SECOND EDITION

Principles of Enhanced Heat Transfer

Ralph L. Webb
Nae-Hyun Kim

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TABLE OF CONTENTS

PREFACE
CHAPTER 1: INTRODUCTION TO ENHANCED HEAT TRANSFER
1.1 INTRODUCTION
1.2 THE ENHANCEMENT TECHNIQUES
1.2.1 Passive Techniques
1.2.2 Active Techniques
1.2.3 Technique vs. Mode
1.3 PUBLISHED LITERATURE
1.3.1 General Remarks
1.3.2 U.S. Patent Literature
1.3.3 Manufacturer's Information
1.4 BENEFITS OF ENHANCEMENT
1.5 COMMERCIAL APPLICATIONS OF ENHANCED SURFACES
1.5.1 Heat (and Mass) Exchanger Types of Interest
1.5.2 Illustrations of Enhanced Tubular Surfaces
1.5.3 Enhanced Fin Geometries for Gases
1.5.4 Plate-Type Heat Exchangers
1.5.5 Cooling Tower Packings
1.5.6 Distillation and Column Packings
1.5.7 Factors Affecting Commercial Development
1.6 DEFINITION OF HEAT TRANSFER AREA
1.7 POTENTIAL FOR ENHANCEMENT
1.7.1 PEC Example 1.1
1.7.2 PEC Example 1.2
NOMENCLATURE
REFERENCES
CHAPTER 2: HEAT TRANSFER FUNDAMENTALS
2.1 INTRODUCTION
2.2 HEAT EXCHANGER DESIGN THEORY
2.2.1 Thermal Analysis
2.2.2 Heat Exchanger Design Methods
2.2.3 Comparison of LMTD and NTU Design Methods
2.3 FIN EFFICIENCY
2.4 HEAT TRANSFER COEFFICIENTS AND FRICTION FACTORS
2.4.1 Laminar Flow over Flat Plate
2.4.2 Laminar Flow in Ducts
2.4.3 Turbulent Flow in Ducts
2.4.4 Tube Banks (Single-Phase Flow)
2.4.5 Film Condensation
2.4.6 Nucleate Boiling
2.5 CORRECTION FOR VARIATION OF FLUID PROPERTIES

2.5.1 Effect of Changing Fluid Temperature
2.5.2 Effect Local Property Variation
2.6 REYNOLDS ANALOGY
2.7 FOULING OF HEAT TRANSFER SURFACES
2.8 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 3: PERFORMANCE EVALUATION CRITERIA FOR SINGLE-PHASE FLOW
3.1 PERFORMANCE EVALUATION CRITERIA (PEC)
3.2 PEC FOR HEAT EXCHANGERS
3.3 PEC FOR SINGLE-PHASE FLOW
3.3.1 Objective Function and Constraints
3.3.2 Algebraic Formulation of the PEC
3.3.3 Simple Surface Performance Comparison
3.3.4 Constant Flow Rate
3.3.5 Fixed Flow Area
3.4 THERMAL RESISTANCE ON BOTH SIDES
3.5 RELATIONS FOR St AND *f*
3.6 HEAT EXCHANGER EFFECTIVENESS
3.7 EFFECT OF REDUCED EXCHANGER FLOW RATE
3.8 FLOW NORMAL TO FINNED TUBE BANKS
3.9 VARIANTS OF THE PEC
3.10 COMMENTS ON OTHER PERFORMANCE INDICATORS
3.10.1 Shah
3.10.2 Soland and Colleagues
3.11 EXERGY-BASED PEC ANALYSIS
3.12 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 4: PERFORMANCE EVALUATION CRITERIA FOR TWO-PHASE HEAT EXCHANGERS
4.1 INTRODUCTION
4.2 OPERATING CHARACTERISTICS OF TWO-PHASE HEAT EXCHANGERS
4.3 ENHANCEMENT IN TWO-PHASE HEAT EXCHANGE SYSTEMS
4.3.1 Work-Consuming Systems
4.3.2 Work-Producing Systems
4.3.3 Heat-Actuated Systems
4.4 PEC FOR TWO-PHASE HEAT EXCHANGE SYSTEMS
4.5 PEC CALCULATION METHOD
4.5.1 PEC Example 4.1
4.5.2 PEC Example 4.2
4.6 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 5: PLATE-AND-FIN EXTENDED SURFACES

5.1 INTRODUCTION
5.2 OFFSET-STRIP FIN
5.2.1 Enhancement Principle
5.2.2 PEC Example 5.1
5.2.3 Analytically Based Models for *j* and *f* vs. Re
5.2.4 Transition from Laminar to Turbulent Region
5.2.5 Correlations for *j* and *f* vs. Re
5.2.6 Use of OSF with Liquids
5.2.7 Effect of Percent Fin Offset
5.2.8 Effect of Burred Edges
5.3 LOUVER FIN
5.3.1 Heat Transfer and Friction Correlations
5.3.2 Flow Structure in the Louver Fin Array
5.3.3 Analytical Model for Heat Transfer and Friction
5.3.4 PEC Example 5.2
5.4 CONVEX LOUVER FIN
5.5 WAVY FIN
5.6 THREE-DIMENSIONAL CORRUGATED FINS
5.7 PERFORATED FIN
5.8 PIN FINS AND WIRE MESH
5.9 VORTEX GENERATORS
5.9.1 Types of Vortex Generators
5.9.2 Vortex Generators on a Plate-Fin Surface
5.10 METAL FOAM FIN
5.11 PLAIN FIN
5.11.1 PEC Example 5.3
5.12 ENTRANCE LENGTH EFFECTS
5.13 PACKINGS FOR GAS-GAS REGENERATORS
5.14 NUMERICAL SIMULATION
5.14.1 Offset-Strip Fins
5.14.2 Louver Fins
5.14.3 Wavy Channels
5.14.4 Chevron Plates
5.14.5 Summary
5.15 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 6: EXTERNALLY FINNED TUBES
6.1 INTRODUCTION
6.2 THE GEOMETRIC PARAMETERS AND THE REYNOLDS NUMBER
6.2.1 Dimensionless Variables
6.2.2 Definition of Reynolds Number
6.2.3 Definition of the Friction Factor
6.2.4 Sources of Data
6.3 PLAIN PLATE-FINS ON ROUND TUBES
6.3.1 Effect of Fin Spacing

6.3.2 Correlations for Staggered Tube Geometries
6.3.3 Correlations for Inline Tube Geometries
6.4 PLAIN INDIVIDUALLY FINNED TUBES
6.4.1 Circular Fins with Staggered Tubes
6.4.2 Low Integral-Fin Tubes
6.5 ENHANCED PLATE FIN GEOMETRIES WITH ROUND TUBES
6.5.1 Wavy Fin
6.5.2 Offset Strip Fins
6.5.3 Convex Louver Fins
6.5.4 Louvered Fin
6.5.5 Perforated Fins
6.5.6 Mesh Fins
6.5.7 Vortex Generators
6.6 ENHANCED CIRCULAR FIN GEOMETRIES
6.6.1 Illustrations of Enhanced Fin Geometries
6.6.2 Spine or Segmented Fins
6.6.3 Wire Loop Fins
6.7 OVAL AND FLAT TUBE GEOMETRIES
6.7.1 Oval vs. Circular Individually Finned Tubes
6.7.2 Flat Extruded Aluminum Tubes with Internal Membranes
6.7.3 Plate-and-Fin Automotive Radiators
6.7.4 Vortex Generators on Flat or Oval Fin-Tube Geometry
6.8 ROW EFFECTS—STAGGERED AND INLINE LAYOUTS
6.9 HEAT TRANSFER COEFFICIENT DISTRIBUTION (PLAIN FINS)
6.9.1 Experimental Methods
6.9.2 Plate Fin-and-Tube Measurements
6.9.3 Circular Fin-and-Tube Measurements
6.10 PERFORMANCE COMPARISON OF DIFFERENT GEOMETRIES
6.10.1 Geometries Compared
6.10.2 Analysis Method
6.10.3 Calculated Results
6.11 PROGRESS ON NUMERICAL SIMULATION
6.12 RECENT PATENTS ON ADVANCED FIN GEOMETRIES
6.13 HYDROPHILIC COATINGS
6.14 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 7: INSERT DEVICES FOR SINGLE-PHASE FLOW
7.1 INTRODUCTION
7.2 TWISTED TAPE INSERT
7.2.1 Laminar Flow Data
7.2.2 Predictive Methods for Laminar Flow
7.2.3 Turbulent Flow
7.2.4 PEC Example 7.1
7.2.5 Twisted Tapes in Annuli
7.2.6 Twisted Tapes in Rough Tubes

7.3 SEGMENTED TWISTED TAPE INSERT
7.4 DISPLACED ENHANCEMENT DEVICES
7.4.1 Turbulent Flow
7.4.2 Laminar Flow
7.4.3 PEC Example 7.2
7.5 WIRE COIL INSERTS
7.5.1 Laminar Flow
7.5.2 Turbulent Flow
7.6 EXTENDED SURFACE INSERT
7.7 TANGENTIAL INJECTION DEVICES
7.8 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 8: INTERNALLY FINNED TUBES AND ANNULI
8.1 INTRODUCTION
8.2 INTERNALLY FINNED TUBES
8.2.1 Laminar Flow
8.2.2 Turbulent Flow
8.2.3 PEC Example 8.1
8.3 SPIRALLY FLUTED TUBES
8.3.1 Spirally Fluted Tube
8.3.2 Spirally Indented Tube
8.4 ADVANCED INTERNAL FIN GEOMETRIES
8.5 FINNED ANNULI
8.6 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 9: INTEGRAL ROUGHNESS
9.1 INTRODUCTION
9.2 ROUGHNESS WITH LAMINAR FLOW
9.2.1 Laminar Flow in Roughened Circular Tubes
9.2.2 Laminar Flow in Roughened Flat Tubes
9.3 HEAT-MOMENTUM TRANSFER ANALOGY CORRELATION
9.3.1 Friction Similarity Law
9.3.2 PEC Example 9.1
9.3.3 Heat Transfer Similarity Law
9.4 TWO-DIMENSIONAL ROUGHNESS
9.4.1 Transverse Rib Roughness
9.4.2 Integral Helical-Rib Roughness
9.4.3 Wire Coil Inserts
9.4.4 Corrugated Tube Roughness
9.4.5 PEC Example 9.2
9.5 THREE-DIMENSIONAL ROUGHNESS
9.6 PRACTICAL ROUGHNESS APPLICATIONS
9.6.1 Tubes with Inside Roughness
9.6.2 Rod Bundles and Annuli

9.6.3 Rectangular Channels
9.6.4 Outside Roughness for Cross Flow
9.7 GENERAL PERFORMANCE CHARACTERISTICS
9.7.1 St and f vs. Reynolds Number
9.7.2 Other Correlating Methods
9.7.3 Prandtl Number Dependence
9.8 HEAT TRANSFER DESIGN METHODS
9.8.1 Design Method 1
9.8.2 Design Method 2
9.9 PREFERRED ROUGHNESS TYPE AND SIZE
9.9.1 Roughness Type
9.9.2 PEC Example 9.3
9.10 NUMERICAL SIMULATION
9.10.1 Predictions for Transverse-Rib Roughness
9.10.2 Effect of Rib Shape
9.10.3 The Discrete-Element Predictive Model
9.11 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 10: FOULING ON ENHANCED SURFACES
10.1 INTRODUCTION
10.2 FOULING FUNDAMENTALS
10.2.1 Particulate Fouling
10.3 FOULING OF GASES ON FINNED SURFACES
10.4 SHELL-SIDE FOULING OF LIQUIDS
10.4.1 Low Radial Fins
10.4.2 Axial Fins and Ribs in Annulus
10.4.3 Ribs in Rod Bundle
10.5 FOULING OF LIQUIDS IN INTERNALLY FINNED TUBES
10.6 LIQUID FOULING IN ROUGH TUBES
10.6.1 Accelerated Particulate Fouling
10.6.2 Long-Term Fouling
10.7 LIQUID FOULING IN PLATE-FIN GEOMETRY
10.8 CORRELATIONS FOR FOULING IN ROUGH TUBES
10.9 MODELING OF FOULING IN ENHANCED TUBES
10.9.1 Example Problem 10.1
10.10 FOULING IN PLATE HEAT EXCHANGERS
10.11 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 11: POOL BOILING AND THIN FILM EVAPORATION
11.1 INTRODUCTION
11.2 EARLY WORK ON ENHANCEMENT (1931–1962)
11.3 SUPPORTING FUNDAMENTAL STUDIES
11.4 TECHNIQUES EMPLOYED FOR ENHANCEMENT
11.4.1 Abrasive Treatment

11.4.2 Open Grooves
11.4.3 Three-Dimensional Cavities
11.4.4 Etched Surfaces
11.4.5 Electroplating
11.4.6 Pierced Three-Dimensional Cover Sheets
11.4.7 Attached Wire and Screen Promoters
11.4.8 Nonwetting Coatings
11.4.9 Oxide and Ceramic Coatings
11.4.10 Porous Surfaces
11.4.11 Structured Surfaces (Integral Roughness)
11.4.12 Combination of Structured and Porous Surfaces
11.4.13 Composite Surfaces
11.5 SINGLE-TUBE POOL BOILING TESTS OF ENHANCED SXURFACES
11.6 THEORETICAL FUNDAMENTALS
11.6.1 Liquid Superheat
11.6.2 Effect of Cavity Shape and Contact Angle on Superheat
11.6.3 Entrapment of Vapor in Cavities
11.6.4 Effect of Dissolved Gases
11.6.5 Nucleation at a Surface Cavity
11.6.6 Bubble Departure Diameter
11.6.7 Bubble Dynamics
11.7 BOILING HYSTERESIS AND ORIENTATION EFFECTS
11.7.1 Hysteresis Effects
11.7.2 Size and Orientation Effects
11.8 BOILING MECHANISM ON ENHANCED SURFACES
11.8.1 Basic Principles Employed
11.8.2 Visualization of Boiling in Subsurface Tunnels
11.8.3 Boiling Mechanism in Subsurface Tunnels
11.8.4 Chien and Webb Parametric Boiling Studies
11.9 PREDICTIVE METHODS FOR STRUCTURED SURFACES
11.9.1 Empirical Correlations
11.9.2 Nakayama et al. [1980b] Model
11.9.3 Chien and Webb Model
11.9.4 Ramaswamy et al. Model
11.9.5 Jiang et al. Model
11.9.6 Other Models
11.9.7 Evaluation of Models
11.10 BOILING MECHANISM ON POROUS SURFACES
11.10.1 O'Neill et al. Thin Film Concept
11.10.2 Kovalev et al. Concept
11.11 PREDICTIVE METHODS FOR POROUS SURFACES
11.11.1 O'Neill et al. Model
11.11.2 Kovalov et al. Model
11.11.3 Nishikawa et al. Correlation
11.11.4 Zhang and Zhang Correlation
11.12 CRITICAL HEAT FLUX

11.13 ENHANCEMENT OF THIN FILM EVAPORATION
11.14 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 12: VAPOR SPACE CONDENSATION
12.1 INTRODUCTION
12.1.1 Condensation Fundamentals
12.1.2 Basic Approaches to Enhanced Film Condensation
12.2 DROPWISE CONDENSATION
12.3 SURVEY OF ENHANCEMENT METHODS
12.3.1 Coated Surfaces
12.3.2 Roughness
12.3.3 Horizontal Integral-Fin Tubes
12.3.4 Corrugated Tubes
12.3.5 Surface Tension Drainage
12.3.6 Electric Fields
12.4 SURFACE TENSION DRAINED CONDENSATION
12.4.1 Fundamentals
12.4.2 Adamek [1981] Generalized Analysis
12.4.3 "Practical" Fin Profiles
12.4.4 Prediction for Trapezoidal Fin Shapes
12.5 HORIZONTAL INTEGRAL-FIN TUBE
12.5.1 The Beatty and Katz [1948] Model
12.5.2 Precise Surface Tension-Drained Models
12.5.3 Approximate Surface Tension-Drained Models
12.5.4 Comparison of Theory and Experiment
12.6 HORIZONTAL TUBE BANKS
12.6.1 Condensation with Vapor Shear
12.6.2 Condensate Inundation without Vapor Shear
12.6.3 Condensate Drainage Pattern
12.6.4 Prediction of the Condensation Coefficient
12.7 CONCLUSIONS
APPENDIX 12.A: THE KEDZIERSKI AND WEBB FIN PROFILE SHAPES
APPENDIX 12.B: FIN EFFICIENCY IN THE FLOODED REGION
NOMENCLATURE
REFERENCES
CHAPTER 13: CONVECTIVE VAPORIZATION
13.1 INTRODUCTION
13.2 FUNDAMENTALS
13.2.1 Flow Patterns
13.2.2 Convective Vaporization in Tubes
13.2.3 Two-Phase Pressure Drop
13.2.4 Effect of Flow Orientation on Flow Pattern
13.2.5 Convective Vaporization in Tube Bundles
13.2.6 Critical Heat Flux
13.3 ENHANCEMENT TECHNIQUES IN TUBES

13.3.1 Internal Fins
13.3.2 Swirl Flow Devices
13.3.3 Roughness
13.3.4 Coated Surfaces
13.3.5 Perforated Foil Inserts
13.3.6 Porous Media
13.3.7 Coiled Tubes and Return Bends
13.4 THE MICROFIN TUBE
13.4.1 Early Work on the Microfin Tube
13.4.2 Recent Work on the Microfin Tube
13.4.3 Special Microfin Geometries
13.4.4 Microfin Vaporization Data
13.5 MINICHANNELS
13.6 CRITICAL HEAT FLUX (CHF)
13.6.1 Twisted Tape
13.6.2 Grooved Tubes
13.6.3 Corrugated Tubes
13.6.4 Mesh Inserts
13.7 PREDICTIVE METHODS FOR IN-TUBE FLOW
13.7.1 High Internal Fins
13.7.2 Microfins
13.7.3 Twisted Tape Inserts
13.7.4 Corrugated Tubes
13.7.5 Porous Coatings
13.8 TUBE BUNDLES
13.8.1 Convective Effects in Tube Bundles
13.8.2 Tube Bundle Convective Vaporization Data
13.8.3 Effect of Spacing between Tubes
13.8.4 Convective Vaporization Models
13.8.5 Starting Hysteresis in Tube Bundles
13.9 PLATE-FIN HEAT EXCHANGERS
13.10 THIN FILM EVAPORATION
13.10.1 Horizontal Tubes
13.10.2 Vertical Tubes
13.11 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 14: CONVECTIVE CONDENSATION
14.1 INTRODUCTION
14.2 FORCED CONDENSATION INSIDE TUBES
14.2.1 Internally Finned Geometry
14.2.2 Twisted-Tape Inserts
14.2.3 Roughness
14.2.4 Wire Coil Inserts
14.2.5 Coiled Tubes and Return Bends
14.3 MICROFIN TUBE

14.3.1 Microfin Geometry Details
14.3.2 Optimization of Internal Geometry
14.3.3 Condensation Mechanism in Microfin Tubes
14.3.4 Convective Condensation in Special Microfin Geometries
14.4 FLAT TUBE AUTOMOTIVE CONDENSERS
14.4.1 Condensation Data for Flat, Extruded Tubes
14.4.2 Other Predictive Methods of Condensation in Flat Tubes
14.5 PLATE-TYPE HEAT EXCHANGERS
14.6 NONCONDENSIBLE GASES
14.7 PREDICTIVE METHODS FOR CIRCULAR TUBES
14.7.1 High Internal Fins
14.7.2 Wire Loop Internal Fins
14.7.3 Twisted Tapes
14.7.4 Roughness
14.7.5 Microfins
14.8 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 15: ENHANCEMENT USING ELECTRIC FIELDS
15.1 INTRODUCTION
15.2 ELECTRODE DESIGN AND PLACEMENT
15.3 SINGLE-PHASE FLUIDS
15.3.1 Enhancement on Gas Flow
15.3.2 Enhancement on Liquid Flow
15.3.3 Numerical Studies
15.4 CONDENSATION
15.4.1 Fundamental Understanding
15.4.2 Vapor Space Condensation
15.4.3 In-Tube Condensation
15.4.4 Falling Film Evaporation
15.4.5 Correlations
15.5 BOILING
15.5.1 Fundamental Understanding
15.5.2 Pool Boiling
15.5.3 Convective Vaporization
15.5.4 Critical Heat Flux
15.5.5 Correlations
15.6 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 16: SIMULTANEOUS HEAT AND MASS TRANSFER
16.1 INTRODUCTION
16.2 MASS TRANSFER RESISTANCE IN THE GAS PHASE
16.2.1 Condensation with Noncondensible Gases
16.2.2 Evaporation into Air
16.2.3 Dehumidifying Finned-Tube Heat Exchangers

16.2.4 Water Film Enhancement of Finned-Tube Exchanger
16.3 CONTROLLING RESISTANCE IN LIQUID PHASE
16.4 SIGNIFICANT RESISTANCE IN BOTH PHASES
16.5 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 17: ADDITIVES FOR GASES AND LIQUIDS
17.1 INTRODUCTION
17.2 ADDITIVES FOR SINGLE-PHASE LIQUIDS
17.2.1 Solid Particles
17.2.2 PEC Example 17.1
17.2.3 Gas Bubbles
17.2.4 Suspensions in Dilute Polymer and Surfactant Solutions
17.3 ADDITIVES FOR SINGLE-PHASE GASES
17.3.1 Solid Additives
17.3.2 Liquid Additives
17.4 ADDITIVES FOR BOILING
17.5 ADDITIVES FOR CONDENSATION AND ABSORPTION
17.6 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 18: MICROCHANNELS
18.1 INTRODUCTION
18.2 FRICTION IN SINGLE MICROCHANNELS
18.3 FRICTION IN A SINGLE CHANNEL VS. MULTICHANNELS
18.4 SINGLE-PHASE HEAT TRANSFER IN MICROCHANNELS
18.4.1 Single Channel Flow
18.4.2 Heat Transfer in Multiple Microchannels
18.5 MANIFOLD SELECTION AND DESIGN
18.5.1 Single-Phase Flow
18.5.2 Two-Phase Flow
18.6 NUMERICAL SIMULATION OF FLOW IN MANIFOLDS
18.7 TWO-PHASE HEAT TRANSFER IN MICROCHANNELS
18.8 CONCLUSIONS
NOMENCLATURE
REFERENCES
CHAPTER 19: ELECTRONIC COOLING HEAT TRANSFER
19.1 INTRODUCTION
19.2 COMPONENT THERMAL RESISTANCES
19.3 LIMITS ON DIRECT HEAT REMOVAL WITH AIR-COOLING
19.3.1 PEC Example 19.1, Enhanced Fin Geometry Heat Sink
19.4 SECOND GENERATION IndHR DEVICES FOR HEAT REMOVAL AT HOT SOURCE
19.4.1 Single-Phase Fluids
19.4.2 Two-Phase Fluids
19.4.3 Heat Pipe

19.4.4 Nucleate Boiling
19.4.5 Forced Convection Vaporization
19.4.6 Spray Cooling
19.5 DISCUSSION OF ADVANCED HEAT REMOVAL CONCEPTS
19.5.1 Jet Impingement/Spray Cooling Devices
19.5.2 Single-Phase Microchannel Cooling
19.5.3 Two-Phase Microchannel Cooling
19.5.4 Enhanced Two-Phase Forced Convection Cooling
19.6 REMOTE HEAT-EXCHANGERS FOR IndHR
19.6.1 Air-Cooled Ambient Heat Exchangers
19.6.2 Condensing Surfaces
19.6.3 Design for Multiple Heat Sources
19.7 SYSTEM PERFORMANCE FOR THE IndHR SYSTEM
19.8 CONCLUSIONS
NOMENCLATURE
REFERENCES
PROBLEM SUPPLEMENT
INDEX

PREFACE

Many changes have occurred in the field of enhanced heat transfer since the publication of the first edition in 1993. While the use of relatively complex geometries was initially limited by manufacturing processes, new manufacturing methods now allow the assemblage of many complex surface geometries. Some enhanced surfaces (e.g., boiling and condensing tubes) are in their fourth generation. As noted in Chapter 1, nearly all heat exchangers used in the air-conditioning and automotive industries are enhanced geometries. Further inroads are being seen in the electronic cooling, process, and power industries.

Fundamental and even graduate level courses in heat transfer typically focus on simple geometries, such as flat plates and circular tubes. Enhanced heat transfer surfaces typically involve complex geometries. Study of enhanced heat transfer concepts will improve the student's ability to address such complex geometries. A key element of the material in the book is the inclusion of tools to address or analyze such geometries. This textbook is primarily directed at practitioners of high-performance heat transfer concepts. However, it is also suitable for use as a text in a second level (or graduate level) heat transfer course. I taught a course on Enhanced Heat Transfer for a number of years at Pennsylvania State University. The present book and included problems are products of this course.

The second edition provides a general update to the initially published 17 chapters and adds two new chapters: Micro-Channels (Chapter 18) and Cooling of Electronic Equipment (Chapter 19). The net additions to the book include 126 new figures, 18 tables, and a doubling of the number of problems at the end of the book. These problems can be used by readers to test their understanding or by instructors teaching an academic course. The book also contains an increased number of solved examples.

A major addition to the book is the inclusion of a Reference CD-ROM containing 9,500 references. We have attempted to include all published journal papers since 1960, with a number of earlier references. The bibliography on the CD is in a pdf format, listing references alphabetically by the first author's last name. One may use the CD to search by author, journal, year, or key word in the article title. The format of a typical reference is given below:

Adamek, T.A., and Webb, R.L., 1990. "Prediction of Film Condensation on Horizontal Integral-fin Tubes," *Int. J. Heat Mass Transfer,* Vol. 33, pp. 1721–1735.

The book is now jointly co-authored with Professor Nae-Hyun Kim, who com-pleted his Ph.D. with me in 1987. Prof. Kim has devoted his research to enhanced heat transfer and is now an international authority on the subject. I am extremely grateful to Prof. Kim for his collaboration and hard work on this second edition.

Ralph L.Webb

CHAPTER 1
INTRODUCTION TO ENHANCED HEAT TRANSFER

1.1 INTRODUCTION

The subject of enhanced heat transfer has developed to the stage that it is of serious interest for heat exchanger application. The refrigeration and automotive industries routinely use enhanced surfaces in their heat exchangers. The process industry is aggressively working to incorporate enhanced heat transfer surfaces in its heat exchangers. Virtually every heat exchanger is a potential candidate for enhanced heat transfer. However, each potential application must be tested to see if enhanced heat transfer "makes sense." The governing criteria are addressed later.
Heat exchangers were initially developed to use plain (or smooth) heat transfer surfaces. An "enhanced heat transfer surface" has a special surface geometry that provides a higher hA value, per unit base surface area than a plain surface. The term "enhancement ratio" (E_h), is the ratio of the hA of an enhanced surface to that of a plain surface. Thus,

$$E_h = \frac{hA}{(hA)_p}$$

(1.1)

Consider a two-fluid counterflow heat exchanger. The heat transfer rate for a two-fluid heat exchanger is given by

$Q=UA\Delta T_m$
(1.2)

To illustrate the benefits of enhancement, we will multiply and divide Equation 1.2 by the total tube length, L

$$Q = \frac{UA}{L} L\Delta T_m$$

(1.3)

The term L/UA is the overall thermal resistance, per unit tube length, and is given by

$$\frac{L}{UA} = \frac{L}{\eta_1 h_1 A_1} + \frac{Lt_w}{k_w A_m} + \frac{L}{\eta_2 h_2 A_2}$$

(1.4)

where subscripts 1 and 2 refer to fluids 1 and 2, respectively. The term η is the surface efficiency, should extended surfaces be employed. For simplicity, Equation 1.4 does not include fouling resistances, which may be important. The performance of the heat exchanger will be enhanced if the term UA/L is increased. An enhanced surface geometry may be used to increase either or both of the hA/L terms, relative to that given by plain surfaces. This will reduce the thermal resistance per unit tube length, L/UA. This reduced L/UA may be used for one of three objectives:
1. Size reduction: If the heat exchange rate (Q) is held constant, the heat exchanger

length may be reduced. This will provide a smaller heat exchanger.

2. Increased *UA:* This may be exploited either of two ways:

a. Reduced ΔT_m: If Q and the total tube length (L) are held constant, the ΔT_m may be reduced. This provides increased thermodynamic process efficiency, and yields a savings of operating costs.

b. Increased heat exchange: Keeping L constant, the increased UA/L will result in increased heat exchange rate for fixed fluid inlet temperatures.

3. Reduced pumping power for fixed heat duty. Although it may seem surprising that enhanced surfaces can provide reduced pumping power, this is theoretically possible. However, this will typically require that the enhanced heat exchanger operates at a velocity smaller than the competing plain surface. This will require increased frontal area, which is normally not desired.

The important principle to be learned is that *an enhanced surface can be used to provide any of three different performance improvements*. Which improvement is obtained depends on the designer's objectives. Thus, Designer A may seek a smaller heat exchanger, and Designer B may want improved thermodynamic process efficiency.

Although the size reduction of Objective 1 may be valued, the more important objective may be cost reduction. In many cases, the designer requires that the size reduction be accompanied by cost reduction. Another factor to consider for Objective 1 is that the fluid volume in the heat exchanger will also be reduced. This may be an important consideration for a manufacturer of refrigeration equipment, because a smaller volume of expensive refrigerant will be required.

Objectives 2 and 3 are important if "life cycle" costing is of interest. For example, Objective 2 for refrigeration condensers and evaporators will result in reduced compressor power costs. Objective 3 is important for upgrading the capacity of an existing heat exchanger. This may allow plant output to be increased.

Pressure drop (or pumping power) is always of concern to the heat exchanger designer. Hence, a practical enhanced surface must provide the desired heat transfer enhancement and meet the required flow rate and pressure drop constraints. Chapters 3 and 4 discuss how one meets these objectives for single-phase (Chapter 3) and two-phase (Chapter 4) flows. A surface geometry that provides a given heat transfer enhancement level with the lowest pressure drop is definitely preferred.

1.2 THE ENHANCEMENT TECHNIQUES

Bergles et al. [1983] have identified 13 enhancement techniques. These techniques are segregated into two groupings: "passive" and "active" techniques. Passive techniques employ special surface geometries, or fluid additives for enhancement. The active techniques require external power, such as electric or acoustic fields and surface vibration. The techniques are listed in Table 1.1, and are described below.

1.2.1 Passive Techniques

Coated surfaces involve metallic or nonmetallic coating of the surface. Examples include a nonwetting coating, such as Teflon, to promote dropwise condensation, or a hydrophilic coating that promotes condensate drainage on evaporator fins, which reduces the wet air pressure drop. A fine-scale porous coating may be used to enhance nucleate boiling. Figure 1.1 a shows the cross section of a sintered porous metal coating for nucleate boiling. The particle size is on the order of 0.005 mm. Figure 1.1b shows larger particles (approximately 0.5 mm) sintered to the surface. These may be used to enhance single-phase convection or condensation.

Rough surfaces may be either integral to the base surface, or made by placing a "roughness" adjacent to the surface. Integral roughness is formed by machining, or "restructuring" the surface. For single-phase flow, the configuration is generally chosen to promote mixing in the boundary layer near the surface, rather than to increase the heat transfer surface area. Figure 1.2a shows two examples of integral roughness. Formation of a rough surface by machining away metal is generally not an economically viable approach. Figure 1.2b shows an enhanced "rough" surface for nucleate boiling. The surface structuring forms artificial nucleation sites, which provide much higher performance than a plain surface. Figure 1.2c shows the wire coil insert, which periodically disturbs the boundary layer. A wire coil insert is an example of a non-integral roughness.

Extended surfaces are routinely employed in many heat exchangers. As shown in Equation 1.4, the thermal resistance may be reduced by increasing the heat transfer coefficient *(h)*, or the surface area *(A)*, or both *h* and *A*. Use of a plain fin may provide only area increase. However, formation of a special shape extended surface