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Energy conservation through heat transfer enhancement techniques

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SUMMARY

Enhanced heat transfer surfaces have been successfully used in the heat transfer industry to obtain more compact and efficient units. Conservation of the useful part of energy (exergy) can be reached through higher heat transfer than that for standard surfaces and flow configurations. There are numerous ways for heat transfer augmentation which have been marketed or tested in laboratories. In this paper, we review some of the passive techniques for single-phase flow. One of the most promising heat transfer enhancement techniques is the compound augmentation method, in which different enhancement techniques are used simultaneously. Extended performance evaluation criteria (PEC) equations for enhanced heat transfer surfaces based on the entropy production theorem are developed to include the effect of fluid temperature variation along the length of a tubular heat exchanger and assess two objectives simultaneously. The need for this more comprehensive treatment of PEC compared to previous references is illustrated by the analysis of heat transfer and friction characteristics of ten spirally corrugated tubes. These PEC are used to assess the multiplicative effect when a corrugated tube is combined with twisted tape insert. Copyright © 2002 John Wiley & Sons, Ltd.

KEY WORDS: enhancement (augmentation) of heat transfer; conservation of useful energy (exergy); single-phase flow; performance evaluation criteria; entropy generation minimization

1. INTRODUCTION

The 'energy conservation' opportunity recognized by all of us is in fact an opportunity for conserving 'useful part of energy' or exergy (available work). The thermal engineers have recognized for a long time the opportunity for conserving energy through heat transfer enhancement techniques.

The heat exchanger is an essential unit in heat extraction and recovery systems. The heat exchangers are devices that provide heat transfer between two fluids at different temperatures without their physical contact. They include power production process, chemical and food

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industries, electronics, environmental engineering, waste heat recovery, manufacturing industry, air-conditioning, refrigeration and space applications. In these applications, thermal-hydraulics and energy usage play dominant roles. Energy and materials savings considerations, as well as economic incentives, have led to efforts to produce more efficient heat exchange equipment.

On the other hand, it is well established that the minimization of the entropy generation in any process leads to the conservation of useful part of energy. In a heat exchanger unit, entropy is generated by the heat transferred due to finite temperature difference and by the irreversible dissipation of kinetic energy due to fluid friction. Heat transfer enhancement devices increase the rate of heat transfer, but they also increase the friction factor associated with the flow. This raises the question of how to employ enhancement techniques in order to minimize the overall entropy generation associated with the heat exchanger operation.

The irreversibility reduction associated with thermal design changes is generally recognized under the name 'techniques for the enhancement (augmentation) of heat transfer'. The traditional view is that the objective of all heat transfer techniques is to enhance the thermal performance by increasing the surface heat transfer coefficient that characterizes the standard (untouched) heat transfer surface. A parallel objective of the technique is to improve the thermal contact without introducing a significant increase in the fluid pumping power required for the heat exchange job. These parallel objectives outline the fundamental conflict that governs the application of any augmentation technique: a design modification that improves the thermal contact will, most likely, cause a parallel increase in the mechanical power dissipated in the heat exchange apparatus. Given this conflict, it is useful to know in advance which augmentation technique will lead to an improvement in thermodynamic performance (Bejan, 1982, 1996).

The performance of conventional heat exchangers can be substantially improved by many augmentation techniques applied in the design systems. A classification of enhancement techniques is presented in Table I (Webb and Bergles, 1983a). Heat transfer enhancement devices are commonly employed to improve the performance of an existing heat exchanger or to reduce the

Table I. Classification of enhancement techniques.

Passive techniques
Treated surfaces
Rough surfaces
Extended surfaces
Displaced enhancement devices
Swirl flow devices
Coiled tubes
Surface tension devices
Additives for fluids
Active techniques
Mechanical aids
Surface vibration
Fluid vibration
Electrostatic fields
Suction or injection
Jet impingement
Compound technique
Rough surface with twisted-tape insert, for example

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size and cost of a proposed heat exchanger. An alternative goal is to use such techniques to increase the system thermodynamic efficiency, which allows to reduce the operating cost. A description and representative developments of these techniques can be found in Bergles (1985, 1988, 1997).

In general, to improve the heat transfer performance means to increase the heat transfer coefficient. Attempts to increase 'normal' heat transfer coefficients have been recorded for more than a century, and there is a large store of information. A recent survey (Bergles *et al*., 1995) cites 5676 technical publications, excluding patents and manufacturers' literature.

Enhancement techniques can basically be classified as the passive methods, which require no direct application of external power, and the active methods, which require external power. Two or more of the above techniques may be utilized simultaneously to produce an enhancement that is larger than either of the techniques operating separately. This is termed compound enhancement.

It should be emphasized that one of the motivation for studying enhanced heat transfer is to assess the effect of an inherent condition on heat transfer. The purpose of this paper is to review some passive augmentation techniques mostly for single-phase forced convection in tubes and the development of the evaluation criteria to evaluate the merit of the conservation of useful part of energy.

2. SINGLE-PHASE FORCED CONVECTION ENHANCEMENT

The enhancement of single-phase flow is important because that flow usually represents the dominant thermal resistance in a two-fluid heat exchanger, especially if it is a gas (as compared to a liquid). Balaras (1990) selects to group the augmentation methods for single-phase heat exchangers into three categories, based on whether the enhancement is caused by either the heat exchanger surface (surface methods), or the working fluid (fluid methods), or a combination of the two (combined methods).

Surface methods include any techniques which directly involve the heat exchanger surface. They are used on the side of the surface that comes into contact with a fluid of low heat transfer coefficient in order to reduce the thickness of the boundary layer and to introduce better fluid mixing. The primary mechanisms for thinning the boundary layer are increased free stream velocity and turbulent mixing. Secondary recirculation flows can further enhance the convective heat transfer. Flows from the core to the wall reduce the thickness of the boundary layer in the wash of the flow, and secondary flows from the wall to the core promote mixing. Flow separation and reattachment within the flow channel also contribute to heat transfer enhancement.

The present discussion emphasizes on some of the surface methods, namely roughened surfaces, swirl flow devices, and a combination of a roughened surface with a twisted tape insert as a compound augmentation technique.

An enormous variety of surface roughness has been tested in both single- and two-phase flows under both natural and forced circulation. The roughness may be applied to any of the usual prime or extended heat exchanger surfaces, e.g. tubes, plates or fins. Both two- and three-dimensional integral roughness elements have been produced by the traditional processes of machining, forming, or welding. Various inserts or wrap-around structures can also provide surface protuberances. Considerable research has been done on tube-side enhancement for forced convection. Sketches of typical surface configuration are shown in Figure 1 (Webb, 1982).

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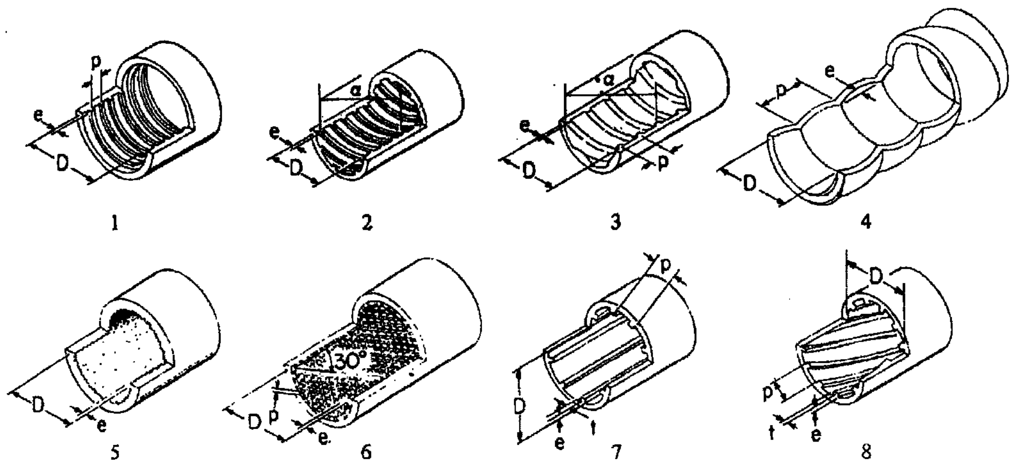

Figure 1. Sketches of typical surface configuration (Webb, 1982) (1—transverse-rib roughness, 2—helical-rib roughness, 3—'Turbochil' type roughness, 4—corrugated roughness, 5—sand-grain roughness, 6—three-dimensional roughness, 7—axial internal finned tube, 8—helical internal finned tube).

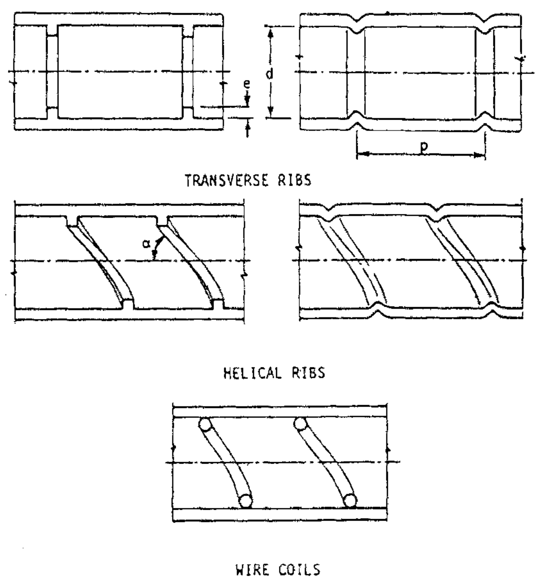

Figure 2. Examples of repeated rib-roughness (Ravigururajan and Bergles, 1985).

Attention is focused here on the helical repeated rib roughness that is readily manufactured and results in good heat transfer performance (up to 300 per cent increases over the smooth tube) (Figure 2) (Ravigururajan and Bergles, 1985).

In general, the maximum enhancement of laminar flow with many of the techniques is of the same order of magnitude and seems to be independent of the wall boundary condition. The enhancement with some rough tubes, corrugated tubes, inner-fin tubes, various static mixers and twisted-tape inserts is about 200 per cent. The improvements in heat transfer coefficient with turbulent flow in rough tubes (based on nominal surface area) are as much as 250 per cent. Correlations for heat transfer and friction coefficients for tubes with transverse or helical repeated ribs, with turbulent flow, are recommended (Ravigururajan and Bergles, 1985). All promoters produce a sizeable elevation in the Nusselt number or heat transfer coefficient but there is an accompanying large increase in the friction factor.

Many proprietary surface configurations have been produced by deforming the basic tube. The 'convoluted', 'corrugated', 'spirally fluted' configurations have multiple-start corrugations, which add area along the tube length. A systematic survey of the single-tube performance of condenser tubes indicates up to 400 per cent increase in the nominal inside heat transfer coefficient (based on the diameter of a smooth tube of the same maximum inside diameter); however, pressure drops on the water side are about 20 times higher.

Displaced enhancement devices are typically in the form of inserts, with the elements arranged to promote transverse mixing. Twisted-tape inserts have been widely used to improve the heat transfer in both laminar and turbulent flows. Laminar and turbulent flow correlations are available in literature (Manglik and Bergles, 1993a, b).

3. COMPOUND HEAT TRANSFER ENHANCEMENT IN SINGLE-PHASE FLOW

It is well known that two or more of the existing techniques can be utilized simultaneously to produce an enhancement larger than that produced by only one technique. The combination of different techniques acting simultaneously is known as compound augmentation. This is an emerging area of interest and holds promise for practical applications. Interactions between different augmentation methods contribute to greater values of the heat transfer coefficients compared to the sum of the corresponding values for the individual techniques used alone. Preliminary studies on compound passive augmentation techniques are encouraging. Some examples of passive compound techniques are: (i) rough tube wall with a twisted-tape insert, (ii) internally finned tubes with a twisted tape insert, (iii) rib-roughened channel with longitudinal vortex generation. The achievements in conserving useful part of energy through compound technique (i) (rough tube with a twisted-tape insert) are discussed below.

Preliminary studies on compound passive augmentation technique of this kind are very encouraging. Some examples are: rough tube wall with a twisted tape (Bergles *et al.*, 1969) and grooved rough tube with a twisted tape (Usui *et al.*, 1984, 1986). The latter reports a particularly high increase in the heat transfer efficiency when an internally grooved rough tube was combined with a tape twisted contrary to the direction of rotation of the grooves.

Many previous surveys indicate that the corrugated (roped) tubes are among the most effective and practical methods for augmenting single-phase heat transfer in tubes. It is reasonable to assume that a combination of a corrugated tube with a twisted tape would be superior to either combination of passive surface techniques. The recent articles (Zimparov, 2000a, b) report on an experimental investigation to see whether or not heat transfer can be enhanced by the multiplicative effect of a corrugated tube combined with a twisted tape. Some of the results representing the variations of the friction and heat transfer coefficients versus Reynolds number are shown in

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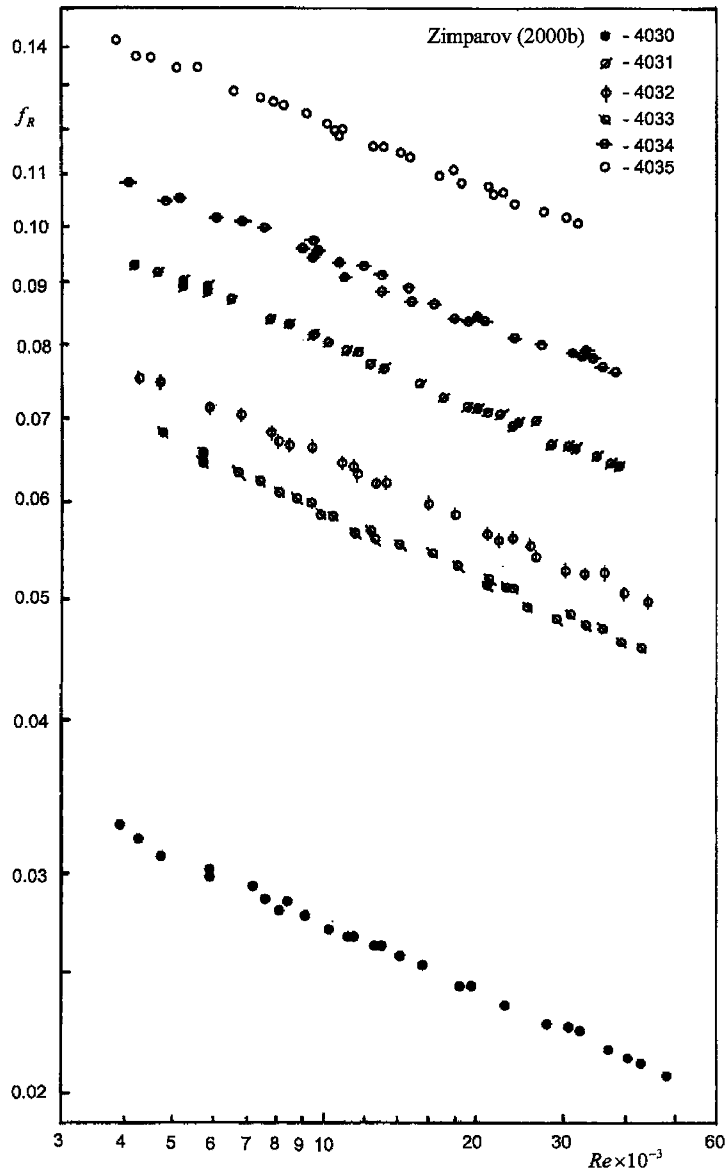

Figure 3. Variation of friction factor versus Reynolds number (Zimparov, 2000b).

Figures 3 and 4. As can be seen, the water-side heat transfer coefficients are particularly high when the height-to-diameter ratio e/D_i increases and the relative pitch H/D_i decreases. This study indicates up to 800 per cent increase in the nominal inside heat transfer coefficient for the tube 4035 ($e/D_i = 0.0569$ and $H/D_i = 4.7$) and Re $\approx 2.3 \times 10^4$; for the same Re, pressure drop on the water side is about 16.5 times higher (Zimparov, 2000b).

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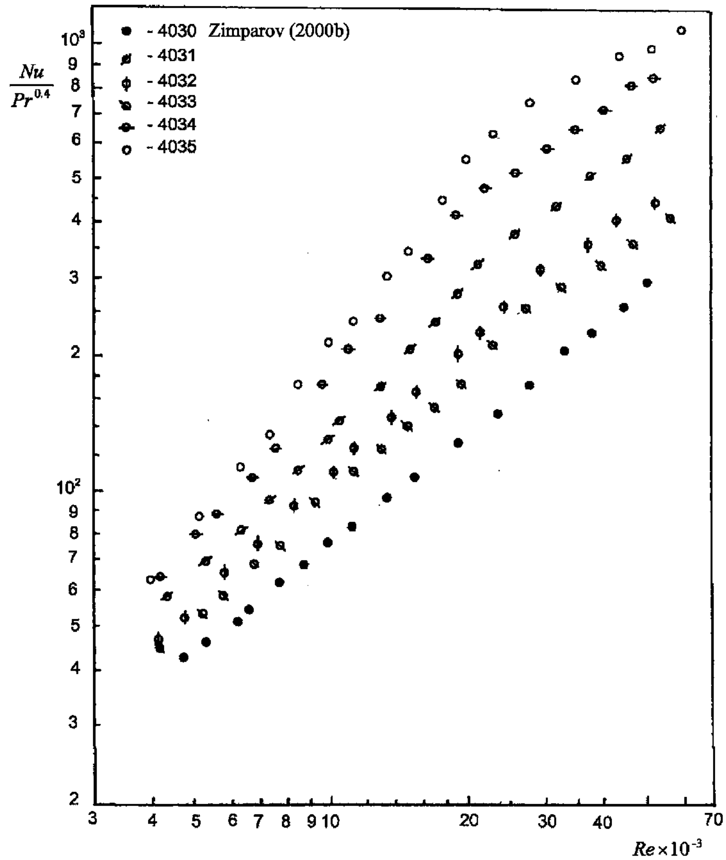

Figure 4. Water-side heat transfer coefficient versus Reynolds number (Zimparov, 2000b).

4. CRITERIA FOR PERFORMANCE EVALUATION

The usual overall heat transfer equation in rate form for an exchanger is given by

$$\dot{Q} = UA\Delta T = \frac{hA\Delta T}{hAR_{\text{ext}} + 1} \tag{1}$$

where R_{ext} includes scale, tube wall and secondary fluid resistances. It is evident that augmentation of one exchanger surface will have the maximum effect when that surface controls the overall heat transfer, i.e. $R_{\text{ext}} \ll 1/hA$. In other words, enhancement should be directed to the side with the low heat transfer coefficient.

The ratio of (hA) of an augmented surface to that of a plain surface is defined as the enhancement ratio, which (Webb, 1994) is

$$E = \frac{(hA)_{\text{R}}}{(hA)_{\text{S}}}. \tag{2}$$

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There are several methods to increase the $(hA)_R$ value:

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

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

$$\dot{Q} = \eta h A (T_w - T_f) \tag{3}$$

where

$$\eta = \left[1 - \frac{A_F}{A}(1 - \eta_F)\right] \tag{4}$$

is called overall surface efficiency. The total surface area on one side is $A = A_U + A_F$.

The inverse of the term UA given in Equation (1) is called the overall thermal resistance which can be expressed (Kakac and Liu, 1997) as

$$R_t = \frac{1}{UA} = \frac{1}{U_o A_o} = \frac{1}{\eta_i h_i A_i} + \frac{R_{fi}}{\eta_i A_i} + R_w + \frac{R_{fo}}{\eta_o A_o} + \frac{1}{\eta_o h_o A_o} \tag{5}$$

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

Today, a number of special surface geometries are commercially available. The designer faces a question, 'Which surface is the best?'. The preferred choice will depend on a number of factors: performance, cost, fouling characteristics or cleanability, and availability in the needed material. The designer may feel frustrated, since much of this information does not appear to be readily available. In this review we attempt to identify the preferred enhancement geometry only for single-phase forced convection inside tubes on the basis of its performance. The evaluation of the performance will be made on the basis of the first and the second law analysis.

On the basis of the first law analysis, several authors (Webb and Eckert, 1972; Bergles *et al*., 1974a, b; Webb, 1981; Webb and Bergles, 1983b) have proposed performance evaluation criteria (PEC) which define the performance benefits of a heat exchanger having enhanced surfaces, relative to a standard heat exchanger subject to various design constraints. The first systematic studies, developed for single-phase flows, which present several alternative ratios, appear to be those of Webb and Eckert (1972) and Bergles *et al*. (1974a, b). Further evolution of the number of PEC was made by Webb (1981) and Webb and Bergles (1983b).

The performance of the heat exchanger will be increased if the term (UA) is increased or the overall thermal resistance is reduced. There are three basic design objectives (Webb, 1981):

(1) To reduce the heat transfer surface material for equal pumping power and heat rate —$A_* = A_R/A_S < 1$ with $\dot{Q}_* = \dot{Q}_R/\dot{Q}_S = 1$ and $P_* = P_R/P_S = 1$.
(2) To increase UA for equal pumping power, $P_* = 1$, and fixed total length of exchanger tubing, $N_* L_* = N_R L_R / N_S L_S = 1$. A higher UA may be exploited in one of the two ways:

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(a) to obtain increased heat rate, $\dot{Q}_* > 1$, for fixed entering fluid temperatures, $\Delta T_i^* = \Delta T_{i,R}/\Delta T_{i,S} = 1$;
(b) to reduce LMTD for fixed heat rate, $\dot{Q}_* = 1$.

(3) To reduce pumping power, $P_* < 1$, for equal heat rate, $\dot{Q}_* = 1$, and total length of the exchanger tubing, $N_* L_* = 1$.

Objectives (1) and (2a) allow a smaller heat exchanger size, and hopefully, reduced capital cost. Objectives (2b) and (3) offer reduced operating cost. The reduced LMTD of (2a) will affect improved system operating cost.

The major operational variables include the heat transfer rate, pumping power (or pressure drop), heat exchanger flow rate and the fluid velocity, which affects the exchanger flow frontal area. A PEC based on the first law are established by selecting one of the operational variables for the performance objective, subject to the design constraints on the remaining variables. The recommended PEC for the several cases of interest are presented in Webb (1981) and Webb and Bergles (1983b) where they are segregated by three different geometrical constraints: FG—fixed geometry, FN—fixed flow area and VG—variable geometry. Within each of the geometry constraint groupings, PEC are established for three performance objectives: (1) reduced surface area $A_* < 1$; (2) increased heat rate, $\dot{Q}_* > 1$, (or $U_R A_R / U_S A_S > 1$) and (3) reduced pumping power, $P_* < 1$.

Quantitative formulation of a PEC requires algebraic relations which: (1) quantify the objective function and (2) define the heat transfer and friction characteristics relative to the reference exchanger. The PEC must account for several important factors:

(i) surface area based on either the nominal (smooth tube) or actual surface area. Typically, the nominal area is used for rough surfaces and the actual area is used for finned surfaces;
(ii) a tube side fouling resistance;
(iii) the thermal resistance across the metal tube wall;
(iv) the possibility of enhancement on the inner and the outer tube surface for a two-fluid heat exchanger.

The equations are developed for tubes of different diameters and heat transfer and friction factors based on the presentation format of performance data for enhanced tubes (Marner *et al.*, 1983). The relative equations for single-phase flow inside enhanced tubes are

$$A_* = N_* L_* D_* \tag{6}$$

$$P_* = W_* \Delta p_* = f_R/f_S \, D_* L_* N_* u_{m,*}^3 = \frac{W_*^3 L_*}{N_*^2 D_*^5} f_R/f_S \tag{7}$$

$$\dot{Q}_* = W_* \varepsilon_* \Delta T_i^* \tag{8}$$

$$W_* = u_{m,*} D_*^2 N_* = \frac{\mathrm{Re}_R}{\mathrm{Re}_S} D_* N_* \tag{9}$$

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$$\Delta p_* = f_R/f_S \frac{L_*}{D_*} u_{m,*}^2 = f_R/f_S \frac{L_*}{D_*^3} \frac{\mathrm{Re}_R^2}{\mathrm{Re}_S^2} \tag{10}$$

$$\frac{(UA)_{i,R}}{(UA)_{i,S}} = \frac{1 + \mathrm{R_S}}{\mathrm{St_S/St_R}\,(f_R/f_S P_*^{-1} A_*^{-2})^{1/3} + R_R A_*^{-1}} \tag{11}$$

where R_S and R_R are the different thermal resistances on both sides of the tube wall. The effect of the thermal resistance external to the surface is also taken into consideration by including the external heat transfer coefficient $h_{o,S}$. The analysis includes the possibility that the augmented exchanger may have an enhanced outer tube surface $E_o = h_{o,R}/h_{o,S}$.

A solid thermodynamic basis to evaluate the merit of augmentation techniques by second-law analysis has been proposed by Bejan (1982, 1996) developing the entropy generation minimization (EGM) method also known as 'thermodynamic optimization'. The ultimate purpose is to evaluate the advantage of a given augmentation technique by comparing the rates of entropy generation in an augmented duct and in a reference smooth one. This method of optimizing was applied to the design of two augmentation techniques: rough surfaces (Bejan and Pfister, 1980) and swirl promoters (Ouelette and Bejan, 1980).

Further expansion of the number of PEC (based on the first-law analysis) is possible if the entropy generation as a variable is included, as suggested by Nelson and Bergles (1986). The analysis of Bejan and Pfister (1980), Ouelette and Bejan (1980) and Bejan (1982) is performed for constraints $W_* = 1$ and $\dot{Q}_* = 1$ which corresponds to the case FG-1b (Webb, 1981). Other publications on this subject are Perez-Blanco (1984) and Chen and Huang (1988).

The criteria proposed by Webb and Bergles (PEC), however, do not allow the assessment of two or more objectives simultaneously and do not take into consideration the entropy generation and one-way destruction of exergy. In many of their cases, Nelson and Bergles (1986) also assess one objective taking the rate of entropy production as a constraint. The method of Bejan (EGM) does not include the effect of variation in fluid temperature similar to that present in tubular heat exchangers.

Nag and Kumar (1989) and Nag and Mukherjee (1987) modified Bejan's entropy generation criterion by including the temperature variation along the heat transfer passage. Prasad and Shen (1993, 1994) have proposed an evaluation method based on exergy analysis. The thermodynamic optimum is obtained by minimizing the exergy destruction number (N_E). In addition, another criterion such as heat transfer improvement number (N_H) is introduced. These numbers permit a comparison of the effect of improved heat transfer with the increased irreversibility. PEC equations based on the first-law analysis and entropy production theorem have been developed by Zimparov and Vulchanov (1994). The latest extensions of these PEC equations including the effect of fluid temperature variation along the length of a tubular heat exchanger are presented by Zimparov (2000c, d). These PEC equations combine the PEC proposed by Webb (1981) with extended Bejan's EGM method, assessing two objectives simultaneously.

In the case of constant wall temperature (more appropriate for two-fluid heat exchangers) the entropy generated along the length of the duct is (Zimparov, 2000c)

$$\dot{S}_{\mathrm{gen}} = \frac{\dot{Q}}{N_t} \frac{\vartheta_o}{T_i T_o} + \frac{32 W^3 f L}{N_t^3 \rho^2 \pi^2 D^5 T_w} \tag{12}$$

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The first and the second terms on the right-hand side of Equation (12) represent the entropy generation due to heat transfer across finite temperature difference and due to friction, respectively. Following Bejan (1982, 1996), the thermodynamic impact of the augmentation technique is defined by the augmentation entropy generation number

$$N_S = \dot{S}_{\mathrm{gen,R}}/\dot{S}_{\mathrm{gen,S}} \tag{13}$$

Augmentation techniques with $N_\mathrm{S} < 1$ are thermodynamically advantageous since in addition to enhancing heat transfer they reduce the degree of irreversibility of the apparatus. The augmentation entropy generation number can be rewritten as (Bejan, 1982, 1996)

$$N_\mathrm{S} = \frac{N_T + \phi_0 N_P}{1 + \phi_0} \tag{14}$$

where (Zimparov, 2000c)

$$N_T = \frac{(\dot{S}_{\mathrm{gen},\Delta T})_\mathrm{R}}{(\dot{S}_{\mathrm{gen},\Delta T})_\mathrm{S}} = \frac{\dot{Q}_* \vartheta_{\mathrm{o,R}}}{N_* \vartheta_{\mathrm{o,S}}} \frac{T_{\mathrm{o,S}}}{T_{\mathrm{o,R}}} \tag{15}$$

$$\frac{T_{\mathrm{o,S}}}{T_{\mathrm{o,R}}} = \left[\frac{T_{\mathrm{i,S}}}{T_{\mathrm{o,S}}} + \frac{\dot{Q}_*}{W_*}\left(1 - \frac{T_{\mathrm{i,S}}}{T_{\mathrm{o,S}}}\right)\right]^{-1} \tag{16}$$

$$\frac{\vartheta_{\mathrm{o,R}}}{\vartheta_{\mathrm{o,S}}} = \exp\left[B\left(1 - \frac{\mathrm{St_R}}{\mathrm{St_S}} \frac{L_*}{D_*}\right)\right] \tag{17}$$

$$B = \frac{4\mathrm{St_S}}{D_\mathrm{S}} L_\mathrm{S} \tag{18}$$

$$N_P = \frac{(\dot{S}_{\mathrm{gen},\Delta P})_\mathrm{R}}{(\dot{S}_{\mathrm{gen},\Delta P})_\mathrm{S}} = \frac{W_*^3 L_*}{N_*^3 D_*^5} \frac{T_{\mathrm{w,S}}}{T_{\mathrm{w,R}}} f_\mathrm{R}/f_\mathrm{S} = P_* \frac{T_{\mathrm{w,S}}}{T_{\mathrm{w,R}}} \tag{19}$$

When the standard heat transfer passage is known, the numerical value of the irreversibility distribution ratio, $\phi_0 = (\dot{S}_{\mathrm{gen},\Delta P}/\dot{S}_{\mathrm{gen},\Delta T})_\mathrm{S}$, describes the thermodynamic mode in which the passage is meant to operate (Zimparov, 2000c)

$$\phi_0 = \frac{2f_\mathrm{S}L_\mathrm{S}}{D_\mathrm{S}} \left(\frac{u_\mathrm{m}^2}{c_\mathrm{p} T_\mathrm{w}}\right)_\mathrm{S} \frac{T_{\mathrm{i,S}} T_{\mathrm{o,S}}}{\vartheta_{\mathrm{o,S}} (T_\mathrm{o} - T_\mathrm{i})_\mathrm{S}} \tag{20}$$

The solution of the PEC equations requires algebraic relations which

(i) Define correlations for St(Nu) and f of the augmented surfaces as a function of Re.
(ii) Quantify performance objectives and design constraints. This means that the designer should define clearly his or her goal and then solve the equations corresponding to the algebraic relations based on the first law of thermodynamics, to obtain the values of

improved heat transfer rate $\dot{Q}_* = \dot{Q}_R/\dot{Q}_S$, reduced heat transfer area $A_* = A_R/A_S$, or reduced pumping power $P_* = P_R/P_S$ as a function of Re.

(iii) Calculate the irreversibility distribution ratio ϕ_0 as a function of Re for the reference (smooth) passage.

The results of the development of PEC can be illustrated by the characteristics of spirally corrugated (roped) tubes for steam condensers obtained through several experimental programs (Anon., 1982; Withers, 1980; Zimparov *et al*., 1991). The studies of Ravigururajan and Bergles (1986, 1995) based on the first law analysis indicated that the optimum rib height-to-diameter ratio (e/D_i) for spirally indented tubes is around 0.02. This conclusion has been verified by using the PEC equations. The geometrical parameters of the tubes considered in this study are presented in Table II.

The operating conditions of the reference (smooth) passage have been chosen as follows: $T_{i,S} = 12°C$, $T_{o,S} = 21°C$, Pr = 7.8, $10^4 < \text{Re} < 10^5$. The corresponding values of ϕ_0 are in the range $5.4 \times 10^{-4} < \phi_0 < 1.7 \times 10^{-2}$, which shows that the channel is dominated by heat transfer irreversibility. The effect of the thermal resistance external to the surface is also taken into consideration by including external heat transfer coefficient $h_{o,S} = 10\,100\ \text{W}\,(\text{m}^2\text{K})^{-1}$. The analysis includes the possibility that the augmented exchanger may have an enhanced outer tube surface, $E_o = h_{o,R}/h_{o,S}$. The values of E_o which have been assigned for each tube are presented in Table II. The fouling resistances on both sides of the tube wall are neglected.

Figure 5 represents the case FG-2a (Zimparov, 2000c) where the objective is to increase the heat rate, $\dot{Q}_* > 1$, of an existing heat exchanger, with the constraints for equal pumping power, $P_* = 1$, and heat transfer area, $A_* = 1$. The augmentation entropy generation number N_S is

$$N_S = \frac{1}{1+\phi_0}\left\{\dot{Q}_* \exp\left[B\left(1 - \frac{\text{St}_R}{\text{St}_S}(f_R/f_S)^{0.073} D_*^{-1.145}\right)\right]\right.$$

$$\left. \times \left[\frac{T_{i,S}}{T_{o,S}} + \dot{Q}_*(f_R/f_S)^{0.364} D_*^{-1.727}\left(1 - \frac{T_{i,S}}{T_{o,S}}\right)\right]^{-1} + \phi_0\right\} = f(\text{Re}_R) \tag{21}$$

Table II. Geometrical characteristics of the tubes*.

Tube no.	D_o (mm)	D_i (mm)	e (mm)	p (mm)	e/D	p/e	β_*	E_o
1	25.40	23.57	0.271	2.54	0.012	9.4	0.935	1.64
2	25.40	23.57	0.393	6.35	0.017	16.1	0.840	1.29
3	25.40	23.57	0.751	12.70	0.032	16.9	0.698	1.09
4	25.30	23.39	0.886	9.75	0.038	11.0	0.906	1.07
5	25.30	23.42	0.775	9.40	0.033	12.1	0.919	1.10
6	24.61	21.90	0.439	6.35	0.020	14.5	0.941	1.25
7	25.38	22.00	0.947	13.46	0.043	14.2	0.878	1.04
8	27.27	25.20	1.022	10.95	0.041	10.7	0.919	1.07
9	27.56	25.62	0.447	6.55	0.017	14.7	0.952	1.20
10	27.62	25.78	0.628	8.48	0.024	13.5	0.938	1.08

**Note*: Tubes no. 1, 2, 3 (Anon., 1982); 4, 5, 6, 7 (Withers, 1980); 8, 9, 10 (Zimparov *et al*., 1991).

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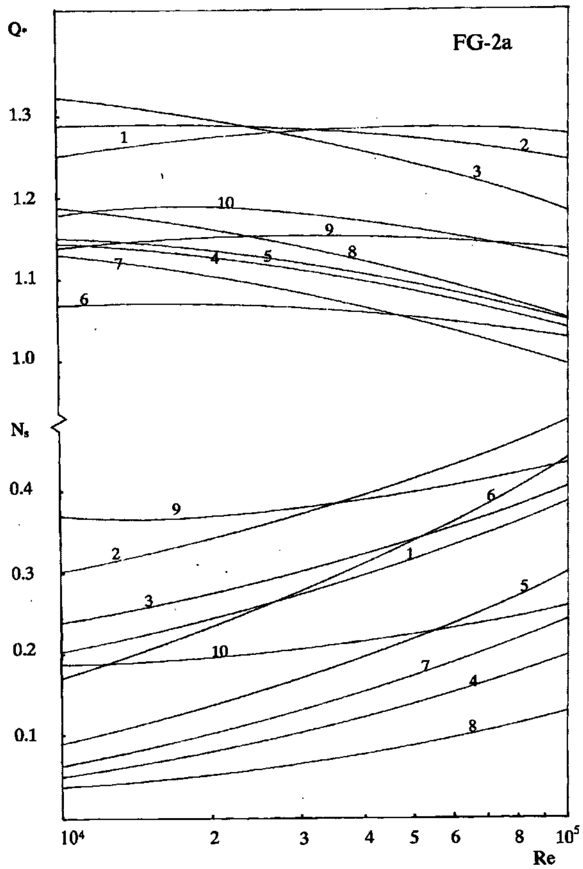

Figure 5. Increased heat transfer rate and augmentation entropy generation number versus Reynolds number (Zimparov, 2000c).

As can be seen from Figure 5, the corrugated tubes 4, 7 (Withers, 1980) and 8 (Zimparov *et al.*, 1991) have the smallest values of $N_S < 1$ but do not guarantee the largest heat transfer rate increase. On the other hand, the corrugated tubes 1, 2 and 3 (Anon., 1982) guarantee the largest values of $\dot{Q}_*$ but they do not guarantee a minimum rate of entropy generation.

Another case considered is VG-1, Figure 6, where the objective is to reduce the surface area, $A_* < 1$, with constraints $W_* = 1$, $\dot{Q}_* = 1$ and $P_* = 1$ (Zimparov, 2000c). The entropy generation number is calculated from

$$N_S = \frac{1}{1+\phi_0}\left\{\left(\frac{f_R}{f_S}A_*\right)^{-0.364} D_*^{2.091} \exp\left[B\left(1 - \frac{St_R}{St_S}(f_R/f_S)^{-0.291} \times A_*^{0.709} D_*^{-0.127}\right)\right] + \phi_0\right\} = f(\mathrm{Re_R}) \quad (22)$$

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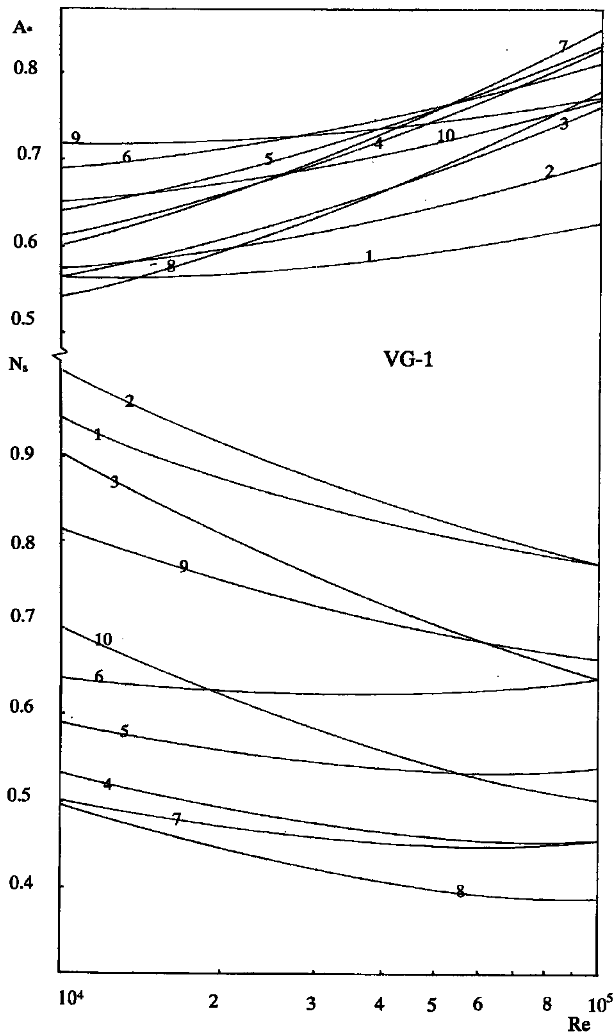

Figure 6. Reduced surface area and augmentation entropy generation number versus Reynolds number (Zimparov, 2000c).

In this case, the preferable tubes which have the largest reduction of heat transfer area, $A_* < 1$ are 1, 2, 3 (Anon., 1982) and 8 (Zimparov *et al.*, 1991). On the other hand, the tubes 4, 5, 7 (Withers, 1980) and 8 (Zimparov *et al.*, 1991) have the smallest value of $N_S < 1$.

These results show that the evaluation and comparison of the heat transfer augmentation techniques should be made on the basis of both first- and second- law analysis. Thus, it is possible to determine the thermodynamic optimum in a heat exchanger by minimizing the augmentation entropy generation number compared with the relative increase of heat transfer rate $\dot{Q}_* > 1$ or relative reduction of heat transfer area $A_* < 1$ ($L_* < 1$) or pumping power $P_* < 1$. Consequently, the ratio $N_S/\dot{Q}_*$, and the groups $N_S A_*$, $N_S L_*$, $N_S P_* = f(\mathrm{Re})$ could be defined to connect the two objectives pursued by the first- and second-law analysis.

Figures 7 and 8 show $N_S/\dot{Q}_* - \mathrm{Re_R}$ for the case FG-2a and $N_S A_* - \mathrm{Re_R}$ for the case VG-1. For all the cases considered the best performance have one and the same tubes—8 (Zimparov *et al.*, 1991), 7, 4 (Withers, 1980) being far superior to others. Therefore, the analysis using new

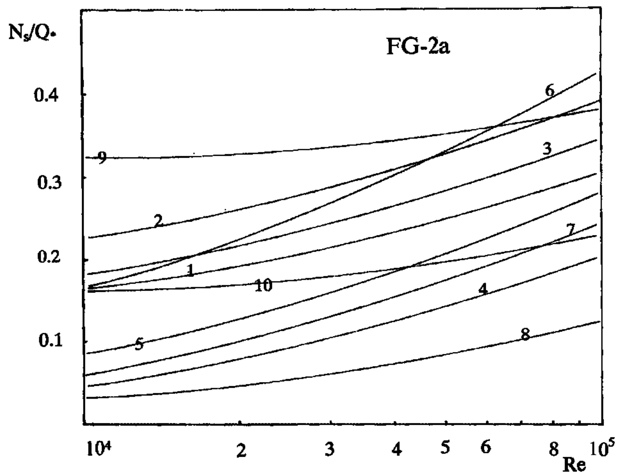

Figure 7. The ratio N_S/Q_* versus the Reynolds number (Zimparov, 2000c).

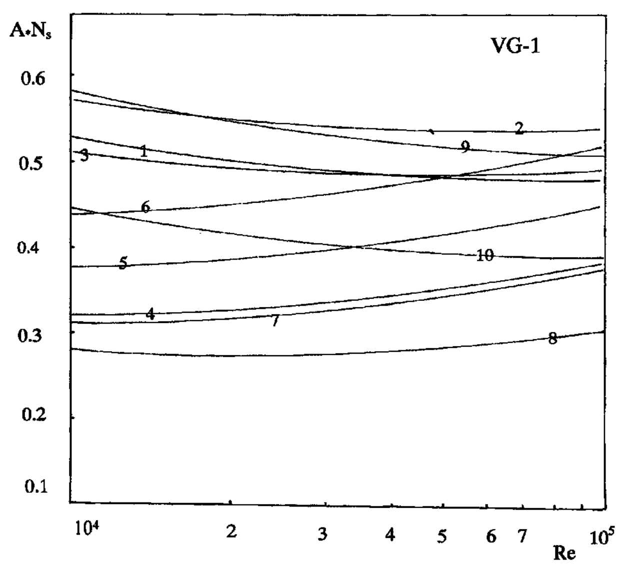

Figure 8. The group $A_* N_S$ versus the Reynolds number (Zimparov, 2000c).

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performance evaluation criteria shows that the optimum rib height-to-diameter ratio (e/D_i) for spirally corrugated tubes is about 0.04 (Zimparov, 2000c, d).

As preliminary design guidance to the selection of an augmentation technique, the heat transfer efficiency can be evaluated based on the power consumption per unit mass of fluid. The criterion i_E (Sano and Usui, 1982) is defined as the ratio between the heat transfer coefficient for the tube using a heat transfer promoter and the value for a smooth tube at the same level of power. Following (Sano and Usui, 1982) the criterion i_E can be defined as

$$i_E = \frac{Nu_R/Nu_S}{(f_R/f_S)^{0.291}} = f(Re_S) \tag{23}$$

This criterion can be used to demonstrate the achievements in this particular kind of compound technique discussed here—rough surface with a twisted tape insert. With the twisted tape

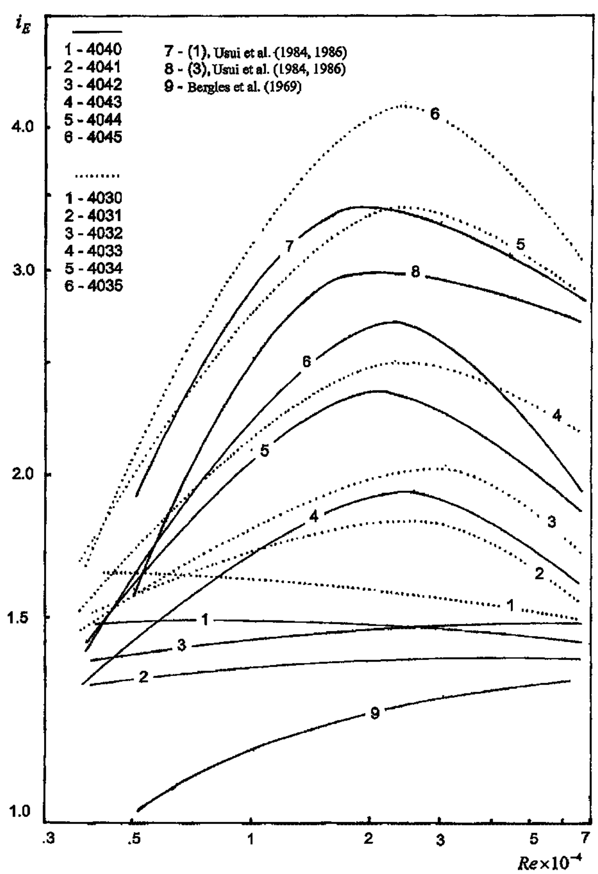

Figure 9. Evaluation of heat transfer efficiency based on the constant energy dissipation (Zimparov, 2000b).

alone, i_E was in the general flow in value of the order of 1.0–1.2. When an internally grooved or corrugated tube was used alone $i_E = 1.5$–2.0 for Re $> 10^4$. When a counterclockwise twisted tape was used in conjunction with an internally grooved tube (Usui *et al.*, 1984, 1986), i_E attained values of 2.5–3.0. Figure 9 represents the values of i_E for all the combinations of a corrugated tube and a twisted tape insert investigated by Zimparov (2000b), together with those obtained in the previous studies (Bergles *et al.*, 1969; Usui *et al.*, 1984, 1986). As can be seen from Figure 9 with an appropriate selection of the geometric parameters e/D_i and H/D_i a very great improvement in the heat transfer efficiency ($i_E = 2.7$–4.2) could be reached.

The real evaluation of the conservation of useful part of energy can be obtained using the extended PEC (Zimparov, 2000b). For the same cases discussed before, Figures 10 and 11 demonstrate the huge amount of exergy which could be conserved by compound augmentation technique of this kind.

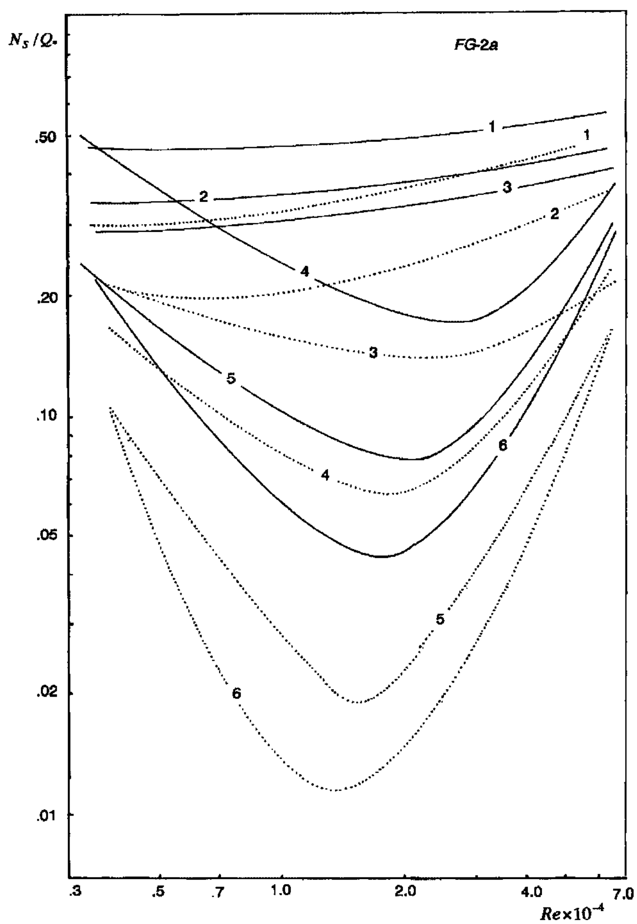

Figure 10. The ratio N_S/Q_* versus the Reynolds number (Zimparov, 2000b).

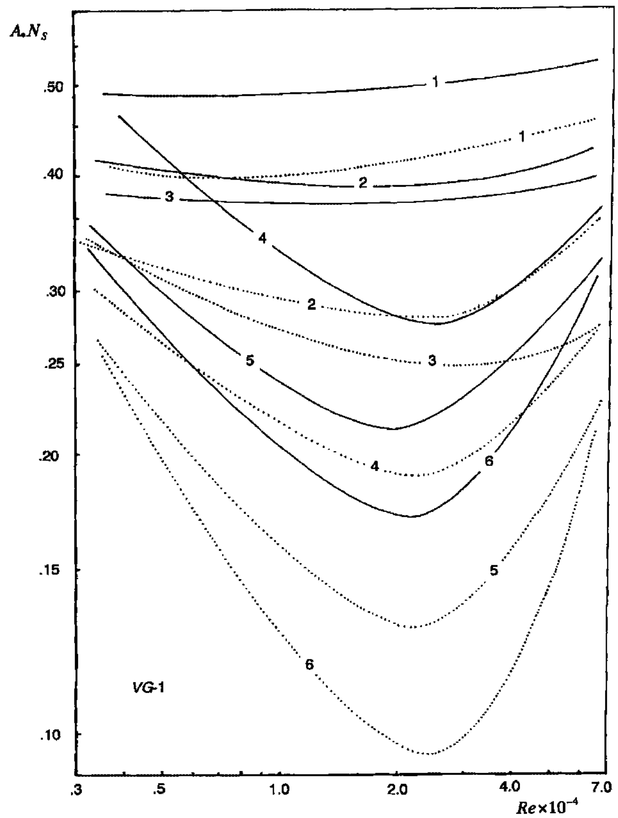

Figure 11. The group A_*N_S versus the Reynolds number (Zimparov, 2000b).

5. CONCLUDING REMARKS

Heat transfer enhancement technology is well established. It is used routinely in the power and process industries. New applications of the techniques are steadily being found. Among the most promising heat transfer enhancement techniques for conserving of the useful part of energy is the compound augmentation method, in which different enhancement techniques are used simultaneously.

To assess the merit of the conservation of useful part of energy (exergy), extended PEC equations have been developed. They include the effect of fluid temperature variation along the length of a tubular heat exchanger at constant wall temperature and constant heat flux and evaluate heat transfer enhancement techniques based on the entropy production theorem with various constraints imposed. These equations add new PEC for enhanced heat transfer surfaces developed by first-law analysis with criteria assessing the benefits of augmentation techniques in connection with the entropy generation and exergy destruction.

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The general evaluation criteria add new information to Bejan's EGM method assessing two objectives simultaneously. They may help to display inappropriate enhanced surfaces and assist the engineer to design better heat transfer equipment.

NOMENCLATURE

A	= heat transfer surface area (m^2)
D	= tube diameter (m)
e	= ridge height (m)
h	= heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
H	= helical pitch defined for 360° rotation of twisted tape (m)
k	= thermal conductivity ($W\,m^{-1}\,K^{-1}$)
L	= tube length (m)
$\dot{m}$	= mass flow rate in tube ($kg\,s^{-1}$)
N_t	= number of tubes
P	= pumping power (W)
p	= pitch of ridging (m)
Δp	= pressure drop (Pa)
$\dot{Q}$	= heat transfer rate (W)
R	= thermal resistance ($K\,W^{-1}$)
$\dot{S}_{gen}$	= rate of entropy generation ($W\,K^{-1}$)
T	= temperature (K)
ΔT	= temperature difference (K)
u_m	= mean flow velocity ($m\,s^{-1}$)
U	= overall heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
W	= mass flow rate in heat exchanger ($kg\,s^{-1}$)

Greek letters

β	= helix angle of rib (°)
η	= surface efficiency (-)
ϑ	= temperature difference between wall and fluid, $T_w - T$ (K)
μ	= dynamic viscosity (Pa s)
ρ	= fluid density ($kg\,m^{-3}$)

Dimensionless groups

A_*	= dimensionless heat transfer surface (A_R/A_S)
D_*	= dimensionless tube diameter (D_R/D_S)
L_*	= dimensionless tube length (L_R/L_S)
f	= Fanning friction factor ($2\tau_w/(\rho u_m^2)$)
Nu	= Nusselt number ($h_i D/k_f$)
N_S	= augmentation entropy generation number
N_*	= ratio of number of tubes ($N_{t,R}/N_{t,S}$)
Pr	= Prandtl number ($\mu c_p/k_f$)
P_*	= dimensionless pumping power (P_R/P_S)

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Δp_*	= dimensionless pressure drop ($\Delta p_R/\Delta p_S$)
Q_*	= dimensionless heat transfer rate ($\dot{Q}_R/\dot{Q}_S$)
Re	= Reynolds number ($\rho u_m D/\mu$)
St	= Stanton number ($h_i/(\rho u_m c_p)$)
ΔT_i^*	= dimensionless inlet temperature difference between hot and cold streams ($\Delta T_{i,R}/\Delta T_{i,S}$)
$u_{m,*}$	= dimensionless flow velocity ($u_{m,R}/u_{m,S}$)
W_*	= dimensionless mass flow rate (W_R/W_S)
β_*	= $\beta/90$
ε_*	= ratio of heat exchanger effectiveness ($\varepsilon_R/\varepsilon_S$)
ϕ_0	= irreversibility distribution ratio

Subscripts

ext	= external
F	= fin
f	= fluid
i	= inside
i	= value at $x = 0$
m	= mean value
R	= rough tube
S	= smooth tube
o	= outside
0	= value at $x = L$
U	= unfinned
w	= wall

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