Heat Transfer Enhancement of Heat Exchangers

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Preface

This volume contains an archival record of the NATO Advanced study Institute on Heat Transfer Enhancement of Heat Exchangers held in Çeşme-İzmir Turkey, May 25 - June 5, 1998. The NATO ASIs are intended to be high level teaching activities in scientific and technical areas of current concern. Certainly, the subject of heat transfer enhancement needs no justification in this regard.

Energy and material-saving considerations, as well as economical and ecological incentives, have led to concerted efforts during the past 20-25 years to produce more efficient heat-exchange equipment. Because of the environmental considerations, design of new heat-exchange equipment for recycling is becoming more important. Therefore, if thermal energy can be conserved, the economical and ecological handling of thermal energy through heat exchangers will be possible. The usual goals are to reduce the size of a heat exchanger required for a specific heat duty and to upgrade the capacity of an existing heat exchange equipment. The present trend is moving toward compact heat exchangers, i.e., toward components with augmentation, enhancement or intensification. Therefore, the heat transfer enhancement in heat-exchange equipment becomes a challenge for the scientists and engineers in industry. World wide demand for efficient, reliable and economical heat-exchange equipment is accelerating rapidly, particularly in large-scale power and process industry, refrigeration and air-conditioning systems. Great improvements in energy conservation and the protection of the environment are possible if the fundamentals, methods of enhancement for heat transfer and design of efficient equipment with augmented surfaces were better understood.

During the ten working days of the Institute, the invited lecturers revive the current state-of-knowledge of heat transfer enhancement of heat exchangers. They discussed many techniques that have been developed to enhance heat transfer in single-phase forced convection, flow boiling, condensation, including modern advances in optical measuring techniques and tools to support energy conservation.

The sponsorship of the NATO Scientific Affairs Division is gratefully acknowledged; in person, we are very thankful to Dr. L.V. da Cunha, Director of ASI Programs who continuously supported and encouraged us at every phase of our activity. Our special gratitude goes to Drs. Egrican, M.Ünsal and T.Ayhan for their help in coordinating sessions during the Institute. We are very thankful to Liping Cao and A.Ümit Coşkun for their invaluable efforts in making the Institute a success. A word of appreciation is also due to the members of the sessions chairmen for their efforts in expediting the technical sessions. We are also grateful to the Kluwer Academic Publishers for their cooperation in preparing this archival record of the Institute, and F.Arınç, Secretary General of ICHMT for his guidance and help during the entire process of organization of this ASI.

Finally, our heartfelt thanks to all invited lecturers and authors, who provided the substance of the Institute, and to the participants for their attendance, questions, and comments.

Sadık Kakaç
Art E. Bergles
Franz Mayinger
Hafit Yüncü

TABLE OF CONTENTS

INTRODUCTION TO HEAT TRANSFER ENHANCEMENT

-Preview of Contributions

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Abstract. This lecture provides a general introduction to the subject of the Institute. Introductory remarks are made about heat transfer enhancements, surface augmentation or intensification in heat exchangers. The preview of lecture topics is introduced, indicating their contributions to the institute.

1. Introduction

Heat exchangers are devices that provide the flow of thermal energy between two or more fluids at different temperatures. These include power production, process, chemical and food industries, electronics, environmental engineering, waste heat recovery, manufacturing industry, and air-conditioning, refrigeration, and space applications.

In these applications, thermal-hydraulics and energy usage play dominant roles. Energy, space, and materials saving considerations, as well as the present-day global economics, have led to the expansion of efforts to produce more efficient heat exchange equipment for minimizing cost, which is to reduce the physical size of heat exchange equipment for a given heat duty. Therefore the main thermal-hydraulic objectives are to reduce the size of a heat exchanger required for a specific heat duty, to upgrade the capacity of an available heat exchanger and its operation with smaller approach temperature differences, or to reduce the pumping power.

There are various techniques used for heat transfer enhancement, which are segregated in two groups: (1) activate (2) passive techniques. The active techniques require external power to the surface(surface vibration, acoustic or electric fields). Passive techniques use specific surface geometries with surface augmentation.

1.1 BASICS OF HEAT TRANSFER ENHANCEMENT

In heat exchangers, depending on the applications, plain or enhanced/augmented heat transfer surfaces are used. Heat transfer between a wall and the fluid is given by

$$Q = hA(T_w - T_f) \tag{1}$$

or

S. Kakaç et al. (eds.), Heat Transfer Enhancement of Heat Exchangers, 1–11.

$$Q = (hA)_p (T_w - T_f) \tag{2}$$

The ratio of the (hA) of an augmented surface to that of a plain surface is defined as the enhancement ratio which is [1]

$$E = \frac{hA}{(hA)_p} \tag{3}$$

There are several methods to increase the hA value:

a) Heat transfer coefficient can be increased without an appreciable increase in the surface area.
b) Surface area can be increased without appreciable changes in heat transfer coefficient.
c) Both the heat transfer coefficient and the surface area can be increased.

These methods will be discussed and examples will be given in the following chapters.

Heat transfer from a finned surface (augmented) is given by

$$Q = \eta h A (T_w - T_f) \tag{4}$$

or

$$Q = \eta h A \Delta T \tag{5}$$

where

$$\eta = [1 - \frac{A_f}{A}(1 - \eta_f)] \tag{6}$$

is called overall surface efficiency. ΔT is the temperature difference between the fluid stream and the wall temperature, and the total surface area on one side is $A = A_u + A_f$ (Figure 1)

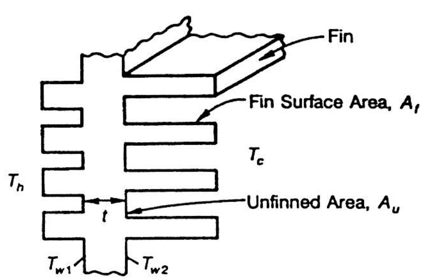

Figure 1. Finned Wall

Consider a two-fluid separated by a wall (heat exchanger). The total heat transfer rate, Q, between the fluids can be determined from the following equation:

$$Q = UA\Delta T_m \tag{7}$$

The term *1/UA* is the overall thermal resistance which can be expressed as [2]:

$$R_t = \frac{1}{UA} = \frac{1}{U_o A_o} = \frac{1}{\eta_i h_i A_i} + \frac{R_{fi}}{\eta_i A_i} + R_w + \frac{R_{fo}}{\eta_o A_o} + \frac{1}{\eta_o h_o A_o} \tag{8}$$

which includes fouling on both sides of the heat transfer surface. The fouling characteristics of an enhanced surface are important to be considered.

1.2 PERFORMANCE EVALUATION

The performance of the heat exchanger will be increased if the term (UA) is increased or the overall thermal resistance is reduced. The details of the principles of performance evaluation for single-phase and two-phase flows can be found in [1, 3].

The heat transfer enhancement increases with either or both of the (hA) terms, that is decreased convective resistances on both sides.

It can be seen from Eq. (7), the increased UA will contribute to

a) Reduction of the size of the heat exchanger: if the heat duty is kept constant, under the same temperature conditions, the length of the heat exchanger may be reduced. If the length and the inlet temperatures are kept constant, then the heat exchange rate will increase.

b) If Q and the total length (L) are kept constant, then ΔT_m may be reduced. This provides increased thermodynamic process efficiency (the second law efficiency) and yields lower system operational cost [4].

c) For fixed heat duty, Q, the pumping power can be reduced. However, this will dictate that the enhanced heat exchanger operates at a smaller velocity than the plain surface. This will require increased frontal area.

The selection one of the above improvements will depend on the objectives of designing the heat exchanger. The most important factor, in general, is the size of reduction, which results in cost reduction.

1.3 PUMPING POWER-SINGLE PHASE

Pumping power expenditure (pressure drop) is always to be considered by the designer.

$$P = \frac{\dot{m}\Delta p}{\eta_p \rho} \tag{9}$$

In practice, the selected surface must satisfy the process specifications including pressure drop constraints. A surface that provides an enhancement level and satisfies all the specifications with the lowest possible pumping power expenditure is certainly preferred. The pressure drop limitation is a very important consideration of an enhanced surface versus a plain surface.

Let us consider a single-phase side passage (a duct) of a heat exchanger where the flow is turbulent and the surface is smooth. It can be shown that the pumping power expenditure per unit heat transfer area (W/m^2) can be expressed as [2]

$$\frac{P}{A} = C\frac{h^{3.5}\mu^{1.83}D_h^{1/2}}{k^{2.33}C_p^{1.17}\rho^2\eta_p} \tag{10}$$

where $C = 1.25\times10$ 4

As can be seen from Eq. (10), the pumping power expenditure per heat transfer area depends strongly on fluid properties, as well as on the heat transfer coefficient, and hydraulic diameter of the flow passage.

Some important conclusions can be drawn from Eq. (10) (see Table 1):

1. With a high-density fluid such as a liquid, the heat exchanger surface can be operated at large values of h without excessive pumping power requirements.
2. A gas with its very low density results in high values of pumping power for even very moderate values of heat transfer coefficient.
3. A large value of viscosity causes friction power to be large even though density may be high. Thus, heat exchangers using oils must be designed for relatively low values of h to hold the pumping power within acceptable limits.
4. The thermal conductivity, k, also has a very strong influence; and therefore for liquid metals with very large values of thermal conductivity, the pumping power is seldom of significance.
5. Small values of hydraulic diameter, D_h, tend to minimize the pumping power.

TABLE 1 Pumping power expenditure for various fluid conditions

Fluid Conditions	**Power Expenditure (W/m^2)**
Water at 300 K	
h=3850 W/m^2·K	3.85
Ammonia at 500 K, atm pressure	
h=100 W/m^2·K	29.1
h=248 W/m^2·K	697
Engine oil at 300 K	
h=250 W/m^2·K	0.270×10^4
h=500 W/m^2·K	3.06×10^4
h=1200 W/m^2·K	65.5×10^4

Note: η_p=80%, D_p=0.0241 m.

Heat transfer coefficient of an enhanced surface which is given by the Colburn modulus, J, and friction factor of the surface, f, are usually presented as a function of Reynolds number (D_hG/μ):

$$J = \frac{h}{Gc_p} Pr^{2/3} \tag{11}$$

The friction factor of an enhanced surface in a single-phase flow is higher than that of the smooth surface, when operated at the same Reynolds number. Therefore the enhanced surface would be allowed to operate at a higher pressure drop. The plain surface would give a higher heat transfer coefficient if it were also allowed to operate at a higher velocity, giving the same pressure drop as the enhanced surface. Therefore, as it can be seen from Eq. (10) and Table 1, the actual performance improvement cannot be obtained by calculating h/h_p and f/f_p at the same Reynolds number (at equal velocities).

Performance evaluation allows us to make a simple surface performance comparison and compare the thermal resistances of both fluid streams. The performance analyses allow the designer to quickly identify the most effective surface or other enhancement techniques of variable choices. These will be discussed in the various chapters of this volume.

If one of the fluids in the heat exchange undergoes a phase change, then this exchanger is called as a two-phase heat exchanger. Some of the examples are evaporators, boilers, reboilers and condensers.

In two-phase flow heat exchangers, pressure drop of phase changing fluid may affect the mean temperature difference in the heat exchanger, and the performance evaluation must take this effect into account. In a single-phase flow, the reduced pumping power can be considered as an objective function, which is not applicable to the two-phase flow situation. Detailed analysis of performance evaluation criteria for two-phase flow heat exchangers is presented in [1].

1.4 PUBLISHED LITERATURE ON HEAT TRANSFER ENHANCEMENT

Bergles et al. [5, 6, 7] have reported the journal publications on enhanced heat transfer. Bergles and Webb [8] summarize the information presented in the technical and patent literature. Figure 2, taken from Bergles et al. [7], shows the journal and conference publications per year between 1900 and 1990, which total 4345, as of December 1990. As August 1995, then total number of papers and reports increased to 5676 [10]. The period of rapid growth began about 1960. The patent literature is an equally important source of information on enhancement techniques. Figure 3 taken from Webb et al. [9] survey of the U.S. patent literature, shows 486 patent issued as of June 1982.

2. The Program of the Institute

During the 10 working days of this Institute, invited lecturers and papers as supplementary to the lectures will be presented. During the institute various heat enhancement/augmentation techniques and their performances will be introduced. This section introduces some of the topics that will be considered in detail in the following lectures.

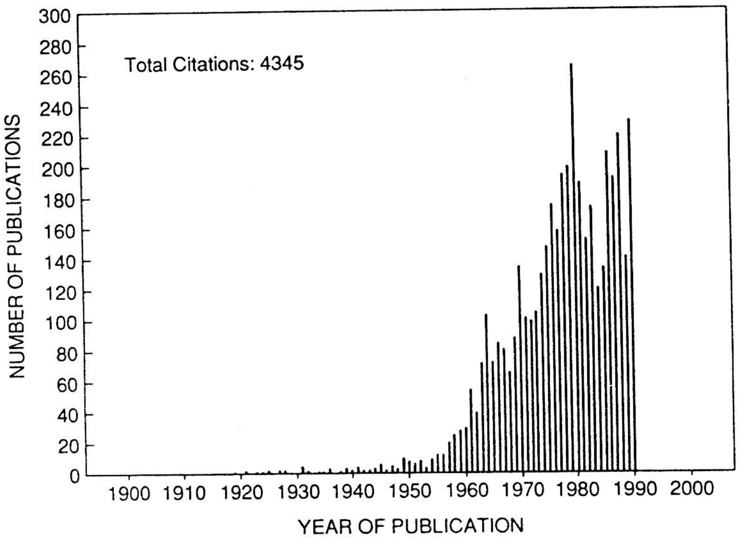

Figure 2. Citations on heat transfer augmentation versus year of publications. Status as of December 1990 is illustrated (From Bergles et al. [6]).

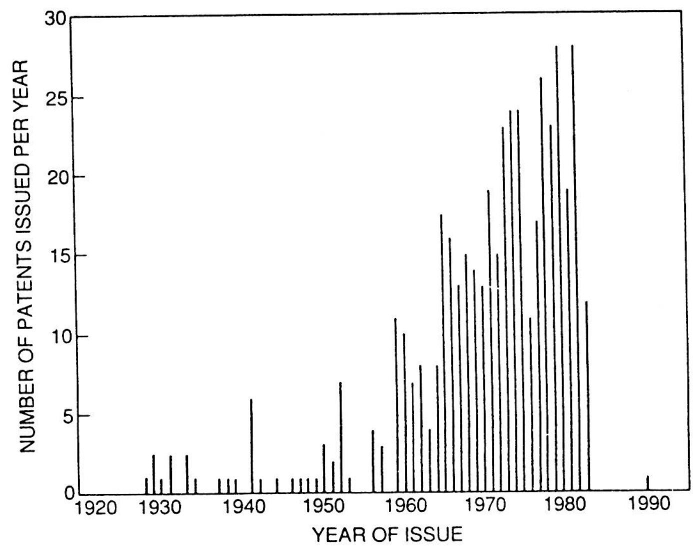

Figure 3. Patents on heat transfer augmentation versus year of publication (From Webb et al. [9]).

Bergles presents **the imperative to enhance heat transfer** and considers many techniques that have been developed to enhance convective heat transfer, and examples are given for the application to various modes of heat transfer. This is an introduction to our entire program. Bergles then presents micro-fin tube technology where he outlines a comprehensive review of the enhanced "micro-fin" tubes that have been effectively applied to single-phase flows, flow boiling/evaporation and convective condensation. Microfin tubes are characterized by a large number of longitudinal or helical fins inside small diameter tubes with fin heights in the range of only 0.1-0.4 mm. They are widely used in air-conditioning and refrigeration. **The most recent experimental research on evaporation in microfin tubes** is reviewed and the latest prediction methods are critically surveyed by Thome.

Webb presents **a survey on the prediction of condensation and evaporation in microfin and micro-channel tubes**. There are several correlations to predict heat transfer of flow boiling (Shah, Kandlikar and Güngör and Winterton) and condensation (Traviss et al., Shah, Cavallini and Zecchin) in plain tubes. In this work, an improved predictive model based on the two-phase heat momentum transfer analogy is described which is called "equivalent Reynolds number model". For condensation in micro-fin tubes, both vapor shear and surface tension forces contribute to the condensing coefficient. It is shown that the vapor shear model is also applicable to evaporation inside tubes, and the nucleate boiling contribution is less than 15% and exists only at vapor qualities less than 50%.

In his second lecture, Webb decribes a series of studies to understand the mechanism of **boiling in structured surfaces** having sub-surface tunnels and surface pores.

Performance Enhancement of finned oval tubes with finned longitudinal vortex generators is introduced by Fiebig and Chen. As it is mentioned above, fins are used to reduce the thermal resistance on the gas side of a gas-liquid heat exchanger as in the case of a tube-fin heat exchanger. The heat transfer coefficients on the gas side is very small (30-300 $W/m^2 \cdot K$) compared with the liquid side heat transfer coefficient. To balance the resistance on both sides, a very large heat transfer area ratio of fin to tube would be needed; then the overall surface efficiency of the finned surface will decrease, Eq. (6), and the economy of fins decreases. Therefore heat transfer enhancement is necessary on the fins side. Fiebig and Chen introduce the delta Winglet pair (DWP) in a finned oval tube to enhance heat transfer on the gas side, and present flow structures and heat transfer analysis. Vortex geometries can be used as fins in plate heat exchangers, and as fin surface modifications in any finned heat exchangers.

There are a number of lectures on heat exchangers used in air-conditioning and refrigeration systems.

Successful design of evaporators and condensers in air-conditioning systems using the new refrigerants as working fluids would require heat transfer and pressure drop characteristics and heat transfer enhancement inside the tubes.

Ebisu and Torikoshi present **evaporation and condensation heat transfer characteristics of R-407C which is an alternative refrigeratnt for R-22**. They discuss a new heat transfer tube having a special inner surface which is named a "W-design heat transfer tube", and they provide information on the heat exchanger performance improvement by the use of the W-design heat transfer tubes in air-conditioning condensers and evaporators. Comparisons are made for the data of the W-design tube with those for the existing inner grooved tube.