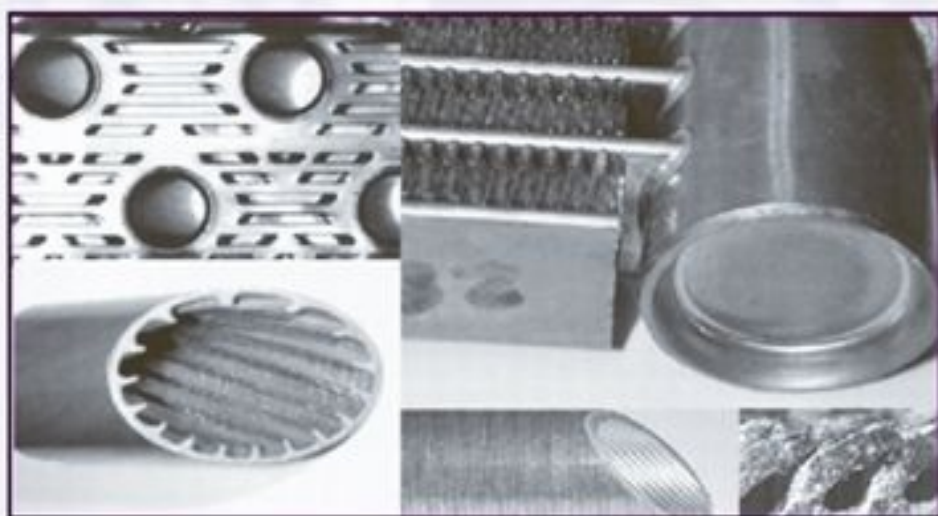


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Principles of Enhanced Heat Transfer



Ralph L. Webb
Nae-Hyun Kim

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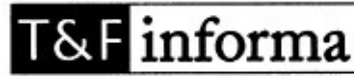
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PREFACE

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 completed 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,

$$E_h = \frac{hA}{(hA)_p} \quad (1.1)$$

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

$$Q = UA\Delta T_m \quad (1.2)$$

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

$$Q = \frac{UA}{L} L\Delta T_m \quad (1.3)$$

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

$$\frac{L}{UA} = \frac{L}{\eta_1 h_1 A_1} + \frac{L t_w}{k_w A_m} + \frac{L}{\eta_2 h_2 A_2} \quad (1.4)$$

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](#) 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