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Performance evaluation criteria for enhanced heat transfer surfaces

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Abstract—Performance evaluation equations for enhanced heat transfer surfaces based on the entropy production theorem are reported. The equations originate from various design constraints and generalize the performance evaluation criteria (PEC) for enhanced heat transfer techniques obtained by means of first law analysis. The application of this more comprehensive treatment of PEC compared to previous references is illustrated by the analysis of heat transfer and fluid friction characteristics of spirally corrugated tubes, assessing the benefit of these tubes by an augmentation technique subject to various design constraints.

INTRODUCTION

THE PERFORMANCE of conventional heat exchangers can be substantially improved by a number of augmentation techniques resulting in the design of high-performance thermal systems. A performance comparison of effectiveness for various types of enhanced surfaces may lead to selection criteria for designers and users. On the basis of first law analysis several authors [1–3] have proposed performance evaluation criteria (PEC) which define the performance benefits of an exchanger having enhanced surfaces, relative to a standard exchanger with smooth surfaces subject to various design constraints. Three basic design objectives have been discussed and applied to 11 cases of interest: (1) reduced heat transfer surface (total length of exchanger tubing) for equal pumping power and heat duty; (2) increased product UA for equal pumping power and fixed total length of exchanger tubing; and (3) reduced pumping power for equal heat duty and total length of exchanger tubing.

On the other hand it is well established that the minimization of entropy generation in any process leads to the conservation of energy. In a heat exchanger unit entropy is generated by the heat transferred due to 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 or at least decrease the overall entropy generation associated with the heat exchanger operation.

A solid thermodynamic basis to evaluate the merit of augmentation techniques by second law analysis has been proposed by Bejan [4]. 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. A second law approach to PEC analysis is described in refs. [4–6] where the entropy generated per unit time and unit length of the duct is analysed for two augmentation techniques: rough surfaces and swirl promoters. The analysis is performed for constraints $W_* = 1$ and $Q_* = 1$ which corresponds to case *FG*-1*b* of Table 1 in ref. [3]. Other publications on this subject are refs. [7, 8]. One of the problems discussed in refs. [7, 8] is how to enhance heat transfer in order to reduce the temperature difference which is the driving force for the heat transfer process with the constraints of the case *FG*-1*b* [3] or with the following constraints: fixed basic geometry, heat duty and pressure drop [8]. Another problem discussed in ref. [7] is to enhance heat transfer in order to reduce the surface area of the unit with the constraints corresponding to the case *FN*-1 [3].

The purpose of this paper is to develop additional PEC equations to evaluate the heat transfer enhancement techniques based on the entropy production theorem with the constraints [3]. These equations add new information to PEC for enhanced heat transfer surfaces developed by first law analysis and implemented in heat exchangers [1–3] with criteria assessing the merits of augmentation techniques in connection with the entropy generation and one-way destruction of exergy. The validity of the generalized

NOMENCLATURE

A	heat transfer surface area [m^2]
c_p	specific heat capacity [J kg^{-1} K^{-1}]
D	tube diameter [m]
h	heat transfer coefficient [W m^{-2} K^{-1}]
k	thermal conductivity [W m^{-1} K^{-1}]
L	tube length [m]
P	pumping power [W]
Δp	pressure drop [Pa]
q'	heat transfer rate per unit length [W m^{-1}]
$\dot{S}_g$	rate of entropy generation [W K^{-1}]
T	fluid temperature [K]
ΔT	wall-to-fluid temperature difference [K]
u_m	mean fluid velocity [m s^{-1}]
U	overall heat transfer coefficient [W m^{-2} K^{-1}]
W	mass flow rate [kg s^{-1}].

Greek symbols

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_iD/k_f
N_S	augmentation entropy generation number, equation (4)
Pr	Prandtl number, $\mu c_p/k_f$
P_*	dimensionless pumping power, P_R/P_S
Q_*	dimensionless heat transfer rate, Q_R/Q_S
Re	Reynolds number, $\rho u_m D/\mu$
St	Stanton number, $h_i/(\rho u_m c_p)$
W_*	dimensionless mass flow rate, W_R/W_S
ϕ_0	irreversibility distribution ratio, equation (8).

Subscripts

f	fluid
i	inside
L	value at $x = L$
m	mean value
R	rough tube
S	smooth tube
0	value at $x = 0$.

set of PEC equations is illustrated by the analysis of heat transfer and fluid friction characteristics of spirally corrugated tubes.

EQUATIONS BASED ON THE ENTROPY PRODUCTION THEOREM

In what follows the effect of heat transfer enhancement techniques on the fluid flowing inside a circular duct of diameter D_h and length L is considered. The rate of entropy generation for a steady-flow system per unit length [4] is

$$\frac{\mathrm{d}\dot{S}_g}{\mathrm{d}x} = \frac{q'\Delta T}{T^2} + \frac{W}{\rho T}\left(-\frac{\mathrm{d}p}{\mathrm{d}x}\right), \tag{1}$$

where it is assumed that the wall–fluid temperature difference, ΔT, is considerably smaller than the local absolute temperature of the fluid. Consider a heat exchanger of length L with a constant wall heat flux and thermally and hydraulically fully developed turbulent flow. Equation (1) can be expressed in the form

$$\dot{S}_g = \frac{\dot{Q}\Delta T}{T_0^2} + \frac{W}{\rho T_0}\Delta p \tag{2}$$

or

$$\dot{S}_g = \frac{\dot{Q}^2 D_h}{k_f T_0^2 A}\frac{1}{Nu} + \frac{32L^3 f W^3}{D_h^3 \rho^2 T_0 A^2}, \tag{3}$$

where it is assumed that $T_0 \gg T_L - T_0$. Following ref. [4] the thermodynamic impact of the augmentation technique is defined by the augmentation entropy generation number

$$N_S = \dot{S}_{g,R}/\dot{S}_{g,S}. \tag{4}$$

Augmentation techniques with $N_S < 1$ are thermodynamically advantageous, since, in addition to enhancing heat transfer, they reduce the degree of irreversibility of the unit's performance. Substituting equation (3) into (4), N_S can be rewritten as

$$N_S = \frac{N_T + \phi_0 N_P}{1+\phi_0}. \tag{5}$$

where

$$N_T = \frac{Q_*^2 D_*}{A_*}\frac{Nu_S}{Nu_R} \tag{6}$$

$$N_P = \frac{A_*}{D_*^3}\frac{f_R}{f_S}\left(\frac{Re_R}{Re_S}\right)^3 \equiv P_*. \tag{7}$$

When the heat transfer passage is known, the numerical value of the irreversibility distribution ratio, $\phi_0 = (\dot{S}_{\Delta p}/\dot{S}_{\Delta T})_S$, describes the thermodynamic mode in which the passage is meant to operate:

$$\phi_0 = (\dot{S}_{g,\Delta p}/\dot{S}_{g,\Delta T}) = \frac{k_f T_0 A^2 \mu^3}{2\rho^2 D_h^4 Q_S^2} f_S Re_S^3 Nu_S. \tag{8}$$

The expression (8) for the irreversibility distribution ratio ϕ_0 can be simplified [4]:

$$\phi_0 = \left(\frac{T_0}{\Delta T_m}\right)_S^2 \left(\frac{u_m^2}{c_p T_0}\right)_S \frac{f_S/2}{St_S}, \tag{9}$$

which yields a straightforward estimate of ϕ_0. For example, in the experimental program described in ref. [9], for $Re_S = 13\,000$, $Pr = 3.23 - \phi_0 = 0.00023$ while for $Re = 51\,000$, $Pr = 2.34 - \phi_0 = 0.00356$. In this case, the reference passage is dominated by heat transfer irreversibility, $\phi_0 \ll 1$.

The design constraints imposed on the exchanger flow rate and velocity cause key differences among the possible PEC relations [3]. The increased friction factor due to augmented surfaces may require reduced velocity to satisfy a fixed pumping power (or pressure drop) constraint. If the exchanger flow rate is held constant, it may be necessary to increase the flow frontal area to satisfy the pumping power constraint. However, if the mass flow rate is reduced, it is possible to maintain a constant flow frontal area at reduced velocity. In many cases the heat exchanger flow rate is specified and a flow rate reduction is not permitted. The PEC discussed in this paper will account for these various possibilities.

Fixed geometry criteria (FG)

These criteria involve a one-for-one replacement of smooth tubes by augmented ones of equal length. The *FG*-1 cases seek increased heat duty or *UA* for constant exchanger flow rate and velocity. The pumping power of the augmented tube exchanger will increase due to the increased fluid friction characteristics of the augmented surface. For these cases the constraints $W_* = 1$, $N_* = 1$ and $L_* = 1$ require $Re_S = D_* Re_R$ and $P_* > 1$. One of the most common and well documented heat transfer augmentation techniques is the surface promoters or 'in-tube roughness'. Wall roughness has a negligible impact on the flow cross-section and hydraulic diameter D_h: thus we assume $D_* = 1$ in what follows. When the objective is increased heat duty $Q_* > 1$, this corresponds to the case *FG*-1*a* [3], and the augmentation entropy generation number N_S, equation (5), becomes

$$N_S = \frac{1}{1+\phi_0}\left\{\frac{Nu_S}{Nu_R} Q_*^2 + \phi_0 (f_R/f_S)\right\}, \tag{10}$$

where

$$N_T = Q_*^2 \frac{Nu_S}{Nu_R} \quad \text{and} \quad N_P = (f_R/f_S).$$

If the objective is $U_R A_R / U_S A_S > 1$ for $Q_* = 1$, the driving temperature difference ΔT_m may be reduced. This case corresponds to *FG*-1*b* [3]. The constraints $N_* = 1$, $L_* = 1$, $W_* = 1$ and $Q_* = 1$ require $Re_R = Re_S$ and $P_* > 1$. The objective is $\Delta T_m^* < 1$. The augmentation entropy generation number N_S, equation (5), can be written in the form [4]

$$N_S = \frac{Nu_S}{Nu_R} + \frac{\phi_0}{1+\phi_0}\left\{\frac{f_R}{f_S} - \frac{Nu_S}{Nu_R}\right\}$$

or

$$N_S = \frac{1}{1+\phi_0}\left\{\frac{Nu_S}{Nu_R} + \phi_0 (f_R/f_S)\right\}. \tag{11}$$

The *FG*-2 criteria have the same objectives as *FG*-1, but require that the augmented tube unit operates at the same pumping power as the reference smooth tube unit. The pumping power is maintained constant by reducing the tube-side velocity and thus the exchanger flow rate. The constraints are: $N_* = 1$, $L_* = 1$ and $P_* = 1$ requiring $W_* < 1$ and $Re_R < Re_S$. When the objective is $Q_* > 1$, the augmentation entropy generation number N_S becomes

$$N_S = \frac{1}{1+\phi_0}\left\{\frac{Nu_S}{Nu_R} Q_*^2 + \phi_0\right\}. \tag{12}$$

In this equation, however, the values of Nu_S and Nu_R are calculated at different Reynolds numbers, Nu_S for Re_S and Nu_R for Re_R. Figure 1, pertaining to the important area of internal, single-phase, forced convection flow, demonstrates the friction factor and heat transfer coefficients using the presentation format of performance data for enhanced tubes [10]. Having in mind that the comparable heat exchangers must produce equal energy dissipation, the problem can readily be solved considering the ratio of heat transfer coefficients at constant pumping power, [11]. If the friction factor and Nusselt number characterizing the smooth surface in the turbulent flow region are fitted by

$$f_S = 0.079 Re^{-0.25} \quad \text{and} \quad Nu_S = 0.023 Re^{0.8} Pr^{0.4}$$

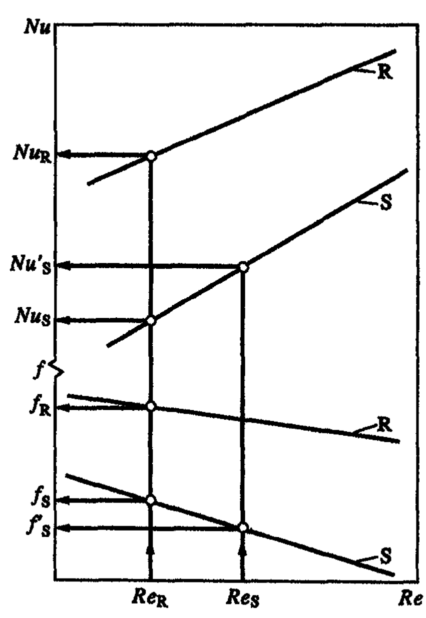

FIG. 1. Sketches of Nusselt number and friction factor vs Reynolds number.

the ratio Nu_S/Nu_R in equation (12) can be expressed (Fig. 1) by

$$\frac{Nu_S(Re_S)}{Nu_R(Re_R)} = \frac{Nu_S(Re_R)}{Nu_R(Re_R)}\frac{Nu_S(Re_S)}{Nu_S(Re_R)} = \frac{Nu_S}{Nu_R}(f_R/f_S)^{0.291} = fnc(Re_R)$$

recalling also that the constraint $P_* = 1$ imposes

$$f_S(Re_S)\,Re_S^3 = f_R(Re_R)\,Re_R^3.$$

Consequently, the term representing the irreversibility due to heat transfer yields

$$Nu_T = \frac{Nu_S}{Nu_R}(f_R/f_S)^{0.291}Q_*^2 = fnc(Re_R). \quad (13)$$

In equation (13) the Nusselt numbers and friction factor coefficients are calculated for one and the same Reynolds number, $Re = Re_R$, and equation (12) can be rewritten in the form

$$N_S = \frac{1}{1+\phi_0}\left\{\frac{Nu_S}{Nu_R}(f_R/f_S)^{0.291}Q_*^2+\phi_0\right\} = fnc(Re_R). \quad (14)$$

When the objective is $\Delta T_m^* < 1$ with the additional constraint $Q_* = 1$, the augmentation entropy generation number N_S is

$$N_S = \frac{1}{1+\phi_0}\left\{\frac{Nu_S}{Nu_R}(f_R/f_S)^{0.291}+\phi_0\right\} = fnc(Re_R). \quad (15)$$

In this case, which corresponds to *FG*-2*b* [3], the Reynolds numbers maintained in the comparable unit are defined by

$$Re_S = Re_R(f_R/f_S)^{0.364} = fnc(Re_R). \quad (16)$$

The case (called *FG*-2*c* here) where the objective is $\Delta T_m^* < 1$ with the constraints $N_* = 1, L_* = 1, Q_* = 1$ and $\Delta p_* = 1$ (pressure drop fixed) is an extension of cases *FG*-2. The consequences are $W_* < 1$, $P_* < 1$, $Re_R < Re_S$ and this case corresponds to case *B* [8]. Now the terms representing the impact of the heat transfer and fluid friction on the entropy generation are

$$N_T = \Delta T_m^* = (\Delta T_{m,R}/\Delta T_{m,S}) = \frac{Nu_S}{Nu_R}(f_R/f_S)^{0.457} = fnc(Re_R) \quad (17a)$$

$$N_P = (W_R/W_S) = (f_R/f_S)^{-0.571} = fnc(Re_R) \quad (17b)$$

and equation (5) yields

$$N_S = \frac{1}{1+\phi_0}\left\{\frac{Nu_S}{Nu_R}(f_R/f_S)^{0.457} + \phi_0(f_R/f_S)^{-0.571}\right\} = fnc(Re_R). \quad (18)$$

The third criterion, *FG*-3 [3], attempts to reduce the pumping power for constant heat duty. The constraints and consequences are $N_* = 1$, $L_* = 1$, $Q_* = 1$, $W_* < 1$, $Re_R < Re_S$. The objective is $P_* < 1$. In this case

$$N_T = \Delta T_{m,R}/\Delta T_{m,S} = \frac{Nu_S}{Nu_R}(f_R/f_S)^{0.291} = fnc(Re_R) \quad (19a)$$

$$N_P = P_* \quad (19b)$$

and equation (5) yields

$$N_S = \frac{1}{1+\phi_0}\left\{\frac{Nu_S}{Nu_R}(f_R/f_S)^{0.291}+\phi_0 P_*\right\} = fnc(Re_R). \quad (20)$$

Fixed flow area criteria (FN)

These criteria maintain constant flow area. The objective of *FN*-1 case is reduced surface area by reduced tube length, $L_* < 1$, for constant pumping power, $P_* = 1$. The additional constraints are $N_* = 1$, $Q_* = 1$ requiring $W_* < 1$ and $Re_R < Re_S$. In this case

$$N_T = \Delta T_{m,R}/\Delta T_{m,S} = \frac{Nu_S}{Nu_R}(f_R/f_S)^{0.291}L_*^{-0.709} = fnc(Re_R) \quad (21a)$$

$$N_P = 1 \quad (21b)$$

and the augmentation entropy generation number N_S becomes

$$N_S = \frac{1}{1+\phi_0}\left\{\frac{Nu_S}{Nu_R}(f_R/f_S)^{0.291}L_*^{-0.709}+\phi_0\right\} = fnc(Re_R). \quad (22)$$

The Reynolds numbers maintained in the comparable unit are defined by

$$Re_S = Re_R\left(\frac{f_R}{f_S}L_*\right)^{0.364} = fnc(Re_R). \quad (23)$$

The objective of *FN*-2 case [3], is to reduce pumping power, $P_* < 1$, with constant heat duty, $Q_* = 1$, and flow rate, $W_* = 1$. Another constraint is $N_* = 1$ and consequently $L_* < 1$ and $Re_R = Re_S$. The P_* equation shows that it is not possible to obtain $P_* < 1$ for a two-fluid heat exchanger [3]. This is because $f_R/f_S > U_R/U_S$. For a prescribed heat flux boundary condition $U_R/U_S = h_R/h_S$. Therefore, $P_* < 1$ would be obtained if $f_R/f_S < Nu_R/Nu_S$. Rough surfaces typically yield $f_R/f_S > Nu_R/Nu_S$. In this case

$$N_T = \frac{Nu_S}{Nu_R}L_*^{-1}, \qquad N_P = \frac{f_R}{f_S}L_* \quad (24)$$

and the equation for the augmentation entropy generation number is

$$N_S = \frac{1}{1+\phi_0}\left\{\frac{Nu_S}{Nu_R}L_*^{-1}+\phi_0 L_*(f_R/f_S)\right\}. \quad (25)$$

Variable geometry criteria (VG)

In most cases a heat exchanger is designed for a required thermal duty with a specified flow rate. Because the tube-side velocity must be reduced to accommodate the higher friction characteristics of the augmented surface, it is necessary to increase the flow area to maintain constant flow rate. All of the *VG* cases maintain $W_* = 1$ and permit the exchanger flow frontal area to vary in order to meet the pumping power constraint: $N_* > 1$; $L_* < 1$; $Re_R < Re_S$. Case *VG*-1 [3] yields reduced surface area $A_* < 1$, for $Q_* = 1$ and $P_* = 1$. Now

$$N_T = \Delta T_{m,R}/\Delta T_{m,S}$$

$$= \frac{Nu_S}{Nu_R}(f_R/f_S)^{0.291}A_*^{-0.709} = fnc(Re_R) \quad (26a)$$

$$N_P = 1 \quad (26b)$$

and the augmentation entropy generation number N_S is

$$N_S = \frac{1}{1+\phi_0}\left\{\frac{Nu_S}{Nu_R}(f_R/f_S)^{0.291}A_*^{-0.709}+\phi_0\right\}$$

$$= fnc(Re_R). \quad (27)$$

After fixing Re_R, Re_S can be calculated from

$$Re_S = Re_R\left(\frac{f_R}{f_S}A_*\right)^{0.364} = fnc(Re_R). \quad (28)$$

The cases *VG*-2 [3] aim at increased thermal performance ($U_R A_R/U_S A_S$ or $Q_* > 1$) for $A_* = 1$ and $P_* = 1$. They are similar to the cases *FG*-2 and the equations for the augmentation entropy generation number N_S have the same forms as equations (14) and (16).

Case *VG*-3 [3] aims at reduced pumping power, $P_* < 1$, for $A_* = 1$ and $Q_* = 1$. It is similar to case *FG*-3 and the equation for N_S has the same form as equation (20).

APPLICATION OF PEC EQUATIONS AND DISCUSSION

The solution of the PEC equations described above requires algebraic relations which:

1. Define correlations for *St* and *f* of the augmented surfaces as a function of *Re*.
2. 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 [3] based on the first law of thermodynamics, to obtain the values of Q_*, A_* or P_* as a function of *Re*.

From an engineering point of view it is important to know under what conditions the wall-roughening technique leads to a reduction in entropy generation. It is evident that the ability of roughened walls to reduce the degree of irreversibility depends on the thermo-fluid operating regime, ϕ_0 and Re_R. The results of this study can be illustrated by the characteristics of the spirally corrugated tubes obtained through the experimental program [9]. Twenty-five spirally corrugated brass tubes with different geometrical parameters were investigated. The variations of *Re* and *Pr* were in the range $10^4 < Re < 6\times10^4$ and $2.2 < Pr < 3.4$ and the corresponding values of ϕ_0 were in the range $0.0002 < \phi_0 < 0.004$. The numerical values of ϕ_0 show that the channel is dominated by heat transfer irreversibility.

Figures 2 and 3 represent case *FG*-1*a* where the augmentation objective is to increase the heat duty, $Q_* > 1$. The variations of the augmentation entropy generation number N_S as a function of Re_R (Fig. 2) are obtained from equation (12) with the values for Nu_S/Nu_R, f_R/f_S and ϕ_0 taken from the experimental program [9]. The values of Q_*, as a function of Re_R are obtained following Webb's treatise on PEC [3] (Fig. 3). In this case, the unit with heat transfer enhancing spirally corrugated tubes increases its heat duty, $Q_* > 1$, significantly, but does not reduce the destruction of exergy. All corrugated tubes (in the range of Reynolds numbers studied) lead to an increase in the rate of entropy generation, despite the fact that the heat duty has increased (Fig. 2).

Figures 4 and 5 represent the case *FG*-1*b* where the

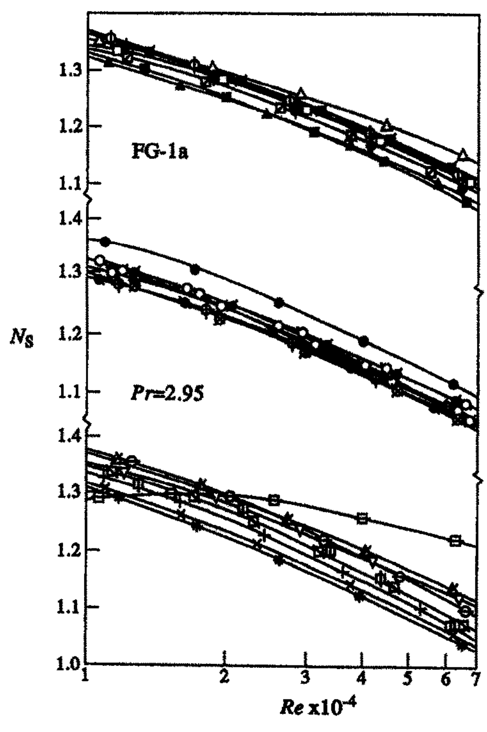

FIG. 2. Augmentation entropy generation number vs Reynolds number.

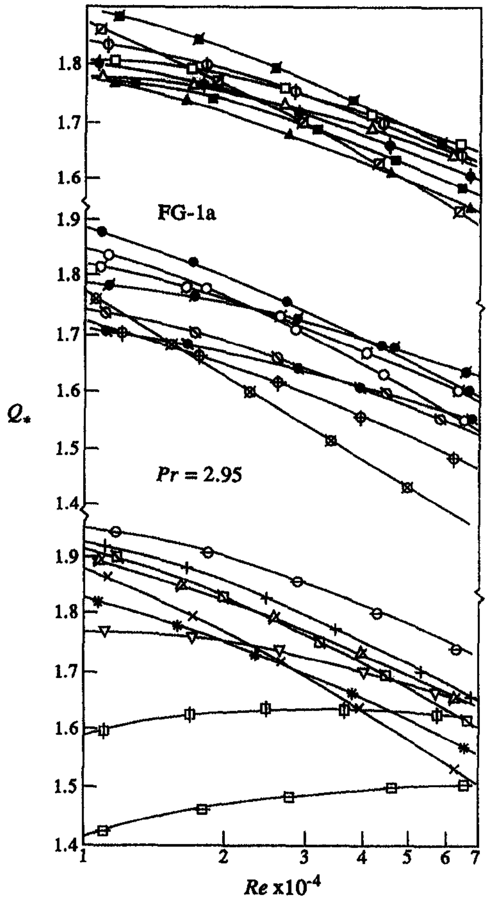

FIG. 3. Increased heat transfer rate vs Reynolds number.

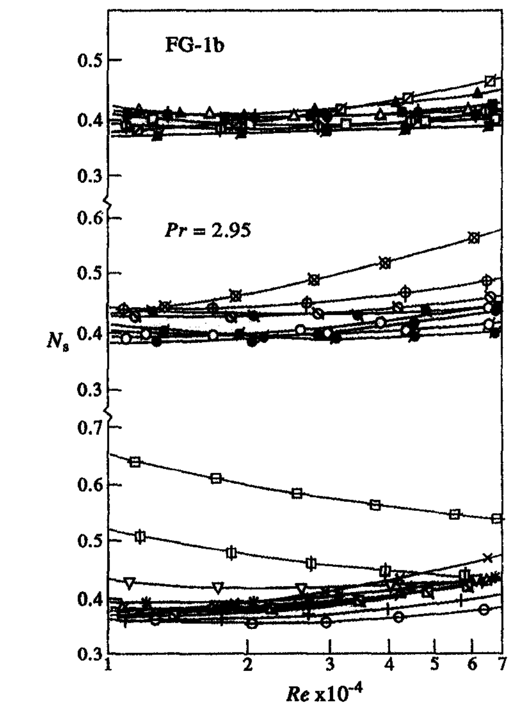

FIG. 4. Augmentation entropy generation number vs Reynolds number.

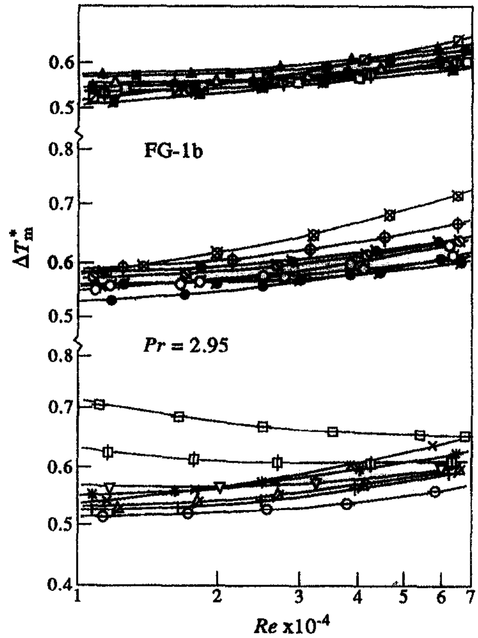

FIG. 5. Reduced driving temperature difference vs Reynolds number.

augmentation techniques may be used to reduce the driving temperature difference, $\Delta T^*_m < 1$. The variations of ΔT^*_m with Re_R for the spirally corrugated tubes studied in [9] are shown in Fig. 5. As seen, a heat exchanger utilizing tubes 32, 21, 23, 13, 18, 28 may operate at 45% lower driving temperature difference than the unit designed with smooth tubes. The variation of the augmentation entropy generation number N_S as a function of Re_R is plotted in Fig. 4. Several spirally corrugated tubes, 32, 18, 28, 27 have $N_S < 0.4$ and will yield significant savings in exergy. These values of N_S can be compared with those for sand-grain roughness and repeated ribs for $\phi_0 = 0$ [4, 5]. None of the roughened surfaces [4, 5] reach $N_S = 0.4$, e.g. the exergy payoff associated with implementing the spirally corrugated tubes will be larger.

In the case *FG-2a* the goal is increased heat transfer rate, $Q_* > 1$, for equal pumping power, $P_* = 1$. The variations of Q_* as a function of *Re* are shown in Fig. 7 [9], whereas Fig. 6 represents the variations of N_S. Examining carefully Fig. 7 [9] and Fig. 6 one may find out that the corrugated tubes 35, 32 and 18 which have the smallest values of N_S do not guarantee the largest heat transfer rate increase. On the other hand, the corrugated tubes 13, 14, 15, 16, 23 guarantee the largest values of Q_* and have $N_S < 1$. This implies that the evaluation of the heat transfer augmentation techniques should be made on the basis of both first and second law analysis.

Figures 7 and 8 illustrate case *FG-2b* where the objective is to decrease the driving temperature difference, $\Delta T^*_m < 1$, for $A_* = P_* = Q_* = 1$. The variations of ΔT^*_m as a function of *Re* are plotted in Fig. 7 and

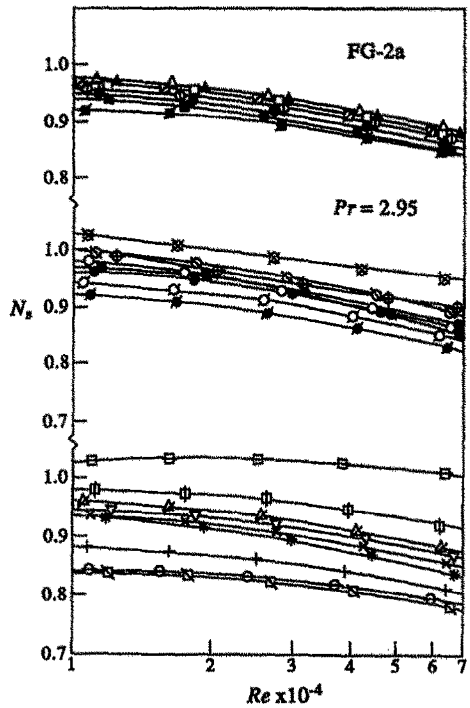

FIG. 6. Augmentation entropy generation number vs Reynolds number.

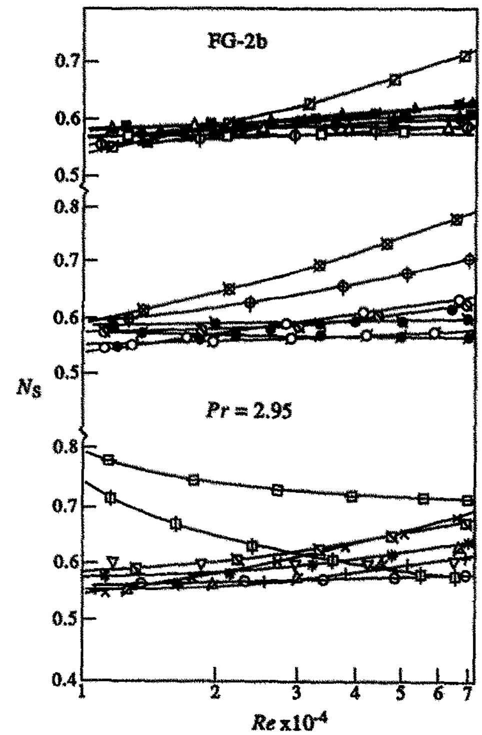

FIG. 8. Augmentation entropy generation number vs Reynolds number.

the corresponding values of N_S in Fig. 8. The tubes 13, 23, 14, 16, 28 show the best characteristics and guarantee simultaneously the smallest values of ΔT_m^* and N_S.

Figure 9 represents the variations of N_S with Re_R for case FG-$2c$ which is an extension of the cases FG-

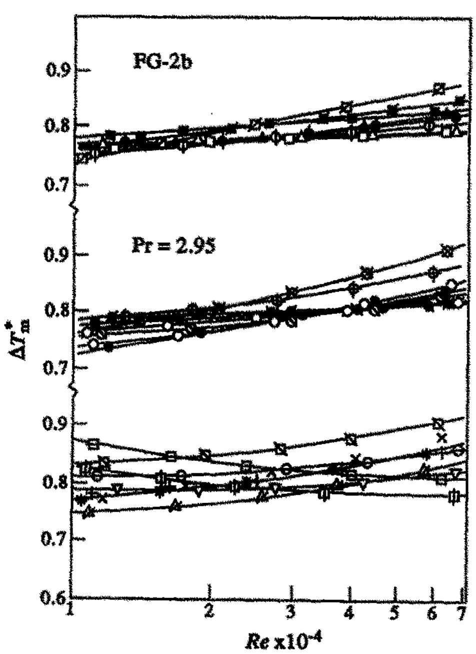

FIG. 7. Reduced driving temperature difference vs Reynolds number.

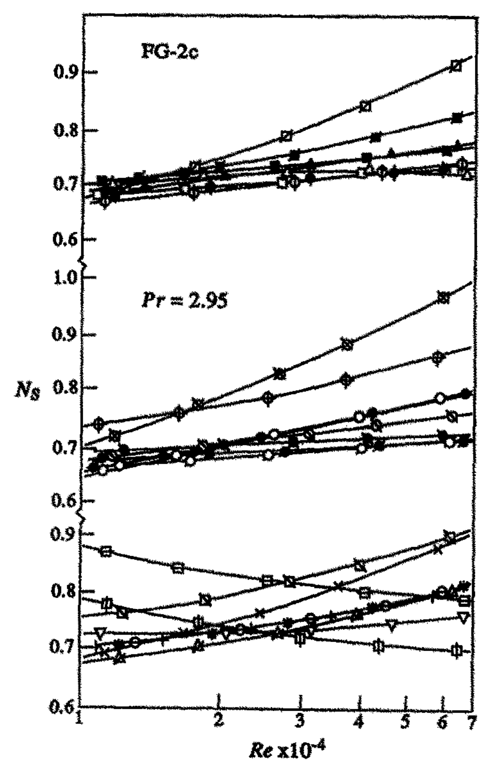

FIG. 9. Augmentation entropy generation number vs Reynolds number.

2. The objective here is also $\Delta T_m^* < 1$, but the variations of ΔT_m^* with Re_R are not presented here because the equations needed to calculate ΔT_m^* are not included in the procedure based on the first law analysis. In this case the behaviour of the tube 34 [9] for $Re > 3 \times 10^4$ is interesting.

The values of N_S for case FG-3 in the range of Re studied are plotted in Fig. 10. This figure should be considered together with Fig. 8 [9] where the variations of P_* with Re for all tubes are presented. Examining the figures one finds that tubes 32, 23, 14 and 16, have the smallest values of N_S (approximately equal values) but the values of P_* are different—for 14 and 16, $P_* = 0.39$; for 23, $P_* = 0.43$; for 32, $P_* = 0.50$.

The reduction of heat transfer surface through reduced tube length $L_* < 1$ for $Q_* = P_* = 1$ as a function of Re is shown in Fig. 11. The corresponding values of N_S are presented in Fig. 12. Considerable tube length reduction can be achieved for lower Reynolds numbers. When the Reynolds number increases this benefit decreases except for tubes 34 and 33. Examining simultaneously Figs. 11 and 12 one can find out that tubes 13, 23 and 32 have the best characteristics for this case.

Figure 13 represents the variations of N_S with Re_R for case VG-1. In this case the objective is to reduce the surface area $A_* < 1$ with $W_* = 1$ for $Q_* = P_* = 1$. This information can be found in ref. [9] (Fig. 9). The preferable tubes for this case are tubes 14, 16, 24 and 32. They guarantee reduction of surface area of 38–48% in the range of Reynolds numbers studied while the values of N_S are 0.82–0.90.

Figures 14 and 15 illustrate case VG-2a where the objective is increased heat rate $Q_* > 1$ for $W_* = 1$ and

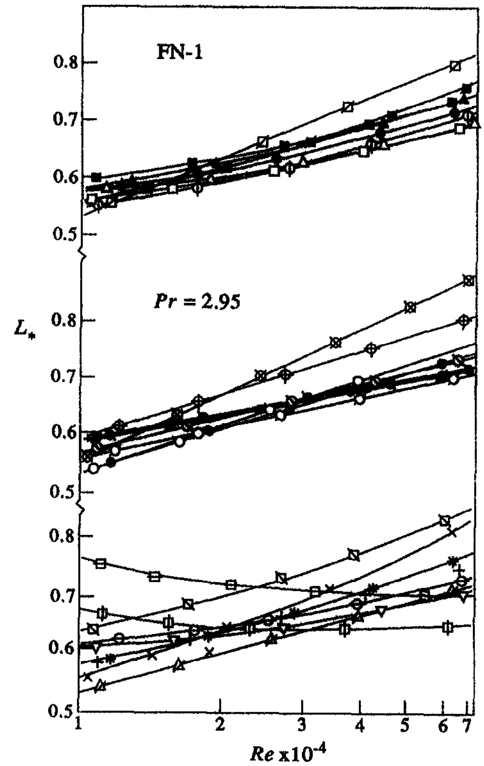

FIG. 11. Reduced tubing length vs Reynolds number.

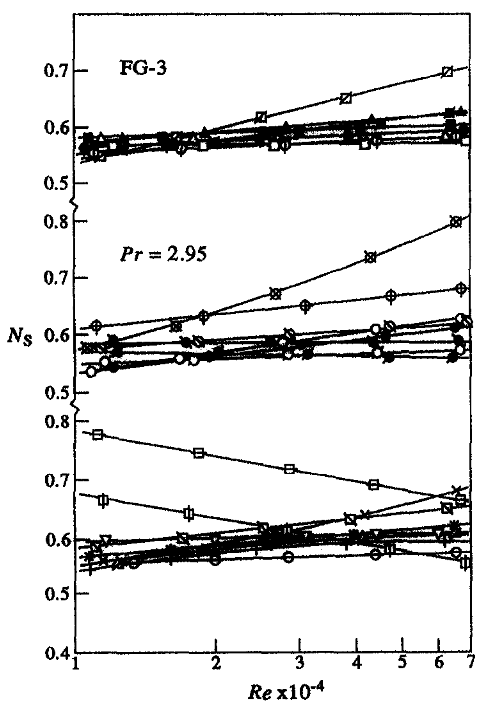

FIG. 10. Augmentation entropy generation number vs Reynolds number.

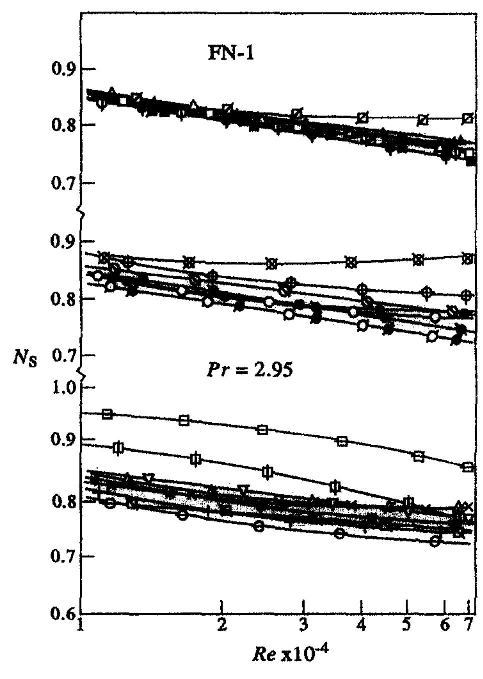

FIG. 12. Augmentation entropy generation number vs Reynolds number.

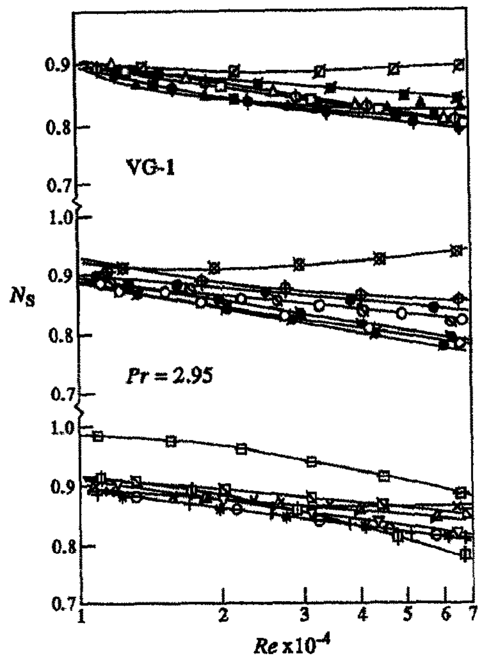

FIG. 13. Augmentation entropy generation number vs Reynolds number.

$A_* = P_* = 1$. Compared to case *FG-2a* [9], Fig. 7, the heat rate can be increased by 20% more compared to case *FG-2a*. This energy payoff however is associated with additional destruction of exergy, Fig. 15.

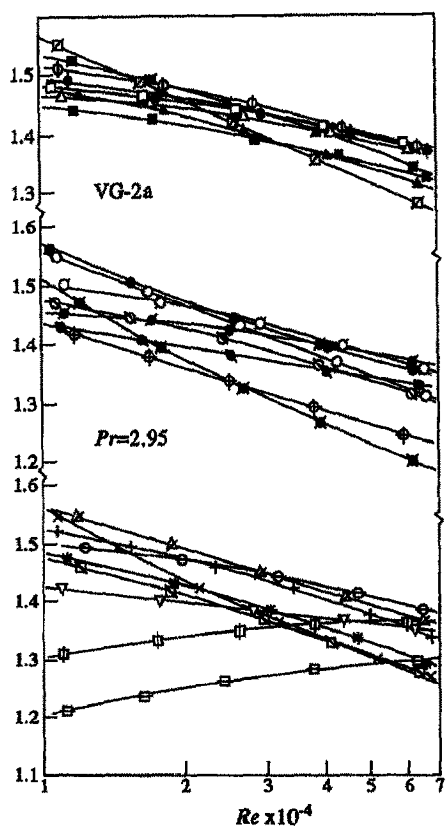

FIG. 14. Increased heat rate vs Reynolds number.

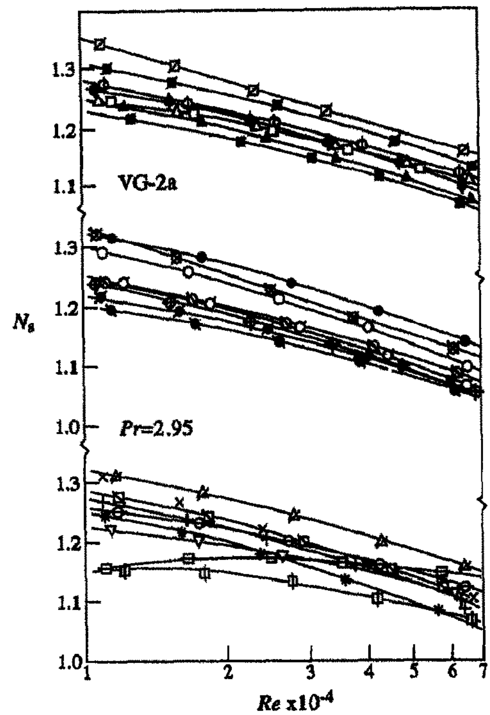

FIG. 15. Augmentation entropy generation number vs Reynolds number.

The best tubes for this case—13, 14, 23, 32—have augmentation entropy generation number $N_S = 1.26$–1.08 in the range of Reynolds numbers studied. The same tubes for case *FG-2a* have $N_S = 0.95$–0.80 for the same range of Reynolds numbers.

CONCLUSIONS

The results of the present study can be summarized as follows:

1. PEC equations have been developed to assess 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 merits of augmentation techniques in connection with the entropy generation and exergy destruction.
2. The heat transfer and fluid friction characteristics of 25 spirally corrugated tubes are used to illustrate the application of the PEC equations. The results for different design constraints show that the evaluation of the heat transfer augmentation techniques should be made on the basis of first and second law analysis simultaneously.
3. The general evaluation criteria may help to display inappropriate enhanced surfaces and assist the engineer to design better heat transfer equipment.

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