HIGH-FLUX PROCESSES THROUGH ENHANCED HEAT TRANSFER

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ABSTRACT

Phase-change processes, such as pool and flow boiling, are generally very effective modes of heat transfer. However, the demands of modern thermal systems have required the development of methods to enhance boiling systems. While heat fluxes above 10^8W/m^2 have been accommodated in carefully controlled situations, the required fluid and the convective conditions usually dictate maximum heat fluxes several orders of magnitude lower. Two major contemporary areas, enhanced surfaces for pool boiling and enhanced surfaces and inserts for forced convection boiling/vaporization, are discussed, as they facilitate the attainment of high heat fluxes. In addition to these passive techniques, active techniques and compound techniques are mentioned. The taxonomy of enhanced heat transfer is covered, and recommendations are given for future work.

1.0 INTRODUCTION

1.1 High-Flux Processes

Modern technology is characterized by the tendency to package larger power conversion or transfer devices in smaller volumes. Ranging from the largest to the smallest, examples of such devices abound in nuclear technology, aerospace, and microelectronics. The heat transfer objectives can usually be stated as either a) removing large rates of energy generation through small surface areas with moderate surface temperatures rises or b) reducing the size of a boiler for a given rating. Both objectives involve higher heat fluxes. This desire to accommodate or promote high heat fluxes has been a major driving force for the study of boiling heat transfer, in general, and the development of methods to enhance boiling heat transfer, in particular. This paper is directed at applications involving both boiling/evaporation and enhanced heat transfer.

Nucleate boiling has long been recognized as a very efficient mode of heat transfer. The pioneering work of Jakob and Fritz (1931) documented the high heat fluxes that could be obtained with pool boiling of water at low temperature differences, in effect, representing very high heat transfer coefficients. This also demonstrated a strong effect of surface condition. Thus, it was established over 70 years ago that nucleate boiling had the promise to satisfy the two main objectives noted above; furthermore, the possibility of enhancing boiling performance was good. As chronicled in Bergles (1981) and Nishikawa (1987), there was intense interest in boiling heat transfer during the next 50 years, for the academic challenges as well as industrial applications. At this point, it is helpful to examine the spectrum of heat fluxes that must be accommodated or are typically achieved. Table 1, an extension of that given by Gambill and Greene (1958), gives some heat flux ranges, and Fig. 1 presents a graphic display of heat fluxes.

TABLE 1. HEAT FLUX RANGES (W/m^2)

Process industry heat exchangers; low for subcooling in condensers, high for some vaporizers.	$1.6 \times 10^3 – 1.6 \times 10^5$
Microelectronic chip dissipating 10W through a 5 mm x 5 mm side.	4×10^5
Peak flux for water in saturated pool boiling at one atmosphere.	1.3×10^6
Range for 12 thermal nuclear reactors of various types, maximum core heat flux (avg. = 1/3 to 1/6 of maximum).	$6.8 \times 10^5 – 3.6 \times 10^6$
Available parabolic reflector solar furnaces, maximum at focus.	6.3×10^6

Attained successfully in nozzle throats of operational liquid-propellant rockets.	1.6×10^7
Commercial plasma jet (water stabilized arc).	1.9×10^7
Highest burnout for a twisted-tape, Tong et al. (1996)	1.4×10^8
Highest burnout flux with spiral ramp inlet vortex generator	1.7×10^8
Maximum mentioned in literature for burnout with water-butanol in linear flow, uniform heating, Ornatskii and Vinyarskii (1965)	2.3×10^8
Highest mentioned in literature for uneven heating of a circular tube, water-butanol in linear flow, Ornatskii and Vinyarskii (1965)	3.7×10^8
Highest estimated flux (stagnation point, non-CHF, melted plate) for water-jet cooling of a plasma-arc heated plate. Liu and Lienhard (1993)	4.2×10^8
Carbon sublimation cooling (air plasma heat input), millisecond duration.	3.7×10^9

As mentioned in Table 1, the highest steady heat flux reported for flow in a tube is 3.7×10^8 W/m^2. Ornatskii and Vinyarskii (1965) obtained this by high velocity and highly subcooled flow of water in a small diameter tube with one-sided heating. A small percentage of alcohol was added to enhance heat transfer. This heat flux would, of course, satisfy virtually any system requirement; however, it is difficult to achieve this. Furthermore, there are many applications where water is neither an acceptable coolant nor an appropriate working fluid. Such is the case with direct immersion cooling of microelectronic devices. An array of microelectronic chips, representing a memory or central processing component of a computer, is simply submerged in the liquid. The devices are readily exposed to the coolant, and boiling occurs at higher power levels. Such phase-change cooling is desirable from the standpoint of the electronics, because the devices are then subject to rather small changes in temperature, even for large changes in power.

The fluids used for immersion cooling must be dielectric and inert; in practice, this means using a refrigerant-type fluid having a moderate boiling point, such as R-113. R-113 is, in fact, used for power electronic devices; however, it is not suitable for computers because of probable long-term corrosion. Special fluids, such as the Fluorinert^{TM} or Novec^{TM} "electronic liquids" developed by 3-M, are compatible with chips, wiring, etc. Unfortunately, all of these fluids have poor heat transfer characteristics compared to water because of low thermal conductivity and low latent heat of vaporization. There is also the problem of boiling-curve hysteresis due to the extreme wettability of these fluids. As noted in Fig. 1, there is a particular need to elevate the peak nucleate or burnout heat flux for these fluids. Some enhancement techniques are indicated that accomplish this and, additionally, reduce the wall superheat.

The important point in the preceding discussion is that *high heat flux is a relative term*. Depending on the coolant and the convective conditions, maximum heat fluxes may vary widely, and in most cases these fluxes are several orders of magnitude below the ultimate limit that has been recorded to date. Some general remarks on enhancement follow.

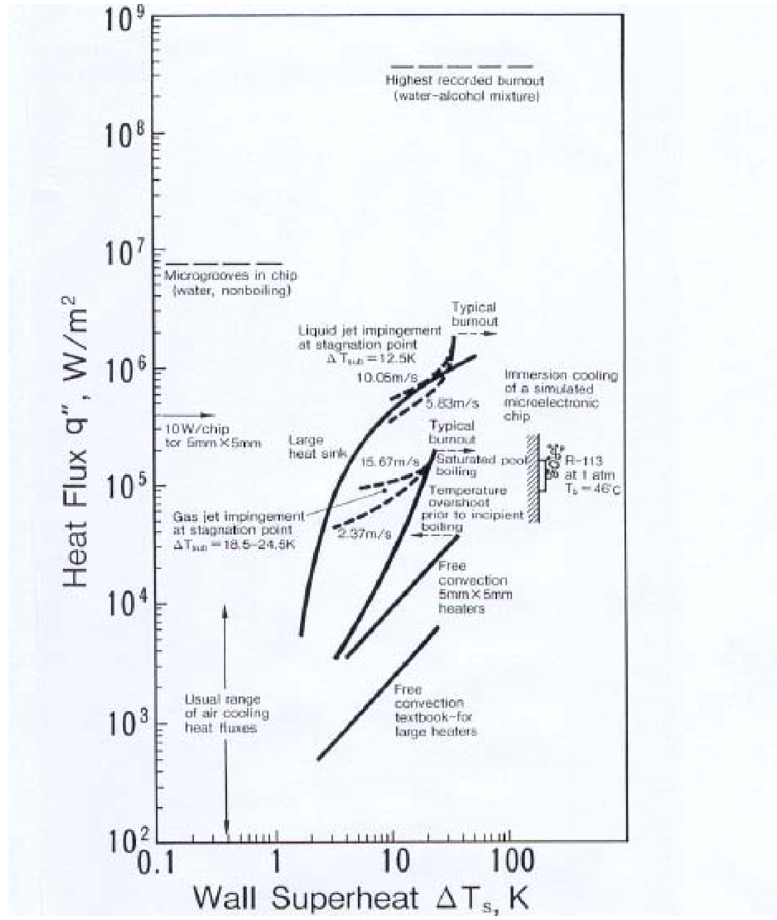

Fig. 1. Heat flux (chip level) ranges for immersion cooling of microelectronic chips; air and water capabilities are provided for reference (Bergles, 1989).

1.2 General Discussion of Enhancement Techniques

The study of improved heat transfer performance is referred to as heat transfer *enhancement, augmentation, or intensification.* In general, this means an increase in heat transfer coefficient. Attempts to increase "normal" heat transfer coefficients have been recorded in the technical literature for well over a century, and there is a large store of information. By "normal" is meant standard flow, including jets and sprays, with plain surfaces.

Enhancement techniques can be classified either as *passive*, which require no direct application of external power, or as *active*, which require external power. The effectiveness of both types of technique is strongly dependent on the mode of heat transfer, which may range from single-phase free convection to dispersed-flow film boiling. A listing of enhancement techniques is shown in Table 2. Two or more of the above techniques may be utilized simultaneously to produce an enhancement that is usually larger than the individual techniques applied separately. This is termed *compound enhancement.*

TABLE 2. CLASSIFICATION OF ENHANCEMENT TECHNIQUES

Passive Techniques	Active Techniques
Treated Surfaces	Mechanical aids
Rough surfaces	Surface vibration
Extended surfaces	Fluid vibration
Displaced enhancement devices	Electrostatic fields
Swirl flow devices	Suction or Injection
Coiled tubes	Additives for fluids
Surface-tension devices	

Compound Enhancement

Extended surfaces and electrostatic fields, for example.

A description of these techniques can be found in Bergles (1997) and Bergles (1998). With 13 enhancement techniques (plus compound enhancement) and 6 modes of convective heat transfer (single-phase, boiling, and condensing – natural or forced flow), it is clear that there are a great many opportunities for research in this field. Not surprisingly, the current emphasis is on effective and cost-competitive (proved or potential) techniques that have been made the transition from the laboratory to commercial heat exchangers. Broad reviews of developments in enhanced heat transfer are available in books: Thome (1990), Webb (1994a) and Kalinin et al. (2002). In any discussion of enhancement, it is necessary to restrict the citations, because the literature is absolutely overwhelming.

A 1995 survey cites 5676 technical publications, excluding patents and manufacturers' literature. The rather recent growth of activity in this area is clearly evident from the yearly distribution of these publications presented in Fig. 2. The data suggest an S-shaped characteristic, and there is no real decline. The apparent decline is because there is always a delay in logging in new literature, as is conspicuous in the case of the 1993-1995 entries. This delay ranges from months to years, depending on the publication media and the efficiency of the search. A current search of the literature for 2001 indicates that publication continues at a rate of about 400 papers and reports (excluding patents) per year. The literature generation in enhanced heat transfer seems to have reached a steady state.

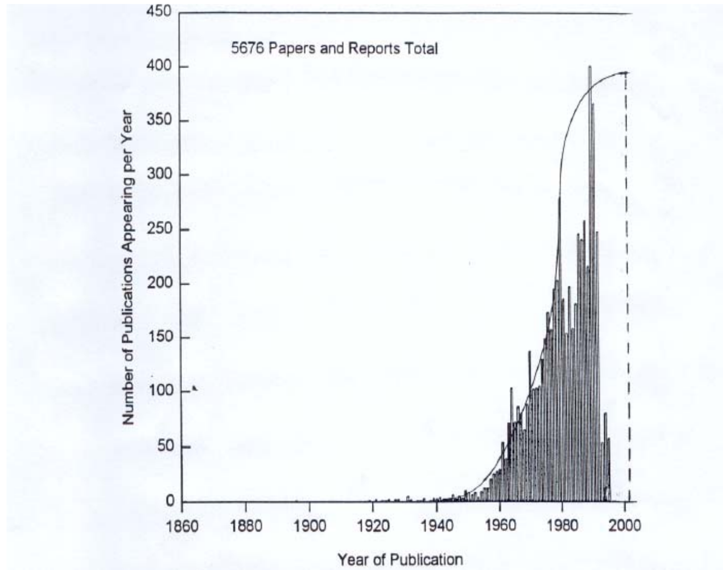

Fig. 2. References on heat transfer enhancement versus year of publication, Bergles et al., 1995), with continuation in 2001 (Manglik and Bergles, 2003).

2.0 ENHANCEMENT OF POOL BOILING

2.1. Applications

Selected passive and active enhancement techniques have been shown to be effective for pool boiling. Most techniques apply to nucleate boiling; however, some techniques are applicable to transition and film boiling. It should be noted that phase-change heat transfer coefficients are relatively high. The main thermal resistance in a two-fluid heat exchanger often lies on the non-phase-change side. (Fouling of either side can, of course, represent the dominant thermal resistance.) On the other hand, the overall thermal resistance may then be reduced to the point where significant improvement in the overall performance can be achieved by enhancing the two-phase flow. Two-phase enhancement is also important in double-phase-change (boiling/condensing) heat exchangers, such as thermosyphon reboilers.

2.2. Enhanced Surfaces

Surface material and finish have a strong effect on nucleate and transition pool boiling. However, reliable control of nucleation on plain surfaces is not easily accomplished. Accordingly, since the earliest days of boiling research, there have been attempts to relocate the boiling curve through use of relatively gross modifications of the surface. For many years, this was accomplished simply by area increase in the form of low helical fins. The subsequent tendency was to improve the nucleate boiling characteristics by fundamental changes in the boiling process. Many of these advanced surfaces are being used in commercial shell-and-tube boilers. Furthermore, they are being considered for use in other areas, such as immersion cooling of microelectronic chips (Fig. 1). This passive technology has found much more favor than active technology.

Several manufacturing processes have been employed: machining, forming, layering, and coating. In Fig. 3 (a), standard low-fin tubing is shown. Figure 3 (c) depicts a tunnel-and-pore arrangement produced by rolling, upsetting, and brushing. An alternative modification of the low fins is shown in Fig. 3 (d), where the rolled fins have been split and further rolled to a T-shape. Further modification of the internal, Fig. 3 (e), or external, Fig. 3 (f), surface is possible. Knurling and rolling are involved in producing the surface shown in Fig. 3(g). The earliest example of a commercial structured surface, shown in Fig. 3 (b) is the porous metallic matrix produced by sintering or brazing small particles to the base surface.

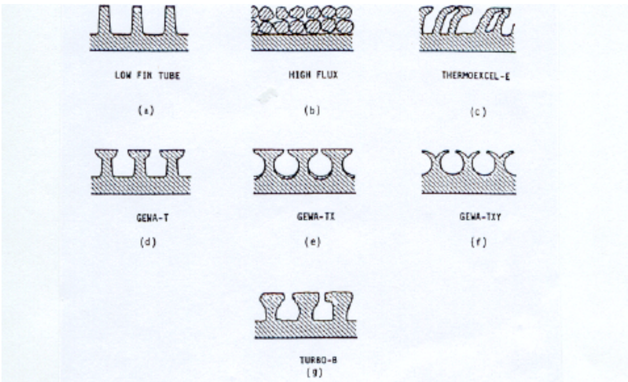

Fig. 3. Examples of structured boiling surfaces (Pate et al.,1990).

The relative performance of three of these surfaces, tested as single tubes, is shown in Fig. 4. Here, as usual, the heat flux is based on the area of the equivalent smooth tube for a particular outside diameter. Wall superheat reductions of up to a factor of ten are common with these surfaces, and there is often an increase in the peak nucleate heat flux. The advantage is seen to be not only high nucleate boiling heat transfer coefficients, but also the fact that boiling can take place at very low temperature differences. Thus, these surfaces can be characterized as low temperature difference surfaces as well as high heat flux surfaces. In all cases, a complex liquid-vapor exchange is involved. The liquid flows in at random locations around the helical aperture or through selected pores to the interior of the structure, where thin film evaporation occurs over a large surface area. The vapor is then ejected through other paths by bubbling. The latent heat transport is complemented by agitated free convection from the exposed surfaces.

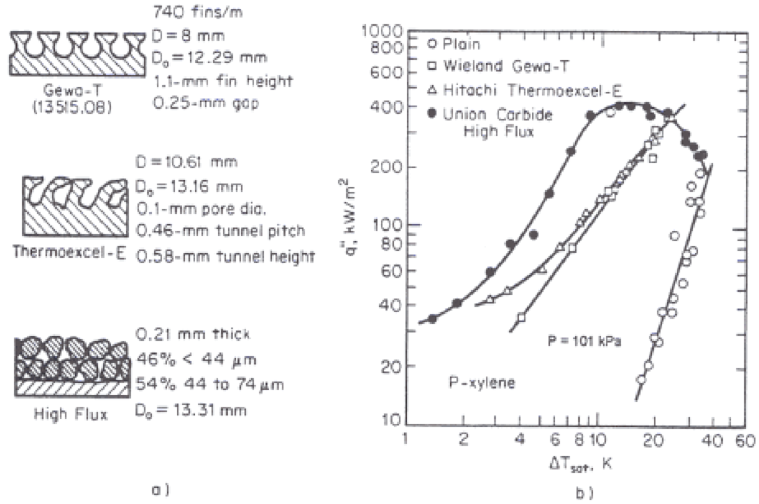

Fig. 4. Pool boiling from smooth and structured surfaces on the same apparatus (Yilmaz et al., 1980).

a. Sketches of cross-sections of the three enhanced heat transfer surfaces tested
b. Boiling curves for the enhanced tubes and a smooth tube.

The behavior of structured surfaces is not yet understood to the point where correlations are available to guide custom production of the surfaces for a particular fluid and pressure level. For example, the Nakayama et al. (1980) model for nucleate boiling from Thermoexcel-E surfaces requires eight empirically determined constants. Such models are useful in confirming the probable physics of the boiling process, but are of no use in engineering design. The problems encountered by these investigators have deterred researchers from undertaking similar studies during the last 25 years. Some manufacturers have accumulated sufficient experience to provide optimized surfaces for important applications, such as flooded evaporators for refrigerant dry-expansion chillers and thermosyphon reboilers for hydrocarbon distillation columns. In spite of the difficulty, it can be said that considerable progress has been made in the analytical/numerical descriptions of some of the enhanced pool boiling surfaces. Webb (1994b) documents the general state-of-the art as of about 10 years ago.

2.3. Temperature Overshoots

Structured surfaces are not exempt from temperature overshoots and resultant boiling curve hysteresis. However, the superheats required for incipient boiling with the highly wetting liquids, such as the "electronic liquids." are generally lower than for plain surfaces due to the greater probability of finding active sites over the larger surface area. An example of such hysteresis for a single tube is shown in Fig. 5. The practical problem is to provide, at least on a transient basis, a trigger to initiate boiling. In some cases, the stimulus can be provided by injected vapor, either naturally as with a dry-expansion chiller or intentionally through sparging – the artificial introduction of vapor. The latter method was used by Bergles and Kim (1988) to improve the boiling performance of simulated microelectronic chips.

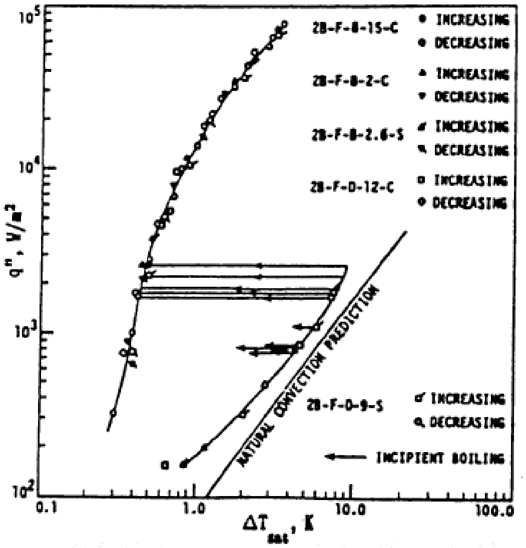

Fig. 5. Illustration of boiling-curve hysteresis with R-113 at 1 atm and a High Flux surface (Bergles and Chyu, 1982).

Certain enhancement techniques are effective in reducing temperature overshoots: surface treatment (painting), O'Connor and You (1995); electrostatic fields, Zaghdoudi and Lallemand (2002).

2.4. Application to Cooling of Microelectronic Devices via Pool Boiling

Structured boiling surfaces developed for process and refrigeration evaporation have been used as "heat sinks" for microelectronic chips. A small section of the surface is attached to the chip with a thin layer of industrial thermal grease or epoxy. Park and Bergles (1986) tested a variety of detachable heat sinks, including several with commercial enhanced boiling surfaces. The heat sinks were epoxyed to the foil heater so that the heat transfer was well defined; the heat flux in Fig. 6 is referenced to the projected area of the heat sink (heater area). It is seen that the order of performance improvement is: microfin, microhole, High Flux (Linde), and Thermoexcel-E. The boiling curve shift relative to the plain heater is substantial, especially at lower heat fluxes. The enhanced heat sinks permit higher heat fluxes than those expected for the plain surface; however, the wall superheat advantage is less at high heat fluxes. This seems to be observed with many enhanced boiling surfaces. Note that these data were taken with decreasing heat flux, so as to avoid temperature overshoots.

The greatest improvements in boiling performance reported to date were obtained by Nakayama et al. (1984) who tested the massive heat sink stud depicted in Fig. 7. The wall superheats for this surface are shown in Fig. 1 are comparable to those noted in Fig. 6; however, very high heat fluxes are attained. Unfortunately, the highest heat fluxes occur at superheats or surface temperature that are above 85ºC, which is usually considered to be the upper level of semiconductor junction temperature for acceptable reliability. The goal, unrealized to date, is to develop enhanced boiling heat sinks that permit high peak heat fluxes at modest superheats.

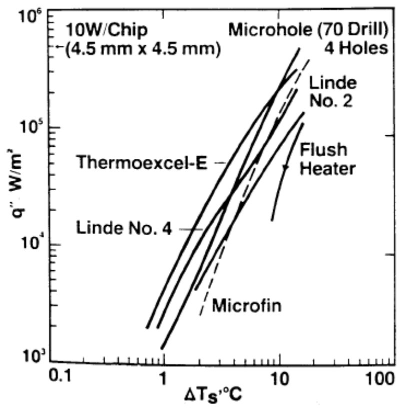

Fig. 6. Composite of data for heat sinks boiling in R-113.

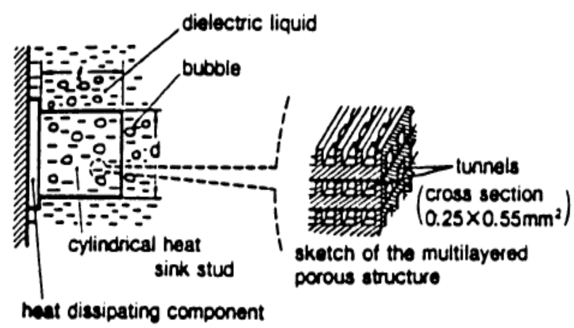

Fig. 7. Porous stud attached to the face of a heat dissipating component.

2.5. Application to Tube Bundles

The performance of enhanced tubes in flooded, horizontal bundles has received considerable attention, because this is the major application for these tubes. The issue at hand can be characterized as the relationship of "separate effects" or single-tube performance to the actual bundle performance. As in most other applications of enhancement technology, the understanding of enhanced tube behavior depends on the mechanism of the normal situation. Plain tube bundles often exhibit substantial increases in average heat transfer coefficients above plain single tubes, as shown in Fig. 8. This "bundle factor" is attributed to the convective effect in the upper portion of the bundle when the vapor quality is high. In the case of enhanced tubes, nucleate boiling overshadows the convective effect, with the result that the bundle performance tends to be reasonably close to that of the single tube, again shown in Fig. 8. The result is a lesser increase in performance than would be expected from simply comparing the single-tube data.

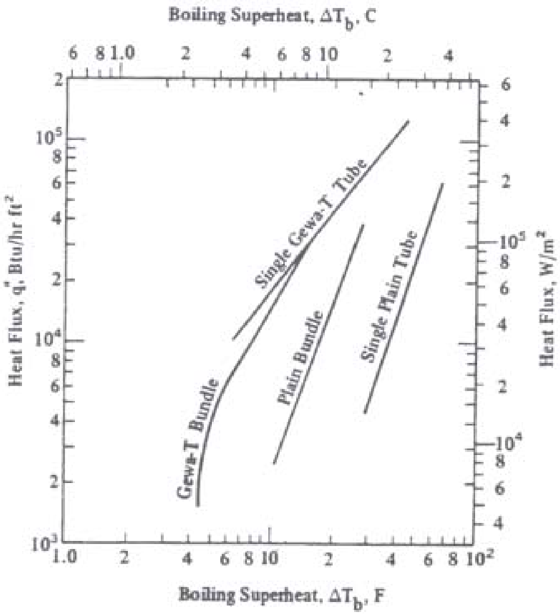

Fig. 8. Comparison of performance for single enhanced and plain tubes to that of enhanced and plain tube bundles with p-xylene at 103 kPa (Yilmaz et al., 1980).

Hsieh et al. (2003) recently reported an extensive study of boiling from plasma-coated tube bundles flooded with saturated R-134a. The bundle-average heat transfer coefficient (at constant heat flux) was found to be up to three times greater than for a smooth-tube bundle.

Flooded natural circulation, recirculation reboilers (thermosyphons) can be of two types: horizontal (kettle reboilers) and vertical. Enhanced tubes have been applied to both types; however, in the former, OD enhancement is used, whereas in the latter, ID enhancement (covered in the next section), is employed. Enhancement affects the vapor generation rate, which, in turn, controls the circulation.

Enhanced bundles are not immune from the problem of boiling curve hysteresis with highly wetting fluids. As reported by Lewis and Sather (1978), an ammonia flooded evaporator with enhanced tubes, intended for closed-cycle ocean thermal energy conversion, came up to full performance only after three days! While this problem does not occur in all systems, because of normal vapor injection or favorable transients, the consequences of the inability to initiate boiling can be severe.

For horizontal bundles where falling-film or spray evaporation is routinely used (absorption refrigeration, chemical processes), there is the possibility of reducing the working-fluid inventory. This is important for the expensive, ozone-friendly refrigerants in large commercial and industrial chillers. Moeykens et al. (1996) reported tests of R-123 with plain and enhanced tubes, and found dramatic increases in heat transfer coefficients with the enhanced surfaces.

2.6. Boiling of Multicomponent Mixtures

Boiling of multicomponent mixtures is the norm, rather than the exception, in the chemical process industry. Also, the air conditioning industry is vigorously investigating mixtures of refrigerants for use in air conditioners and heat pumps.

Enhanced tubes maintain their advantage over plain tubes when boiling mixtures. However, the single-tube characteristics of enhanced tubes, as a function of concentration, do not agree with those of plain tubes. In pool boiling, the effect of a mixture is to cause the wall superheat to increase above that for a linear interpolation based on mixture composition between the single- component data points. For smooth tubes the deviation in wall superheat can be very large. On the other hand, for a sintered porous surface with a mixture of a volatile fluid in a relatively nonvolatile fluid, Uhlig and Thome (1985), for example, showed that the deviation is much smaller than that of a smooth tube. They concluded that the boiling processes for mixtures are different for the smooth and enhanced surfaces.

Thome (1988), reported tests of a plain tube and the Gewa-TX tube ((e) in Fig. 3) with a five-component hydrocarbon mixture (boiling ranges of 68 and 75 K). In addition to the T-shaped fins, the interior channels have small notches around the circumference. The performance gains are similar to those shown in Fig. 4 for a pure fluid. There do not appear to be any reports in the open literature of boiling of mixtures in tube bundles with enhanced tubes.

2.7. Fouling of Structured Surfaces

The final issue that is very important with surfaces having small interior channels is fouling. It would be reasonable to expect solids to precipitate, or less volatile components to "hide out," in the pores, and render the normal boiling process ineffective. Only a few studies have considered fouling on either single tubes or tube bundles, and the results are mixed.

In discussions relating to the application of heat transfer enhancement techniques, it is common to raise concerns about their performance under fouling conditions. This may have a basis in experience, or may derive from imaginary concerns. It so happens that experience and imagination may be poor guides in this area. The conventional wisdom is that the performance of enhancement devices is reduced under fouling conditions, as suggested above. Furthermore, it is felt that these devices promote fouling. The truth, in fact, is often different from the perception, and some enhanced heat transfer devices may also be anti-fouling devices. Somerscales and Bergles (1997) review the empirical evidence and theoretical background of the performance of enhanced heat transfer devices that also mitigate fouling.

In their review, it is shown that enhanced heat transfer and reduced fouling are observed with: a sintered surface, (b) in Fig. 3, and calcium sulfite scale. The same behavior was observed with a structured surface, (g) in Fig. 3. Under potential fouling conditions, it is important to control the liquid chemistry and use a pore size that keeps up the liquid circulation.

Compressor lubricating oil is a likely contaminant in flooded refrigeration evaporators; thus, this might be considered a fouling situation. Alternatively, it could be considered a mixture problem. The three surfaces in Fig. 4 have been tested with refrigerant-oil mixtures. These are conventional refrigerants with soluble mineral oils. As discussed by Bergles (1989), degradation in the performance of enhanced surfaces is frequently observed with 3-9% oil; however, the enhanced surface with an oil mixture still has a better performance than the plain surface with an oil mixture. Moeykens et al. (1996)

reported that oil considerably reduced the performance of an enhanced bundle with spray evaporation (polyester oil with R-113).

2.8. Active Enhancement Techniques

Active enhancement techniques for pool boiling include heated-surface rotation, surface wiping, surface vibration, fluid vibration, electrostatic fields, and suction at the heated surface. They are reviewed by Ohadi et al. (1996). Because of its greater potential for practical application, electrohydrodynamic-enhanced boiling has attracted considerable attention from academic and industrial researchers. Pool boiling enhancements of heat transfer coefficients up to 900% have been reported for refrigerants. Although active techniques are effective in reducing the wall superheat and/or increasing the critical heat flux, the practical applications may be restricted, largely because of the necessity to develop reliable, low-cost transducers or power supplies and associated enhanced heat exchangers.

2.9. Compound Techniques

Recall that compound enhancement involves two (or more) of the techniques applied simultaneously to produce an enhancement that is larger than the individual techniques applied separately. For pool boiling, this has been demonstrated with :

- fins and electric fields,
- electric fields in a bundle of treated tubes,
- radially grooved rotating disk,
- extended surfaces that are treated,
- and rough surfaces with additives.

3.0 ENHANCEMENT OF FORCED-CONVECTION BOILING AND EVAPORATION

3.1. Subcooled Flow Boiling

A variety of surfaces and devices for passive enhancement of forced-convection boiling and evaporation are depicted in Fig. 9. The following discussion will refer to these sketches.

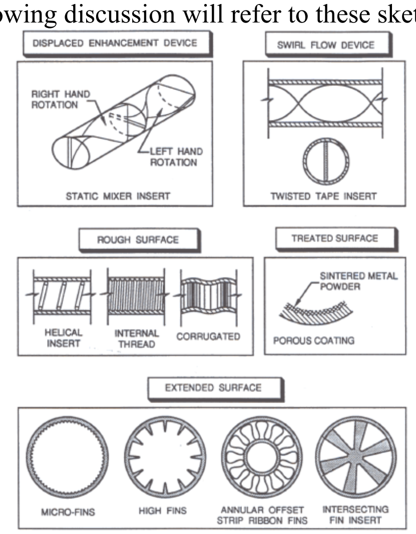

Fig. 9. Examples of passive enhancement techniques for forced-convection boiling and evaporation, Schlager et al. (1989).

Subcooled boiling is of interest when cooling high-power devices. The object is to accommodate high heat fluxes with moderate pressure drop penalty, *not* to generate vapor for use in an energy-conversion device (e.g., Rankine cycle power plant) or for absorption of a high heat load (e.g., vapor-compression-cycle air conditioner). The fixed heat flux boundary condition invariably is imposed. This applies to pressurized water nuclear-fission reactors, nuclear-fusion reactors, and electrical and electronic devices (e.g., high-field electromagnets, liquid-cooled radar tubes, and chips).

Although enhancement of subcooled boiling has been sought using almost all of the methods shown in Fig. 9, the twisted tape is the simplest and most effective. The twisted-tape insert was recently reviewed in much detail by Manglik and Bergles (2002). Twisted tapes are very effective in elevating the critical heat flux in subcooled boiling, or accommodating very high heat fluxes The swirl-induced radial pressure gradient promotes vapor removal from the heated surface, thereby permitting higher heat fluxes before vapor blanketing occurs. To demonstrate this, it is instructive to look at early data of Gambill et al. (1961), shown in Fig. 10. It is seen that the maximum critical (burnout) heat flux is about 1.18 x 10^8 W/m^2, which tends toward the high end of Fig. 1. More recently, Tong et al. (1996) ran experiments, also with water, in tubes with inside diameters varying from 2.44 – 6.54 mm and twist ratios from 1.9 to ∞ (straight tape). The maximum heat flux obtained was 1.44 x 10^8 W/m^2. For gently twisting tapes, CHF values with tape inserts were lower than those for empty tubes. This was due to the thermal-insulating effect between the tightly fitted tape and the wall of the electrically heated tube.

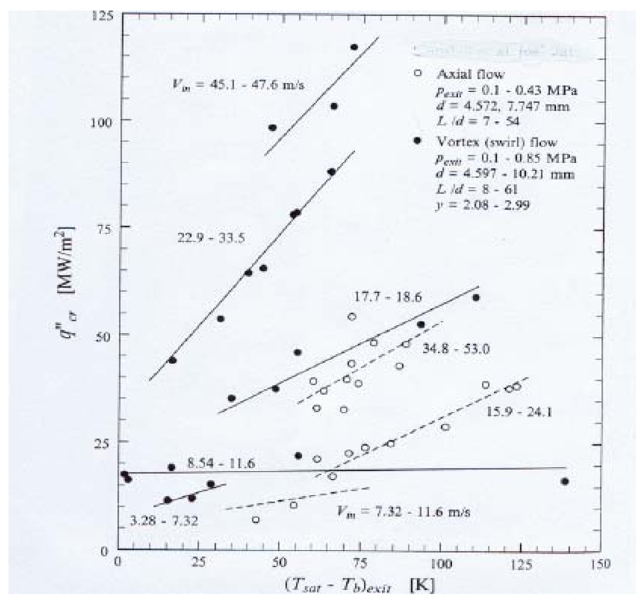

Fig. 10. Influence of twisted-tape-generated swirl flow on subcooled-boiling critical heat flux with water.

Under certain circumstances, subcooled CHF to water can be improved with a volatile additive, according to and Bergles and Scarola (1966). For instance, the addition of about 2% 1-pentanol to water elevated the CHF in plain tubes at high subcooling by as much as 15%. However, the opposite trend was observed at low subcooling. Note that this is in general agreement with the results of Ornatskii and Vinyarskii (1965) mentioned in Table 1.

None of the active techniques are especially effective in subcooled boiling. The reason is that subcooled boiling is a very efficient heat transfer process, and a very large disturbance is required to alter it.

Finally, one of the few compound techniques that has been used with subcooled flow boiling is water-alcohol plus the twisted tape. Pabisz and Bergles (1996), contrary to expectations using the "magic potion" of 2% 1-pentanol in water, found no significant enhancement with the mixture.

3.2. Vaporization

The devices and surfaces depicted in Fig. 9 have also been tested for passive enhancement of in-tube, forced-convection vaporization. The structured surfaces in Fig. 3 are generally not used for in-tube vaporization, due to the difficulty of manufacture. One notable exception is the porous metallic matrix surface in a vertical thermosyphon reboiler, because the coating can be applied to the inside of larger diameter tubes. Thome (1990) discusses the application of these tubes, as a retrofit, indicating that the overall heat transfer coefficient was increased at least 80 percent.

The behavior of vertical tubes in bundles can be studied with single tubes. The ID enhancement can be very effective, because reboilers usually are driven by condensing steam, and the in-tube fluid is typically a hydrocarbon with relatively low heat transfer coefficient. To get the circulation going, it is often necessary to sparge the flow. This artificial introduction of vapor is equivalent to boiling at very low superheat, which is possible with enhanced tubes.

Helical repeated ribs and helically coiled wire inserts have been used to increase vaporization coefficients and dryout heat flux in once-through boilers. Numerous tubes with internal fins, either integral or attached, are available for refrigerant evaporators. Original configurations were tightly packed, copper, offset strip fins inserts soldered to a copper tube or aluminum, star-shaped inserts secured by drawing the tube over the insert. Examples are shown in Fig. 9. Average heat transfer coefficients (based on the surface area of smooth tube of the same diameter) for typical evaporator conditions are increased by as much as 200%, as shown in Fig. 11.

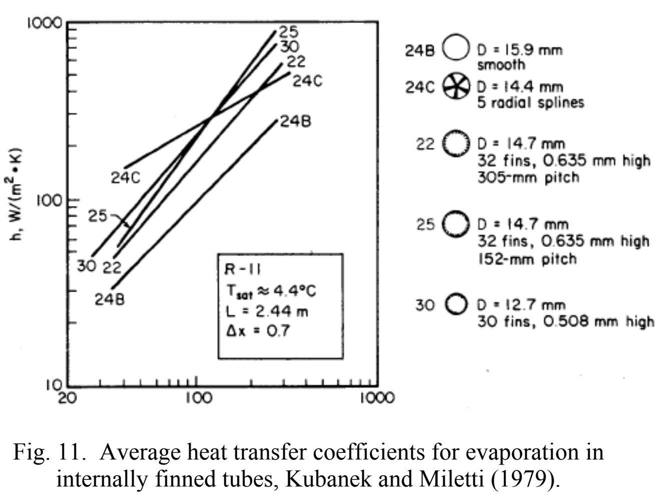

Fig. 11. Average heat transfer coefficients for evaporation in internally finned tubes, Kubanek and Miletti (1979).

For reasons of high cost and pressure drop, the composite tubes and high fin tubes of Fig. 9, and even the medium fins of Fig. 11, have been largely replaced with micro-fins. For the popular 9.5-mm-OD tubes, they involve numerous (~70), low (~0.15 mm), spiraling (~18 deg.) fins inside tubes. They are usually produced inside circular tubes (commonly copper) by swaging. They are widely used for convective vaporization of refrigerants, and improve heat transfer coefficients by about 50-100%.

The one variant in commercial micro-fin tubes is the fin profile. Khanpara et al. (1986) reported a study of the influence of fin profile (Fig. 12) on heat transfer (Fig. 13) and pressure drop (Fig. 14) for evaporating R-22. Regarding geometry, the higher fins perform better, flat valleys are preferred, and flat peaks are better. Tube 10 has the best overall heat transfer performance, but it also has the highest pressure drop. However, it is noted from Fig. 14 that all microfin tubes have a modest pressure drop, so in terms of heat transfer/pressure drop performance, Tube 10 is the best. Single-grooved tubes are generally used, but cross-grooved tubes perform better. Cross-grooves can be produced by enhancing the pattern on a strip, rolling up the strip, and welding it. The application of standard micro-fin tubing to chemical processing is given by Thome (1996).

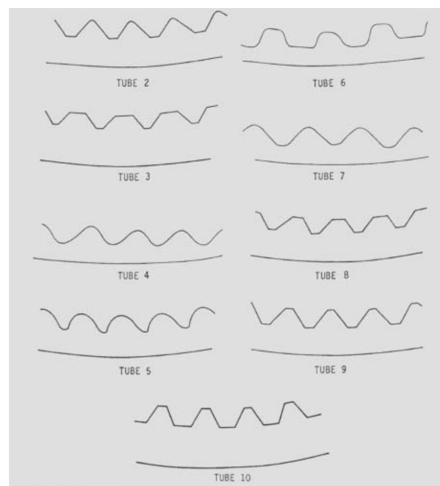

Fig. 12. Fin profiles for micro-fin tubes.

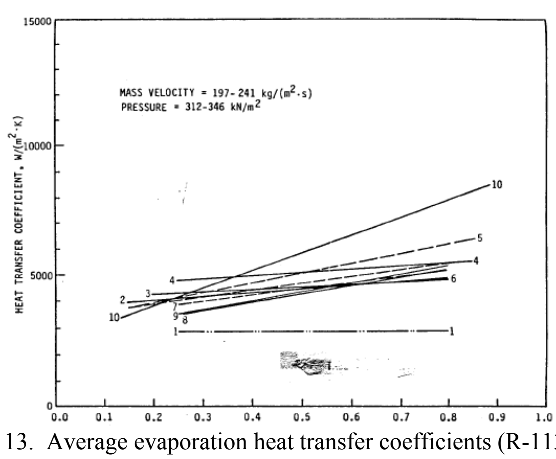

Fig. 13. Average evaporation heat transfer coefficients (R-113) for micro-fin tubes and a plain tube.

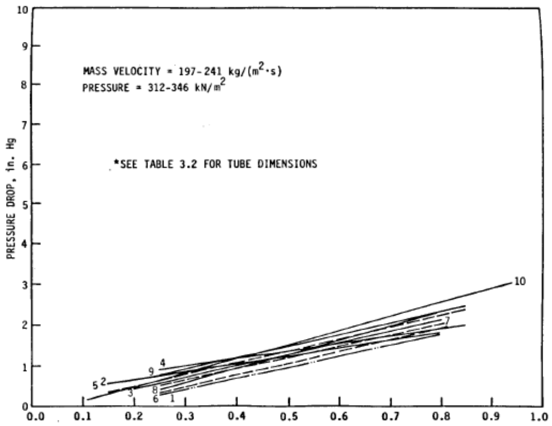

Fig. 14. Evaporation pressure drops (R-113) for micro-fin tubes and a plain tube.

The usual contaminant in refrigeration systems is compressor lubricating oil, which may be present in concentrations up to 10%. Experiments by Schlager et al. (1988) have defined the effects of oil on the performance of a typical micro-fin tube. It is of interest to compare the average heat transfer coefficients (expressed as a ratio to the oil-free performance) with smooth tube and low-fin tube behavior. In Fig. 15 it is seen that oil generally degrades the performance of the low-fin tube while a small amount of oil slightly enhances the micro-fin tube performance at about 2% concentration. In contrast, the plain tube heat transfer coefficient is enhanced by oil throughout the test range, with a particularly sharp enhancement occurring at 3% concentration. While the inner-fin tubes always enhance heat transfer relative to the smooth tube, it is evident that the absolute performance and the enhancement are strongly dependent on the oil concentration.

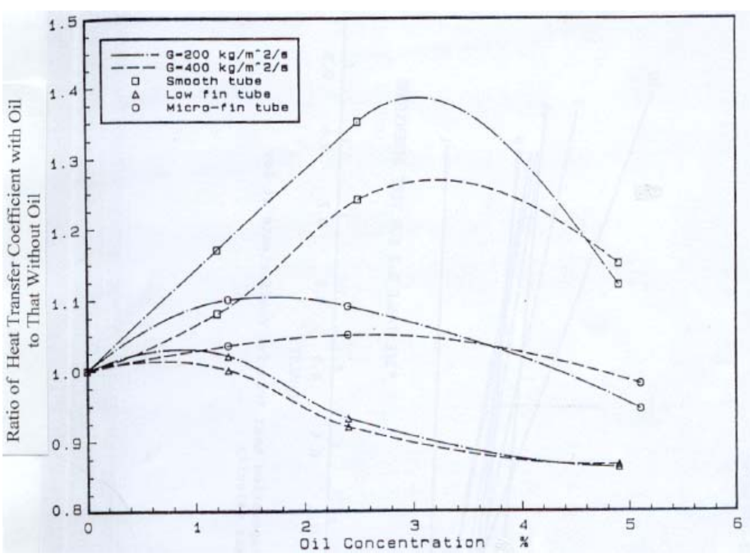

Fig. 15. Influence of oil on average evaporation heat transfer coefficients for plain and internally finned tubes.

The effect of a twisted-tape insert in forced convective boiling is depicted in Fig. 16. Here, the entire length of a once-through boiler tube is considered; the preheating zone, X < 0, involves enhancement of single-phase flow and subcooled boiling. Here we are concerned mainly with vaporization, or X > 0. The general characteristics of vaporization enhancement are discussed by Shatto and Peterson (1996) and Manglik and Bergles (2002).

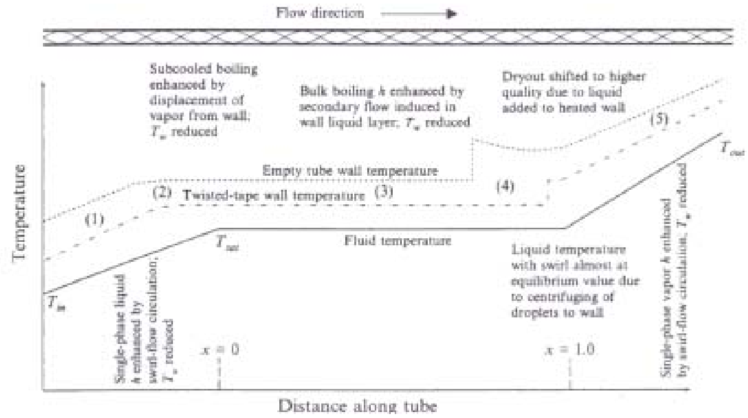

Fig. 16. The influence of twisted-tape-induced swirl on the evolution of fluid-bulk and tube-wall temperatures along the tube length in forced-convection vaporization, Manglik and Bergles (2002).

Twisted tapes are very effective in elevating the critical heat flux in boiling with net vapor generation. The swirl-induced radial pressure gradient promotes vapor removal from the heated surface, and directs liquid to the heated surface, thereby permitting higher heat fluxes before dryout of the surface occurs. Data are reviewed by Shatto and Peterson (1996) and Manglik and Bergles (2002). Of particular note, is an extensive study by Kedzierski and Kim (1997) that shows the effect of a twisted-tape insert on a wide range of vaporizing refrigerants.

Displaced parameters are located away from the heated surface and direct the flow toward the heated surface. One prominent application is to elevate the power level in boiling water nuclear reactors. Although considerable work has been done with fuel-rod bundle liners, special clips are used that separate the rods and typically swirl the flow. Liquid in the core of the subchannels is thereby brought to the rod surfaces, and the CHF is increased. The burnup rate (average heat flux) can then be increased while preserving the same safety margin. This has been an important factor in U.S. nuclear technology; with no new plants being built, the main business has been in reload fuel – preferably fuel that permits operation at a higher power level. This is illustrated in Fig. 17.

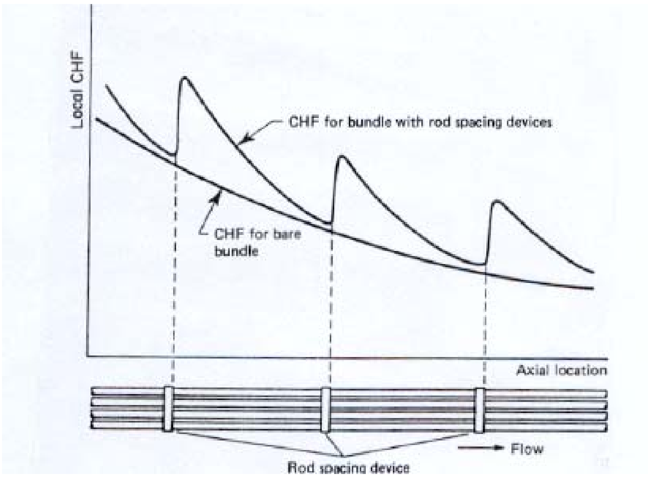

Fig. 17. Effect of rod-spacing devices on CHF, Groeneveld and Yousef (1980).

Some active enhancement techniques have been applied to flow boiling, but the results have not been promising. For instance, neither surface vibration nor fluid vibration (pulsations to ultrasound) have any effect on developed boiling or CHF. The same result has been observed with injection and electrostatic fields. This suggests that active techniques should be reserved for pool boiling, or possibly low velocity flow boiling, where the convective conditions still influence the boiling curve. A sophisticated analysis of horizontal, annular two-phase flow is given by Feng and Seyed-Yagoobi (2002). This is applicable to vaporization, and both enhancement and suppression of heat transfer are addressed.

There are some examples of compound enhancement in flow boiling:

- rough surfaces and additives,
- electrostatic fields with micro-fin tubes

4.0 CONCLUDING REMARKS

4.1. Advanced Enhancement

Heat Transfer enhancement or **second generation heat transfer technology** is used in many industrial applications. What we want now is advanced enhancement or **third generation heat transfer technology.** Some examples are shown in Table 3, where the generation of enhancement is one less than the generation of heat transfer technology. We do not find a chronological relationship among levels of enhancement. For example, the 3rd generation pool-boiling surface employing metallic coatings was patented in 1968.

TABLE 3. THE GENERATIONS OF HEAT TRANSFER TECHNOLOGY

Outside tubes, boiling	
1st generation	smooth tube
2nd generation	2-D fins
3rd generation	3-D fins or metallic coatings
In-tube evaporation	
1st generation	smooth tube
2nd generation	massive fins or inserts
3rd generation	micro-fins

4.2. Surprises

Effects contrary to expectations are sometimes observed when applying enhancement technology. This contributes to the excitement of enhancement. The issue of fouling of enhanced surfaces in pool boiling, and how enhancement performance may actually be better under fouling conditions, was mentioned above. Also mentioned was the reduction of twisted-tape CHF in flow boiling with tightly fitted, loosely twisted tapes, and no enhancement in CHF with twisted tapes and water - with an additive. Other examples are: reduction of critical heat under certain conditions with flow boiling in a tube having a porous coating, and reduction in critical heat flux with twisted-tape inserts in saturated boiling.

4.3. Final Points

This review has given an overview of enhanced heat transfer technology, citing representative developments. About ten percent of the heat transfer literature now concerns enhancement, and a recent year of the *Journal of Heat Transfer* has over 20 percent of the papers directed to various areas of enhancement. The proceedings of an entire conference on this subject were reported recently by Kakaç et al. (1999).

An enormous amount of information is available; what is needed is technology transfer. Many techniques, and variations thereof, have made the transition from the academic, or industrial-research, laboratory to industrial practice. This transition of enhancement technologies must be accelerated; however, the literature should be consulted to take advantage of previous experience. The older literature contains much good data that should be examined before starting expensive physical or numerical experiments. To facilitate this, the bibliographic survey should be updated.

The interest in enhanced heat transfer is closely tied to energy prices. In the 1980s and 1990s, energy prices were low due to oil and natural gas "bubbles". Recently, however, increased demand and inadequate supply or distribution have resulted in large increases in the price of energy. There is now an incentive to save energy, and enhanced heat transfer can be exploited to do so. Whereas in the previous two decades, enhancement was employed to reduce the size of equipment, thereby saving space, it is now applied to save energy costs. It is possible that because of this, the field of enhanced heat transfer will experience another growth phase (see Fig. 2).

When enhancement is applied, manufacturing methods and materials requirements may be overriding considerations. Can the enhancement be produced in a material that will survive any corrosion inherent in the working fluid or environment? Much work needs to be done to define the fouling/corrosion characteristics of enhanced surfaces. Particularly, "antifouling" surfaces need to be developed.

It should be noted that enhancement technology is still largely experimental, although great strides are being made in analytical/numerical descriptions of the various techniques. Accordingly, it is imperative that the craft of experimentation be kept viable. With the wholesale rush to "technology," physical laboratories everywhere are being decommissioned. Hands-on experiences in universities are being decreased or replaced by computer skills and virtual laboratories. Experimentation is still a vital art needed for direct resolution of transport phenomenon in complex enhanced geometries, as well as bench-marking of computer codes. As such, experimental skills should continue to be taught, and conventional laboratories should be maintained.

NOMENCLATURE

D, d	internal diameter of test section, m
D_o	outside diameter of test section, m
G	mass flux, kg/m^{2}s
h	heat transfer coefficient, W/m^{2}K
L	length of test section, m
P	pressure, Pa
P_{exit}	pressure at exit of test section, Pa
q"	heat flux, W/m^2
q_{cr}"	critical heat flux, W/m^2
T_{in}	liquid inlet temperature, K
T_{out}	liquid outlet temperature, K
T_s	liquid saturation temperature, K
T_b,ΔT_s,ΔT_{sat}	boiling superheat, K
ΔT_{sub}, T_{sat}-T_b	liquid subcooling, K
V_{in}	lnlet liquid velocity
x	quality
Δx	quality change
y	tube diameters for 180˚ rotation of twisted tape

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This paper was written with a feeling of considerable nostalgia, as it is one of the last reviews of enhanced heat transfer that I will undertake. After forty years, it is time to put away the collection of enhanced surfaces, and focus on more mundane matters, such as keeping the house heating, plumbing, etc. operational. It has been a very satisfying professional journey, and I have hundreds of colleagues and acquaintances to thank for making it all possible. I am particularly grateful to Prof. Warren Rohsenow and the MIT Heat Transfer Laboratory for the opportunity to undertake the study and review of enhanced heat transfer in 1963. Accordingly, I must emphasize the point made above about reviewing the older literature. As Santyana said, "Those who cannot remember the past are condemned to repeat it."

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