect at fixed inlet



condition.

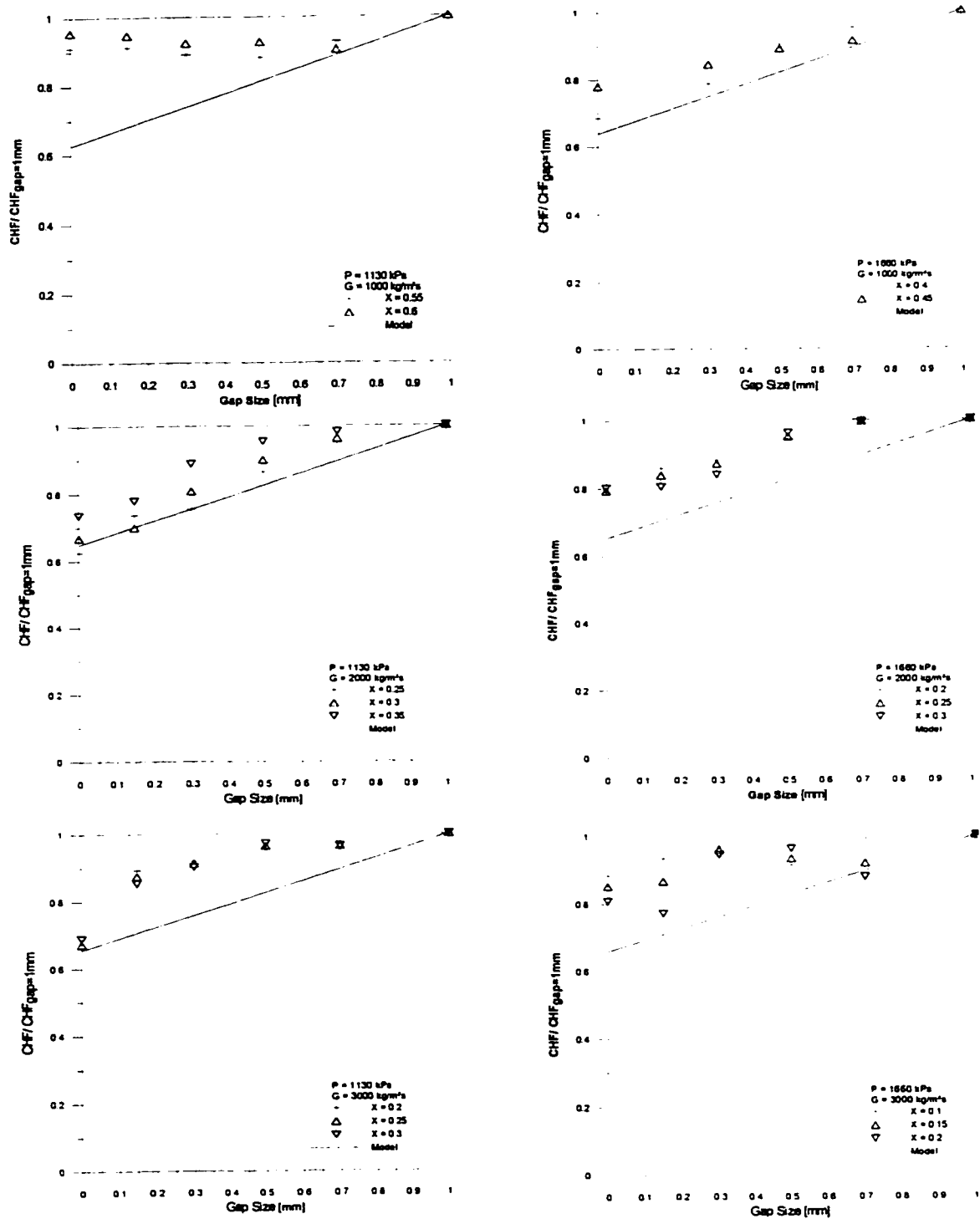


Figure 7.30 Comparison of predicted and measured gap effect at fixed local condition.

7.3 Partition Model Application to Non-uniform Heating Case

7.3.1.1 Partition Model Approach for Non-uniform Heating

As described in the previous section, the heat flux distribution around the periphery of the dumbbell-shaped test section is not uniform. To predict the CHF of the dumbbell-shaped tube more accurately, an approach was developed to take account of the non-uniform heating. By taking the advantage of the partition approach, the model simply applies two different heat fluxes to the two pseudo-channels. The heat flux of each pseudo-channel is the average heat flux of the partition.

As per the computation in section 7.2.4.1, for the dumbbell-shaped tube used in the experiment, the ratio of heat fluxes in the narrow gap and the main flow regions is:

$$\frac{q_g}{q_c} = 0.89$$

where, q_g and q_c are the average heat fluxes for the two pseudo-channels. If the average heat flux of the entire tube q is given, q_g and q_c can be calculated based on the following heat balance equation:

$$q_g l_g + q_c l_c = q l$$

where, l , l_g , l_c are the perimeters for the entire tube and the two pseudo-channels. Note that l_g and l_c vary with gap size.

7.3.1.2 Results and Analysis

Appendix V tabulates the results of the partition model (CHFA++) for the CHF in the dumbbell-shaped tube with non-uniform heating. The comparison with uniform heating as well as the measured data is also shown in the appendix. As shown in the table, the partition model predicts a more accurate CHF by taking account of the non-uniform heat flux.

7.3.1.2.1 CHF Location

As described in the previous section, in the case of uniform heating condition, the predicted CHF always occurs first in the narrow gap region. In the case of non-uniform heating, the predicted

CHF may occur first in the main flow channel region in the cases of big gap. Also, the CHF may occur simultaneously in both the narrow gap region and the main flow channel region. These results are much closer to the measured data.

7.3.1.2.2 *General Trend*

From Appendix V, it is found that the partition model predicts more accurate CHF values in the case of non-uniform heating. The predicted results are closer to the real test data. If compared with the test data, the average relative error of the partition model for the non-uniform heating is -3.2% against -6.6% for the uniform heating.

For non-uniform heating, the CHF was computed using local heat flux values while the final CHF is reported using the average heat flux value over the entire tube. This is exactly the same as the experiment. For the experiment, only the average heat flux can be measured. Similar to the experiment, in the case that CHF occurs first in the narrow gap region, the mean heat flux value (the reported CHF value) is higher than the local heat flux in the narrow gap region. If the CHF occurs first in the main flow channel region, the mean heat flux is lower than the local heat flux in the main flow channel region.

Since the non-uniform model is closer to the actual test condition, the prediction results are more accurate.

7.3.1.2.3 *Gap Effect*

Under non-uniform heating assumption, the model predicts the gap effect more accurately. The comparison of the prediction results between the non-uniform and uniform heating is shown in Appendix V. From the appendix, it is found that the relative errors between non-uniform heating and uniform heating change with the gap size. Generally speaking, the CHF value at zero gap for non-uniform heating is about 4% higher than that for uniform heating. However, at the gap of 1 mm, the difference between the two cases becomes relatively small. This means that when the gap size decreases from 1 mm to zero, the decrease in CHF for the non-uniform heating case is smaller than that for the uniform heating case. Therefore, the CHF ratio between the zero gap

and 1 mm gap cases will be smaller for the non-uniform heating than that for the uniform heating. In other words, the predicted gap effect for the non-uniform heating is about 4% lower than that for uniform heating.

As discussed in the section 7.2.4.4, the uniform heating model over predicted the gap effect about 5%. This implies that the non-uniform model over predicts the gap effect only about 1%. Therefore, it is concluded that predicted gap effect for the non-uniform heating is closer to the experiment results.

7.4 Summary

The partition model has been applied to predict CHF in circular tubes and dumbbell-shaped tubes. As shown in Table 7.4, the partition model predicts the CHF in circular tube and the dumbbell-shaped tube for R-134a flow with good accuracy. For the circular tube, the root mean square error is less than 10% while the average error is about 2%. For the dumbbell-shaped tube, the non-uniform heating results are closer to the experiment results.

Table 7.4 Prediction errors of partition model

Tube \ Error		RMS error	Average error	Mean Error
Circular tube		8.96%	1.99%	7.17%
Dumbbell-shaped tube	Uniform heating	12.67%	-6.63%	10.01%
	Non-uniform heating	10.40%	-3.65%	7.86%

As shown in Figures 7.5 and 7.6, the partition model predicts correct trend of the gap effect on CHF. The predicted CHF decreases with decreasing gap size.

Table 7.5 Summary of modeling results for uniform heating

	Overall Geometry Effect	Pure Gap Effect
At fixed inlet condition	23.0%	11.0%
At fixed local condition	23.5%	35.5%

Table 7.6 Summary of modeling results for non-uniform heating

	Overall Geometry Effect	Pure Gap Effect
At fixed inlet condition	~19%	~7%

Chapter 8

CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE RESEARCH

8.1 Conclusions

8.1.1 Experimental Investigation

The parametric trends in CHF with pressure, mass flux/mass flow rate, quality, and inlet enthalpy/subcooling in the dumbbell-shaped tube are the same as those in the circular tube. No severe adverse effects on CHF were found in the dumbbell-shaped tube for the range of test conditions studied.

At a fixed inlet condition, the CHF decreases with decreasing gap size. As gap size decreases from 1 mm to zero CHF drops by about 17.2% on average. After deducting the flow-area effect, the CHF drop reduces to about 5.2%, assumed to be due to the gap effect alone.

At a fixed local condition, the gap size has a noticeable effect on CHF. In general, the CHF decreases with decreasing gap size. When the gap size decreases from 1 mm to zero, on average, the CHF drops by about 11.5% depending on flow conditions. If the diameter changes are taken into account, the CHF drops by about 23.5%, considering the pure gap effect.

The effect of gap size on CHF is coupled with the effects of flow conditions, but the flow parameters seem to have little influence on the gap effect.

The dumbbell-shaped tube flow visualization test results show that the liquid bridge may form in the gap region in the annular flow regime. In this case, two gas cores are formed in the two subchannels.

8.1.2 Modeling study

The partition model predicts the CHF in a circular tube and the dumbbell-shaped tube for R-134a with good accuracy. For circular tube, the average error is about 2%. For the dumbbell-shaped tube under uniform heating condition, the average error is about -6.6%. For non-uniform heating, the average error is only about -3.5%.

The partition model predicts correct trend of the gap effect on CHF. In the case of non-uniform heating, the partition model predicts the CHF and gap effect more accurately.

8.2 Recommendations for Future Research

8.2.1 Experimental Investigation

The gap effect on CHF was investigated experimentally using a dumbbell-shaped tube. More CHF tests need to be conducted to build a more complete database of the CHF in complex geometries.

- **Curvature effect**

Curvature is another important geometry parameter that may affect the CHF. Further experimental investigation on curvature effect is required to analyze the mechanistic effects of curvature on CHF. Furthermore, the gap effect on CHF is normally coupled with a curvature effect. The coupling of the two effects also needs to be investigated.

- **Circumferential distribution of liquid film**

A measurement of the circumferential distribution of liquid film thickness in the dumbbell-shaped tube is needed as a basis for prediction.

- **Visualization**

In the current study, a simple adiabatic flow visualization test has been conducted to qualitatively determine the flow pattern in a dumbbell-shaped channel. To get

quantitative results and the relationship between the flow patterns and flow parameters, a more sophisticated adiabatic flow visualization test is needed. Flow visualization under uniform heating would help to understand the CHF mechanism in the narrow gap region.

- Control of heating condition

The CHF that occurred first in the main flow region in the dumbbell-shaped tube, during this study, may be due to non-uniform heating. Uniform heating is an ideal condition, but is very difficult to achieve in reality because of limitations in power capacity, manufacturing processes and material quality. In the future experimental investigation using a dumbbell-shaped tube, a better method of controlling the heating condition needs to be considered.

8.2.2 Modeling Study

8.2.2.1 Kernel

The kernel of the partition model consists of the various partition and flow model assumptions. The model could be extended and optimized in the future to:

- CHF other than dryout-type.

Currently, the partition model is limited to dryout-type CHF because of its annular flow assumptions. The partition model could be extended to DNB type CHF in complex flow geometries.

- Deposition and Entrainment Model.

This study used Whalley's deposition and entrainment model. The deposition coefficient was assumed to be a single value function based on surface tension. The partition model could use other models of deposition and entrainment, such as Hewitt and Govan's model (1990), after a comprehensive evaluation.

- Optimization of the deposition factor

In the current study, the deposition factor depends on the shape factor that is a pure geometric parameter. If possible, the deposition could be optimized based on the measurement data of deposition rate distribution.

- Prediction of circumferential distribution of liquid film thickness

In the current model, only the effect of geometry on deposition was considered. Other mechanisms of liquid film spreading are neglected. The model could be enhanced if the geometry effect on the liquid film distribution along the tube wall is predicted. This requires the evaluation of turbulent wave, interfacial shear stress, and surface tension.

8.2.2.2 Application

The partition model is created to handle the CHF in a complex geometry. The present study focuses on the dumbbell-shaped tube. To verify the correctness of the model, its application to a circular tube was also designed and implemented in the software package CHFA++. In the further modeling study of complex geometry CHF, the partition model could be applied to other complex geometries. The shape effect and other geometric parameter effects on CHF could be found based on the partition model.

- **Concentric annuli**

Concentric annuli can be partitioned into two interactive pseudo flow channels, inner and outer pseudo-channels. By taking the advantage of the partition model, the curvature effect on CHF may be obtained. The heating condition effect on CHF can also be found (i.e., single surface heating vs. bilateral heating, different heat flux for the inner surface and outer surface, etc.).

- **Triangular and Rectangular Flow Channels**

By dividing the channel into the main flow and corner pseudo-channels, the corner effect may be obtained using the partition model.

- **Eccentric Annuli**

Like a dumbbell-shaped tube, eccentric annuli could be partitioned into a gap and main flow channel. The coupling of the gap and curvature effects may be found using the partition model.

- **Fuel Bundles**

The optimal goal of this research on geometry effects is to better predict the CHF in fuel-rod bundles. Merging the partition model (or its methodology) into a subchannel code could result in a better prediction of bundle CHF.

REFERENCES

- Andersen, P. S., Jensen, A., Mannov, G. and Olsen, A., 1974, "Burn-out, Circumferential Film Distribution and Pressure Drop for an Eccentric Annulus with Heated Rod," *Int. J. Multiphase Flow*, Vol. 1, pp. 585 -600.
- Andreyevskiy, A.A., Borishanskiy, V.M., Bykov, G.S. and Shleyfer, V.A., 1978, "Threshold Steam Qualities and Cut off Heat Fluxes in Two-Phase Channel Flows," *Heat Transfer-Soviet Research*, Vol.10, No.4, July-August.
- Aoki, S., Inoue, A., Aritomi, M. and Sakamoto, Y., 1982, "Experimental Study on the Boiling Phenomena within a Narrow Gap," *Int. J. Heat Mass Transfer*, Vol. 25, No.7, pp. 985-990.
- Babrin, V. P., Sevast'yanov, R. I. and Alad'yev, I. T., July 1969, "A Special Hydrodynamic Effect on the Boiling Crisis in Tubes," *Heat Transfer - Soviet Research*, Vol. 1, No. 4, pp. 34-48.
- Balino, J.L. and Converti, 1994, "Saturated CHF in Horizontal Eccentric Annuli for Low Mass Fluxes: Measurement and Modelling," *Int. J. Multiphase Flow*, Vol.20, No.5, pp.901-913.
- Barnet, P.G., 1968, "A Comparison of the Accuracy of Some Correlations for Burnout in Annuli and Rod Bundles," AEEW-R-588, Atomic Energy Establishment, Winfrith.
- Barnet, P.G., 1966, "A Correlation of Burnout Data for Uniformly Heated Annuli and Its Use for Predicting Burnout in Uniformly Heated Rod Bundles," AEEW-R-463, Atomic Energy Establishment, Winfrith.
- Barnet, P.G., 1963, "An Investigation into the Validity of Certain Hypothesis Implied by Various Burnout Correlations," AEEW-R-214, Atomic Energy Establishment, Winfrith.
- Becker, K.M, and Letzter, A., 1975, "Burnout Measurements for Flow of Water in Annulus with

Two Sides Heating,” Paper Presented at the European Two-Phase Flow Group Meeting in Haifa, Israel.

Becker, K.M, 1967, “An Analytical and Experimental Study of Burnout Conditions in Vertical Round Ducts,” *Nukleonik*, Vol.9, No.6, pp.257-270

Becker, K.M, Hernborg, G., Bode, M., and Erikson, O., 1965, “Burnout Data for Flow Boiling Water in Vertical Round Ducts, Annuli and Rod Clusters,” AE-177.

Becker, K.M, and Hernborg, G., 1964, “Measurements of Burnout Conditions for Flow of Boiling Water in a Vertical Annulus,” *Journal of Heat Transfer*, Vol.86, pp.393-407.

Bergles, A. E., 1992, “What is the Real Mechanism of CHF in Pool Boiling?” in *Pool and External Flow Boiling*, V.K.Dhir and A.E.Bergles eds., pp.165-170, ASME, New York.

Bergles, A. E., 1979, “Burnout in Boiling Heat Transfer. Part III: High-Quality Forced-Convection Systems,” *Nuclear Safety*, Vol. 20, No.6, pp.671-689.

Bergles, A. E., Lopina, R. F and Fiori, M. P., 1967, “Critical-Heat-Flux and Flow Pattern Observations for Low-Pressure Water Flowing in Tubes,” *Journal of Heat Transfer. Transactions of the ASME*, pp. 69-74.

Bergles, A. E., 1963, “Subcooled Burnout in Tubes of Small Diameter,” ASME Winter Annual Meeting, Philadelphia, paper number 63-WA-182.

Bobkov, V.P., Kozina, N. V., Vinogradov, V.N. and Zyatina, O.A., 1995, “New Model for Burnout Prediction in Channels of Various Cross-Section,” *Proceedings of the 7th International Meeting on Nuclear Reactor Thermal-Hydraulics NURETH-7*, Saratoga Springs, New York, pp.2539-2557.

Bobkov, V. P., et al, February 1994, "Effect of Narrow Gaps on Critical Heat Flux," Report of Heat Mass Transfer Information Center, Institute of Physics and Power Engineering, Obninsk, Russian Federation.

Bobkov, V. P., "New Model for Burnout Prediction in Channels of Various Cross-Section," Report of State Research Center, Institute of Physics and Power Engineering, Obninsk, Russian Federation. (unpublished)

Boltienko, E.A., Pomietko, R.S., Katan, I.B. and Ziatnina, O.A., 1989, "Effect of Diameter on CHF," (Translated by Doerffer S.S, Chalk River , April 1991).

Borishanskiy, V. M. and Fokin, B. S., 1969, "Onset of Heat-Transfer Crisis with Unsteady Increase in Heat Flux," Heat Transfer-Soviet Research, Vol.1, No.5, pp. 27-55.

Bowditch, F.H. and Lee, D.H., 1973, "An Experimental Investigation into Dryout Should an SGHWR Rod Bow into Contact with a Pressure Tube," AEEW-R-903, Atomic Energy Establishment, Winfrith.

Butterworth, D., 1971, "A Model for Prediction Dryout in a Tube with a circumferential Variation in Heat Flux," AERE-M-2436, Atomic Energy Research Establishment.

Butterworth, D. and Hewitt, G.F., 1977 "Two-Phase Flow and Heat Transfer," Oxford University Press.

Celata, G. P., Cumo, M., Mariani, A. and Zummo, G., 1995, "The Prediction of the Critical Heat Flux in Water Subcooled Flow Boiling," Int. J. Heat Mass Transfer, Vol. 38, No. 6, pp. 1111-1119.

Celata, G. P., Cumo, M., Mariani, A., Simoncini, M. and Zummo, G., 1994a, "Rationalization of Existing Mechanistic Models for the Prediction of Water Subcooled Flow Boiling Critical Heat

Flux,” *Int. J. Heat Mass Transfer*, Vol. 37, Suppl.1. pp.347-360.

Celata, G. P., Cumo, M., Mariani, A. and Zummo, G., 1994b, “Assessment of Correlation and Models for the Prediction of CHF in Water Subcooled Flow Boiling,” *Int. J. Heat Mass Transfer*, Vol. 37, No.2, pp.237-255.

Celata, G. P., Cumo, M. and Mariani, A., 1993, “Burnout in highly Subcooled Water Flow Boiling in Small Diameter Tubes,” *Int. J. Heat Mass Transfer*, Vol. 36, No.5, pp.1269-1285.

Celata, G. P., Cumo, M., D'Annibale, F., Farello, G. E. and Abou Said, S., 1988, “Critical Heat Flux Phenomena in Flow Boiling During Step-Wise and Ramp-Wise Power Transients,” *Rev. Gen Therm. Fr.*, No. 317, pp. 296-303.

Cengel, Y.A., and Boles, M.A., 1994, *Thermodynamics – An Engineering Approach*, 2nd edition, McGraw-Hill Inc.

Chernobay, V. A., 1970, “Critical Heat Loads in a Rectangular Duct with a Nonuniformly Heated Perimeter,” *Heat Transfer-Soviet Research*, Vol. 2, No.1, pp. 167-169.

Collier, J.R. and Thome, J. R., 1994, “*Convective Boiling and Condensation*, 3rd Edition,” Oxford University Press.

Dhir, V.K., 1996, “Pool Boiling Heat Transfer: Recent Advance and Expectations for the Future,” in *Process, Enhanced and Multiphase Heat Transfer: A Festschrift for A.E. Bergles*, R. M. Manglik and A.D. Kraus eds., pp. 99-126, Begell House, New York.

Doerffer, S., Groeneveld, D.C and Cheng, S.C., 1995, “A Comparison of Critical Heat Flux in Tubes and Bilaterally Heated Annuli,” *Proceedings of the 7th International Meeting on Nuclear Reactor Thermal-Hydraulics NURETH-7*, Saratoga Springs, New York, pp.2583-2607.

- Doerffer, S., Groeneveld, D.C., Cheng, S.C. and Rudzinski, K. F., 1994, "A Comparison of Critical Heat Flux in Tubes and Annuli," *Nuclear Engineering and Design* 149, pp.167-175.
- Doerffer, S. and Cheng, S. C., August 1994, "The Effect Of Flow Cross-Section on the Critical Heat Flux - CHF Characteristics in Concentric and Eccentric Annuli with Uni- and Bilateral Heating," Technical Report, University of Ottawa, Dept. of Mechanical Engineering.
- Doerffer, S., Groeneveld, D. C., Tain, R., Cheng, S. C. and Zeggle, W., 1991, "Fluid-to-Fluid Modelling of the Critical Heat Flux in Simple and Complex Geometries," *Trends in Heat, Mass & Momentum Transfer*, 1, pp. 45-64
- Doroshchuk, V. Y., Levitan, L. L. and Lantsman, F. P., 1980, "Boiling Crisis in Evaporation Tubes," *Heat Transfer-Soviet Research*, Vol. 12, No. 6, pp. 21-31
- El-Genk, M.S., Hayness, S.J., and Ki, S.H., 1988, "Experimental studies of critical heat flux for low flow of water in vertical annuli at near atmospheric pressure," *Int. J. Heat Mass Transfer* 31 (11), 2291-2304.
- Fukano, T., and Ousaka, A., 1989, "Predication of the Circumferential Distribution of Film Thickness in Horizontal and Near-Horizontal Gas-Liquid Annular Flows," 1989, *Int. J. Multiple Phase Flow*, Vol.15, No.3, pp.403-419.
- Fukano, T., Goto, A., Tsurusaki, Y., Morooka, S., 1996, "Dryout of Water Film on a Heated Tube Surface caused by an Obstruction in a Boiling Two-phase Vertical Upward Flow," *Chem. Eng. Comm.*, Vols. 141-142, pp. 191-206.
- Fukano, T., Nakagawa, H., Mori, Y., and Watanabe, 1997, "Liquid Film Formation on the Inner Surface of the Horizontal Channel", *Nuclear Engineering and Design*, 175, pp 3-13.
- Fukano, T., and Fukukawa, T., 1998, "Prediction of the Effects of Liquid Viscosity on Interfacial

- Shear Stress and Frictional Pressure Drop in Vertical Upward Gas-Liquid Annular Flow”, *Int. J. Multiphase Flow*, Vol.24, No.4, pp.587-603.
- Fukano, T., Egashira, R., and Naitoh, K., 1999, “Upward Two-Phase Flow in a Vertical Annular Channel (Effect of Spacer Geometry),” *Heat Transfer Research*, Vol.30, pp.411-421.
- Fujita, Y., Ohta, H., Uchida, S, and Nishikawa, K., 1988, “Nucleate Boiling Heat Transfer and Critical Heat Flux in Narrow Space Between Rectangular Surfaces,” *Int. J. Heat Mass Transfer*, Vol.31, No.2, pp. 229-239.
- Gambill, W. R., 1963, “Generalized Prediction of Burnout Heat Flux for Flowing, Subcooled, Wetting Liquid,” *Chemical Engineering Progress Symposium Series, Heat Transfer-Houston*.
- Gambill, W. R., Bundy, R. D. And Wansbrough, R. W., 1960, “Heat Transfer, Burnout, and Pressure Drop for Water in Swirl Flow Through Tubes with Internal Twisted Tapes,” *Chemical Engineering Progress Symposium Series, Heat Transfer-Buffalo*.
- Gallow, J. E. And Mudawar, I, 1993, “CHF mechanism in Flow Boiling From a Short Heated Wall - I. Examination of Near-Wall Conditions with the Aid of Photomicrography and High-Speed Video Imaging,” *Int. J. Heat Mass Transfer*, Vol. 36, No.10, pp.2511-2526.
- Gallow, J. E. And Mudawar, I, 1993, “CHF mechanism in Flow Boiling From a Short Heated Wall - II. Theoretical CHF Model,” *Int. J. Heat Mass Transfer*, Vol. 36, No.10, pp.2527-2540.
- Goldmann, K., Firstenberg, H. and Lombardi, C., May 1961, “Burnout in Turbulent Flow - A Droplet Diffusion Model,” *Transaction of ASME , Journal of Heat Transfer*, pp.158-162.
- Govan, A.H., Hiwitt, G.F., Owen D.G., and Bott, T.R., 1988, “An Improved CHF Modelling Code,” 2nd U.K. National Heat Transfer Conf., Glasgow.

- Green, W. J. and Beattie, D.R.H., 1982, "An Investigation Into Nature and Significance of a New Critical Heat Flux Correlation," Nuclear Engineering and Design, pp.33-41.
- Griffel, J. and Bonilla, C. F., 1965, "Forced-Convection Boiling Burnout for Water in Uniformly Heated Tubular Test Section," Nuclear Structure Engineering 2, pp. 1-35
- Groeneveld, D. C., Doerffer, S.D., and Tain R.M., et al., 1997, "Fluid-to-Fluid Modelling of the Critical Heat Flux and Post-dryout Heat Transfer," Proc. of the 4th World Congress on Experimental Heat Transfer, Fluid Mechanics and Thermodynamics, Brussels, June 2-6, pp.859-867.
- Groeneveld, D.C., Leung, L.K.H., Kirillov, P.L. et al., 1996, "The 1995 Look-Up Table for Critical Heat Flux in Tubes," Nucl. Eng. Des. 163, pp.1-23.
- Groeneveld, D. C., Joobar, K., Leung, L.K.H., Doerffer, S., et al., 1992, "Effect of Fuel Subchannel Geometry on CHF," Proc. 5th, Int. NURETH Conf., Salt Lake City, pp.683-690.
- Groeneveld, D. C., Joobar, K., Doerffer, S., Wong, W., Leung, L.K.H. and Cheng, S.C., 1992, [The Effect of Fuel Subchannel Geometry on CHF], Proceedings of NURETH-5, Salt Lake City, Utah, Vol. 3, pp. 683-690.
- Groeneveld, D. C., Joobar, K., Leung, L.K.H., Doerffer, S., Rudzinski, K.F., and Wong W., 1992, "Effect of Flow Geometry on Critical Heat Flux: Phase I: Observed Effects and Preliminary Prediction Method for Subchannels," AECL report ARD-TD-338, COG-91-286, (unpublished).
- Groeneveld, D.C., Cheng, S.C. and Doan, T., 1986, "1986 AECL-UO Critical Heat Flux Lookup Table," Heat Transfer Engineering, vol.7 nos.1-2, pp.46-62
- Haramura, Y. and Katto, Y., 1983, "A New Hydrodynamic Model of Critical Heat Flux,

Applicable Widely to Both Pool and Forced Convection Boiling on Submerged Bodies in Saturated Liquids," *Int. J. Heat Mass Transfer*, Vol. 26, No.3, pp.389-399.

Herer, Ch., 1995,"A Comparison of CHF Between Tubes and Annuli under PWR Thermal-Hydraulic Condition," *Proceedings of the 7th International Meeting on Nuclear Reactor Thermal-Hydraulics NURETH-7*, Saratoga Springs, New York, pp.2519-2538.

Herrero, V.A., Guido-Lavalle, G. and Clausse, A., 1995, "Void Fraction Distribution in a Heated Rod Bundle under Flow Stagnation Conditions," *Proceedings of the 7th International Meeting on Nuclear Reactor Thermal-Hydraulics NURETH-7*, Saratoga Springs, New York, pp.2762-2771.

Hetsroni, G., 1982, "Handbook of Multi-Phase Systems," Hemisphere Pub. Corp.

Hewitt, G.F. and Govan, A.H., 1990, "Phenomenological Modeling of Non-Equilibrium Flows with Phase Change," *Int.J. Heat Mass Transfer*, Vo.33, No.2, pp.229-242.

Hewitt, G.F. and Hall-Taylor, N.S., 1970, "Annular Two-Phase Flow," Pergamon Press.

Hill,K.W., Motley,F.E., Cadek,F.F. and Casterline, J.E., 1975, "Effect of a Rod Bowed to Contact on Critical Heat Flux in Pressurized Water Reactor Rod Bundles,"an ASME publication, the Winter Annual Meeting. Huston, Texas

Hung, Y. H. and Yao, S. C., 1983, "Critical Heat Flux of Convective Freon-113 in Very Narrow Annular," ASME Paper, No. 83-HT-10.

Isbin, H.S., Vanderwater,R., Fauske, H. and Singh, S., May 1961, "A Model for Correlating Two-Phase, Stem-Water, Bourn-out Heat Transfer Fluxes," ASME J. of Heat Transfer, pp.149-157.

Israel, S., Casterline, J. and Matzner, B., 1969, "Critical Heat Flux Measurements in a 16-Rod

- Simulation of a BWR Fuel Assembly,” ASME Journal of Heat Transfer, Vol.91, pp.355-363.
- Jacket, H.S., Roarty, D.J., Serbe, J.E., 1958, “Investigation of Burnout Heat Flux in Rectangular Channels at 2000 Psia,” Transactions of ASME, No.2, pp.391-401.
- Jafri, T., Dougherty, T.J., and Yang, B.W., 1995, “Correlation of Critical Heat Flux Data for Uniform Tubes,” Proceedings of the 7th International Meeting on Nuclear Reactor Thermal-Hydraulics NURETH-7, Saratoga Springs, New York, pp.3197-3217.
- Janssen, E., and Kervinen, J.A., 1964, “The Critical Boiling Heat Flux at High Flows and High Qualities,” APED-4554, Atomic Power Equipment Department.
- Janssen, E., and Kervinen, J.A., 1963, “Burnout Conditions for Single Rod in Annular Geometry, Water at 600 to 1400 PSIA,” GEAP-3899, General Electric Atomic Power Equipment Department.
- Janssen, E., Levy, S., and Kervinen, J.A., 1963, “Investigations of Burnout in Internally Heated Annulus Cooled by Water at 600 to 1450 PSIA,” ASME Paper No. 63-WA-149.
- Jayanti, S., Hewitt, G.F., White, S.P., 1990, “Time-dependent Behavior of the Liquid Film in Horizontal Annular Flow,” Int. J. Multiphase Flow 16, pp.97-116.
- Jayanti, S., Hewitt, G.F., 1993, “Response of Turbulent Flow to Abrupt Changes in Surface Roughness and its Relevance in Horizontal Annular Flow,” Proc. First International CFDS User Congerence, pp.1-19.
- Joobar, K., 1993, “The Effect of Flow Geometry on Critical Heat Flux,” University of Ottawa, Dept. of Mechanical Engineering, M.A.Sc. Thesis.
- Kalliadasis, S., Chang, H., 1996, “Dynamics of Liquid Spreading on Solid Surfaces,” Ind. Eng.

Chem 35, pp.2860-2874.

Kamei, T., and Serizawa, A., 1998, "Measurement of 2-dimensional local instantaneous liquid film thickness around simulated nuclear fuel rod by ultrasonic transmission technique", Nuclear Engineering and Design 184 (2-3) pp. 349-362.

Katto, Y., 1994, "Critical Heat Flux," Int. J. Multiphase Flow, Vol.20, Suppl., pp53-90.

Katto, Y., 1992a, "Critical Heat Flux in Pool Boiling," in Pool and External Flow Boiling, V.K.Dhir and A.E.Bergles eds., pp.151-164, ASME, New York.

Katto, Y., 1992b, "A Prediction Model of Subcooled Water Flow Boiling CHF for Pressure in the Range 0.1-2.0 MPa," Int. J. Heat Mass Transfer, Vol. 35, No.5, pp.1115-1123.

Katto, Y., 1990a, "Prediction of Critical Heat Flux of Subcooled Flow Boiling in Round Tubes," Int. J. Heat Mass Transfer, Vol.33, No.9, pp. 1527 - 1542.

Katto, Y., 1990b, "A Physical Approach to Critical Heat Flux of Subcooled Flow Boiling in Round Tubes," Int. J. Heat Mass Transfer, Vol.33, No.4, pp. 611-620.

Katto, Y. and Ohno, H., 1984, "An Improved Version of the Generalized Correlation of Critical Heat Flux for Convective Boiling in Uniformly Heated Vertical Tubes," Int. J. Heat Mass Transfer, Vol 27, pp. 1641-1648.

Katto, Y., 1981, "Generalized Feature of CHF Forced Convection Boiling in Uniformly Heated Rectangular Channels," Int. J. Heat Mass Transfer, Vol. 24, pp. 1413 - 1419.

Katto, Y., 1979a, "Generalized Correlations of Critical Heat Flux for the Forced Convection Boiling in Vertical Uniformly Heated Annuli," Int. J. Heat Mass Transfer, Vol. 22, pp. 575-584

- Katto, Y., 1979b, "A Generalized Correlation of Critical Heat Flux for the Forced Convection Boiling in Vertical Uniformly Heated Round tubes - A Supplementary Report," *Int. J. Heat Mass Transfer*, Vol. 22, pp. 783 - 794.
- Katto, Y., 1979c, "An Analysis of the Effect of Inlet Subcooling on Critical Heat Flux of Forced Convection Boiling in Vertical Uniformly Heated Round tubes," *Int. J. Heat Mass Transfer*, Vol. 22, pp. 1567 - 1575.
- Katto, Y., 1978, "A Generalized Correlation of Critical Heat Flux for the Forced Convection Boiling in Vertical Uniformly Heated Round tubes," *Int. J. Heat Mass Transfer*, Vol. 21, pp. 1527 - 1542.
- Kenning, D.B.R., 1977, "Pool Boiling," in *Two-Phase Flow and Heat Transfer*, edited by Butterworth and Huwitt, Oxford University Press, pp. 128-152.
- Kirillov, P. L. and Smogalev, I.P., 1995, "On the Look-Up Tables for the Critical Heat Flux in Tubes (History and Problems)," *Proceedings of the 7th International Meeting on Nuclear Reactor Thermal-Hydraulics NURETH-7*, Saratoga Springs, New York, pp. 2558-2582.
- Kirillov, P. L. and Smogalev, I. P., 1972, "Calculation of Heat-Transfer Crisis for Annular Two-Phase Flow of a Steam- Liquid Mixture Through an Annular Channel," Translated from report FEI-297 of the Physics and Power-Engineering Institute, Obninsk, USSR. AECL Report-4752. February 1974.
- Kitto, J.B., 1980, "Critical Heat Flux and the Limiting Quality Phenomenon," *AIChE Symposium, Heat Transfer* 76, pp. 57-78.
- Kozhelupenko, Yu. D., Smirnov, G. F. and Koba, A. L., 1985, "Boiling Crisis (Dryout) in Low-Velocity Flow of Subcooled Water in Narrow Annuli," *Heat Transfer-Soviet Research*, Vol. 17, No. 5, pp. 93-96.

- Kruzhilin, G. N. and Labuntsov, D. A., 1969, "Boiling Heat Transfer Crisis for Flow Water in Forced Flow Through Channels," Heat Transfer-Soviet Research, Vol. 1, No. 6, pp.30-39.
- Laurinat, J.E., Hanratty, T.J., Jepson, W.P., 1985, "Film Thickness Distribution for Gas-Liquid Annular Flow in a Horizontal Pipe," PCH PhysicoChem Hydrodynam, 6., pp.197-205.
- Lee, C. H. and Mudawwar, I., 1988, "A Mechanism Critical Heat Flux Model for Subcooled Flow Based on Local Bulk Flow Conditions," Int. J. Multiphase Flow, Vol.14, No.6, pp.711-728.
- Lee, D.H., and Little, R.B., 1962, "Experimental Studies into the Effect of Rod Spacing on Burnout in a Simulated Rod Bundle," UK Rep. AEEW, Atomic Energy Establishment, Winfrith, England.
- Lee, K.H. and Lee, S.Y., 1995, "An Investigation of Transition Boiling Mechanisms of Subcooled Water under Forced Convective Conditions," Proceedings of the 7th International Meeting on Nuclear Reactor Thermal-Hydraulics NURETH-7, Saratoga Springs, New York, pp.2489-2505.
- Levy, S., J.M. Healzer, and D. Abdollahian, 1980, "Prediction of Critical Heat Flux for Annular Flow in Vertical Pipes," EPRI NP-1619, EPRI, Palo Alto, CA.
- Levy, S., Polomik, E. E., Swan, C. L. and Mckinney, A. W., 1962, "Eccentric Rod Burnout at 1000 lbf/in² with Net Steam Generation," Int. J. Heat Mass Transfer, Vol.5, pp.595-614.
- Levy, S., Fuller, R. A. and Niemi, R. O., 1959, "Heat Transfer to Water in Thin Rectangular Channels," ASME Journal of Heat Transfer, pp.129-143
- Lim, J. C. and Weisman, J., 1990, "A Phenomenologically Based Prediction of the Critical Heat

Flux in Channels Containing an Unheated Wall,” *Int. J. Heat Mass Transfer*, Vol. 33, No.1, pp.203-205.

Lim, J. C. and Weisman, J., 1988, “A Phenomenologically Based Prediction of Rod-Bundle Dryout,” *Nuclear Engineering and Design* 105, pp.363-371.

Lombardi, C., 1995, “A Formal Approach for the Prediction of the Critical Heat Transfer in Subcooled Water,” *Proceedings of the 7th International Meeting on Nuclear Reactor Thermal-Hydraulics NURETH-7*, Saratoga Springs, New York, pp.2506-2518.

Macbeth, R.V., 1963, “Burn-out Analysis. Part 4: Application of a Local Conditions Hypothesis to World Data for Uniformly Heated Round Tubes and Rectangular Channels,” Report No. UKAEA-AEEW-R 267, Reactor Group, Reactor Development Division.

Marinelli, V., 1977, “Critical Heat Flux: A review of Recent Publications,” *Nuclear Technology*, Vol. 34, pp.135-170.

Markowski, E.S., Lee, L., Biderman, R., and Casterline, J.E., 1977, “Effect of Rod Bowing on CHF in PWR Fuel Assemblies”, ASME Paper No. 77-TH-91.

McPherson, G. D. and Ahmad, S. Y., 1971, “Fluid Model of Critical Heat Flux in an 18-Element Bundle,” *Nuclear Engineering and Design*, p.409-427.

Merilo, M., 1977, “Critical Heat Flux Experiments in a Vertical and Horizontal Tube With Both Freon-12 and Water as Coolant,” *Nuclear Engineering and Design*, pp.1-16.

Mihail, R. and Straja, S., 1986, “A Theoretical Model Concerning Bubble Size Distributions,” *The Chemical Engineering Journal*, pp.71-77.

Mishima, K. and Nishihara, H., 1985, “The Effect of Flow Direction and Magnitude on CHF for

Low Pressure Water in the Rectangular Channels,” Nuclear Engineering and Design, pp.165-181.

Mitsutake, T., Terasaka, H., Yoshimura, K., Oishi, M., Inoue, A. and Akiyama, M., 1990, “Subchannel Analysis of a Critical Power Test, Using Simulated BWR8*8 Fuel Assembly,” Nuclear Engineering and Design 122, pp.235-254.

Moeck, E.O., Matzner, B., Casterline, J.E., and Yuill, G.K., 1966, “Critical Heat Fluxes in Internally Heated Annuli of Large Diameter Cooled by Boiling Water at 1000 PSIA,” Proceedings of the 3rd International Heat Transfer Conference, Vol. III, pp.86-97, A.I.Ch.E., New York

Nagino, Y., Shodai, N. Matsumoto, C., Shodai, N., and Ueji, M., 1978, “Rod Bowed to Contact Departure From Nucleate Boiling Tests in Cold Wall Thimble Cell Geometry”, Journal of Nuclear Science & Technology, vol.15, no.8, pp.568-73.

Nariai, H., et al, 1989, “Critical Heat Flux in Narrow Tubes with Uniform Heating,” Heat Transfer: Japanese Research, Vol. 18, pp21-30.

Ornatskiy, A. P., Chernobay, V. A., Vasil'yev, A. F. and Struts, G.V., 1977a, “Variation in the Critical Heat Flux in the case of Very Nonuniform Heat Release along an Annulus,” Heat Transfer-Soviet Research, Vol. 9, No. 1, pp.105-111.

Ornatskiy, A. P., Chernobay, V. A., Vasil'yev, A. F. and Perkoy, S. V., 1977, “Boiling Crisis in Annuli with Different Lengthwise Heated Flux Distributions,” Heat Transfer-Soviet Research, Vol. 9, No. 3, pp.1-8.

Ornatskiy, A. P., 1969a, “The Effect of Basic Regime Parameters and Channel Geometry on Critical Heat Flux in Forced Convection of Subcooled Water,” Heat Transfer-Soviet Research, Vol. 1, No. 3, pp.17-22.

Ornatskiy, A. P., Glushchenko, L. F. and Kichigin, A. M., 1969b, "Burnout in Annular Channels with Two-Side Heating," Heat Transfer-Soviet Research, Vol. 1, No. 6, pp.12-16.

Ornatskiy, A. P., Chernobay, V. A., Lazarev, N. A. and Puraev, V. S., 1969c, "Boiling Crisis in Annuli with Different Lengthwise Heated Flux Distributions," Teploenergetika.

Ornatskiy, A. P., Chernobai, V. A., Lazarev, N. A. and Furaev, V. S., 1969d, "Investigation of the Influence of Eccentricity on burnout Heat Transfer in Annular Channels," Teploenergetika

Park, J., Baek, W., and Soon H.C., 1997, "Critical heat flux and flow pattern for water flow in annular geometry," Nuclear Engineering and Design 172 (1-2) pp. 137-155.

Pei, B. S., Chen, Y. B., Shih, C. and Lin, W. S., 1986, "A Further Evaluation of Contact Effects on CHF's for PWR Fuel Elements," 2nd International Topic Meeting on Nuclear Power Plant Thermal Hydraulics and Operation, April 1986, Tokyo, Japan

Pernica, R. and Cizek, J., 1995, "General Correlation for Prediction of Critical Heat Flux Ratio in Water Cooled Channels," Proceedings of the 7th International Meeting on Nuclear Reactor Thermal-Hydraulics NURETH-7, Saratoga Springs, New York, pp.2636-2653.

Rehme, Klaus, 1992, "The Structure of Turbulence in Rod Bundles and the Implications on Natureal Mixing Between the Subchannels," Int J. Heat Mass Transfer, Vol.35, No.2, pp.567-581.

Rogers, J. T., 1966, "Heat Transfer Limitation on Nuclear reactor Fuel Elements," The Canadian Journal of Chemical Engineering, pp.289-298.

Rohsenow, W.M., Hartnett, J.P. and Cho, Y.I., 1997, "Handbook of Heat Transfer," Third Edition, McGraw-Hill.

- Rumbaugh, J., Blaha, M., Premerlani, W., and Lorensen, W., 1995, "Object-Oriented Modeling and Design," New York, Prentice Hall Inc.
- Ryabov, A.N., et al, 1977, "Boiling Crisis and Pressure Drop in Rod Bundles with Heated Transfer Enhancement Devices," Heat Transfer-Soviet Research, Vol.9, No.1, pp.112-122.
- Sadatomu M., Kawahara, A., Sato, Y., 1996, "Prediction of the Single-Phase Turbulent Mixing Rate Between Two Parallel Subchannels Using a Subchannel Geometry Factor," Nuclear Engineering and Design 162, pp.245-256.
- Sadatomu, Y., Sato, Y., Saruwatari, S., 1982, "Two-phase Flow in Vertical Noncircular Channels," Int. J. Multiphase Flow 8, pp641-655.
- Saito, T., Hughes, E.D. and Carbon, M.W., 1978, "Multi-Fluid Model of Annular Two-Phase Flow," Nuclear Engineering and Design, Vol. 50, pp225-271.
- Sudo, Y. and Kaminaga, 1993, "A New CHF Correlation Scheme Proposed for Vertical Rectangular Channels Heated From Both sides in Nuclear Research Reactors," ASME Journal of Heat Transfer, pp.426-434.
- Sudo, Y. and Kaminaga, 1989, "A CHF Characteristic for Downward Flow in a Narrow Vertical Rectangular Channels Heated From Both Sides," Int. J. Multiphase Flow, Vol.15, No.5, pp.755-766.
- Sugawara, S., 1990a, "Analytical Prediction of CHF by FIDAS Code Based on Three-Fluid and Film-Dryout Model," Nuclear Science and Technology, 27[1], pp.12-99.
- Sugawara, S. and Miyamoto, Y., 1990b, "FIDAS: Detailed Subchannel Analysis Code Based on the Three-Fluid and Three-Field Model," Nuclear Engineering and Design 120, pp.147-161.

Tain, R.M., 1994, "An Investigation of CHF Fluid-to-Fluid Scaling and Multi-Fluid Prediction Techniques," University of Ottawa, Dept. of Mechanical Engineering , Ph.D. Thesis.

Tain, R. M., Groenveld, D. C., and Cheng, S. C., 1994, "Limitation of the Fluid-to-Fluid Scaling Technique for Critical Heat Flux in Flow Boiling," *Int. J. Heat Mass Transfer*, Vol. 38, No. 12, pp. 2195-2208.

Theofanous, T. G., 1980, "The Boiling Crisis in Nuclear Reactor Safety and Performance," *Int. J. Multiphase Flow*, Vol.6, pp.69-95.

Tippets, F.E., February 1964, "Critical Heat Flux and Flow Patterns in High-Pressure Boiling Water Flows," *Transaction of ASME, J. of Heat Transfer*, pp.12-22.

Tippets, F.E., February 1964, "Analysis of the Critical Heat-Flux Condition in High-Pressure Boiling Water Flows," *Transaction of ASME, J. of Heat Transfer*, pp.23-38.

Tolubinskiy, V. I., Domashev, Ye. D., Litoshenko, A. K. and Matorin, A. S., 1977, "Boiling Crisis in Concentric and Eccentric Annuli," *Heat Transfer-Soviet Research*, Vol. 9, No.1

Tolubinskiy, V. I., Ostrovskiy Yu. N. and Pisarev, V. Ye., 1977, "Effect of Heater Diameter on the Transient Critical Heat Flux," *Heat Transfer-Soviet Research*, Vol. 9, No.2

Tolubinskiy, V. I., Litoshenko, A. K., Shevtsov, V. L., Domashev, Ye.D. and Struchenko, G.Ye., May-June 1973, "Heat Transfer Crisis at the Inward Surface of Annuli Heated from Both Sides," *Heat Transfer-Soviet Research*, Vol. 5, No.3, pp93-96.

Tolubinskiy, V. I., Litoshenko, A. K. and Shevtsov, V. L., January-February 1973, "Effect of Curvature on the Critical Heat Flux Density in Forced-Convection Flow," *Heat Transfer-Soviet Research*, Vol. 5, No.1, pp21-25.

Tolubinskiy, V. I., Litoshenko, Domashev, E. D., A. K. and Chirov, D. A., 1971, "Critical Heat Flux in Eccentric Annuli," *Teplopenergetika*, pp.93-95.

Tolubinskiy, V. I., Mazka, S. A. and Vasil'Yev, A. A., 1970, "Boiling Crisis in Longitude Flow Past Rod Bundles," *Heat Transfer-Soviet Research*, Vol. 2, No.1, pp.1-7.

Tolubinskiy, V. I., Litoshenko, A. K. and Shevtsov, V. L., November 1970, "Critical Heat Flux Densities in Internally-Heated Annuli," *Heat Transfer-Soviet Research*, Vol. 2, No.6, pp.183-190.

Tolubinskiy, V.I., Litoshenko, A. K. and Shevtsov, V. L., 1969, "Correlation of Experimental Data on Critical Heat Flux in Annular Channels," *Heat Transfer-Soviet Research*, Vol. 1, No.1, pp.80-87

Tong, L.S. and Tang, Y.S., 1997, "Boiling Heat Transfer and Two-Phase Flow," Second Edition, Taylor & Francis Publishers.

Tong, L.S., 1968, "An Evaluation of the Departure from Nucleate Boiling in Bundles of Reactor Fuel Rods," *Nuclear Science and Engineering*: 33, pp.7-15

Towell, R.H., 1965, "Effect of Rod Spacing on Heat Transfer Burnout in Rod Bundles," USAEC Rep. DP-1013, E.I.duPont de Nemours and Co., Wilmington, DE.

Triplett, K.A., Ghiaasiaan, S.M., Abdel-Khalik, S.I., and Sadowski D.L, 1999a, "Gas-liquid two-phase flow in microchannels Part I: two-phase flow patterns," *International Journal of Multiphase Flow* 25 (3) pp. 377-394.

Triplett, K.A, Ghiaasiaan, S.M, Abdel-Khalik, S.I, LeMouel, A, and McCord, B.N., 1999b, "Gas-liquid two-phase flow in microchannels Part II: void fraction and pressure drop,"

International Journal of Multiphase Flow 25 (3) pp. 395-410.

Turian, R., and Kessler, F.D., 2000, "Capillary Flow in a Noncircular Tube," AIChE J, Vol.46. No.4, pp.695-706.

Ueda, T., Tsunenari, S. and Koyanagi, M., 1983, "An Investigation of Critical heat Flux and Surface Rewet in Flow Boiling System," Int. J. Heat Mass Transfer, Vol. 26, No.8, pp.1189-1198.

Ulrych, G., 1985, "Application of the Tube Critical Heat Flux Tables to Annuli and Rod Bundles," Proceedings of Third International Topic Meeting on Reactor Thermal Hydraulics, Vol.2

Vaitekunas, D.A., Tain, R.M., and Cheng, S.C., 1996, "Assessment of Critical Heat Flux Models," Technical Report UO-MCG-TH-95002, University of Ottawa

Wallis, G.B., 1969, "One-dimensional Two-Phase Flow," McGraw-Hill Book Company.

Wallis, G. B., 1970, "Annular Two-Phase Flow Part I: A simple Theory," ASME Journal of Basic Engineering, pp.59-81.

Weisman, J., 1992, "The Current Status of Theoretically Based Approaches to the Prediction of the Critical Heat Flux in Flow Boiling," Nuclear Technology, Vol. 99, pp1-21

Weisman, J. and Ying, S.H., 1985, "A Theoretically Based Predictions of Critical Heat Flux Prediction for Rod Bundles at PWR Condition," Nuclear Engineering and Design 85, pp.239-250

Weisman, J., 1985, "Theoretical Based Predictions of Critical Heat Flux in Rod Bundles," Proceedings of Third International Topic Meeting on Reactor Thermal Hydraulics, Vol.2

- Weisman, J. and Pei, B. S., 1983, "Prediction of Critical Heat Flux in Flow Boiling at low Qualities," *Int. J. Heat Mass Transfer*, Vol.26, No.10, pp.1463-1477.
- Whalley, P. B., Hutchinson, P. and Hewitt, G. F., 1974, "The Calculation of Critical Heat Flux in Forced Convection Boiling," 5th International Heat Transfer Conference, Heat Transfer, Vol.4, pp .290-294.
- Whalley, P.B., 1977, "The Calculation of Dryout in a Rod Bundle," *Int.J. Multiphase Flow*, Vol.3, pp.501-515.
- Whalley, P.B., 1977, "The Calculation of Dryout in a Rod Bundle-A Comparison of Experimental and Calculated Results," *Int.J. Multiphase Flow*, Vol.4, pp.427-431.
- Whalley, P. B., Hutchinson, P. and James, P. W., 1978, "The Calculation of Critical Heat Flux in Complex Situations Using an Annular Flow Model," 6th International Heat Transfer Conference, pp.65-70.
- Wilmarth, T., Ishii, M., 1994, "Two-phase Flow Regimes in Narrow Rectangular Vertical and Horizontal Channels," *Int. J. Heat Mass Transfer* 37, 1749-1758.
- Xu, J.L., Cheng, P., and Zhao T.S., 1999, "Gas-Liquid Two-Phase Flow Regimes in Rectangular Channels with Mini/Micro Gaps," *Int. J. Multiphase Flow* 25, pp.411-432.
- Yang, J. Y. and Weisman, J., 1991, "A Phenomenological Model of Subcooled Flow Boiling in the Detached Bubble Region," *Int. J. Multiphase Flow*, Vol.17, No.1, pp.77-94.
- Yefimov, V. A., Trutnev, D. P. and Marchenko, L. D., 1969, "A Study of the Boiling Crisis with Water in Rod Bundles," *Heat Transfer-Soviet Research*, Vol.1, No.4, pp.48-54.

- Yefimov, V. A., Marchenko, L. D. and Trutnev, D. P., 1969, "Study of the Effect of Rod Bundles on Boiling Crisis," Heat Transfer-Soviet Research, Vol.1, No.4, pp.42-47.
- Zeitoun, O. and Shoukri, M., 1995, "Bubble Behaviour and Mean Diameter in Subcooled Flow Boiling," Proceedings of the 7th International Meeting on Nuclear Reactor Thermal-Hydraulics NURETH-7, Saratoga Springs, New York, pp.3175-3196.
- Zeigarnik, Yu.A., Privalov, N.P., and Klimov, A.I., "Critical Heat Flux with Boiling of Subcooled Water in Rectangular Channels with One-Side Supply of Heat," Thermal Engineering, Vol.28, No.1, pp.40-43, 1981.

Appendix I

Dumbbell-Shaped Test Section

A dumbbell-shaped tube is consisted of two subchannels interconnected via a narrow gap. This geometry is close to the two neighboring subchannels and a connecting gap in nuclear fuel bundle. The narrow gap region of the dumbbell-shaped tube simulates the space between two adjacent rods. Figure I.1 shows a sample of the dumbbell-shaped tube test section.

I.1 Construction of Dumbbell-Shaped Test Section

The test section of the dumbbell-shaped tube was constructed from a circular tube using a die. Figures I.2 and I.3 illustrate the detailed structure of the die designed. The Die has two slots that hold two balls and guide the balls moving in the radial direction of the circular tube. By turning the die, the balls squeeze the circular tube symmetrically. The dumbbell-shaped tube is formed by moving the entire die in the axial direction of the circular tube.

The curvature of gap region is determined by the ball diameter. The gap size is controlled by ball displacement which is determined by turning angle of the die.

I.2 Gap Size

Gap size is the main geometric parameter of dumbbell-shaped tube. A correlation was developed to predict the gap size of the dumbbell-shaped tube test section to be made. The gap size is the function of turning angle and the slope of internal surface of the die. Figure I.4 gives the correlation between gap size and die turning angle.

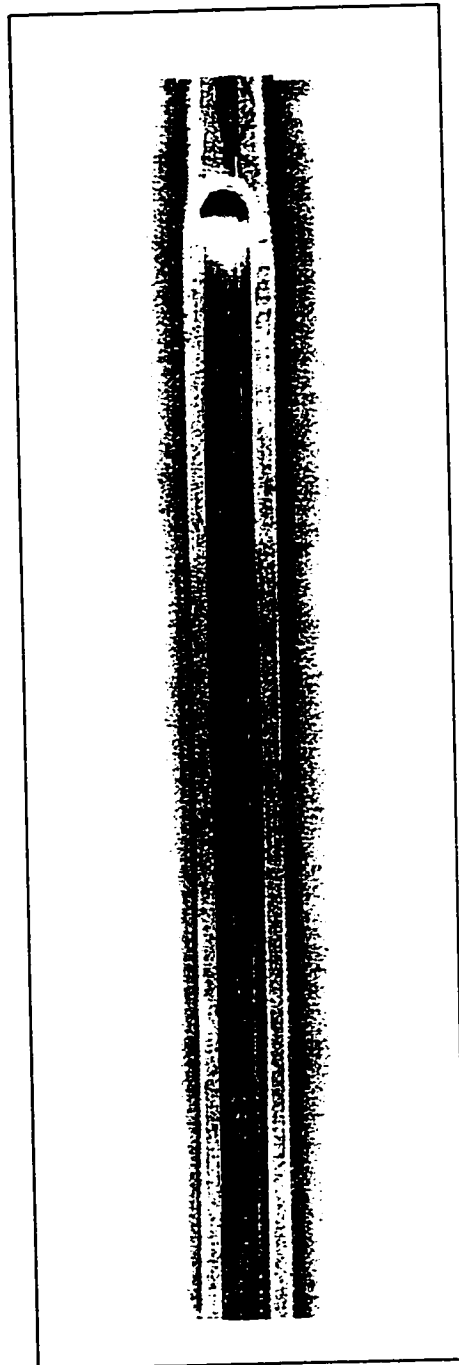


Figure I.1 Dumbbell-shaped test section.

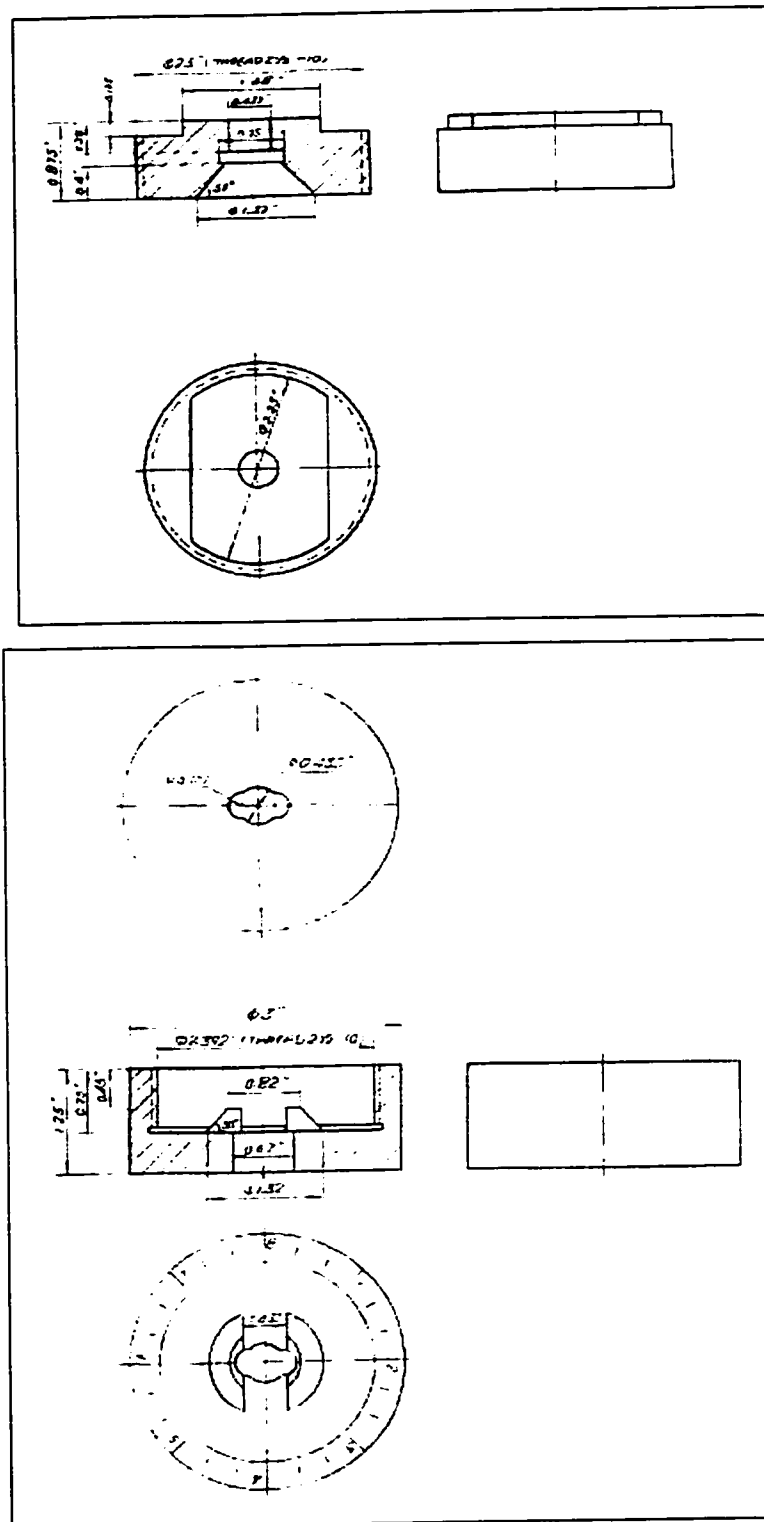


Figure I.2 Die for dumbbell-shaped tube.

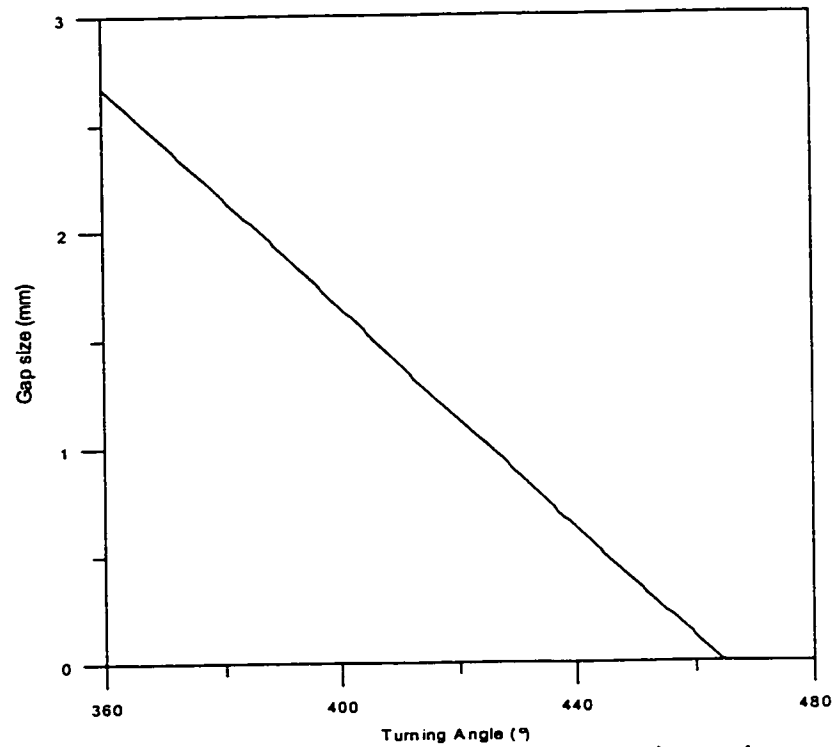


Figure I.3 Variation of gap size against turning angle.

I.3 Cross Section of Dumbbell-Shaped Tube

As shown in Figure 4.2, the dumbbell-shaped tube cross section is composed of three sets of tangential circles and two sets of tangential lines. Three sample test sections were made and the six characteristic geometric parameters were measured. The test results are shown in Table I.1.

Table I.1 Dumbbell-shaped tube characteristic geometric parameters.

Sample Tube	Turning Angle	Gap Size (mm)	Characteristic Geometric Parameters (mm)					
			w_g	w_x	w_y	R_g	R_s	R_t
1	400°	1.65	2.35	12.09	5.60	3.0	3.0	1.6
2	430°	0.87	1.57	12.07	5.49	3.0	3.0	1.6
3	450°	0.34	1.04	12.04	5.30	3.0	3.0	1.6

I.4 Temperature and Pressure Effects on the Shape of Test Section

Critical heat flux experiment is conducted at pressures up to 1.66 MPa (saturation temperature of 59.5°C). Because both temperature and pressure may affect the shape of test section, a test for temperature and pressure effects on test section shape was performed before CHF experiments. A special test section of 600mm long was made for this purpose. At room temperature and atmosphere pressure, the gap size of the test section is 0.52 mm. Low-voltage, high-current power is supplied to the sample dumbbell-shaped tube to control the wall temperature. The power supplied to the tube was predicted by using empirical heat transfer equation for free convection. The system pressure of the tube was controlled by the amount of nitrogen gas charged. The characteristic geometric parameters were measured with micrometer. Table I.2 lists the test results.

Table I.2 Temperature and pressure effects on the shaped of dumbbell tube.

p (kPa)	T (°C)	w _g (mm)	w _x (mm)	w _y (mm)
101	23.7	1.22	12.083	5.601
	49.5	1.225	12.083	5.601
	57.9	1.225	12.083	5.601
	82.2	1.228	12.083	5.601
	102.2	1.23	12.083	5.601
	121.9	1.228	12.083	5.601
965	51.5	1.378	12.089	5.654
	80.6	1.376	12.089	5.654
	93.7	1.378	12.089	5.654
	126.7	1.378	12.089	5.654
1931	42.9	1.566	12.091	5.72
	81.3	1.573	12.091	5.72
	114.6	1.575	12.091	5.72
	132.1	1.573	12.091	5.72

The test result shows that the temperate effect on the shape of dumbbell test section is negligible. However, the pressure has very strong effect on the minimum gap size of the dumbbell-shaped tube. The result suggests that a clamp is needed to prevent the gap size changing during the CHF test.

Appendix II

ERROR ANALYSIS

II.1 Introduction

All measurements of physical quantities are subject to uncertainties. It is never possible to measure anything exactly. In order to draw valid conclusions the error must be indicated and dealt with properly. The following sections analyze the errors associated with the CHF experiment.

The degree of uncertainty associated with the measurement data is presented in either as an RMS error or standard deviation (σ). The error adds both systematic and random errors together. The RMS error and standard deviation are defined as:

$$\sigma = \sqrt{\frac{\sum_k (X_k - \bar{X})^2}{N}}$$

If one assumes these errors are perfectly random and normally distributed about the mean value measured, then 68% of the data will lie within $\pm 1\sigma$, and 95% within $\pm 2\sigma$. This is an important characteristic when considering the significance of measured data: if the error estimates are correct, about 70% of the data should fall within $\pm 1\sigma$, and all the data should fall within $\pm 2\sigma$, otherwise there is a systematic error unaccounted for.

II.2 Uncertainties in Basic Measurements

II.2.1 Temperature

K-type thermocouples were used to measure the fluid temperature and monitor the wall temperature. Before the CHF experiment, all the thermocouples were calibrated using a digital RTD thermometer and a water bath ranging in temperature from 0-100°C. The following table lists the accuracy of the two measuring methods.

Method	RMS	MAX
RTD	$\pm 0.03\text{ }^{\circ}\text{C}$	$\pm 0.05\text{ }^{\circ}\text{C}$
Thermocouple	$\pm 0.06\text{ }^{\circ}\text{C}$	$\pm 0.2\text{ }^{\circ}\text{C}$

In this study, an RMS value of $\pm 0.1^{\circ}\text{C}$ is used for error analysis for all the thermocouples.

II.2.2 Pressure

The absolute system pressures were measured with the absolute pressure transducer installed at the outlet of the test section. The pressure transducer was calibrated before the CHF measurements. The transducer is characterized as:

Calibration range: 0-3.45 MPa

Pressure calibrator: $\pm 0.01\%$

Max calibration error: 1.834 kPa

The errors associated with the pressure measurement are:

$$\Delta p = \sqrt{(0.345)^2 + (1.834)^2} = 1.866 \text{ kPa}$$

II.2.3 Mass Flow Rate

The mass flow rate was measured using a Coriolis-type flow meter (Model D65 by Micro Motion). The meter has been calibrated before the CHF measurements. The mass flow rate meter is characterized as:

Accuracy = $\pm 0.2\%$

Repeatability = $\pm 0.05\%$

Zero stability = $\pm 0.23\text{ g/s}$

A/D conversion = $\pm 0.1\%$ F.S. (F.S. = 1.333 kg/s)

The RMS error of mass flow rate is calculated by:

$$\begin{aligned}\frac{\Delta W}{W} &= \sqrt{\left(\frac{0.2}{100}\right)^2 + \left(\frac{0.05}{100}\right)^2 + \left(\frac{0.23}{W}\right)^2 + \left(\frac{1.33}{W}\right)^2} \\ &= \sqrt{4.25 \times 10^{-6} + \left(\frac{1.82}{W}\right)^2}\end{aligned}$$

which ranges from 0.2%-0.27% (1000 – 5000 g/s)

II.2.4 Power

The power (Q) applied to the test section is the product of the voltage drop across the test section (U) and the current through the shunt resistor (I). The mass flow rate meter is characterized as:

maximum offset voltage = $\pm 0.2\%$

uncertainty in current = $\pm 2.9\text{A}$

A/D conversion = $\pm 0.14\%$

The RMS error is given by:

$$\frac{\Delta Q}{Q} = \sqrt{\left(\frac{\Delta U}{U}\right)^2 + \left(\frac{\Delta I}{I}\right)^2 + \left(\frac{0.14}{100}\right)^2}$$

In this study, the power applied to the test section ranges from 2000 to 8000 W. For the purpose of the error analysis, assume that the test section is of an average electrical resistance of 0.09Ω .

In case of $Q = 2000\text{ W}$,

$$I = \sqrt{\frac{Q}{R}} = \sqrt{\frac{2000}{0.09}} = 149.1\text{A}$$

$$U = \frac{Q}{I} = \frac{2000}{149.1} = 13.4\text{V}$$

$$\frac{\Delta Q}{Q} = \sqrt{\left(\frac{0.2}{100}\right)^2 + \left(\frac{2.9}{149.1}\right)^2 + \left(\frac{0.14}{100}\right)^2} = 0.020$$

In case of $Q = 8000\text{ W}$,

$$I = \sqrt{\frac{Q}{R}} = \sqrt{\frac{8000}{0.09}} = 298.1\text{A}$$

$$U = \frac{Q}{I} = \frac{8000}{298.1} = 26.8\text{V}$$

$$\frac{\Delta Q}{Q} = \sqrt{\left(\frac{0.2}{100}\right)^2 + \left(\frac{2.9}{298.1}\right)^2 + \left(\frac{0.14}{100}\right)^2} = 0.010$$

Therefore, the RMS error of power ranges from 1% to 2% for the CHF experiment (2-8kW).

II.2.4 Geometry Parameters

II.2.4.1 Heated Length

The measurement error of heated length is given by:

$$\Delta z = 0.002m = 2mm$$

The heated length z ranges from 0.7 m to 1.4 m. Therefore, the error $\Delta z/z$ ranges from 0.14% to 0.28%.

II.2.4.2 Perimeter

As described in Appendix I, the dumbbell-shaped tube was made by squeezing a circular tube. Assuming that the perimeter of the tube does not change after deformation, the dumbbell-shaped tube perimeter is the same as the circular tube. The perimeter of the dumbbell-shaped tube was obtained by measuring the inner diameter of the circular tube. The error of the diameter is given by:

$$\Delta d = 0.000002m = 0.002mm$$

thus, the error of the perimeter is given by:

$$\Delta l_p = 0.0063mm$$

$$\frac{\Delta l_p}{l_p} = \frac{\Delta d}{d} = \frac{0.002}{9.8} = 0.02\%$$

II.2.4.3 Gap Size

As shown in Figure 4.6, to obtain the gap size, a chain of pin, rod and wall thickness was measured. The gap size is the length of the chain minus the chain length with zero gap. The error of the gap size is given by:

$$\Delta \delta = \sqrt{0.002^2 + 0.002^2} = 0.0028mm$$

II.2.4.4 Flow Area

To determine the flow area of a dumbbell-shaped tube is not a simple work due to the complicity of the geometry and the small size of the test section. As described in Chapter 4, a sophisticated method was developed to calculate the flow area based on six characteristic geometry parameters measured. The measurement errors of these six geometry parameters are as follows:

Gap, longitude, and latitude width:

$$\Delta l_{w_g} = 0.002 \text{ mm} \quad \Delta l_{w_x} = 0.002 \text{ mm} \quad \Delta l_{w_y} = 0.002 \text{ mm}$$

Radii,

$$\Delta l_{R_g} = 0.25 \text{ mm} \quad \Delta l_{R_l} = 0.25 \text{ mm} \quad \Delta l_{R_c} = 0.25 \text{ mm}$$

To find out the error range of the measurement, the possible maximum and minimum flow was calculated areas based on the range of each gap size. The real flow area should be within this range. Based on the computation, the maximum error of the flow area is:

$$\Delta A = 0.23 \text{ mm}^2$$

II.3 Inlet Subcooling

The inlet subcooling is the liquid saturation enthalpy minus inlet enthalpy.

$$h_{sub} = h_l - h_{in}$$

where,

$$h_l = f(p) \quad h_{in} = f(T_{in})$$

Thus, the error of inlet subcooling is calculated by:

$$\Delta h_{sub} = \sqrt{\left(\frac{\partial h_{sub}}{\partial h_l} \Delta h_l \right)^2 + \left(\frac{\partial h_{sub}}{\partial h_{in}} \Delta h_{in} \right)^2}$$

where,

$$\frac{\partial h_{sub}}{\partial h_l} = \frac{\partial h_{sub}}{\partial h_{in}} = 1$$

$$\Delta h_l = \sqrt{\left(\frac{\Delta f(p)}{f(p)} h_l\right)^2 + \left(\frac{\partial h_f}{\partial p} \Delta p\right)^2}$$

$$\Delta h_m = \sqrt{\left(\frac{\Delta f(T)}{f(T)} h_m\right)^2 + \left(\frac{\partial h_f}{\partial T} \Delta T\right)^2}$$

We assume that

$$\frac{\Delta f(p)}{f(p)} = \frac{\Delta f(T)}{f(T)} = 0.2\%$$

To give the degree of the uncertainty, we give the two cases of the CHF test. In case of $P = 1130$ kPa, $G = 2000$ kg/m²s, and $T_{in} = 40$ °C, $h_l = 112060$ J/kg. $h_{in} = 106014$ J/kg, based on the equations given above, the error of the inlet subcooling is: $\Delta h_{sub} = 385$ J/kg.

In case of $P = 1660$ kPa, $G = 3000$ kg/m²s, and $T_{in} = 55$ °C, $h_l = 136058$ J/kg. $h_{in} = 128790$ J/kg, based on the equations given above, the error of the inlet subcooling is: $\Delta h_{sub} = 447$ J/kg.

II.4 Mass Flux

Mass flux is the mass flow rate of unit flow area.

$$G = \frac{W}{A}$$

The error of mass flux is calculated by:

$$\frac{\Delta G}{G} = \sqrt{\left(\frac{\Delta W}{W}\right)^2 + \left(\frac{\Delta A}{A}\right)^2}$$

According to the previous analysis on mass flow rate, the mass flow rate error ranges from 0.2%-0.27%. In case of the zero gap, the flow area is of a minimum value of 26.4 mm². Therefore, the uncertainty degree of mass flux is:

$$\frac{\Delta G}{G} = \sqrt{\left(\frac{\Delta W}{W}\right)^2 + \left(\frac{\Delta A}{A}\right)^2} = \sqrt{(0.27\%)^2 + \left(\frac{0.23}{26.4}\right)^2} = 0.91\%$$

II.5 Quality

The exit quality is calculated by:

$$X = \frac{h_{out} - h_l}{h_{fg}}$$

where,

$$h_{out} = h_m + \frac{Q}{W}$$

thus,

$$X = \frac{h_m + \frac{Q}{W} - h_l}{h_{fg}}$$

The error of the exit quality is given by:

$$\Delta X = \sqrt{\left(\frac{\partial X}{\partial h_m} \Delta h_m\right)^2 + \left(\frac{\partial X}{\partial Q} \Delta Q\right)^2 + \left(\frac{\partial X}{\partial W} \Delta W\right)^2 + \left(\frac{\partial X}{\partial h_f} \Delta h_f\right)^2 + \left(\frac{\partial X}{\partial h_{fg}} \Delta h_{fg}\right)^2}$$

where,

$$\begin{aligned} \frac{\partial X}{\partial h_m} &= \frac{1}{h_{fg}} & \frac{\partial X}{\partial h_{fg}} &= \frac{1}{h_{fg}} & \frac{\partial X}{\partial h_f} &= -\frac{h_m + \frac{Q}{W} - h_l}{h_{fg}^2} \\ \frac{\partial X}{\partial W} &= -\frac{Q}{W^2 h_{fg}} & \frac{\partial X}{\partial Q} &= \frac{1}{W h_{fg}} \end{aligned}$$

To give the degree of the uncertainty, we give the two cases of the CHF test. In case of $P = 1130$ kPa, $G = 2000$ kg/m²s, and $T_{in} = 40$ °C, $h_l = 112060$ J/kg, $h_m = 106014$ J/kg, $h_{fg} = 158503$ J/kg, $\delta = 0.5$ mm, $X = 0.319$, $Q = 3433$ kW, based on the equations given above, the error of the exit is: $\Delta X = 0.012$.

In case of $P = 1660$ kPa, $G = 3000$ kg/m²s, and $T_{in} = 55$ °C, $h_l = 136058$ J/kg, $h_m = 128790$ J/kg, $h_{fg} = 140056$ J/kg, $\delta = 0.5$ mm, $X = 0.336$, $Q = 4714$ kW, based on the equations given above, the error of the exit is: $\Delta X = 0.009$.

II.6 Critical Heat Flux

The critical heat flux is the heat flux when the CHF phenomenon occurs at exit of the test section.

$$q = \frac{Q}{z l_p}$$

where, z is the heated length and l_p is the perimeter of the dumbbell-shaped tube.

$$\Delta q = \sqrt{\left(\frac{\partial q}{\partial Q} \Delta Q\right)^2 + \left(\frac{\partial q}{\partial z} \Delta z\right)^2 + \left(\frac{\partial q}{\partial l_p} \Delta l_p\right)^2}$$

$$\frac{\Delta q}{q} = \sqrt{\left(\frac{\Delta Q}{Q}\right)^2 + \left(\frac{\Delta z}{z}\right)^2 + \left(\frac{\Delta l_p}{l_p}\right)^2}$$

where,

$$\frac{\partial q}{\partial Q} = \frac{1}{z l_p}$$

$$\frac{\partial q}{\partial z} = -\frac{Q}{z^2 l_p}$$

$$\frac{\partial q}{\partial l_p} = -\frac{Q}{z l_p^2}$$

According to the error analysis for Q , z , and l_p ,

$$\frac{\Delta q}{q} = \sqrt{(2\%)^2 + (0.28\%)^2 + (0.02\%)^2} = 2.1\%$$

Appendix III

DUMBBELL-SHAPED TUBE CHF EXPERIMENTAL DATA

Gap (mm)	P (kPa)	G (kg/m ² s)	T _b (°C)	I _h (A)	V _h (V)	X _h	CHF (kW/m ²)	Q (W)	L _h (m)	First CHF
0	1130.1	993	21.4	222	14.3	0.541	147.3	3174.6	0.7	G
0	1130.1	1010	29.4	216	13.9	0.574	139.3	3002.4	0.7	G
0	1130.1	1004	37.2	208	13.4	0.596	129.3	2787.2	0.7	G
0	1130.1	991	22.2	196	17.8	0.614	113.3	3488.8	1	G
0	1130.1	991	28.7	191	17.4	0.644	107.9	3323.4	1	G
0	1130.1	1010	37.4	184	16.8	0.67	100.4	3091.2	1	GC
0	1130.1	991	21.6	171	21.6	0.666	85.7	3693.6	1.4	GC
0	1130.1	991	28.4	165	20.9	0.67	80	3448.5	1.4	GC
0	1130.1	1000	38.3	155	19.7	0.67	70.8	3053.5	1.4	GC
0.15	1130.1	999	20.4	224	14.3	0.511	148.6	3203.2	0.7	G
0.15	1130.1	1006	28.7	221	14.1	0.568	144.6	3116.1	0.7	G
0.15	1130.1	1016	38.1	211	13.5	0.595	132.2	2848.5	0.7	G
0.15	1130.1	1015	36.6	186	16.9	0.648	102.1	3143.4	1	G
0.15	1130.1	1002	21.4	175	22.1	0.672	89.7	3867.5	1.4	GC
0.15	1130.1	1016	29.9	169	21.3	0.69	83.5	3599.7	1.4	GC
0.15	1130.1	1016	38.4	160	20.2	0.685	75	3232.0	1.4	GC
0.3	1130.1	1001	20.2	227	14.5	0.504	152.7	3291.5	0.7	G
0.3	1130.1	994	29.2	223	14.2	0.559	146.9	3166.6	0.7	G
0.3	1130.1	1016	38.6	210	13.4	0.57	130.6	2814.0	0.7	G
0.3	1130.1	1008	22.3	198	17.9	0.579	115.1	3544.2	1	G
0.3	1130.1	1011	29.2	194	17.6	0.614	110.9	3414.4	1	G
0.3	1130.1	1016	38.6	187	16.9	0.646	102.6	3160.3	1	G
0.3	1130.1	1002	19.1	179	22.7	0.664	94.3	4063.3	1.4	G
0.3	1130.1	995	27.3	172	21.9	0.674	87.4	3766.8	1.4	GC
0.3	1130.1	996	37.3	162	20.6	0.673	77.4	3337.2	1.4	G
0.5	1130.1	1013	18.2	229	14.7	0.47	156.2	3366.3	0.7	G
0.5	1130.1	1008	25.6	223	14.3	0.5	148	3188.9	0.7	G
0.5	1130.1	1016	33.3	217	13.9	0.535	140	3016.3	0.7	G
0.5	1130.1	1004	40	214	13.7	0.581	136	2931.8	0.7	G
0.5	1130.1	999	19	182	22.9	0.646	96.7	4167.8	1.4	C
0.5	1130.1	1008	20.8	180	22.7	0.645	94.8	4086	1.4	C
0.5	1130.1	1007	29.1	172	21.7	0.647	86.6	3732.4	1.4	C
0.5	1130.1	1013	33.8	168	21.1	0.652	82.2	3544.8	1.4	GC

Gap (mm)	PP (kPa)	PG (kg/m ³)	TH (°C)	TA (°C)	V (m)	X (m)	CHF (kW/m ²)	D (W)	L (m)	First CHF
0.5	1130.1	1015	40	162	20.4	0.66	76.7	3304.8	1.4	GC
0.7	1130.1	1019	14.1	243	15.4	0.478	173.3	3734.9	0.7	G
0.7	1130.1	1007	21.7	234	14.8	0.491	160.4	3456.2	0.7	G
0.7	1130.1	998	28.7	229	14.5	0.527	154	3318.2	0.7	G
0.7	1130.1	1015	14.9	213	19.1	0.554	132.3	4074.7	1	G
0.7	1130.1	1005	23.8	205	18.5	0.576	122.9	3784.3	1	G
0.7	1130.1	1001	29.8	200	18	0.595	117	3602	1	C
0.7	1130.1	1004	35	195	17.6	0.608	111.3	3428.1	1	C
0.7	1130.1	1019	18.4	184	23.4	0.632	99.8	4303.8	1.4	C
0.7	1130.1	996	27.4	176	22.3	0.637	90.9	3919.5	1.4	C
0.7	1130.1	1005	33.9	169	21.5	0.638	84.1	3625.1	1.4	C
0.7	1130.1	1011	40.5	163	20.7	0.65	78.4	3377.4	1.4	C
1	1130.1	993	18.9	242	15.5	0.481	174	3751	0.7	G
1	1130.1	1018	26.8	235	15	0.51	163.6	3525	0.7	G
1	1130.1	1017	38.5	223	14.3	0.556	148	3188.9	0.7	GC
1	1130.1	1006	19.5	214	19.4	0.563	134.8	4151.6	1	G
1	1130.1	996	27.6	206	18.7	0.58	125.1	3852.2	1	GC
1	1130.1	1013	19.6	185	23.5	0.601	100.9	4347.5	1.4	C
1	1130.1	993	29.8	176	22.3	0.614	91.1	3924.8	1.4	C
1	1130.1	1015	38.3	167	21.3	0.624	82.5	3557.1	1.4	C
0	1130.1	1994	21.8	242	15.6	0.24	175.2	3775.2	0.7	G
0	1130.1	1988	28.9	231	14.9	0.266	159.7	3441.9	0.7	G
0	1130.1	1985	36.9	222	14.3	0.309	147.3	3174.6	0.7	G
0	1130.1	2027	21.9	219	20.1	0.315	143	4401.9	1	G
0	1130.1	2035	27.3	212	19.2	0.325	132.2	4070.4	1	G
0	1130.1	2019	22.8	193	24.5	0.362	109.7	4728.5	1.4	G
0	1130.1	1985	31.8	189	23.1	0.402	101.3	4365.9	1.4	G
0	1130.1	2001	37.3	182	23	0.433	97.1	4186	1.4	GC
0.15	1130.1	2031	21.7	253	16.2	0.26	190.2	4098.6	0.7	G
0.15	1130.1	2023	29.4	238	15.2	0.276	167.9	3617.6	0.7	G
0.15	1130.1	2039	38.5	227	14.5	0.324	152.7	3291.5	0.7	G
0.15	1130.1	2003	21.9	227	20.6	0.327	151.9	4676.2	1	G
0.15	1130.1	2026	30.2	209	18.9	0.321	128.3	3950.1	1	G
0.15	1130.1	2028	36.2	198	17.9	0.331	115.1	3544.2	1	G
0.3	1130.1	2027	20.1	258	16.5	0.246	197.5	4257	0.7	G
0.3	1130.1	1981	29.2	240	15.4	0.268	171.5	3696	0.7	G
0.3	1130.1	2033	37.4	232	14.8	0.316	159.3	3433.6	0.7	G

SP (mm)	P (kPa)	G (kg/m ³)	T (°C)	I (A)	V (V)	X (m)	CHF (kW/m ²)	Q (W)	L ₀ (m)	First CHE
0.3	1130.1	2017	22.1	206	26.1	0.388	124.7	5376.6	1.4	G
0.3	1130.1	2027	31.4	196	24.9	0.419	113.2	4880.4	1.4	IC
0.3	1130.1	2021	36.9	188	23.9	0.428	104.2	4493.2	1.4	C
0.5	1130.1	2027	18.8	268	17.2	0.251	213.9	4609.6	0.7	G
0.5	1130.1	2014	25.5	256	16.4	0.269	194.8	4198.4	0.7	G
0.5	1130.1	2032	32.6	245	15.7	0.298	178.5	3846.5	0.7	G
0.5	1130.1	1982	39.4	232	14.8	0.319	159.3	3433.6	0.7	G
0.5	1130.1	2038	25.3	207	26.2	0.397	125.8	5423.4	1.4	GC
0.5	1130.1	1995	30.7	200	25.3	0.408	117.4	5060	1.4	GC
0.5	1130.1	2008	37.2	190	24	0.417	105.8	4560	1.4	C
0.5	1130.1	2000	40.6	186	23.5	0.429	101.4	4371	1.4	C
0.7	1130.1	2007	13.9	283	17.9	0.233	235.1	5065.7	0.7	C
0.7	1130.1	2019	20.6	273	17.2	0.257	218.1	4701.1	0.7	C
0.7	1130.1	2033	26.6	264	16.7	0.282	204.6	4408.8	0.7	C
0.7	1130.1	2019	30.9	256	16.2	0.295	192	4137	0.7	C
0.7	1130.1	2022	36.8	242	15.3	0.305	171.5	3695.3	0.7	G
0.7	1130.1	2020	42.5	233	14.7	0.332	159	3427.4	0.7	G
0.7	1130.1	1987	15.8	222	28.2	0.371	145.2	6258.2	1.4	G
0.7	1130.1	1998	21.9	213	27	0.376	133.6	5759.5	1.4	C
0.7	1130.1	1991	26.7	206	26.2	0.383	125.2	5397.2	1.4	C
0.7	1130.1	2011	30.8	201	25.5	0.394	118.9	5125.5	1.4	C
0.7	1130.1	1999	36.1	193	24.4	0.402	109.4	4716.9	1.4	C
0.7	1130.1	2032	40.7	186	23.6	0.413	101.9	4391.5	1.4	C
1	1130.1	2015	19.2	279	17.8	0.242	230.4	4966.2	0.7	C
1	1130.1	2038	28.4	268	17.1	0.29	212.6	4582.8	0.7	C
1	1130.1	2026	36.4	249	16	0.308	184.9	3984	0.7	G
1	1130.1	1995	20.2	246	22.4	0.303	179	5510.4	1	G
1	1130.1	2013	27.6	236	21.4	0.327	164	5050.4	1	G
1	1130.1	2024	36	221	20.1	0.347	144.3	4442.1	1	C
1	1130.1	2002	22.1	216	27.4	0.359	137.3	5918.4	1.4	C
1	1130.1	1996	29	207	26.3	0.377	126.3	5444.1	1.4	C
1	1130.1	2006	38.6	194	24.7	0.405	111.2	4791.8	1.4	GC
0	1130.1	3045	22.8	265	17.1	0.159	210.3	4531.5	0.7	G
0	1130.1	2970	31.2	242	15.6	0.177	175.2	3775.2	0.7	G
0	1130.1	3052	24.2	233	21.1	0.203	159.7	4916.3	1	G
0	1130.1	2982	28.4	213	26.9	0.306	132.9	5729.7	1.4	G
0	1130.1	3025	38.2	201	25.5	0.35	118.9	5125.5	1.4	GC

Gap (mm)	P (kPa)	G (kg/m ² s)	T _m (°C)	I (A)	V (V)	X	CHF (kW/m ²)	Q (W)	I _{he} (m)	First CHF
0.15	1130.1	2992	23.9	278	17.8	0.188	229.6	4948.4	0.7	GC
0.15	1130.1	2971	29.7	259	16.6	0.193	199.5	4299.4	0.7	G
0.15	1130.1	3024	38.8	248	15.9	0.251	183	3943.2	0.7	G
0.15	1130.1	2973	26.5	247	17.2	0.257	179.7	5532.8	1	GC
0.15	1130.1	2978	31.1	241	21.8	0.278	170.6	5253.8	1	GC
0.15	1130.1	3017	35.3	228	20.7	0.282	153.3	4719.6	1	GC
0.3	1130.1	3051	23.1	281	18	0.176	234.7	5058	0.7	C
0.3	1130.1	3043	30.1	262	16.8	0.193	204.2	4401.6	0.7	G
0.3	1130.1	2982	38.9	254	16.2	0.254	190.9	4114.8	0.7	GC
0.3	1130.1	3046	30.9	243	22	0.269	173.6	5346	1	GC
0.3	1130.1	3027	39.1	229	20.8	0.304	154.7	4763.2	1	G
0.3	1130.1	2993	34.4	219	27.8	0.357	141.2	6088.2	1.4	GC
0.5	1130.1	3038	21	284	18.2	0.148	239.8	5168.8	0.7	C
0.5	1130.1	3023	26.8	273	17.5	0.174	221.7	4777.5	0.7	C
0.5	1130.1	2998	37.6	259	16.6	0.242	199.5	4299.4	0.7	GC
0.5	1130.1	2986	26.1	238	30.1	0.336	166.2	7163.8	1.4	GC
0.5	1130.1	3047	30.9	228	28.8	0.338	152.3	6566.4	1.4	GC
0.5	1130.1	3002	35.2	220	27.8	0.347	141.9	6116	1.4	GC
0.7	1130.1	3038	14.6	299	18.9	0.108	261.8	5642.1	0.7	C
0.7	1130.1	3053	23.4	281	17.8	0.143	231.4	4987.8	0.7	C
0.7	1130.1	3026	30.1	267	16.9	0.173	209	4504.3	0.7	C
0.7	1130.1	3023	17.1	245	31.1	0.264	176.8	7622	1.4	C
0.7	1130.1	2971	26.5	233	29.5	0.299	159.6	6878.2	1.4	C
0.7	1130.1	3029	32.7	226	28.6	0.329	150	6465.9	1.4	C
0.7	1130.1	2972	39.3	213	27	0.343	133.4	5748.9	1.4	C
1	1130.1	2998	20.3	289	18.5	0.118	248.1	5346.5	0.7	C
1	1130.1	3042	26.9	277	17.8	0.152	228.8	4930.6	0.7	C
1	1130.1	2987	38	265	17	0.229	209	4505	0.7	C
1	1130.1	2990	27.4	262	23.8	0.24	202.5	6235.6	1	C
1	1130.1	3047	39.6	238	21.7	0.286	167.7	5164.6	1	GC
1	1130.1	2979	36.5	224	28.5	0.334	148.1	6384	1.4	GC
0	1660.3	1000	20.6	218	13.9	0.394	140.6	3030.2	0.7	G
0	1660.3	1017	29.8	205	13.1	0.397	124.6	2685.5	0.7	G
0	1660.3	993	39.9	199	12.7	0.463	117.3	2527.3	0.7	G
0	1660.3	1015	49.3	184	11.7	0.464	99.9	2152.8	0.7	C
0	1660.3	992	53.9	176	11.3	0.471	92.3	1988.8	0.7	C
0	1660.3	1009	20.5	165	20.9	0.505	80	3448.5	1.4	GC

Gap (mm)	P (kPa)	G (kg/m ³ s)	T _f (°C)	I _f (A)	V (V)	X	CHF (kW/m ²)	Q (W)	I _{he} (m)	First CHF
0	1660.3	1009	28.4	157	20	0.505	72.8	3140	1.4	GC
0	1660.3	995	37.4	146	18.5	0.482	62.7	2701	1.4	GC
0	1660.3	1009	48.4	135	17.1	0.496	53.6	2308.5	1.4	GC
0	1660.3	990	57.2	122	15.5	0.481	43.9	1891	1.4	GC
0.15	1660.3	999	21.4	222	14.3	0.41	147.3	3174.6	0.7	G
0.15	1660.3	1005	29.5	214	13.7	0.432	136	2931.8	0.7	G
0.15	1660.3	1011	38.4	205	13.1	0.463	124.6	2685.5	0.7	G
0.15	1660.3	1016	48.3	194	12.4	0.498	111.6	2405.6	0.7	GC
0.15	1660.3	998	55.5	181	11.6	0.498	97.4	2099.6	0.7	GC
0.15	1660.3	1012	22.1	166	21	0.498	80.9	3486	1.4	G
0.15	1660.3	1010	29.7	159	20.1	0.503	74.1	3195.9	1.4	G
0.15	1660.3	1017	36.4	151	19.1	0.493	66.9	2884.1	1.4	G
0.15	1660.3	1003	46.1	140	17.8	0.496	57.8	2492	1.4	G
0.15	1660.3	1012	57.4	128	16.2	0.513	48.1	2073.6	1.4	G
0.3	1660.3	1007	20.3	226	14.5	0.396	152.1	3277	0.7	G
0.3	1660.3	1010	28.8	214	13.7	0.399	136	2931.8	0.7	G
0.3	1660.3	998	35.7	206	13.2	0.419	126.2	2719.2	0.7	G
0.3	1660.3	1007	46.3	196	12.6	0.47	114.6	2469.6	0.7	G
0.3	1660.3	1004	56.6	180	11.5	0.484	96	2070	0.7	G
0.3	1660.3	998	19.6	170	21.6	0.488	85.2	3672	1.4	G
0.3	1660.3	1011	26.8	164	20.8	0.497	79.1	3411.2	1.4	G
0.3	1660.3	1005	34.5	154	19.6	0.481	70	3018.4	1.4	G
0.3	1660.3	997	46.8	141	18	0.493	58.9	2538	1.4	GC
0.3	1660.3	990	57.7	127	16.2	0.494	47.7	2057.4	1.4	C
0.5	1660.3	991	17.8	235	15	0.394	163.6	3525	0.7	G
0.5	1660.3	1003	26.7	224	14.4	0.415	149.7	3225.6	0.7	G
0.5	1660.3	1013	35.5	212	13.6	0.426	133.8	2883.2	0.7	G
0.5	1660.3	991	43.6	200	12.8	0.436	118.8	2560	0.7	G
0.5	1660.3	992	50.9	189	12.1	0.45	106.1	2286.9	0.7	G
0.5	1660.3	1010	57.8	179	11.4	0.468	94.7	2040.6	0.7	G
0.5	1660.3	1011	18	176	22.1	0.485	90.4	3896.6	1.4	G
0.5	1660.3	1004	28.1	166	20.9	0.486	80.3	3461.1	1.4	G
0.5	1660.3	991	37.7	155	19.5	0.483	70.1	3022.5	1.4	GC
0.5	1660.3	1005	46.4	146	18.2	0.489	61.6	2655.7	1.4	C
0.5	1660.3	1008	56	133	16.7	0.491	51.6	2222.4	1.4	C
0.7	1658.9	1004	13.4	246	15.6	0.387	178.1	3837.6	0.7	GC
0.7	1658.9	998	18.9	240	15.2	0.399	169.3	3648	0.7	GC
0.7	1660.3	1018	26.8	231	14.6	0.418	156.5	3372.6	0.7	G

Grp (mm)	SP (kPa)	KG (kg/m ³)	TL (°C)	I (A)	V (V)	X	CHF (kW/m ²)	IQ (W)	L (m)	First CHF
0.7	1660.3	999	15.9	211	19	0.451	130.2	4009	1	G
10.7	1660.3	1014	24.1	202	18.2	0.459	119.4	3676.4	1	G
0.7	1660.3	1011	38.3	186	16.7	0.479	100.9	3106.2	1	G
10.7	1660.3	1005	22.8	174	22.1	0.485	89.3	3847.1	1.4	GC
0.7	1660.3	990	29.4	167	21.1	0.481	81.9	3530.4	1.4	C
10.7	1660.3	1007	38.2	155	19.7	0.466	70.9	3056.6	1.4	C
0.7	1660.3	1019	45.8	146	18.5	0.468	62.8	2705.4	1.4	C
10.7	1660.3	994	54.7	134	16.7	0.458	51.9	2235.1	1.4	C
15.7	1660.3	1006	27.2	237	15.2	0.427	167.2	3602.4	0.7	C
1	1660.3	1002	34.8	227	14.5	0.44	152.7	3291.5	0.7	G
15.7	1660.3	992	17.7	213	19.4	0.443	134.2	4132.2	1	G
1	1660.3	1001	26.6	203	18.4	0.449	121.3	3735.2	1	G
15.7	1660.3	992	35.1	193	17.5	0.462	109.7	3377.5	1	G
1	1660.3	1008	44.2	180	16.3	0.464	95.3	2934	1	GC
15.7	1660.3	998	35.5	163	20.7	0.465	78.3	3374.1	1.4	C
1	1660.3	999	44.3	152	19.4	0.468	68.4	2948.8	1.4	C
15.7	1660.3	992	54.7	138	17.5	0.467	56	2415	1.4	C
0	1660.3	2004	22.4	260	16.6	0.179	200.3	4316	0.7	G
0	1660.3	2024	31.3	246	15.8	0.214	180.4	3886.8	0.7	G
0	1660.3	2012	38.1	233	14.9	0.231	161.1	3471.7	0.7	G
0	1660.3	2038	44.9	212	13.6	0.249	133.8	2883.2	0.7	G
0	1660.3	2024	23	203	25.8	0.308	121.5	5237.4	1.4	G
0	1660.3	2038	31.9	190	24.2	0.316	106.7	4598	1.4	G
0	1660.3	1993	39.3	179	22.7	0.323	94.3	4063.3	1.4	G
0	1660.3	2025	54.4	155	19.7	0.352	70.8	3053.5	1.4	GC
0.15	1660.3	1997	22.8	267	17.2	0.198	213.1	4592.4	0.7	GC
0.15	1660.3	2028	30	252	16.2	0.207	189.4	4082.4	0.7	G
0.15	1660.3	2034	37.8	236	15.1	0.223	165.4	3563.6	0.7	G
0.15	1660.3	1984	45.2	220	14.1	0.243	143.9	3102	0.7	G
0.15	1660.3	2008	21.8	206	26.1	0.289	124.7	5376.6	1.4	G
0.15	1660.3	1992	28.7	199	25.1	0.311	115.9	4994.9	1.4	G
0.15	1660.3	2024	36.6	187	23.6	0.32	102.4	4413.2	1.4	G
0.15	1660.3	2037	44.8	170	21.5	0.31	84.8	3655	1.4	G
0.15	1660.3	1990	56.4	154	19.5	0.353	69.7	3003	1.4	C
0.3	1660.3	1987	21.5	273	17.5	0.187	221.7	4777.5	0.7	C
0.3	1660.3	2003	30.4	257	16.5	0.213	196.8	4240.5	0.7	C
0.3	1660.3	1996	37.4	241	15.4	0.221	172.2	3711.4	0.7	G
0.3	1660.3	2035	44	224	14.3	0.229	148.6	3203.2	0.7	G

Depth (mm)	Pressure (kPa)	Age (yr)	Temp (°C)	Humidity (%)	Moisture (g/g)	Heat Flux (kW/m²)	Power (W)	Height (m)	First CHF	
0.3	1660.3	1993	20.4	209	26.6	0.273	129	5559.4	1.4	G
0.3	1660.3	2017	34.7	189	24.1	0.286	105.7	4554.9	1.4	G
0.3	1660.3	2037	51.3	164	20.9	0.336	79.5	3427.6	1.4	G
0.3	1660.3	2024	56.1	158	20	0.356	73.3	3160	1.4	C
0.5	1660.3	2008	18.4	281	18	0.163	234.7	5058	0.7	C
0.5	1660.3	1991	25.7	268	17.2	0.184	213.9	4609.6	0.7	C
0.5	1660.3	2008	32.3	256	16.4	0.205	194.8	4198.4	0.7	C
0.5	1660.3	2006	41.1	241	15.5	0.244	173.3	3735.5	0.7	GC
0.5	1660.3	1991	21.5	216	27.1	0.289	135.8	5851.4	1.4	G
0.5	1660.3	1983	31.8	203	25.5	0.317	120.2	5182.6	1.4	G
0.5	1660.3	1999	38.6	194	24.4	0.335	109.7	4727.8	1.4	GC
0.5	1660.3	2024	43.6	185	23.3	0.339	99.9	4306.8	1.4	GC
0.5	1660.3	1981	54.2	165	20.8	0.35	79.6	3432	1.4	C
0.5	1660.3	2007	56.2	162	20.5	0.359	77	3317.8	1.4	C
0.7	1660.3	1982	15.3	294	18.6	0.152	253.2	5456.6	0.7	C
0.7	1660.3	2034	19.9	286	18.1	0.166	239.8	5168	0.7	C
0.7	1660.3	1985	25.2	277	17.5	0.183	224.7	4842	0.7	C
0.7	1660.3	2027	31.8	262	16.6	0.196	201.7	4346.6	0.7	C
0.7	1660.3	2015	35.4	258	16.3	0.218	195.1	4205.4	0.7	C
0.7	1660.3	1991	16.3	226	28.6	0.278	150.1	6470.4	1.4	C
0.7	1660.3	2029	21.5	219	27.8	0.288	141.4	6094.8	1.4	G
0.7	1660.3	2034	26.8	212	26.9	0.299	132.4	5704.9	1.4	G
0.7	1660.3	2013	33.1	204	25.8	0.315	122.3	5269.3	1.4	G
0.7	1660.3	2030	37.1	198	25.1	0.324	115.3	4971.8	1.4	G
0.7	1660.3	2031	44	186	23.6	0.332	101.9	4391.5	1.4	C
0.7	1660.3	2032	51.3	173	21.9	0.342	87.9	3788.7	1.4	C
1	1660.3	1981	20.5	287	18.4	0.15	245	5280.8	0.7	C
1	1660.3	1986	27.3	274	17.6	0.171	223.8	4822.4	0.7	C
1	1660.3	2005	39.2	251	16.1	0.213	187.5	4041.1	0.7	C
1	1660.3	2023	27	246	22.3	0.239	178.2	5485.8	1	C
1	1660.3	2015	32.4	240	21.8	0.269	169.9	5232	1	C
1	1660.3	2031	20.4	227	28.9	0.287	152.2	6560.3	1.4	C
1	1660.3	2014	29.1	213	27.1	0.292	133.9	5772.3	1.4	C
1	1660.3	1984	39.2	198	25.2	0.315	115.8	4989.6	1.4	C
1	1660.3	2004	46.7	183	23.3	0.318	98.9	4263.9	1.4	C
1	1660.3	1980	54.8	170	21.6	0.343	85.2	3672	1.4	C
0	1660.3	3034	23.4	294	18.8	0.104	256.5	5527.2	0.7	G
0	1660.3	3005	30.1	280	17.9	0.128	232.6	5012	0.7	G

Gap (mm)	P (kPa)	G (kg/m ³)	T (°C)	I (A)	V (V)	X	CHF (kW/m ²)	Q (W)	$\frac{Q}{A}$ (m)	First CHF
0	1660.3	3025	39.6	255	16.3	0.153	192.9	4156.5	0.7	G
0	1660.3	3058	24.7	225	28.6	0.199	149.3	6435	1.4	G
0	1660.3	3043	33.7	209	26.5	0.214	128.5	5538.5	1.4	G
0	1660.3	3041	41.1	197	32.5	0.238	114.3	4925	1.4	G
0	1660.3	2979	53.4	181	23	0.304	96.6	4163	1.4	G
0.15	1660.3	3046	24.3	294	18.9	0.098	257.8	5556.6	0.7	C
0.15	1660.3	3035	32.7	275	17.6	0.125	224.6	4840	0.7	C
0.15	1660.3	3051	39.8	259	16.6	0.154	199.5	4299.4	0.7	C
0.15	1660.3	2976	24.9	227	28.8	0.189	151.7	6537.6	1.4	G
0.15	1660.3	3013	31.6	218	27.6	0.215	139.6	6016.8	1.4	G
0.15	1660.3	2974	40.5	201	25.5	0.233	118.9	5125.5	1.4	G
0.15	1660.3	3043	45.2	191	24.2	0.24	107.2	4622.2	1.4	C
0.3	1660.3	2982	23.6	304	19.5	0.106	275.1	5928	0.7	C
0.3	1660.3	3033	30.6	286	18.4	0.124	244.2	5262.4	0.7	C
0.3	1660.3	3037	38.3	268	17.2	0.151	213.9	4609.6	0.7	C
0.3	1660.3	3039	23.8	233	29.7	0.19	160.5	6920.1	1.4	GC
0.3	1660.3	3020	32.7	223	28.3	0.233	146.4	6310.9	1.4	GC
0.3	1660.3	2997	39.1	212	26.9	0.251	132.3	5702.8	1.4	G
0.3	1660.3	2997	46.9	196	24.9	0.267	113.2	4880.4	1.4	GC
0.3	1660.3	3011	55.9	186	23.7	0.326	102.3	4408.2	1.4	GC
0.5	1660.3	3006	21.2	311	20	0.082	288.6	6220	0.7	C
0.5	1660.3	2981	27.7	294	18.9	0.097	257.8	5556.6	0.7	C
0.5	1660.3	3004	23.7	282	18.1	0.103	236.8	5104.2	0.7	C
0.5	1660.3	3020	36.6	271	17.4	0.125	218.8	4715.4	0.7	C
0.5	1660.3	2977	24.2	243	29.8	0.196	168.2	7251.1	1.4	C
0.5	1660.3	2988	34.8	226	28.4	0.241	149	6422.9	1.4	C
0.5	1660.3	3042	35.1	225	28.3	0.239	147.5	6356.3	1.4	C
0.5	1660.3	3015	41.1	217	27.2	0.268	137	5906.7	1.4	GC
0.5	1660.3	2978	50.3	200	25.1	0.297	116.4	5018	1.4	GC
0.5	1660.3	3055	56	194	24.3	0.336	109.4	4714.2	1.4	G
0.7	1660.3	3049	21.2	316	20	0.069	293.1	6316.8	0.7	C
0.7	1660.3	2977	27.6	300	19	0.088	264.1	5691	0.7	C
0.7	1660.3	3018	31	291	18.4	0.099	248.6	5357.3	0.7	C
0.7	1660.3	3011	34.8	285	18	0.122	238.3	5135.7	0.7	C
0.7	1660.3	3046	41.5	264	16.7	0.138	204.6	4408.8	0.7	C
0.7	1660.3	3034	16.4	252	31.9	0.152	186.6	8041.3	1.4	C
0.7	1660.3	2982	19.6	248	31.5	0.166	181.1	7804.6	1.4	C
0.7	1660.3	3053	23.8	242	30.7	0.181	172.4	7429.4	1.4	C

GDP (mm)	P (kPa)	G (kg/m ³)	T _f (°C)	I (A)	V (V)	X (m)	CHF (kW/m ²)	Q (W)	f _h (m)	First CHF
0.7	1660.3	3015	29.4	233	29.5	0.198	159.6	6878.2	1.4	C
10.7	1660.3	2991	43.5	1214	27.1	0.265	134.3	5790.8	1.4	GC
1	1660.3	2992	23.2	319	20.4	0.074	36.1	6487.2	0.7	C
1	1660.3	2990	29.6	300	19.3	0.091	268.7	5790	0.7	C
1	1660.3	3035	35.5	285	18.3	0.132	242	5215.5	0.7	C
1	1660.3	3051	22.6	279	25.4	0.111	230.2	7086.6	1	C
1	1660.3	3044	35.2	252	22.9	0.149	187.4	5770.8	1	C
1	1660.3	3040	23.9	249	31	0.17	179.1	7719	1.4	C
1	1660.3	2974	30.3	235	29.9	0.193	163	7026.5	1.4	C
1	1660.3	3006	36.6	227	28.9	0.221	152.2	6560.3	1.4	C
1	1660.3	3024	44.46	213	27.1	0.265	133.9	5772.3	1.4	GC
1	1660.3	3015	55.6	199	25.3	0.318	116.8	5034.7	1.4	GC
m = 1 kg/min										
0	1130.1	635	20.2	209	13.3	0.83	129	2779.7	0.7	G
0	1130.1	633	28.2	201	12.8	0.825	119.4	2572.8	0.7	G
0	1130.1	630	37.7	194	12.4	0.851	111.6	2405.6	0.7	GC
0	1130.1	621	20.8	179	16.3	0.888	94.8	2917.7	1	GC
0	1130.1	629	27.7	174	15.9	0.894	89.9	2766.6	1	GC
0	1130.1	630	36.3	167	15.2	0.888	82.4	2538.4	1	GC
0	1130.1	636	19.9	155	19.6	0.925	70.5	3038	1.4	GC
0	1130.1	633	29.7	148	18.8	0.919	64.6	2782.4	1.4	GC
0	1130.1	636	38.8	141	17.9	0.906	58.6	2523.9	1.4	GC
0.15	1130.1	600	19.7	210	13.4	0.838	130.6	2814	0.7	G
0.15	1130.1	610	28.4	200	12.8	0.822	118.8	2560	0.7	G
0.15	1130.1	610	36.4	193	12.3	0.826	110.2	2373.9	0.7	G
0.15	1130.1	612	20.7	178	16.1	0.867	93.1	2865.8	1	G
0.15	1130.1	616	28	172	15.6	0.865	87.2	2683.2	1	G
0.15	1130.1	603	38.8	164	14.9	0.875	79.4	2443.6	1	G
0.15	1130.1	615	19.7	156	19.6	0.93	70.9	3057.6	1.4	GC
0.15	1130.1	602	28.9	150	18.9	0.93	65.8	2835	1.4	GC
0.15	1130.1	600	38.2	143	18.1	0.924	60	2588.3	1.4	GC
0.3	1130.1	587	19.4	207	13.3	0.812	127.7	2753.1	0.7	G
0.3	1130.1	587	27.5	203	13	0.844	122.5	2639	0.7	G
0.3	1130.1	590	38	191	12.2	0.825	108.1	2330.2	0.7	G
0.3	1130.1	585	20.3	183	16.6	0.928	98.7	3037.8	1	GC
0.3	1130.1	590	29.5	174	15.8	0.904	89.3	2749.2	1	G
0.3	1130.1	592	37.8	166	15	0.884	80.9	2490	1	G
0.3	1130.1	582	18.1	155	19.7	0.914	70.8	3053.5	1.4	G

Gap (mm)	IP (kPa)	G (kg/m ²)	T (°C)	I (A)	V (V)	IX (A)	CHF (kW/m ²)	Q (W)	L (m)	First CHF
0.3	1130.1	591	28	148	18.8	0.903	64.6	2782.4	1.4	G
0.3	1130.1	588	37.2	142	18	0.903	59.3	2556	1.4	G
0.5	1130.1	564	18.2	212	13.6	0.851	133.8	2883.2	0.7	G
0.5	1130.1	567	28.9	203	13	0.856	122.5	2639	0.7	G
0.5	1130.1	558	34.2	197	12.6	0.847	115.2	2482.2	0.7	G
0.5	1130.1	556	41.8	190	12.2	0.856	107.6	2318	0.7	GC
0.5	1130.1	568	18.8	155	19.6	0.915	70.5	3038	1.4	GC
0.5	1130.1	554	28.3	149	18.8	0.912	65	2801.2	1.4	GC
0.5	1130.1	554	39.7	141	17.7	0.904	57.9	2495.7	1.4	GC
0.7	1130.1	537	11.9	217	13.7	0.829	138.1	2977.2	0.7	G
0.7	1130.1	536	18.3	213	13.5	0.848	133.4	2875.5	0.7	GC
0.7	1130.1	531	26.9	208	13.2	0.874	127	2737.3	0.7	GC
0.7	1130.1	532	40.2	197	12.5	0.894	114.2	2460.5	0.7	GC
0.7	1130.1	534	13.6	184	16.6	0.875	99.3	3058.1	1	C
0.7	1130.1	530	23.6	178	16	0.887	92.6	2851.6	1	C
0.7	1130.1	533	37.9	167	15	0.891	81.5	2510	1	C
0.7	1130.1	533	19.4	154	19.5	0.908	69.8	3007.6	1.4	C
0.7	1130.1	537	22.9	152	19.3	0.913	68.1	2936.6	1.4	C
0.7	1130.1	542	28.8	148	18.8	0.91	64.6	2783.9	1.4	C
0.7	1130.1	531	34.8	143	18.2	0.898	60.4	2605.5	1.4	C
1	1130.1	506	17.3	214	13.7	0.86	136	2931.8	0.7	C
1	1130.1	513	28.6	206	13.2	0.883	126.2	2719.2	0.7	C
1	1130.1	506	39.2	197	12.6	0.893	115.2	2482.2	0.7	C
1	1130.1	503	17.1	184	16.7	0.912	99.8	3072.8	1	C
1	1130.1	508	26.6	177	16	0.907	92	2832	1	C
1	1130.1	502	36.6	168	15.3	0.902	83.5	2570.4	1	C
1	1130.1	511	25	151	19.2	0.918	67.3	2899.2	1.4	C
1	1130.1	508	36.5	141	18.3	0.905	59.9	2580.3	1.4	C
m = 3 kg/min										
0	1130.1	1878	21.4	238	15.4	0.251	170.1	3665.2	0.7	G
0	1130.1	1902	29.1	228	14.7	0.282	155.5	3351.6	0.7	G
0	1130.1	1914	36.9	222	14.3	0.333	147.3	3174.6	0.7	G
0	1130.1	1869	21.9	218	19.8	0.338	140.2	4316.4	1	G
0	1130.1	1887	29.9	206	18.6	0.35	124.5	3831.6	1	G
0	1130.1	1862	35.8	195	17.7	0.358	112.1	3451.5	1	G
0	1130.1	1881	22.5	191	24.1	0.379	106.8	4603.1	1.4	G
0	1130.1	1885	30	187	23.7	0.427	102.8	4431.9	1.4	G
0	1130.1	1911	37.4	180	22.7	0.453	94.8	4086	1.4	GC

Gap (mm)	P (kPa)	G (kg/m ³ s)	T _m (°C)	I (A)	V (V)	R _X	CHF (kW/m ²)	Q (W)	L _m (m)	First CHF
0.15	1130.1	1802	20.4	224	16	0.231	166.3	3584	0.7	G
0.15	1130.1	1805	28.7	221	15.2	0.279	155.9	3359.2	0.7	G
0.15	1130.1	1847	38.1	211	14.3	0.325	140	3017.3	0.7	G
0.15	1130.1	1810	28.2	210	19	0.354	129.6	3990	1	GC
0.15	1130.1	1828	22.9	201	25.4	0.446	118.4	5105.4	1.4	GC
0.15	1130.1	1831	31.1	191	24.1	0.459	106.8	4603.1	1.4	GC
0.15	1130.1	1805	38.2	180	22.8	0.463	95.2	4104	1.4	C
0.3	1130.1	1746	20.3	249	15.9	0.278	183.7	3959.1	0.7	G
0.3	1130.1	1755	29.3	237	15.2	0.316	167.2	3602.4	0.7	G
0.3	1130.1	1769	37.9	225	14.4	0.351	150.3	3240	0.7	G
0.3	1130.1	1747	20.4	197	25.1	0.403	114.7	4944.7	1.4	G
0.3	1130.1	1752	28.7	193	24.5	0.452	109.7	4728.5	1.4	C
0.3	1130.1	1743	37.5	182	23.1	0.469	97.5	4204.2	1.4	C
0.5	1130.1	1664	18.1	263	16.8	0.316	205	4418.4	0.7	G
0.5	1130.1	1666	25.8	248	15.9	0.326	183	3943.2	0.7	G
0.5	1130.1	1691	32.5	237	15.2	0.346	167.2	3602.4	0.7	G
0.5	1130.1	1689	39.3	225	14.4	0.364	150.3	3240	0.7	G
0.5	1130.1	1703	21.6	202	25.5	0.44	119.5	5151	1.4	GC
0.5	1130.1	1680	28.7	192	24.3	0.444	108.2	4665.6	1.4	GC
0.5	1130.1	1696	33.2	187	23.6	0.455	102.4	4413.2	1.4	GC
0.5	1130.1	1675	38.3	180	22.8	0.464	95.2	4104	1.4	GC
0.7	1130.1	1594	16.9	268	17	0.321	210.9	4545.3	0.7	C
0.7	1130.1	1594	25.8	253	15.9	0.338	187.2	4035.3	0.7	C
0.7	1130.1	1590	30.5	245	15.5	0.352	176.3	3799.9	0.7	GC
0.7	1130.1	1614	35.9	234	14.8	0.361	160.8	3465.5	0.7	G
0.7	1130.1	1589	41.8	225	14.2	0.383	148.7	3204	0.7	G
0.7	1130.1	1592	14.9	208	26.4	0.422	127.3	5487	1.4	C
0.7	1130.1	1632	21.2	201	25.5	0.435	119.1	5133.5	1.4	C
0.7	1130.1	1591	27.1	193	24.5	0.439	109.9	4738.1	1.4	C
0.7	1130.1	1625	32.3	187	23.8	0.45	103	4441.3	1.4	C
0.7	1130.1	1604	40.8	176	22.3	0.466	91.2	3931.8	1.4	C
1	1130.1	1534	19.6	265	16.9	0.337	207.8	4478.5	0.7	C
1	1130.1	1536	26.4	252	16.1	0.346	188.3	4057.2	0.7	C
1	1130.1	1520	36.4	236	15.1	0.377	165.4	3563.6	0.7	G
1	1130.1	1524	18.9	231	21	0.378	157.6	4851	1	G
1	1130.1	1536	26.3	223	20.2	0.402	146.3	4504.6	1	GC
1	1130.1	1500	36.2	209	18.9	0.424	128.3	3950.1	1	C

SP (mm)	P (kPa)	G (kg/min)	T (°C)	I (A)	V (V)	X (mm)	CHF (kW/m²)	Q (W)	h _m (m)	First CHF
1	1130.1	1502	19.7	204	25.9	0.44	122.6	5283.6	1.4	GC
1	1130.1	1530	28.2	193	24.5	0.448	109.7	4728.5	1.4	C
1	1130.1	1523	37.8	180	22.8	0.459	95.2	4104	1.4	C
m = 5 kg/min										
0	1130.1	3163	22.8	269	17.4	0.156	217.2	4680.6	0.7	G
0	1130.1	3124	30.3	246	15.9	0.167	181.5	3911.4	0.7	G
0	1130.1	3104	35.9	236	15.3	0.196	167.5	3610.8	0.7	G
0	1130.1	3119	35.7	221	20.1	0.257	144.3	4442.1	1	G
0	1130.1	3119	32.2	211	26.7	0.315	130.7	5633.7	1.4	G
0	1130.1	3118	37.1	204	25.8	0.333	122.1	5263.2	1.4	G
0.15	1130.1	3011	23.9	278	17.8	0.188	229.6	4948.4	0.7	GC
0.15	1130.1	3010	29.7	259	16.6	0.193	199.5	4299.4	0.7	G
0.15	1130.1	3037	38.8	248	15.9	0.251	183	3943.2	0.7	G
0.15	1130.1	3000	26.5	247	22.4	0.257	179.7	5532.8	1	GC
0.15	1130.1	3046	31.1	241	21.8	0.278	170.6	5253.8	1	GC
0.15	1130.1	3031	35.8	228	20.7	0.282	153.3	4719.6	1	GC
0.3	1130.1	2953	22.7	281	18	0.183	234.7	5058	0.7	GC
0.3	1130.1	2915	30.6	259	16.6	0.199	199.5	4299.4	0.7	G
0.3	1130.1	2890	38.3	252	16.1	0.253	188.3	4057.2	0.7	GC
0.3	1130.1	2951	28.9	245	22.2	0.269	176.7	5439	1	GC
0.3	1130.1	2923	38.6	226	20.5	0.299	150.5	4633	1	GC
0.3	1130.1	2932	34.1	216	27.5	0.356	137.8	5940	1.4	GC
0.5	1130.1	2769	19.5	284	18.2	0.162	239.8	5168.8	0.7	C
0.5	1130.1	2798	26.3	273	17.4	0.193	220.4	4750.2	0.7	C
0.5	1130.1	2790	30.9	264	16.9	0.214	207	4461.6	0.7	C
0.5	1130.1	2808	38.2	253	16.2	0.255	190.2	4098.6	0.7	C
0.5	1130.1	2779	33.3	217	27.5	0.35	138.4	5967.5	1.4	GC
0.5	1130.1	2823	39.5	208	26.3	0.371	126.9	5470.4	1.4	GC
0.7	1130.1	2657	16.5	290	18.4	0.147	246.9	5321.5	0.7	C
0.7	1130.1	2663	20.2	285	18	0.167	238.6	5141.4	0.7	C
0.7	1130.1	2688	26.6	271	17.1	0.187	215	4634.1	0.7	C
0.7	1130.1	2711	31.1	266	16.8	0.216	207.4	4468.8	0.7	C
0.7	1130.1	2681	17.2	236	30	0.286	164	7070.6	1.4	C
0.7	1130.1	2669	28.2	225	28.5	0.336	148.7	6408	1.4	C
0.7	1130.1	2674	32.3	218	27.6	0.345	139.8	6025.5	1.4	C
0.7	1130.1	2684	34.4	215	27.2	0.352	135.8	5852.3	1.4	G
0.7	1130.1	2662	37.2	209	26.6	0.356	128.8	5553.1	1.4	G

Gap (mm)	P (kPa)	P/G (kg/m ² s)	T (°C)	ΔT (°C)	V (m)	X (m)	CHF (kW/m ²)	Q (W)	L ₀ (m)	First CHF
1	1130.1	2566	20.3	283	18.1	0.166	237.7	5122.3	0.7	C
1	1130.1	2535	26.9	274	17.5	0.202	222.5	4795	0.7	C
1	1130.1	2527	23.4	257	23.4	0.262	195.3	6013.8	1	GC
1	1130.1	2510	32.5	240	21.8	0.287	169.9	5232	1	GC
1	1130.1	2540	37.9	231	21	0.309	157.6	4851	1	C
1	1130.1	2510	28.1	223	28.3	0.328	146.4	6310.9	1.4	C
1	1130.1	2529	38.3	208	26.4	0.362	127.4	5491.2	1.4	GC
m = 9 kg/min										
0	1660.3	634	20.1	197	12.6	0.64	115.2	2482.2	0.7	C
0	1660.3	634	30.3	187	11.9	0.636	103.3	2225.3	0.7	GC
0	1660.3	635	40.7	177	11.3	0.65	92.8	2000.1	0.7	GC
0	1660.3	631	50.3	166	10.6	0.652	81.6	1759.6	0.7	GC
0	1660.3	634	53.4	162	10.4	0.654	78.2	1684.8	0.7	GC
0	1660.3	631	19.4	144	18.3	0.698	61.1	2635.2	1.4	GC
0	1660.3	636	30.9	137	17.3	0.704	55	2370.1	1.4	GC
0	1660.3	638	39.5	130	16.5	0.699	49.8	2145	1.4	GC
0	1660.3	632	48.8	123	15.6	0.703	44.5	1918.8	1.4	GC
0	1660.3	633	56.6	115	14.6	0.687	39	1679	1.4	GC
0.15	1660.3	606	19.8	201	12.9	0.683	120.3	2592.9	0.7	G
0.15	1660.3	605	27.3	193	12.3	0.667	110.2	2373.9	0.7	G
0.15	1660.3	615	37.4	182	11.6	0.662	98	2111.2	0.7	G
0.15	1660.3	610	48.6	170	10.9	0.672	86	1853	0.7	G
0.15	1660.3	606	55.4	162	10.4	0.675	78.2	1684.8	0.7	GC
0.15	1660.3	612	20.1	144	18.2	0.698	60.8	2620.8	1.4	GC
0.15	1660.3	608	30.8	137	17.3	0.703	55	2370.1	1.4	GC
0.15	1660.3	605	41.8	129	16.4	0.711	49.1	2115.6	1.4	GC
0.15	1660.3	604	50.2	122	15.4	0.701	43.6	1878.8	1.4	GC
0.15	1660.3	600	55.7	118	14.9	0.71	40.8	1758.2	1.4	GC
0.3	1660.3	590	19.1	199	12.8	0.657	118.2	2547.2	0.7	G
0.3	1660.3	578	28.2	190	12.2	0.653	107.6	2318	0.7	G
0.3	1660.3	584	37.7	181	11.6	0.66	97.4	2099.6	0.7	G
0.3	1660.3	592	47.6	172	11	0.678	87.8	1892	0.7	GC
0.3	1660.3	584	56.6	160	10.3	0.673	76.5	1648	0.7	GC
0.3	1660.3	580	17.9	145	18.4	0.696	61.9	2668	1.4	C
0.3	1660.3	582	27.3	139	17.6	0.699	56.8	2446.4	1.4	GC
0.3	1660.3	579	35.5	133	17	0.706	52.5	2261	1.4	GC
0.3	1660.3	589	46.8	126	16.1	0.728	47.1	2028.6	1.4	GC
0.3	1660.3	588	56.3	118	15	0.722	41.1	1770	1.4	GC

δ (mm)	T (K)	$\dot{m}$ (kg/min)	T_c (°C)	T_h (K)	V (V)	X (mm)	CHF (kW/m ²)	Q (W)	h_{eff} (m)	First CHF
0.5	1660.3	567	16.8	203	13	0.673	122.5	2639	0.7	G
0.5	1660.3	559	24.4	195	12.5	0.665	113.1	2437.5	0.7	G
0.5	1660.3	561	33.9	188	12.1	0.695	105.6	2274.8	0.7	G
0.5	1660.3	560	42.9	177	11.3	0.674	92.8	2000.1	0.7	G
0.5	1660.3	560	50	170	10.9	0.688	86	1853	0.7	GC
0.5	1660.3	557	57.9	161	10.3	0.692	76.9	1658.3	0.7	GC
0.5	1660.3	556	18.8	147	18.5	0.727	63.1	2718	1.4	C
0.5	1660.3	562	28.9	139	17.5	0.709	56.4	2431.1	1.4	C
0.5	1660.3	567	38	135	17	0.744	53.1	2288.3	1.4	GC
0.5	1660.3	553	48.2	127	16	0.743	47.1	2028.2	1.4	GC
0.5	1660.3	562	56.7	120	15.1	0.742	41.9	1806	1.4	GC
0.7	1660.3	544	10.5	213	13.4	0.705	132.9	2864.8	0.7	GC
0.7	1660.3	541	22.6	202	12.8	0.707	119.8	2581.6	0.7	GC
0.7	1660.3	534	36.7	188	11.9	0.709	103.9	2239.1	0.7	GC
0.7	1660.3	536	45.3	179	11.4	0.713	94.4	2033.4	0.7	GC
0.7	1660.3	536	24.7	168	15.2	0.714	82.7	2546.9	1	C
0.7	1660.3	531	12.4	150	19	0.722	66.3	2856	1.4	C
0.7	1660.3	542	30.8	138	17.5	0.722	56.1	2417.8	1.4	C
0.7	1660.3	545	40.3	131	16.6	0.721	50.5	2177.2	1.4	C
0.7	1660.3	534	51.1	122	15.6	0.718	44	1897.1	1.4	C
1	1660.3	501	17.2	207	13.2	0.716	126.8	2732.4	0.7	C
1	1660.3	499	27.2	199	12.7	0.731	117.3	2527.3	0.7	C
1	1660.3	509	39.2	186	11.9	0.724	102.7	2213.4	0.7	C
1	1660.3	510	47.6	176	11.2	0.712	91.5	1971.2	0.7	C
1	1660.3	501	28.8	165	15	0.726	80.4	2475	1	C
1	1660.3	504	39.9	156	14.2	0.733	72	2215.2	1	C
1	1660.3	508	17.2	149	19	0.758	65.7	2831	1.4	C
1	1660.3	499	28.4	141	18	0.749	58.9	2538	1.4	C
1	1660.3	513	38.6	135	17.2	0.764	53.9	2322	1.4	C
1	1660.3	504	46.8	129	16.4	0.765	49.1	2115.6	1.4	C
1	1660.3	506	57	119	15.1	0.741	41.7	1796.9	1.4	C
$m = 3 \text{ kg/min}$										
0	1660.3	1907	21.9	254	16.3	0.186	192.1	4140.2	0.7	G
0	1660.3	1902	29.1	247	15.8	0.227	181.1	3902.6	0.7	G
0	1660.3	1907	38.3	228	14.6	0.243	154.5	3328.8	0.7	G
0	1660.3	1873	47.4	206	13.2	0.253	126.2	2719.2	0.7	G
0	1660.3	1910	22.7	199	25.3	0.322	116.8	5034.7	1.4	G
0	1660.3	1875	30.9	189	23.9	0.334	104.8	4517.1	1.4	G

Depth (m)	Pressure (kPa)	Temperature (K)	Salinity (g)	Density (kg)	Velocity (m/s)	Acceleration (g)	Heat Flux (W/m²)	Power (W)	Depth (m)	Salinity (g)
0	1660.3	1893	47.4	162	20.6	0.342	77.4	3337.2	1.4	G
0	1660.3	1885	53.2	152	19.3	0.348	68.1	2933.6	1.4	GC
0.15	1660.3	1811	22.6	259	16.6	0.215	199.5	4299.4	0.7	G
0.15	1660.3	1795	30.2	245	15.7	0.23	178.5	3846.5	0.7	G
0.15	1660.3	1828	39.5	227	14.6	0.253	153.8	3314.2	0.7	G
0.15	1660.3	1801	47.6	212	13.6	0.279	133.8	2883.2	0.7	G
0.15	1660.3	1834	21.7	202	25.6	0.331	120	5171.2	1.4	G
0.15	1660.3	1842	30.1	189	23.9	0.325	104.8	4517.1	1.4	G
0.15	1660.3	1796	35.5	181	22.9	0.329	96.2	4144.9	1.4	G
0.15	1660.3	1832	45.8	162	20.6	0.325	77.4	3337.2	1.4	G
0.3	1660.3	1740	20.5	265	17	0.223	209	4505	0.7	C
0.3	1660.3	1748	28.7	250	16	0.237	185.6	4000	0.7	G
0.3	1660.3	1732	36.9	232	14.9	0.246	160.4	3456.8	0.7	G
0.3	1660.3	1769	45.6	215	13.8	0.27	137.7	2967	0.7	G
0.3	1660.3	1746	18.4	205	26.1	0.322	124.1	5350.5	1.4	G
0.3	1660.3	1780	25.8	194	24.7	0.319	111.2	4791.8	1.4	G
0.3	1660.3	1750	35.1	178	22.6	0.308	93.3	4022.8	1.4	G
0.3	1660.3	1744	45.1	164	20.8	0.328	79.1	3411.2	1.4	G
0.3	1660.3	1763	55.3	149	19	0.357	65.7	2831	1.4	C
0.5	1660.3	1671	18.3	267	17.1	0.209	211.9	4565.7	0.7	C
0.5	1660.3	1688	25.2	254	16.3	0.23	192.1	4140.2	0.7	C
0.5	1660.3	1685	32.1	248	15.9	0.264	183	3943.2	0.7	GC
0.5	1660.3	1685	38.1	237	15.2	0.28	167.2	3602.4	0.7	G
0.5	1660.3	1678	43.8	226	14.5	0.294	152.1	3277	0.7	G
0.5	1660.3	1656	21.2	205	25.7	0.339	122.1	5264.4	1.4	G
0.5	1660.3	1663	33.9	187	23.5	0.348	101.9	4392.6	1.4	G
0.5	1660.3	1662	40.8	178	22.4	0.363	92.4	3983.6	1.4	G
0.5	1660.3	1703	47.7	167	21	0.369	81.3	3505.3	1.4	GC
0.5	1660.3	1696	54.6	151	19.1	0.357	66.9	2884.1	1.4	C
0.7	1660.3	1602	13.8	275	17.4	0.195	221.8	4779.5	0.7	C
0.7	1660.3	1589	19.6	267	16.9	0.213	208.8	4499	0.7	C
0.7	1660.3	1631	26.5	256	16.2	0.235	192.4	4147.2	0.7	C
0.7	1660.3	1623	33.9	245	15.5	0.262	176	3792.6	0.7	C
0.7	1660.3	1612	39.8	239	15.1	0.298	167.2	3604.1	0.7	C
0.7	1660.3	1631	46.6	228	14.4	0.327	152.7	3290	0.7	GC
0.7	1660.3	1606	15.6	211	26.7	0.336	130.9	5640	1.4	G
0.7	1660.3	1594	21.7	203	25.8	0.341	121.6	5241.5	1.4	G

Gap (mm)	P (kPa)	G (kg/m ³)	T _f (°C)	R (kA)	V (V)	X (kV)	CHF (kW/m ²)	Q (W)	h ₀ (m)	First CHF
0.7	1660.3	1610	30.7	183	24.5	0.356	109.8	4732.4	1.4	G
0.7	1660.3	1619	37.7	182	23.1	0.362	97.7	4209.7	1.4	G
1.0	1660.3	1634	47.7	166	21.1	0.369	81.2	3499.3	1.4	C
1.0	1660.3	1514	19.7	266	16.7	0.217	209.8	4522	0.7	C
1	1660.3	1523	26.8	257	16.4	0.247	195.6	4214.8	0.7	C
1	1660.3	1510	34.3	245	15.7	0.274	178.5	3846.5	0.7	C
1	1660.3	1516	19.2	242	22	0.327	172.9	5324	1	C
1	1660.3	1520	26.9	231	14.1	0.339	157.6	4851	1	GC
1	1660.3	1525	35.3	215	19.5	0.334	136.2	4192.5	1	G
1	1660.3	1506	19.8	208	26.4	0.357	127.4	5491.2	1.4	G
1	1660.3	1508	27.8	197	25.1	0.362	114.7	4944.7	1.4	G
1	1660.3	1520	34.3	188	23.9	0.366	104.2	4493.2	1.4	G
1	1660.3	1522	43.8	173	22	0.37	88.3	3806	1.4	G
1	1660.3	1517	49.3	163	20.7	0.368	78.3	3374.1	1.4	GC
1	1660.3	1523	56.2	150	19.1	0.372	66.5	2865	1.4	GC
m = 5 kg/min										
0	1660.3	3166	32.6	294	18.8	0.085	256.5	5527.2	0.7	G
0	1660.3	3184	30.8	280	17.9	0.117	232.6	5012	0.7	G
0	1660.3	3137	39.8	254	16.3	0.138	192.1	4140.2	0.7	G
0	1660.3	3173	24.9	225	28.6	0.177	149.3	6435	1.4	G
0	1660.3	3121	32.6	213	27.1	0.201	133.9	5772.3	1.4	G
0	1660.3	3193	41.3	203	25.7	0.246	121	5217.1	1.4	G
0	1660.3	3109	49.3	193	24.5	0.291	109.7	4728.5	1.4	G
0	1660.3	3171	53.8	185	23.5	0.308	100.9	4347.5	1.4	GC
0.15	1660.3	3063	24.3	294	18.9	0.098	257.8	5556.6	0.7	C
0.15	1660.3	3001	32.7	275	17.6	0.125	224.6	4840	0.7	C
0.15	1660.3	2995	39.8	259	16.6	0.154	199.5	4299.4	0.7	C
0.15	1660.3	3059	24.9	227	28.8	0.189	151.7	6537.6	1.4	G
0.15	1660.3	3004	31.6	218	27.6	0.215	139.6	6016.8	1.4	G
0.15	1660.3	3054	40.5	201	25.5	0.233	118.9	5125.5	1.4	C
0.15	1660.3	3023	45.2	191	24.2	0.24	107.2	4622.2	1.4	C
0.3	1660.3	2927	23.2	302	19.4	0.11	271.9	5858.8	0.7	C
0.3	1660.3	2936	32.7	279	17.9	0.136	231.7	4994.1	0.7	C
0.3	1660.3	2965	38.8	264	16.9	0.155	207	4461.6	0.7	C
0.3	1660.3	2966	23.3	233	29.6	0.2	180	6896.8	1.4	GC
0.3	1660.3	2939	33.2	220	27.9	0.239	142.4	6138	1.4	C
0.3	1660.3	2955	39.7	208	26.4	0.246	127.4	5491.2	1.4	GC
0.3	1660.3	2952	47.6	193	24.6	0.275	110.2	4747.8	1.4	GC

Gap (mm)	P (kPa)	G (kPa)	T _g (°C)	T _m (°C)	V (m/s)	X (m)	CHF (kW/m ²)	Q (W)	I _{ac} (m)	First CHF
0.3	1660.3	2887	55.3	185	23.5	0.325	100.9	4347.5	1.4	C
0.5	1660.3	2762	21.3	304	19.5	0.096	275.1	5928	0.7	C
0.5	1660.3	2788	28	287	18.4	0.11	245	5280.8	0.7	C
0.5	1660.3	2831	33.3	273	17.5	0.123	221.7	4777.5	0.7	C
0.5	1660.3	2835	36.8	264	17	0.135	207.6	4474.8	0.7	C
0.5	1660.3	2825	24.2	237	29.7	0.221	163.3	7038.9	1.4	C
0.5	1660.3	2793	33	223	28	0.246	145	6248.5	1.4	C
0.5	1660.3	2793	41.1	212	26.6	0.281	131.1	5649.8	1.4	GC
0.5	1660.3	2787	44.1	207	26.2	0.294	125.8	5423.4	1.4	GC
0.5	1660.3	2834	51.2	194	24.5	0.315	110.3	4753	1.4	GC
0.5	1660.3	2795	57.8	184	23.3	0.348	99.5	4287.2	1.4	GC
0.7	1660.3	2712	16.9	316	20	0.083	292.5	6304.2	0.7	C
0.7	1660.3	2721	20.5	306	19.3	0.087	274.6	5918	0.7	C
0.7	1660.3	2721	25.6	294	18.6	0.101	253.7	5468.4	0.7	C
0.7	1660.3	2696	30.1	285	18	0.12	238.3	5135.7	0.7	C
0.7	1660.3	2657	36.1	269	17	0.136	212.2	4573	0.7	C
0.7	1660.3	2688	42.2	253	16	0.156	187.8	4048	0.7	C
0.7	1660.3	2718	16.7	245	31.1	0.194	176.8	7619.5	1.4	C
0.7	1660.3	2690	22.4	238	30.2	0.215	166.6	7180.5	1.4	G
0.7	1660.3	2695	30.6	227	28.7	0.244	151.4	6524	1.4	G
0.7	1660.3	2713	41.4	210	26.6	0.28	129.7	5590.2	1.4	GC
0.7	1660.3	2671	46.6	200	25.4	0.293	118	5088	1.4	GC
0.7	1660.3	2715	49.3	195	24.8	0.301	112.2	4836	1.4	GC
1	1660.3	2516	21.9	305	19.6	0.107	277.4	5978	0.7	C
1	1660.3	2504	28.9	288	18.5	0.124	247.2	5328	0.7	C
1	1660.3	2507	21.2	267	24.2	0.141	209.9	6461.4	1	C
1	1660.3	2523	29.4	252	22.9	0.167	187.4	5770.8	1	C
1	1660.3	2511	22.5	238	30.3	0.218	167.3	7211.4	1.4	C
1	1660.3	2506	30.2	227	28.9	0.243	152.2	6560.3	1.4	C
1	1660.3	2563	37.7	218	27.7	0.278	140.1	6038.6	1.4	GC
1	1660.3	2561	45.1	202	25.7	0.285	120.4	5191.4	1.4	C
1	1660.3	2504	56.2	186	23.7	0.34	102.3	4408.2	1.4	GC

Note:

- G represents the CHF occurs at the Gap region first.
- C represents the CHF occurs at the main Channel region first.
- GC represents the CHF occurs at the Gap region and main Channel region simultaneously.

Gap (mm)	P (kPa)	G (kN/m²)	T _{in} (°C)	X (mm)	CHF (kW/m²)	ΔT _{in} (°C)	CHF _{in} (kW/m²)	ΔT _{out} (°C)	CHF _{out} (kW/m²)	ΔT _{avg} (°C)	CHF _{avg} (kW/m²)	ΔT _{diff} (°C)	CHF _{diff} (kW/m²)
0.3	1660.3	1996	37.4	0.177	152	102.1	43.6	172.2	102.1	35.8	0.7	-11.7	
0.3	1660.3	2035	37.4	0.209	138.2	124.4	58.9	156.3	124.4	35.8	0.7	-11.7	
0.3	1660.3	1993	20.4	0.252	123	124.4	77.1	129	126.7	77.1	1.4	-4.7	
0.3	1660.3	2017	34.7	0.331	105.7	138.8	123.2	79.5	153.8	123.2	1.4	-4.7	
0.3	1660.3	2037	51.3	0.383	86.2	156.4	123.2	79.5	153.8	123.2	1.4	-4.7	
0.3	1660.3	2024	56.1	0.403	80	161.8	130.7	73.3	161.8	130.7	1.4	-4.7	
0.5	1660.3	2008	18.4	0.104	209	74.2	-2.2	234.7	74.2	-11.6	0.7	-10.9	
0.5	1660.3	1991	18.4	0.104	209	74.2	-2.2	234.7	74.2	-11.6	0.7	-10.9	
0.5	1660.3	2008	32.3	0.154	172.7	94.5	31.4	194.8	94.5	23.3	0.7	-11.3	
0.5	1660.3	2008	32.3	0.154	172.7	94.5	31.4	194.8	94.5	23.3	0.7	-11.3	
0.5	1660.3	2008	44.1	0.185	151.8	107.7	122.2	78.3	107.7	78.3	1.4	-5.9	
0.5	1660.3	1991	21.5	0.257	127.8	125.4	78.7	135.8	128.3	78.7	1.4	-5.9	
0.5	1660.3	1983	31.8	0.298	115	135.8	103.9	109.7	144	103.9	1.4	-2.7	
0.5	1660.3	1999	38.6	0.328	106.7	142.9	103.9	109.7	144	103.9	1.4	-2.7	
0.5	1660.3	2024	43.6	0.35	100.5	142.9	103.9	109.7	144	103.9	1.4	-2.7	
0.5	1660.3	1981	54.2	0.396	86.9	159.5	127.7	79.6	156.8	127.7	1.4	-9.1	
0.5	1660.3	2007	56.2	0.403	84.3	161.8	130.7	73.3	161.8	130.7	1.4	-9.1	
0.7	1660.3	1982	15.3	0.095	229.5	69.8	-10.1	253.2	69.8	-18.4	0.7	-9.4	
0.7	1660.3	2034	19.9	0.108	215.4	76.8	-10.1	253.2	69.8	-18.4	0.7	-9.4	
0.7	1660.3	1985	25.2	0.126	200.1	84	14.3	224.7	84	5.8	0.7	-10.9	
0.7	1660.3	2027	31.8	0.151	182.8	103.9	103.9	109.7	144	103.9	1.4	-2.7	
0.7	1660.3	2015	35.4	0.167	173	99.1	38.9	195.1	99.1	31.2	0.7	-11.3	
0.7	1660.3	1991	16.3	0.238	140.7	120.2	73.2	150.1	123.5	73.2	1.4	-5.1	
0.7	1660.3	2029	21.5	0.257	134.2	125.4	78.7	141.4	127.9	78.7	1.4	-5.1	
0.7	1660.3	2034	26.8	0.278	127.2	130.7	86.4	132.4	132.5	86.4	1.4	-5.1	
0.7	1660.3	2013	33.1	0.304	119	137.1	95.7	122.3	138.3	95.7	1.4	-2.7	
0.7	1660.3	2030	37.1	0.321	113.9	141.3	101.7	115.3	141.8	101.7	1.4	-2.7	
0.7	1660.3	2031	44	0.352	104.9	148.6	112.1	101.9	147.5	112.1	1.4	-2.7	
0.7	1660.3	2032	51.3	0.383	95.8	156.6	123.2	87.8	153.8	123.2	1.4	-2.7	
1	1660.3	1981	20.5	0.11	227.8	77.2	3.1	245	77.2	-2.5	0.7	-7	
1	1660.3	1986	27.3	0.133	208.1	87.1	19.5	223.8	87.1	14.3	0.7	-6.3	
1	1660.3	2005	39.2	0.183	175.8	104.8	47.6	187.5	104.8	43.8	0.7	-6.3	
1	1660.3	2023	27	0.2	167.1	110	55.8	178.2	111.5	53.8	0.7	-6.3	
1	1660.3	2015	32.4	0.223	156.6	116.5	65.5	169.9	118.3	63.1	1	-7.8	
1	1660.3	2031	20.4	0.252	145.1	124.3	77.1	162.2	126.6	77.1	1.4	-0.8	
1	1660.3	2014	29.1	0.286	132.9	133	89.8	133.9	133.3	89.8	1.4	-0.8	
1	1660.3	1984	39.2	0.33	118.8	143.5	104.8	115.8	142.5	104.8	1.4	-0.8	
1	1660.3	2004	46.7	0.364	108.6	151.5	116.2	98.9	148.3	116.2	1.4	-0.8	

Appendix V

PREDICTED CHF FOR NON-UNIFORM HEATING AND COMPARISON WITH THE RESULTS OF TEST AND UNIFORM HEATING

Gap (mm)	TP (t/s)	G (kg/m²)	T _g (°C)	L (m)	X _g (mm)	CHF _g (kW/m²)	F _{CHF} (N)	CHF _g (kW/m²)	X _g (mm)	E _{CHF} (J/kg)	E _{CHF} (%)	CHF _g (kW/m²)	X _g (mm)	E _{CHF} (J/kg)	E _{CHF} (%)	CHF _g (kW/m²)	X _g (mm)	E _{CHF} (J/kg)	E _{CHF} (%)	
0	1130.1	993	21.4	0.7	0.541	147.3	G	0.37	110.5	0.392	-25	0.392	114.6	G	-21.7	3.7				
0	1130.1	1010	29.2	0.7	0.574	139.3	G	0.403	102.3	0.424	-26.6	0.424	108.8	G	-22.9	4.1				
0	1130.1	1004	37.2	0.7	0.596	129.3	G	0.436	95	0.459	-26.6	0.459	99.1	G	-22.9	4.3				
0	1130.1	991	21.2	1	0.614	113.3	G	0.463	89.4	0.486	-21.1	0.486	85.3	G	-18.2	4.1				
0	1130.1	991	28.7	1	0.644	107.9	G	0.49	84.3	0.516	-21.9	0.516	87.8	G	-18.2	4.1				
0	1130.1	1010	37.4	1	0.674	100.4	GC	0.528	77.8	0.553	-22.2	0.553	74.1	GC	-12	3.4				
0	1130.1	991	21.6	1.4	0.666	85.7	GC	0.556	72.6	0.586	-15.2	0.586	75.1	G	-12	3.4				
0	1130.1	991	28.3	1.4	0.674	80	GC	0.588	68.7	0.608	-14.1	0.608	71.3	GC	-10.4	3.8				
0	1130.1	1000	38.3	1.4	0.67	70.8	GC	0.613	63.3	0.64	-10.5	0.64	65.9	G	-6.5	4.1				
0.15	1130.1	999	20.4	0.7	0.511	148.6	G	0.388	116.7	0.392	-21.3	0.392	120.8	G	-21.7	4.3				
0.15	1130.1	1006	28.7	0.7	0.568	144.6	G	0.403	107.9	0.425	-25.4	0.425	112.5	G	-21.7	4.3				
0.15	1130.1	1016	38.1	0.7	0.595	132.2	G	0.443	98.6	0.465	-25.4	0.465	103.4	G	-21.3	4.3				
0.15	1130.1	1015	36.6	1	0.648	102.1	G	0.524	82.6	0.55	-19.1	0.55	86.2	G	-15.1	4.4				
0.15	1130.1	1002	21.4	1.4	0.672	89.7	GC	0.558	76.4	0.588	-14.8	0.588	79.3	GC	-11.2	3.7				
0.15	1130.1	1016	29.9	1.4	0.69	83.5	GC	0.589	71.1	0.616	-14.8	0.616	74.1	G	-10.8	4.2				
0.15	1130.1	1018	38.4	1.4	0.685	76	GC	0.617	68.2	0.644	-11.7	0.644	69.2	GC	-7.3	4.5				
0.3	1130.1	1001	20.2	0.7	0.504	152.7	G	0.37	121.7	0.393	-20.3	0.393	126.6	G	-16.6	4				
0.3	1130.1	994	26.2	0.7	0.559	146.9	G	0.405	112.5	0.428	-23.4	0.428	117.2	G	-19.8	4.2				
0.3	1130.1	1016	38.6	0.7	0.57	130.6	G	0.447	102.5	0.47	-21.5	0.47	107.2	G	-17.4	4.6				
0.3	1130.1	1008	22.3	1	0.579	115.1	G	0.472	97.1	0.498	-15.6	0.498	101.3	GC	-11.6	4.3				
0.3	1130.1	1011	29.2	1	0.614	110.9	G	0.498	91.8	0.524	-17.2	0.524	95.8	G	-13.1	4.4				
0.3	1130.1	1018	38.6	1	0.648	102.6	G	0.535	84.6	0.562	-17.6	0.562	88.3	G	-13.6	4.3				
0.3	1130.1	1002	19.1	1.4	0.664	94.3	G	0.554	81	0.584	-14.1	0.584	84.1	G	-10.4	3.8				
0.3	1130.1	995	27.3	1.4	0.674	87.4	GC	0.581	76.1	0.61	-12.9	0.61	79.3	GC	-8.8	4.2				
0.3	1130.1	996	37.3	1.4	0.673	77.4	G	0.616	69.8	0.645	-9.8	0.645	72.7	G	-5.7	4.1				
0.5	1130.1	1013	18.2	0.7	0.47	156.2	G	0.363	130.6	0.388	-18.4	0.388	135.6	GC	-12.7	3.8				
0.5	1130.1	1008	25.6	0.7	0.5	148	G	0.391	122.6	0.417	-17.2	0.417	127.3	G	-13.5	3.8				
0.5	1130.1	1016	33.3	0.7	0.535	140	G	0.425	119.7	0.448	-18.8	0.448	119.3	GC	-14.3	4.6				
0.5	1130.1	1004	40	0.7	0.581	136	G	0.455	106.8	0.479	-21.5	0.479	111.7	G	-17.4	4.6				
0.5	1130.1	998	19	1.4	0.646	96.7	C	0.558	85.4	0.588	-11.7	0.588	89.2	GC	-7.3	4.5				

Gap (mm)	P (kPa)	G (kN/m ²)	T _a (°C)	L (m)	X ₀ (mm)	CHF ₀ (kW/m ²)	F _{CHF} (CHF)	X ₀ (mm)	CHF ₁ (kW/m ²)	ET _{CHF} (°C)	CHF ₁ (kW/m ²)	ET _{CHF} (°C)			
0.3	1130.1	1981	29.2	0.7	0.268	171.5	G	0.251	160.8	-6.3	0.268	167.6	G	-1.8	4.3
0.3	1130.1	2033	37.4	0.7	0.316	159.3	G	0.288	145	-14.5	0.305	153.9	G	-1.8	3.8
0.3	1130.1	2017	22.1	1.4	0.388	124.7	G	0.364	117.4	-5.9	0.385	121.9	G	-1.8	3.8
0.3	1130.1	2027	39.1	1.4	0.419	113.2	G	0.397	108.1	-14.1	0.416	110.5	G	-1.8	3.8
0.3	1130.1	2021	36.9	1.4	0.428	104.2	C	0.415	99.3	-4.7	0.435	103.5	G	-0.2	4.2
0.5	1130.1	2027	33.8	0.7	0.251	192.9	G	0.269	178.1	-8.6	0.253	185	G	-4.5	3.9
0.5	1130.1	2014	25.5	0.7	0.269	194.8	G	0.235	178.1	-8.6	0.253	185	G	-4.5	3.9
0.5	1130.1	2032	32.8	0.7	0.288	178.5	G	0.267	162.6	-14.1	0.284	169.3	G	-1.8	4.7
0.5	1130.1	1982	39.4	0.7	0.319	159.3	G	0.298	148.7	-6.6	0.316	155.7	G	-1.8	4.7
0.5	1130.1	2038	25.3	1.4	0.397	125.8	G	0.375	118.9	-14.1	0.398	123.3	G	-1.8	4.2
0.5	1130.1	1995	30.7	1.4	0.408	117.4	GC	0.395	112.8	-3.9	0.415	117.5	G	0.5	4.2
0.5	1130.1	2008	37.2	1.4	0.417	105.8	G	0.418	104.1	-14.1	0.437	109.3	G	-1.8	4.2
0.5	1130.1	2000	40.6	1.4	0.429	101.4	C	0.427	99.4	-2	0.446	103.9	G	2.9	4.5
0.7	1130.1	2007	33.8	0.7	0.233	235.1	G	0.191	214.9	-14.1	0.206	222.2	G	-1.8	3.8
0.7	1130.1	2019	20.6	0.7	0.257	218.1	C	0.217	197.7	-9.4	0.234	205.4	G	-5.3	4
0.7	1130.1	2033	28.8	0.7	0.282	204.8	G	0.224	183.8	-14.1	0.257	191.3	G	-1.8	4.2
0.7	1130.1	2019	30.9	0.7	0.295	192	C	0.26	174	-9.4	0.277	181.6	G	-4.9	4.4
0.7	1130.1	2022	38.9	0.7	0.305	171.5	G	0.288	161.5	-14.1	0.304	168.9	G	-1.8	4.2
0.7	1130.1	2020	42.5	0.7	0.332	159	G	0.314	149.7	-5.9	0.331	157.2	G	-0.6	5.1
0.7	1130.1	1987	15.8	1.4	0.371	145.2	G	0.344	132.8	-14.1	0.366	141.9	G	-1.8	4.2
0.7	1130.1	1998	21.9	1.4	0.376	133.6	C	0.365	129.9	-2.7	0.385	134.9	C	1.5	3.9
0.7	1130.1	2011	26.7	1.4	0.383	125.2	G	0.341	123.7	-14.1	0.402	127.7	G	-1.8	4.2
0.7	1130.1	2011	30.8	1.4	0.394	118.9	C	0.396	118	-0.8	0.413	122.6	C	3.6	3.9
0.7	1130.1	2019	36.1	1.4	0.402	109.4	G	0.413	110.6	-14.1	0.429	114.3	G	-1.8	4.2
0.7	1130.1	2032	40.7	1.4	0.413	101.9	C	0.427	104.3	2.4	0.441	107.4	C	5.9	3
1	1130.1	2015	19.2	0.7	0.242	230.4	G	0.209	216	-6.3	0.217	217.9	G	-1.8	0.9
1	1130.1	2038	28.4	0.7	0.29	212.6	C	0.247	192.7	-9.4	0.251	194.4	C	-8	0.9
1	1130.1	2026	36.4	0.7	0.308	184.9	G	0.284	173.3	-6.2	0.288	174.2	G	-5.3	0.5
1	1130.1	1995	20.2	1	0.303	179	G	0.284	173.4	-3.1	0.288	173.5	C	-2.6	0.1
1	1130.1	2013	27.6	1	0.327	164	G	0.313	160.2	-2.3	0.314	160.3	G	-1.8	0.1
1	1130.1	2024	36	1	0.347	144.3	C	0.349	145.5	0.8	0.35	144.4	C	0.5	-0.7
1	1130.1	2002	22.1	1.4	0.359	137.3	C	0.365	139.1	1.3	0.363	138.6	C	-1.8	-0.3
1	1130.1	1996	29	1.4	0.377	126.3	C	0.389	129.3	2.4	0.386	128.8	C	2.5	-0.4
1	1130.1	2006	38.6	1.4	0.405	111.2	GC	0.42	114.8	3.3	0.417	116.7	C	-1.8	-0.9
0	1130.1	3045	22.8	0.7	0.159	210.3	G	0.156	201.3	-4.3	0.168	208.8	G	-0.2	3.8
0	1130.1	2970	31.2	0.7	0.177	175.2	G	0.19	177	1	0.203	183.7	G	-1.8	3.8

QD (mm)	P (kPa)	G (kgm ⁻³)	T _g (°C)	L (m)	X ₀	GHF ₀ (kW/m ²)	First GHF ₁	X ₁	GHF ₁ (kW/m ²)	Error (%)	X ₂	GHF ₂ (kW/m ²)	GHF ₂ (kW/m ²)
0.3	1660.3	1987	21.5	0.7	0.187	221.7	C	0.113	190.5	-14.1	0.129	195.8	G
0.3	1660.3	2003	30.4	0.7	0.213	186.8	C	0.146	168.3	-11.7	0.192	173.8	G
0.3	1660.3	1996	37.4	0.7	0.221	172.2	G	0.177	152	-11.7	0.192	156.2	G
0.3	1660.3	2035	44	0.7	0.229	148.8	G	0.209	139.2	-11.7	0.225	144.1	G
0.3	1660.3	1993	20.4	1.4	0.273	129	G	0.252	123	-4.7	0.275	126.6	G
0.3	1660.3	2017	34.7	1.4	0.296	105.7	G	0.31	105.7	-0.0	0.333	105.8	G
0.3	1660.3	2037	51.3	1.4	0.336	79.5	G	0.383	86.2	8.4	0.403	89.9	G
0.3	1660.3	2024	56.1	1.4	0.356	73.3	G	0.403	80	9.1	0.421	83.7	G
0.5	1660.3	2008	18.4	0.7	0.163	234.7	C	0.104	209	-10.9	0.119	214.6	G
0.5	1660.3	1991	25.7	0.7	0.184	213.9	C	0.126	188.8	-11.3	0.171	185.8	G
0.5	1660.3	2008	32.3	0.7	0.205	194.8	C	0.154	172.7	-11.3	0.171	178.2	G
0.5	1660.3	2006	41.1	0.7	0.244	173.3	G	0.257	151.6	-11.3	0.281	158.5	G
0.5	1660.3	1991	21.5	1.4	0.289	135.8	G	0.257	127.8	-5.9	0.281	131.7	G
0.5	1660.3	1983	31.8	1.4	0.317	120.2	G	0.286	115.1	-4.1	0.32	119.4	G
0.5	1660.3	1999	38.6	1.4	0.335	109.7	GC	0.328	106.7	-2.7	0.349	111	G
0.5	1660.3	2024	43.8	1.4	0.339	99.9	GC	0.335	100.5	0.6	0.371	104.8	G
0.5	1660.3	1981	54.2	1.4	0.35	79.6	C	0.396	86.9	9.1	0.416	90.6	G
0.5	1660.3	2007	58.2	1.4	0.359	77	C	0.403	83.3	8.6	0.422	88	G
0.7	1660.3	1982	15.3	0.7	0.152	253.2	C	0.095	229.5	-9.4	0.111	233.5	G
0.7	1660.3	2034	19.9	0.7	0.168	239.8	C	0.108	215.4	-10.2	0.124	221.2	G
0.7	1660.3	1985	25.2	0.7	0.183	224.7	C	0.126	200.1	-10.9	0.142	206.4	G
0.7	1660.3	2027	31.8	0.7	0.196	201.7	C	0.151	182.8	-9.4	0.168	189.2	G
0.7	1660.3	2015	35.4	0.7	0.218	195.1	C	0.167	173	-11.3	0.184	180	G
0.7	1660.3	1991	16.3	1.4	0.278	160.1	C	0.238	140.7	-6.2	0.281	145.5	G
0.7	1660.3	2029	21.5	1.4	0.288	141.4	G	0.257	134.2	-5.1	0.281	138.2	G
0.7	1660.3	2034	28.8	1.4	0.299	132.4	G	0.278	127.2	-3.9	0.301	131.5	G
0.7	1660.3	2013	33.1	1.4	0.315	122.3	G	0.304	119	-2.7	0.326	123.6	G
0.7	1660.3	2030	37.1	1.4	0.324	115.3	G	0.321	113.9	-1.2	0.343	118.4	G
0.7	1660.3	2031	44	1.4	0.332	101.9	C	0.352	104.9	3	0.371	109.2	C
0.7	1660.3	2032	51.3	1.4	0.342	87.9	C	0.383	95.8	8.9	0.399	98.9	C
1	1660.3	1981	20.5	0.7	0.15	245	C	0.11	227.8	-7	0.125	235.5	G
1	1660.3	1986	27.3	0.7	0.171	223.8	C	0.183	208.1	-7	0.147	213.4	C
1	1660.3	2005	39.2	0.7	0.213	187.5	C	0.183	175.8	-6.3	0.192	178.8	C
1	1660.3	2023	27	1	0.239	178.2	C	0.2	167.1	-6.2	0.209	168.6	C
1	1660.3	2015	32.4	1	0.269	169.9	C	0.223	156.6	-7.8	0.23	158.1	C
1	1660.3	2031	20.4	1.4	0.287	162.2	G	0.252	145.1	-17	0.256	145.8	C

Gap (mm)	Temp (°C)	CHF _{uni} (kW/m ²)	CHF _{test} (kW/m ²)	X _{uni}	CHF _{uni} (kW/m ²)	CHF _{test} (kW/m ²)	Error (%)	CHF _{uni} (kW/m ²)	CHF _{test} (kW/m ²)	Error (%)		
0.7	1660.3	3018	31	0.7	0.088	0.076	-3.9	0.099	246.8	G	-0.2	3.3
0.7	1660.3	3018	34.8	0.7	0.122	0.099	-6.3	0.143	231.1	G	-2.6	4.2
0.7	1660.3	3046	41.5	0.7	0.138	0.129	-3.9	0.143	204.7	G	0.5	4.2
0.7	1660.3	2982	19.6	1.4	0.166	0.147	-1.6	0.177	182.4	G	1.2	3.2
0.7	1660.3	3015	23.8	1.4	0.181	0.173	-1.6	0.213	163.3	G	2.8	4
0.7	1660.3	3015	29.4	1.4	0.198	0.195	-1.6	0.213	163.3	G	2.8	4
1	1660.3	2992	23.2	0.7	0.074	0.065	-2.3	0.076	301.1	G	0.5	2.5
1	1660.3	3035	35.5	0.7	0.112	0.101	-2.3	0.115	244.5	G	1.5	3.5
1	1660.3	3044	35.2	1	0.149	0.155	1.9	0.166	194.7	C	4.4	2
1	1660.3	3040	23.9	1.4	0.17	0.172	0.7	0.182	182.6	C	3.5	1.6
1	1660.3	2974	30.8	1.4	0.193	0.199	1.5	0.205	167.9	C	3.5	1.6
1	1660.3	3006	36.6	1.4	0.221	0.224	0.8	0.228	164.7	C	0.5	-0.3
1	1660.3	3024	46	1.4	0.265	0.267	0.4	0.267	134	C	0.5	-0.3
1	1660.3	3015	55.8	1.4	0.318	0.31	-2.3	0.309	116.7	C	2.2	10.3

Note:

1. the subscript “test” stands for the measured data
2. the subscript “uni” stands for the partition model data for uniform heating condition
3. the subscript “non” stands for the partition model data for non-uniform heating condition
4. the relative errors are given by:

$$Error_{uni} = \frac{CHF_{uni} - CHF_{test}}{CHF_{test}}$$

$$Error_{test} = \frac{CHF_{non} - CHF_{test}}{CHF_{test}}$$

$$Error_{non} = \frac{CHF_{non} - CHF_{uni}}{CHF_{uni}}$$

Appendix VI

CIRCULAR TUBE CHF EXPERIMENTAL AND MODELING DATA

P (MPa)	G (kg/m ² s)	D (mm)	q'' (MW/m ²)	ΔT_{sub} (°C)	CHF (Model)	CHF (Test)	L (m)	h_{fg} (kJ/kg)	Δh_{fg} (Model)	Δh_{fg} (Test)	Error
4.674	1660	1007	3.6	0.262	140	155	1	53.6	118.2	122.1	-0.0976
4.674	1660	1014	5.8	0.266	138.6	152.8	1	56.6	114.6	118.2	-0.0937
4.674	1660	1022	7.3	0.269	137.4	149.6	1	58.6	112.1	115.1	-0.0819
4.674	1660	1003	19.3	0.281	133.3	147.6	1	61.7	108.5	112.2	-0.0976
4.674	1660	1005	12.6	0.29	130.5	143.9	1	66	103.5	106.9	-0.0937
4.674	1660	1017	16.2	0.301	127.7	138.3	1	71.1	97.1	100.1	-0.0858
4.674	1660	1003	19.8	0.318	122.3	136.6	1	76.2	91.2	94.9	-0.1054
4.674	1660	1002	22.8	0.331	118.9	133.7	1	80.3	88.3	90	-0.1093
4.674	1660	998	25.9	0.342	115.8	129.9	1	85.1	80.8	84.5	-0.1093
4.674	1660	1008	31.9	0.365	110.1	117.3	1	93.9	70.3	72.1	-0.0624
4.674	1660	1028	34.1	0.37	108.8	115	1	97.2	66.1	67.7	-0.0546
4.674	1660	1040	36.3	0.378	107	110.3	1	100.5	62	62.9	-0.0312
4.674	1660	1030	40.1	0.396	102.8	105.6	1	106.2	55.6	56.3	-0.0272
4.674	1660	1029	41.8	0.404	101.2	102.7	1	108.7	52.7	53.1	-0.0155
4.674	1660	1028	43.2	0.411	99.7	99.6	1	110.8	50.2	50.2	0.0001
4.674	1660	1013	45.1	0.425	96.8	96.7	1	113.7	47	46.9	-0.0032
4.674	1660	1030	47.2	0.43	95.8	92.1	1	116.9	43.1	42.2	0.0395
4.674	1660	1057	49.2	0.433	95	90.7	1	120	39.2	38.2	0.0468
4.674	1660	1008	51.2	0.457	90.5	85.6	1	123.1	36.1	34.9	0.0567
4.674	1660	1012	54.2	0.47	87.9	82.4	1	127.7	30.7	29.4	0.0662
4.674	1660	1025	56.1	0.477	86.4	79.6	1	130.7	27.1	25.4	0.0838
4.674	1660	1032	57.1	0.489	85.4	76.8	0.45	133.9	26.2	27.1	-0.0546
4.674	1660	1051	6.2	0.091	251.2	263.5	0.45	57.1	252.9	261.4	-0.0468
4.674	1660	1019	8.4	0.101	238	258.1	0.45	60.1	245.9	260.3	-0.078
4.674	1660	1014	10.5	0.107	231.3	251.9	0.45	63	239	253.9	-0.0819
4.674	1660	1019	13.5	0.114	223.7	246.7	0.45	67.2	228.6	245.1	-0.0936
4.674	1660	1003	16.4	0.126	213.9	238	0.45	71.3	219.9	237.5	-0.1015
4.674	1660	1034	21	0.134	206	230.2	0.45	77.9	203.1	220.2	-0.1054
4.674	1660	1030	24.2	0.146	197.5	219.7	0.45	82.6	193	208.7	-0.1015
4.674	1660	1023	27.8	0.16	187.8	212.6	0.45	87.9	181.7	199.4	-0.1171
4.674	1660	1044	31.1	0.169	181.8	205.8	0.45	92.7	170	186.9	-0.1171
4.674	1660	1034	34.4	0.185	173.3	199.7	0.45	97.6	160.4	179	-0.1327
4.674	1660	1030	38.3	0.204	163.5	191	0.45	103.5	148.1	167.6	-0.1444
4.674	1660	1058	41.6	0.215	158.5	181	0.45	108.4	136.6	152.2	-0.1249
4.674	1660	1021	43.6	0.232	151.4	172.9	0.45	111.5	132.4	147.8	-0.1249
4.674	1660	1020	46.3	0.247	145.4	163.1	0.45	115.6	124.2	136.9	-0.1093
4.674	1660	1024	48.9	0.26	140.6	157.7	0.45	119.5	116.4	128.7	-0.1093
4.674	1660	1001	51.4	0.28	133.9	153.6	0.45	123.4	110	124.4	-0.1288
4.674	1660	1001	53.9	0.294	129.3	148.3	0.45	127.2	102.8	116.7	-0.1288
4.674	1660	1049	56.3	0.3	127.5	146.2	0.45	131	93.5	106.5	-0.1288

Age (mm)	Age (Ga)	Age (Ga)	Age (Ga)	Age (Ga)	Age (Ga)	Age (Ga)	Age (Ga)	Age (Ga)	Age (Ga)	Age (Ga)	Error
4.674	1660	1000	3.7	0.339	116.6	126.9	1.3	53.7	82.5	82.5	-0.0819
4.674	1660	1003	5.7	0.344	115.3	125.5	1.3	56.5	79.7	79.7	-0.0819
4.674	1660	1007	8.4	0.352	113.3	121.8	1.3	60.1	76.1	76.1	-0.0702
4.674	1660	1009	11.1	0.356	111.2	118.5	1.3	63.7	72.5	72.5	-0.0624
4.674	1660	1014	13.7	0.369	109.1	115.3	1.3	67.5	68.7	68.7	-0.0546
4.674	1660	1020	16.3	0.374	107.8	112.2	1.3	70.9	65.3	65.3	-0.039
4.674	1660	995	19.1	0.393	103.6	109	1.3	75.2	61	61	-0.0507
4.674	1660	1015	22.3	0.398	102.5	105.7	1.3	79.8	56.4	56.4	-0.0391
4.674	1660	1015	25.6	0.411	99.6	101.1	1.3	84.6	51.6	51.6	-0.0155
4.674	1660	1019	28.7	0.424	96.9	98.3	1.3	89.2	47	47	-0.0155
4.674	1660	1020	31.8	0.436	94.6	93.9	1.3	93.8	42.4	42.4	0.0065
4.674	1660	1008	35.7	0.454	90.9	90.9	1.3	99.6	36.6	36.6	0.0001
4.674	1660	1012	40.2	0.474	87.1	84	1.3	106.3	29.9	29.9	0.0361
4.674	1660	1003	44.2	0.494	83.3	80.2	1.3	112.4	23.8	23.8	-0.0378
4.674	1660	1004	47.5	0.507	80.9	76	1.3	117.4	18.8	18.8	0.0639
4.674	1660	1019	50.3	0.516	79.3	72.2	1.3	121.7	14.5	14.5	-0.0966
4.674	1660	1009	52.9	0.529	76.9	68.3	1.3	125.7	10.5	10.5	0.1244
4.674	1660	1030	55.4	0.535	73.5	64.5	1.3	129.6	6.6	6.6	-0.1693
4.674	1660	1004	57.3	0.551	72.9	61.9	1.3	132.6	3.6	3.6	0.1765
4.674	1660	2004	59	0.109	227.1	228.8	1	54	111.2	111.5	-0.0077
4.674	1660	2003	6.7	0.115	221.1	226.3	1	57.8	106.7	107.4	-0.0233
4.674	1660	1988	9.1	0.123	213.9	218.9	1	61.1	102.7	103.4	-0.0233
4.674	1660	2051	12.4	0.126	211.3	212.9	1	65.7	96.9	97.1	-0.0077
4.674	1660	2040	16.7	0.114	200	202.2	1	71.8	89.5	89.8	-0.0116
4.674	1660	2025	22.1	0.159	187.1	193	1	79.5	80.4	81.1	-0.0311
4.674	1660	2015	28.3	0.185	172.6	175.2	1	88.3	69.8	70.2	-0.0155
4.674	1660	2001	33.7	0.209	160.3	163.4	1	96.6	60.1	60.5	-0.0194
4.674	1660	1995	42.9	0.254	142	140.1	1	110.4	44	43.8	-0.013
4.674	1660	1970	44.3	0.263	138.5	137.2	1	112.5	41.7	41.5	0.0089
4.674	1660	2004	46.5	0.27	135.8	131.5	1	115.9	37.6	37.1	-0.0322
4.674	1660	2012	49.6	0.286	130.4	123.1	1	120.6	32.2	31.3	0.0589
4.674	1660	2013	52.7	0.302	125	109.8	1	125.4	26.7	24.8	-0.1375
4.674	1660	2004	56.7	0.327	117	93.3	1	131.6	19.5	16.5	0.2529
4.674	1660	2055	43.2	0.116	220.8	219.5	0.45	110.8	103.5	103.1	0.0055
4.674	1660	1993	46.3	0.136	203.8	205.3	0.45	115.6	94.9	95.5	-0.0077
4.674	1660	2068	49.6	0.149	194.2	191.1	0.45	120.6	83.8	82.8	0.0158
4.674	1660	2048	53.1	0.171	180	178.1	0.45	126	74.1	73.4	0.0102
4.674	1660	1997	55.5	0.189	170.1	166.4	0.45	129.7	68.4	67.1	0.0217
4.674	1660	2005	16.8	0.199	165	163.7	1.3	71.9	64.3	64.3	0.0074
4.674	1660	2011	21.5	0.216	157.4	156.1	1.3	78.7	57.5	57.5	0.0078
4.674	1660	2025	27.5	0.236	148.6	148.5	1.3	87.4	48.8	48.8	0.0001
4.674	1660	2008	31.5	0.254	141.6	139.7	1.3	93.3	42.9	42.9	-0.013
4.674	1660	2054	22.6	0.214	158.2	153.9	1.3	80.3	55.9	55.9	0.0274
4.674	1660	2025	24.9	0.225	153.1	150.6	1.3	83.6	52.6	52.6	-0.0161
4.674	1660	1991	35	0.272	135.4	132.3	1.3	98.5	37.7	37.7	0.0229
4.674	1660	2003	36	0.285	130.8	127.1	1.3	103	33.2	33.2	-0.0286

ΔH_{ice} (m)	ΔH_{ice} (m)	ΔH_{ice} (m)	ΔH_{ice} (m)	ΔH_{ice} (m)	ΔH_{ice} (m)	ΔH_{ice} (m)	ΔH_{ice} (m)	ΔH_{ice} (m)	ΔH_{ice} (m)	ΔH_{ice} (m)	Error
4.674	1660	2040	41.2	0.294	127.6	120.3	1.3	107.8	28.4	28.4	0.0602
4.674	1660	2040	43.8	0.311	127.2	120.3	1.3	107.8	28.2	28.2	0.1247
4.674	1660	2026	47.8	0.327	116.9	99.9	1.3	117.8	18.4	18.4	0.1695
4.674	1660	2017	50.8	0.348	114.0	87.2	1.3	124.1	12.2	12.2	0.2638
4.674	1660	1996	56	0.37	103.1	76.3	1.3	130.5	5.7	5.7	0.3494
4.674	1660	2022	55.2	0.36	122.8	121.6	0.9	69.6	103.4	103.7	-0.0077
4.674	1660	2013	18.9	0.131	207.6	210	0.9	74.9	96.6	97	-0.0116
4.674	1660	2005	23.4	0.148	195.1	199.7	0.9	81.4	88.1	88.9	-0.0233
4.674	1660	2033	28	0.161	185.9	186.5	0.9	88.2	79.2	79.4	-0.0038
4.674	1660	2017	34	0.187	171.2	168.7	0.9	97	68.2	67.8	0.0143
4.674	1660	1993	38.6	0.21	159.8	161	0.9	103.9	59.7	59.9	-0.0077
4.674	1660	2017	41.6	0.228	151.8	151.3	0.9	108.4	54.1	54.1	0.0001
4.674	1660	2004	43.6	0.234	149.4	146.3	0.9	111.5	50.2	49.6	0.0207
4.674	1660	2019	48.8	0.245	134.5	142.5	0.9	116	44.4	44	0.017
4.674	1660	2017	50.3	0.269	136.4	130.3	0.9	121.7	37.6	36.6	0.0463
4.674	1660	2015	52.7	0.284	131.1	121.4	0.9	125.4	33	31.4	0.0796
4.674	1660	2002	55.9	0.303	124.7	105.9	0.9	130.4	27.1	23.9	0.1769
4.674	1660	3080	33.2	0.144	188.7	182.2	1	101.8	50.7	49.5	0.079
4.674	1660	3070	40.4	0.16	186.2	169.3	1	106.6	45.1	43.7	0.0993
4.674	1660	3062	43.9	0.178	175.3	159.6	1	111.9	38.9	37.6	0.0979
4.674	1660	3025	49.1	0.208	159.7	146.4	1	119.8	29.9	28.8	0.0905
4.674	1660	3000	50.7	0.218	154.7	139	1	122.3	27.1	25.8	0.1127
4.674	1660	2945	53.7	0.239	145.6	129.2	1	126.9	21.9	20.5	0.126
4.674	1660	2823	56.4	0.255	138.4	119	1	131.2	17.1	15.4	0.1623
4.674	1660	3000	41.6	0.068	288.5	275.3	0.45	108.4	97.7	94.5	0.0475
4.674	1660	3032	46.4	0.086	254.5	241.3	0.45	115.7	81.5	78.3	0.0544
4.674	1660	3100	48.5	0.095	243.2	227.6	0.45	118.9	74.3	70.7	0.0682
4.674	1660	3057	52.2	0.116	219.9	199.5	0.45	124.6	63.9	59	0.102
4.674	1660	3010	55	0.136	203.1	182.6	0.45	129	56.2	51.3	0.1117
4.674	1660	2940	70	0.132	206.5	185.6	1.3	76.5	59.7	59.7	0.0552
4.674	1660	3012	24.5	0.143	198	185.9	1.3	83	53.2	53.2	0.0645
4.674	1660	2980	29.8	0.164	183.4	171.3	1.3	90.8	45.4	45.4	0.0701
4.674	1660	3060	32.4	0.171	179.3	166	1.3	94.7	41.5	41.5	0.0796
4.674	1660	3062	35.5	0.184	172.2	157.7	1.3	99.3	36.9	36.9	0.0915
4.674	1660	3025	39.4	0.205	161.2	146.7	1.3	105.1	31.1	31.1	0.0985
4.674	1660	2994	42.9	0.223	152.5	138	1.3	110.4	25.8	25.8	0.1048
4.674	1660	3045	46.6	0.237	145.7	131.1	1.3	116	20.2	20.2	0.1104
4.674	1660	3045	48.3	0.246	141.6	127.6	1.3	118.6	17.6	17.6	0.1087
4.674	1660	3024	50.8	0.259	135.7	116.3	1.3	122.4	13.8	13.8	0.1661
4.674	1660	2991	53.4	0.274	129.1	106.1	1.3	126.5	9.7	9.7	0.2164
4.674	1660	3011	55.8	0.284	123.4	95.5	1.3	130.2	6	6	0.2913
4.674	1660	3018	59.5	0.068	229.7	283.9	0.9	70.5	98.8	97.8	0.0336
4.674	1660	3011	21.3	0.078	268.4	258.8	0.9	78.4	88.3	87.2	0.0369
4.674	1660	3063	25.7	0.088	252.5	246.4	0.9	84.8	79.6	78.9	0.0243
4.674	1660	3074	28.4	0.096	241.9	237	0.9	88.7	74.4	73.9	0.0202
4.674	1660	3067	31.8	0.108	228.7	219	0.9	93.8	67.9	66.8	0.0437

δ (mm)	ρ (kg/m ³)	ρ_g (kg/m ³)	T_f (°C)	X	CHF (Model)	CHF (Test)	L (m)	h_{fg} (kJ/kg)	Δh_{fg} (r.s.) (Model)	Δh_{fg} (r.s.) (Test)	Error
4.674	1660	3031	36.8	0.129	208.5	195.2	0.9	101.2	58.5	57	0.0675
4.674	1660	3007	42.1	0.158	188.8	174.3	0.9	109.2	48.4	46.8	0.0827
4.674	1660	3052	46.1	0.174	177.8	160.9	0.9	115.3	40.8	38.9	0.1046
4.674	1660	3000	50.2	0.198	164.4	146.3	0.9	121.5	33.4	31.4	0.1234
4.674	1660	3019	54.3	0.221	153.2	132.6	0.9	127.9	25.6	23.3	0.1545
4.674	1660	3003	58.2	0.234	147.5	128.7	0.9	130.8	22.2	20	0.1452