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PERFORMANCE EVALUATION CRITERIA FOR USE OF ENHANCED HEAT TRANSFER SURFACES IN HEAT EXCHANGER DESIGN

R. L. WEBB
Department of Mechanical Engineering, Pennsylvania State University,
University Park, PA 16802, U.S.A.

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Abstract—This study extends previous work of Bergles and Webb to establish a broad range of Performance Evaluation Criteria (PEC) applicable to single phase flow in tubes. The equations include the effects of shell-side enhancement and fouling and are applicable to roughness and internally finned tubes. Detailed procedures are outlined to calculate the performance improvement and to select the 'optimum' surface geometry. PEC are presented for four design cases: (1) reduced heat exchanger material; (2) increased heat duty; (3) reduced log-mean temperature difference; and (4) reduced pumping power. The 11 cases discussed include fixed flow area and variable flow area. Appropriate PEC for two-phase exchangers area also discussed.

NOMENCLATURE

a,	heat transfer surface area per unit length of tubing, for internally finned tubing (a_a), for smooth tube (a_s) [$m^2\,m^{-1}$];
A,	heat transfer surface area [m^2];
A_m,	mean conduction surface area [m^2];
B,	external-to-internal surface area ratio;
C_p,	specific heat [$kJ\,kg^{-1}\,K^{-1}$];
D,	tube inside diameter [m];
D_h,	hydraulic diameter [m];
e,	roughness height or fin height [m];
e^+,	roughness Reynolds number $(e/D)Re\sqrt{f/2}$;
E_f,	f/f_s at constant Re;
E_h,	h/h_s at constant Re;
f,	fanning friction factor;
FG,	fixed geometry criterion (defined in Table 1);
FN,	constant flow area criterion (defined in Table 1);
G,	mass velocity [$kg\,m^{-2}$];
h,	heat transfer coefficient [$W\,m^{-2}\,K^{-1}$];
k,	thermal conductivity [$W\,m^{-2}\,K^{-1}$];
L,	length of flow path in heat exchanger [m];
$LMTD$,	log-mean temperature difference [K];
n,	number of fins in internally finned tube, $\pi D_i/p$;
N,	number of tubes in each pass of heat exchanger;
NTU,	number of heat transfer units;
p,	spacing between roughness elements or fins [m];
ΔP,	pressure drop [Pa];
P,	pumping power [W];
Pr,	Prandtl number;
Q,	heat transfer rate [W];
r,	ratio of resistances (defined in Table 2);
Re,	Reynolds number, smooth or rough tube (Du/ν), internally finned tube ($D_h u/\nu$);
R_f,	fouling factor [$m^2\,K\,W^{-1}$];
S',	entropy generation rate per unit length;
St,	Stanton number;
t,	tube wall thickness [m];
ΔT_i,	inlet temperature difference between hot and cold streams [K];
u,	flow velocity [$m\,s^{-1}$];
U,	overall heat transfer coefficient [$W\,m^{-2}\,K^{-1}$];
VG,	variable geometry criterion (defined in Table 1);
w,	tube-side heat exchanger flow rate [$kg\,s^{-1}$];
β,	sum of resistances (defined in Table 2) [$m^2\,K\,W^{-1}$];
ε,	heat exchanger thermal effectiveness;
ν,	Kinematic viscosity [$m^2\,s^{-1}$].

Subscripts

i,	inner tube surface;
o,	outer tube surface;
s,	smooth tube.

INTRODUCTION

THE SUBJECT of enhanced heat transfer has developed to the stage that it is of serious interest for heat exchanger application. Industry has used enhanced heat transfer surfaces to develop more compact and less expensive heat exchangers. An alternate goal is to use such surfaces to increase the system thermodynamic efficiency, which allows reduced operating costs. Extensive data have been published on the basic heat transfer and friction characteristics of the various augmentation techniques. A recent bibliography [1] gives 1967 literature citations. Webb, Bergles and Junkhan [2] identify 329 U.S. Patents on enhanced heat transfer surfaces.

Bergles *et al.* [3, 4] and Webb *et al.* [5, 6] have attempted to simplify this problem by proposing Performance Evaluation Criteria (PEC) which define the performance benefits of an exchanger having enhanced surfaces, relative to a reference exchanger, e.g. one having smooth surfaces. These PEC consider only the thermal performance and material content of the heat transfer surface used in the exchanger. It does not include the complimentary structural materials and flow losses. This provides a method of screening the various enhancement techniques to identify those offering greatest potential. The PEC forms a fundamental 'building block' which may be incorporated in more complex optimization programs which minimize the heat exchanger total cost or maximize the system performance.

Bergles *et al.* PEC [3, 4] define a method to evaluate the comparative performance of existing surfaces for which St and f data are available. Within a class of enhanced surfaces, e.g. $p/e = 10$ transverse-rib roughness, a family of roughness sizes is available. Only one of these roughness sizes will provide the 'optimum' performance for specified operating conditions. The analysis of Webb and Eckert [5] and Webb and Scott [6] select the 'optimum' geometry for the specified operating conditions. PEC, in general, were recently surveyed by Shah [7]. This paper presents a more comprehensive treatise on PEC than provided in the previous references. This work emphasizes single phase flow and defines

(1) Recommended performance objectives and design constraints.

(2) The required algebraic relations for the several PEC.

(3) A method to use the PEC to select the preferred enhancement type and to calculate the 'optimum' geometrical parameters for the preferred enhancement type.

The present PEC are primarily applicable to single-phase flow, since they relate the heat transfer performance to fluid pumping power. They may not be directly applicable to two-phase flow situations if the two-phase pressure drop affects the log-mean temperature difference. This problem will be discussed in a later section.

PERFORMANCE OBJECTIVES AND DESIGN CONSTRAINTS

The performance objectives may be defined such that the augmented exchanger is required to do a better 'job' than the reference design for established design constraints. There are three basic design objectives

(1) Reduced heat transfer surface material for equal pumping power and heat duty. This may be of little interest unless the material reduction is accompanied by reduced manufacturing cost.

(2) Increased UA for equal pumping power and fixed total length of exchanger tubing. A higher UA may be exploited in one of two ways

(a) to obtain increased heat duty for fixed entering fluid temperatures;

(b) to secure reduced $LMTD$ for fixed heat duty.

(3) Reduced pumping power for equal heat duty and total length of exchanger tubing.

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 thermodynamic efficiency, yielding lower system operating cost. Objective (3) offers reduced pumping power cost. Objectives (2b) and (3) are particularly important if 'life cycle' cost analysis is of interest. A more costly augmented surface will be justified if the operating cost savings are sufficiently high.

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 is established by selecting one of the operational variables for the performance objective, subject to design constraints on the remaining variables.

The design constraints placed on the exchanger flow rate and velocity cause key difference among the possible PEC relations. The increased friction factor of 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. For constant tube diameter, this will increase the shell diameter of a shell-and-tube design. However, if the mass flow-rate is reduced, it is possible to maintain constant flow frontal area at reduced velocity. When the exchanger flow rate is reduced, it must operate at higher thermal effectiveness to provide the required heat duty. This may significantly reduce the performance potential of the augmented surface if the design effectiveness is sufficiently high. In many cases the heat exchanger flow rate is specified and thus flow rate reduction is not permitted. The PEC discussed in the next section will account for these various possibilities.

Table 1 gives the recommended PEC for the several cases of interest. The Table segregates the PEC by three different geometrical constraints (FG, FN and VG). Within each of the geometry constraint groupings, PEC are established for three performance objectives: (1) reduced surface area; (2) increased heat duty (or UA); and (3) reduced pumping power. The applications for the various PEC are described below.

Fixed geometry criteria (FG)

These involve a one-for-one replacement of smooth

Table 1. Performance Evaluation Criteria for $D/D_s = 1$

	Fixed						Consequences						
Case	Geom.	W	P	Q	T_i	Objective	$\frac{N}{N_s}$	$\frac{L}{L_s}$	$\frac{W}{W_s}$	$\frac{Re}{Re_s}$	$\frac{P}{P_s}$	$\frac{Q}{Q_s}$	$\frac{\Delta T_i}{\Delta T_{is}}$
FG-1a	N, L†	×			×	$\uparrow Q$	1	1	1	1§	>1	>1	1
FG-1b	N, L	×		×		$\downarrow \Delta T_i$	1	1	1	1§	1	1	<1
FG-2a	N, L		×		×	$\uparrow Q$	1	1	<1	<1	1	>1	1
FG-2b	N, L		×	×		$\downarrow \Delta T_i$	1	1	<1	<1	1	1	<1
FG-3	N, L			×	×	$\downarrow P$	1	1	<1	<1	<1	1	1
FN-1	N		×	×		$\downarrow L$	1	<1	<1	<1	1	1	1
FN-2	N	×		×		$\downarrow P$	1	<1	1	1§	<1	1	1
VG-1		×	×	×	×	$\downarrow NL$	>1*	<1	1	<1*	1	1	1
VG-2a	NL‡	×	×		×	$\uparrow Q$	>1*	1	1	<1*	1	>1	1
VG-2b	NL	×	×	×		$\downarrow \Delta T_i$	>1*	1	1	<1*	1	1	<1
VG-3	NL	×		×	×	$\downarrow P$	>1	1	1	<1*	<1	1	1

* Roughness with high Pr fluids may not require $N/N_s > 1$ (or $Re/Re_s < 1$).
† N and L are held constant in cases FG-1 through FG-3.
‡ The product of N and L are constant in cases VG-2 and 3.
§ For internal roughness. For internal fins, $Re/Re_s = D_h/D$.

tubes by augmented tubes of equal length. These may be regarded as 'retrofit' applications. 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 higher friction characteristic of the augmented surface. The designer should consider the benefits of operating a smooth tube exchanger design at equal pumping power (via increased tube-side velocity) before adopting this approach. The FG-2 criterion has the same objective as FG-1, but requires that the augmented tube design operate at the same pumping power as the reference smooth tube design. The pumping power is maintained constant by reducing the tube-side velocity, and thus the exchanger flow rate. Because FG-2 has a smaller tube-side flow rate, the $LMTD$ will be reduced resulting in a smaller heat exchange capacity than for FG-1. The third criterion (FG-3) attempts to effect reduced pumping power for constant heat duty.

Fixed flow area criterion (FN)

These criteria maintain constant flow area. For a shell-and-tube exchanger having constant diameter tubes, this means the number of tubes and shell diameter are held constant. The objective of FN-1 is reduced surface area, via reduced tubing length, for constant pumping power. Reduced flow rate will probably be required to satisfy the constant pumping power criterion. Augmented tubes are used in FN-2 to obtain reduced pumping power with constant heat duty and flow rate.

Variable geometry cases (VG)

In most cases, a heat exchanger is 'sized' for a required thermal duty with specified flow rate. In these situations the FG and FN criteria are not applicable. Because the tube-side velocity must be reduced to accommodate the higher friction characteristic of the augmented surface, it is necessary to increase the flow area to maintain constant flow rate. This is accomplished using a greater number of tubes in parallel, or by using the same number of larger diameter tubes. Maintenance of constant exchanger flow rate avoids the penalty due to operating at higher thermal effectiveness encountered in the previous FG and FN cases. However, the shell diameter must be increased which will add to the exchanger cost. It may not be necessary to increase the number of exchanger tubes if the reference exchanger is of multipass design. For example, a two-pass smooth tube design may be replaced by a one-pass augmented design using a 50% velocity reduction. The criteria VG-1, 2 and 3 correspond directly to FN-1, FN-2 and FG-3 with the only difference being the flow rate reduction of the FN cases.

The PEC listed in Table 1 compare to the previously presented criteria by Bergles *et al.* [3] and by Webb and Eckert [5] as follows. The correspondence for the Bergles *et al.* criteria [3] are: FG-1(B-1), FG-2(B-3), FG-3(B-4), VG-2(B-8). Bergles *et al.* criteria B-5 corresponds to FN-1. Bergles *et al.* criteria B-2 and B-6 correspond to FG-2 and FN-1 if pressure drop rather than pumping power is the constraint. Bergles *et al.* B-7 criterion does not correspond to any of those in Table 1. It is a FN-type criterion with fixed heat duty and constant exchanger flow rate. In this case the augmented exchanger will have a higher pressure drop than the reference exchanger, for the same flow area. The A, B and C criteria of Webb and Eckert correspond with the VG criteria as follows: A(VG-2), B(VG-1) and C(VG-3).

ALGEBRAIC FORMATION OF THE PEC

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. For generality, the PEC must account for several important factors:

(1) 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.

(2) A tube side fouling resistance.

(3) The thermal resistance across the metal tube wall.

(4) The possibility of enhancement on the inner and outer tube surfaces for a two-fluid heat exchanger.

The PEC equations of [2–5] do not include all of the above factors. The following development basically parallels the special cases of [5] and [6] but are now presented in the generalized form.

For simplicity, the equations are developed for tubes of constant diameter, and heat transfer and friction factors are based on the nominal diameter. The modifications necessary for internally finned tubes will be discussed in a later section.

The relative friction power equation for the tube side fluid is

$$\frac{P}{P_s} = \frac{f}{f_s}\frac{A}{A_s}\left[\frac{G}{G_s}\right]^3. \tag{1}$$

The UA equations for the reference (smooth tube) and augmented exchangers are

$$\frac{1}{U_sA_s} = \frac{1}{h_sA_s} + \frac{1}{h_{0s}A_{0s}} + \frac{t}{kA_m} + \frac{R_f}{A_s} \tag{2a}$$

$$\frac{1}{UA} = \frac{1}{hA} + \frac{1}{h_0A_0} + \frac{t}{kA_m} + \frac{R_f}{A}. \tag{3a}$$

The resulting development is more easily understood if the various resistance terms are written as ratios. The resistance terms are normalized by the tube side heat conductance (hA product). Table 2 defines the dimensionless ratios.

The analysis includes the possibility that the augmented exchanger may have an enhanced outer tube surface.

$$E_{h0} \equiv h_0/h_{0s}. \tag{4}$$

Table 2. Dimensionless ratios to be used in equations (1), (2a) and (3a)

Definition	Reference H-X	Augmented H-X
Surface area ratio	$B_s = A_{0s}/A_s$	$B = A_0/A$
Outer surface conductance	$r_s = h_s/B_s h_{0s}$	$r = r_s B_s/B$
Metal resistance	$r_{ws} = h_s t\, A_s/kA_m$	$r_w = htA/kA_m$
Fouling resistance	$r_{fs} = h_s R_f$	$r_f = hR_f$
Composite resistance	$\beta_s = r_s + r_{ws} + r_{fs}$	$\beta = \dfrac{r}{E_{h0}} + r_{ws}\dfrac{A}{A_s} + r_{fs}$

The simplified UA equations result from substituting the Table 2 definitions and equation (4) in equations (2a) and (3a).

$$\frac{1}{U_s A_s} = \frac{1}{h_s A_s}[1+r_s+r_{ws}+r_{fs}] \tag{2b}$$

$$\frac{1}{UA} = \frac{1}{hA}\left\{1 + \left[\frac{r_s}{E_{h0}}\cdot\frac{B_s}{B} + r_{ws}\frac{A}{A_s} + r_{fs}\right]\frac{h}{h_s}\right\}. \tag{3b}$$

Dividing equation (2b) by (3b) and including the β-definitions gives UA/U_sA_s.

$$\frac{UA}{U_sA_s} = \frac{hA}{h_sA_s}\left[\frac{1+\beta_s}{1+\dfrac{h}{h_s}\beta}\right]. \tag{5a}$$

Using $h \propto StG$ in equation (5a) we obtain

$$\frac{UA}{U_sA_s} = \frac{1+\beta_s}{\dfrac{St_s}{St}\dfrac{G_s}{G}\dfrac{A_s}{A} + \beta\dfrac{A_s}{A}} \tag{5b}$$

the G_s/G is eliminated from equation (5b) by substituting (1)

$$\frac{UA}{U_sA_s} = \frac{1+\beta_s}{\dfrac{St_s}{St}\left\{\dfrac{f}{f_s}\cdot\dfrac{P_s}{P}\left[\dfrac{A_s}{A}\right]^2\right\}^{1/3} + \beta\dfrac{A_s}{A}}. \tag{6}$$

Equations (1) and (6) give the UA and pumping power (P) ratios for an augmented exchanger, relative to a smooth tube exchanger which operates at specified G_s. The terms β and β_s contain known or specified information. The parameters of interest in equation (6) are A/A_s, P/P_s and UA/U_sA_s. For each case of interest in Table 1, two of the parameters are set as constraints and the third is the objective function. For example, consider case FG-2 for which $P/P_s = A/A_s = 1$ and the objective is $UA/U_sA_s > 1$. Equation (6) becomes

$$\frac{UA}{U_sA_s} = \frac{1+\beta_s}{\dfrac{St_s}{St}\left[\dfrac{f}{f_s}\right]^{1/3} + \beta} \tag{7}$$

the G/G_s is obtained from equation (1)

$$\frac{G}{G_s} = \left[\frac{f}{f_s}\right]^{1/3}. \tag{8}$$

These equations may be solved for specified G_s once appropriate equations are established for f and St of the augmented surface.

The internal surface area ratio (for roughness) and W/W_s are given by equation (9) and (10), respectively.

$$\frac{A}{A_s} = \frac{N}{N_s}\cdot\frac{L}{L_s} \tag{9}$$

$$\frac{W}{W_s} = \frac{N}{N_s}\cdot\frac{G}{G_s}. \tag{10}$$

CALCULATION OF St AND f

Solution of the PEC equations requires correlations for the St and f of the augmented surfaces. Assume that St and f are known as a function of Reynolds number. In most of the Table 1 PEC, $Re \neq Re_s$ and Re/Re_s is dependent on St/St_s and f/f_s. Therefore, it is necessary to solve equations (1) and (6) by iteration.

The general problem is more complex. Assume that the designer has selected a particular type of augmented surface for evaluation, say $p/e = 10$ transverse rib roughness. By varying the roughness height, a wide range of geometrically similar roughness sizes are possible. In conceptual terms, visualize 10 tubes, each having a different e/D with $p/e = 10$. Which of these tubes will yield the greatest performance improvement relative to the smooth tube exchanger designed to operate at Re_s? Several methods may be employed to solve equations (1) and (7).

(1) Rough surface data are normally correlated as a function of the 'roughness Reynolds number' (e^+), rather than the pipe Reynolds number (Re). A correlation does not exist in the form $f(Re, e/D)$ and $St(Re, Pr, e/D)$, to use in equations (1) and (7). Webb and Eckert [5] outline a design procedures for this situation, which employs the generalized St and f correlations in terms of e^+. The procedure defines the specific roughness size (e/D) that yields the maximum (or minimum) value for the objective function.

(2) Alternatively, one may use the generalized St and f vs e^+ correlations to generate curves of St and f vs Re for each roughness size. An empirical curve fit for each specific geometry may then be used to obtain the iterative solution of equations (1) and (7).

(3) Bergles *et al.* [3] propose a procedure which specifies Re rather than Re_s. At the specified Re, the f

and St of a specific augmented surface are obtained by any appropriate means. Algebraic forms for $f_s(Re_s)$ and $St_s(Re_s)$ are used in equations (1) and (6) to solve for Re_s. Although this is a simple method, it does not allow independent specification of Re_s, and Re_s will be different for each roughness size evaluated. The inability to specify constant Re_s as the major disadvantage of this method. In order to make the performance comparison for Re_s = constant, it would be necessary to repeat the solution for each roughness size at several values of Re and obtain an interpolated value for the objective function at the specified Re_s.

(4) If the augmented and smooth surfaces have the same Reynolds number exponent $f/f_s \equiv E_{fi}$ and $St/St_s \equiv E_{hi}$ when $Re = Re_s$. Using the Blasius equation, ($f_s = 0.046\, Re_s^{-0.2}$) and the Dittus–Boelter equation ($Nu_s = 0.023\, Re_s^{0.8}\, Pr^{0.4}$) we may write

$$\frac{f}{f_s} = E_f\left[\frac{Re_s}{Re}\right]^{0.2} \tag{11}$$

$$\frac{St}{St_s} = E_{hi}\left[\frac{Re_s}{Re}\right]^{0.2}. \tag{12}$$

These two equations are sufficient to solve equations (1) and (6) for specified Re_s. Normally, the rough and smooth surfaces do not have the same Reynolds number exponent so this method is of limited usefulness for roughness. However, this method may be applied to internally finned tubes in turbulent flow using correlations for f and St developed by Carnavos [8].

HEAT EXCHANGER EFFECTIVENESS

The augmented and smooth exchangers may not operate at the same effectiveness (ε). For these cases when the objective is increased heat duty, the ε-NTU design method gives

$$\frac{Q}{Q_s} = \frac{W}{W_s}\frac{\varepsilon}{\varepsilon_s}\frac{\Delta T_i}{\Delta T_{is}} \tag{13a}$$

where ΔT_i is the temperature difference between the two inlet streams. For fixed inlet temperatures ($\Delta T_i = \Delta T_{is}$), equation (13a) yields

$$\frac{Q}{Q_s} = \frac{W}{W_s}\cdot\frac{\varepsilon}{\varepsilon_s}. \tag{13b}$$

Since the operating conditions of the smooth tube exchanger are known, its $NTU_s = NTU_s A_s/W_s C_p$ is known and ε_s is calculable. Once UA/U_sA_s and W/W_s for the augmented exchanger are known, its NTU is calculated by

$$NTU = NTU_s\frac{UA}{U_sA_s}\cdot\frac{W_s}{W}. \tag{14}$$

Then the ε of the augmented exchanger may be calculated, and Q/Q_s obtained from equation (13b).

When the objective is increased UA with $Q/Q_s = 1$, $\Delta T_i/\Delta T_{is} < 1$. By equation (13a)

$$\frac{\Delta T_i}{\Delta T_{is}} = \frac{\varepsilon_s}{\varepsilon}\cdot\frac{W_s}{W}. \tag{15}$$

MODIFIED EQUATIONS FOR INTERNALLY FINNED TUBES

Equation (6) requires minor modification for application to internally finned tubes. The differences arise because

(1) The heat transfer coefficient is based on the total internal surface area.

(2) The friction factor and Reynolds number are based on the hydraulic diameter (D_h).

(3) The flow area for equal nominal diameter is reduced due to the cross-sectional area of the fins.

Webb and Scott (6) derived the equations for internally finned tubes. The results are

$$\frac{A}{A_s} = \frac{NL}{N_sL_s}\cdot\frac{a_a}{a_s} \tag{16}$$

$$\frac{UA}{U_sA_s} = \frac{1+\beta_s}{\dfrac{St_s}{St}\cdot\dfrac{G_s}{G} + \beta\dfrac{a_a}{a_s}} \tag{17}$$

$$\frac{UA}{U_sA_s} = \frac{1+\beta_s}{\dfrac{St_s}{St}\left\{\dfrac{f}{f_s}\dfrac{P_s}{P}\left[\dfrac{A_s}{A}\right]^2\right\}^{1/3} + \beta\dfrac{A_s}{A}\dfrac{a_a}{a_s}}. \tag{18}$$

Using Carnavos' correlation for internally finned tubes (8) the St and f are given in the form of equations (11) and (12).

APPLICATION OF PEC EQUATIONS TO TABLE 1 CASES

The PEC equations will now be used to derive algebraic relations for each of the cases in Table 1. The developments are shown only for the roughness case (equations (1) and (6)). Parallel development may be made for internal fins using equation (1) and (16) through (18).

Case FG-1

For this case $W/W_s = N/N_s = L/L_s = 1$. Since $G/G_s = 1$, the f and St are directly calculable. Equations (1) and (5b) give

$$\frac{UA}{U_sA_s} = \frac{1+\beta_s}{\dfrac{St_s}{St} + \beta} \tag{19a}$$

$$\frac{P}{P_s} = \frac{f}{f_s}. \tag{19b}$$

Using UA/U_sA_s and P/P_s from equations (19a) and (19b), the NTU is given by equation (14) after which ε is calculated. If the entering fluid temperatures are fixed, equation (13b) gives $Q/Q_s = \varepsilon/\varepsilon_s$. If the alternate objective ($UA/U_sA_s > 1$ for $Q/Q_s = 1$) is desired, the initial temperature difference (ΔT_i) may be reduced and is given by equation (15).

Case FG-2

This case has the same geometry and objective constraints as FG-1, but is more restrictive since $P/P_s = 1$. Since $N/N_s = L/L_s$, a reduced flow rate is necessary to satisfy $P/P_s = 1$. Equations (1) and (6) yield

$$\frac{G}{G_s} = \left[\frac{f_s}{f}\right]^{1/3} \tag{20a}$$

$$\frac{UA}{U_sA_s} = \frac{1+\beta_s}{\frac{St_s}{St}\left[\frac{f}{f_s}\right]^{1/3} + \beta}. \tag{20b}$$

These equations must be solved by an iterative process, since G/G_s is unknown.

Case FG-3

Here, $A/A_s = Q/Q_s = 1$ and $N/N_s = L/L_s = 1$. To satisfy $Q/Q_s = 1$, the augmented exchanger must operate with $\varepsilon/\varepsilon_s > 1$ since $W/W_s < 1$. The equations to be solved are

$$\frac{UA}{U_sA_s} = \frac{1+\beta_s}{\frac{St_s}{St}\left[\frac{f}{f_s}\frac{P_s}{P}\right]^{1/3} + \beta} \tag{21a}$$

$$\frac{P}{P_s} = \frac{f}{f_s}\left[\frac{G}{G_s}\right]^3 \tag{21b}$$

$$\frac{Q}{Q_s} = 1 = \frac{G}{G_s}\cdot\frac{\varepsilon}{\varepsilon_s}. \tag{21c}$$

The solutions method is: (1) assume G/G_s; (2) solve for P/P_s, (3) obtain UA/U_sA_s from equation (21a); (4) calculate NTU from equation (14); (5) determine ε and ε_s; and (6) calculate Q/Q_s from equation (21c). Repeat the iterative solution using different values of G/G_s until equation (20c) yields $Q/Q_s = 1$. When G/G_s is converged, P/P_s is given by equation (21c).

Case FN-1

This case fixes $P/P_s = Q/Q_s = N/N_s = 1$. The objective is reduced surface area via $L/L_s < 1$. The equations to be solved are

$$\frac{UA}{U_sA_s} = \frac{1+\beta_s}{\frac{St_s}{St}\left[\frac{f}{f_s}\right]^{1/3}\left[\frac{L_s}{L}\right]^{2/3} + \beta\frac{L_s}{L}} \tag{22a}$$

$$\frac{P}{P_s} = 1 = \frac{f}{f_s}\frac{L}{L_s}\left[\frac{G}{G_s}\right]^3 \tag{22b}$$

$$\frac{Q}{Q_s} = 1 = \frac{G}{G_s}\frac{\varepsilon}{\varepsilon_s}. \tag{22c}$$

The solution procedure is similar to that for case FG-3. Substitution of equation (22b) in (22a) gives

$$\frac{UA}{U_sA_s} = \frac{1+\beta_s}{\frac{f}{f_s}\cdot\frac{St_s}{St}\left[\frac{G}{G_s}\right]^2 + \beta\frac{f}{f_s}\left[\frac{G}{G_s}\right]^3}. \tag{22d}$$

The solution method is: (1) assume G/G_s; (2) solve for UA/U_sA_s from equation (22d); (3) calculate NTU from equation (14); and (4) calculate Q/Q_s from equation (22c). Repeat the iterative process on G/G_s until equation (22c) yields $Q/Q_s = 1$. Then L/L_s is given by equation 22b.

Case FN-2

This case fixes $Q/Q_s = N/N_s = 1$ and seeks $P/P_s < 1$. It is similar to case FG-3, except it maintains $W/W_s = 1$ and yields a reduced exchanger length as well as reduced pumping power. Since $G/G_s = 1$, the St and f are directly calculable and equation (5b) gives

$$\frac{U}{U_s} = \frac{1+\beta_s}{\frac{St}{St_s} + \beta} \tag{23a}$$

then

$$\frac{L}{L_s} = \frac{U_s}{U} \tag{23b}$$

and

$$\frac{P}{P_s} = \frac{f}{f_s}\cdot\frac{U_s}{U}. \tag{23c}$$

The P/P_s equation shows it is not possible to obtain $P/P_s < 1$ for a two-fluid heat exchanger. This is because $f/f_s > U/U_s$. For a prescribed heat flux boundary condition $U/U_s = h/h_s$. Therefore, $P/P_s < 1$ would be obtained if $f/f_s < h/h_s$. Rough surfaces typically yield $f/f_s > h/h_s$.

Case VG-1

All of the VA cases maintain $W/W_s = 1$ and permit the exchanger frontal area (N/N_s) to vary in order to meet the pumping power constraint. Case VG-1 yields reduced surface area for $Q/Q_s = P/P_s = 1$ since $\varepsilon = \varepsilon_s$, Q/Q_s is satisfied by $UA/U_sA_s = 1$. The equations to be solved are

$$1 = \frac{1+\beta_s}{\frac{St_s}{St}\left\{\frac{f}{f_s}\left[\frac{A_s}{A}\right]^2\right\}^{2/3} + \beta\frac{A_s}{A}} \tag{24a}$$

$$1 = \frac{f}{f_s}\frac{A}{A_s}. \tag{24b}$$

The solution is obtained by (1) assume G/G_s, and calculate f/f_s and St/St_s; (2) obtain A/A_s from equation (24b); and (3) calculate UA/U_sA_s from equation (24a). When the solution is converged for G/G_s, equations (9) and (10) give N/N_s and L/L_s.

Case VG-2

This case seeks increased thermal performance (UA/U_sA_s or $Q/Q_s > 1$) for $A/A_s = P/P_s = 1$. This case is similar to case FG-2 and requires solution of equation (19a) and (19b). Having obtained UA/U_sA_s, the NTU is given by equation (14) and ε and ε_s may be determined. Equation (15) ($W/W_s = 1$) gives the permissible reduction of ΔT_i and equation (13b) ($W/W_s = 1$) gives Q/Q_s. The N/N_s is again given by equation (9).

For internally finned tubes the geometry constraint is $NL/N_sL_s = 1$ (fixed total tubing length in the exchanger).

Case VG-3

Reduced pumping power is desired for $A/A_s = Q/Q_s = 1$. This case is similar to case FG-3, although considerably more simple to solve since $W/W_s = 1$, the $Q/Q_s = 1$ is satisfied by $UA/U_sA_s = 1$. By equation (5b)

$$1 = \frac{1 + \beta_s}{\frac{St_s}{St}\frac{G_s}{G} + \beta}. \tag{25a}$$

Equation (24a) may be solved for G/G_s

$$\frac{G}{G_s} = \frac{St_s}{St}\frac{\beta}{1 + \beta_s}. \tag{25b}$$

This equation may be solved for G/G_s using the known $St(Re)$ relation for the augmented surface. Then f/f_s is calculated and P/P_s is given by equation (21b).

SELECTION OF 'OPTIMUM' ENHANCEMENT GEOMETRY

The procedure outlined may be employed to establish the performance benefits of an enhanced tube of given dimensions. However, if the 'optimum' performance benefits of a particular enhancement type is to be realized, further evaluation is required. The designer should establish the height (e/D) and spacing (p/e) of the roughness or internal fins. In addition, the helix angle of two-dimensional ribs or internal fins will have a significant effect or performance. The effect of these geometric parameters were evaluated in [5, 9 and 10] for two-dimensional rib roughness and sand-grain type roughness. Webb and Scott [6] establishes preferred fin spacings and heights for internally finned tubes.

It is important to understand that the preferred size of a roughness geometry is dependent on the operational Reynolds number, as discussed by Webb [9]. As the Reynolds number increases, the preferred roughness size becomes smaller.

Finally, the designer may wish to know which basic type of enhancement will yield the greatest performance benefit. To properly answer this question, the comparison of the various enhancement types should be based on the 'optimum' dimensions for each of the enhancement types compared. Little work has been done on this question. Webb and Hong [11] compare the performance of nine 'optimized' enhancement types for one application. And, the performance of 'optimized' $p/e = 10$ transverse-rib roughness and internal fins are compared in [6].

RESULTS OF PEC ANALYSIS

The results of Webb and Hong [11] are presented to illustrate the use of the PEC equations to identify preferred enhancement types and optimum dimensions for each enhancement type. The reference ex-

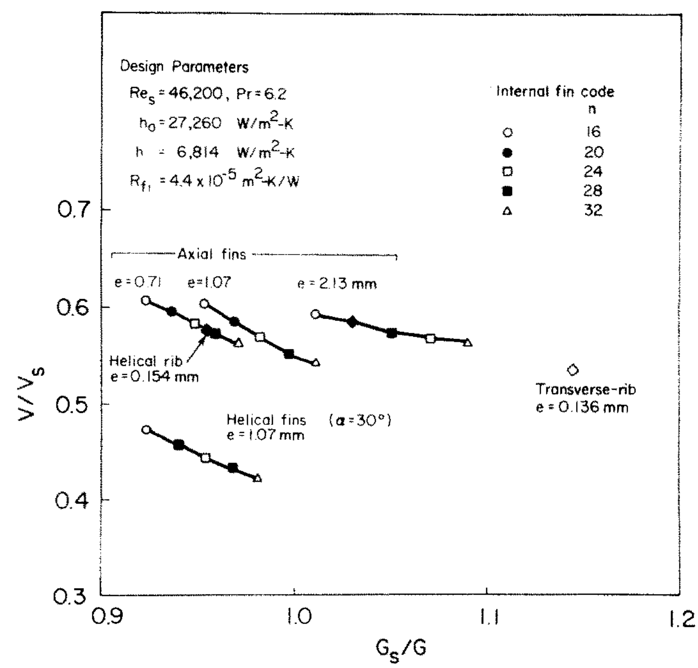

FIG. 1. V/V_s vs G_s/G for several water-side enhancements. $P/P_s = Q/Q_s = 1$, with $w/w_s = 1$.

changer (smooth inner tube surface) boils ammonia at 22.2°C using a porous boiling surface on the tube outside diameter, with 26.7°C cooling water at $Re_s = 46\,200$ on the tube side. The exchanger uses aluminum tubes (25.4 mm I.D. with 1.6 mm wall thickness). The inside-to-outside thermal resistance is $h_0A_0/h_sA_s = 1.13 \times 27\,260/6814 = 4.52$. Because the dominant resistance is on the tube side, enhancement should be beneficial. The study determined the possible heat exchanger tube material reduction due to use of tube-side enhancement for $P/P_s = Q/Q_s = 1$ with constant flow rate ($w/w_s = 1$). This corresponds to case VG-1 of Table 1. The study selected the optimum dimensions of several enhanced surface types using a water-side fouling resistance of $R_{fi} = 4.4 \times 10^{-5}\ \mathrm{m^2\,K\,W^{-1}}$. The enhancement types evaluated are

(1) Two-dimensional transverse-rib roughness with $p/e = 10$ [5].

(2) Helical-rib roughness with $\alpha = 49°$ [10].

(3) Axial internal fins [6].

(4) Helical internal fins with $\alpha = 30°$ [10].

A parametric analysis was performed to establish the optimum fin or roughness height (e) and the number of fins (n) which yields the minimum tube material volume ratio (V/V_s).

Internal-fin analysis

The analysis method is described by Webb and Scott [6]. A solution is generated for each fin height and spacing following the procedure described for case VG-1. Equation (18) (with $P/P_s = 1$) is employed, rather than equation (24), which is for roughness. The f/f_s and St/St_s are obtained from equations (11) and (12), using E_f and E_{hi} values from Carnavos correlation [8]. The fin height-to-thickness ratio is 2.5, and fin efficiency effects are included in the analysis. Figure 1 shows the results. The upper solid curves show V/V_s vs G_s/G for axial fins with three different fin heights. The smallest value of V/V_s (0.54) is obtained using $e = 1.07$ mm and $n = 32$ fins. The lower curve shows the results for 30 helical fins 1.07 mm high. In this case, 32 fins yield $V/V_s = 0.42$. Therefore, the optimum internal fin geometry has 32 helical fins ($\alpha = 30°$) with 1.07 mm fin height. The tube side velocity is approximately equal to that of the smooth tube design ($G_s/G = 0.98$), so no increase of heat exchanger diameter is required.

Internal roughness analysis

The St and f characteristics were obtained from [5] for transverse-rib roughness and from [10] for the helical-rib. These references use the roughness Reynolds number (e^+) rather than the pipe Reynolds number (Re) as the independent variable. Previous analysis [5, 11] has shown that the optimum roughness size (e/D) occurs for $e^+ = 20$. If e^+ is specified, the e/D required to satisfy equations (24a) and (24b) may be directly calculated, although an iterative solution is required [5]. The V/V_s for the two roughness geometries operated at $e^+ = 20$ is shown by the noted points on Fig. 1. Both roughness types provide material savings approximately equal to that of the axial internal fins. The helical-rib roughness may be operated at a higher velocity than the transverse-rib roughness, which allows a smaller diameter. The roughness heights are 0.14 mm (transverse-rib) and 0.15 mm (helical-rib). Note that the optimum roughness size is quite small.

EFFECT OF REDUCED EXCHANGER FLOW RATE

The FN and FG cases maintain $N/N_s = 1$. For $P/P_s = 1$, it is necessary to operate at reduced exchanger flow rate ($W/W_s < 1$). The reduced flow rate of these cases penalize the augmented exchanger. Such a penalty does not occur for the VG-cases, which maintain $W/W_s = 1$. Let us compare this effect for the FN-1 and VG-1 cases which seek reduced surface area for $Q/Q_s = P/P_s = 1$. Because the FN exchanger operates at higher thermal effectiveness, it will require additional surface area to compensate for the reduced log-mean temperature difference. If ΔT_i = constant, equation (13b) shows

$$\frac{Q}{Q_s} = \frac{W}{W_s}\,\frac{\varepsilon}{\varepsilon_s}. \tag{26}$$

Figure 2 may be used to illustrate the penalty associated with the $W/W_s < 1$ exchanger. This figure is constructed for an evaporator or condenser ($C_{min}/C_{max} = 0$) which operates with constant UA and ΔT_i. The ε is given by $\varepsilon = 1 - \exp(-NTU)$. To obtain $Q/Q_s = 1$, for U = constant, the surface area of the $W/W_s < 1$ exchanger must be increased by the ratio Q_s/Q, relative to that for the $W/W_s = 1$ exchanger. The area penalty increases with larger NTU of the reference smooth tube exchanger. The $W/W_s < 1$ and $W/W_s = 1$ augmented exchangers have the same U-value if $\Delta P/\Delta P_s = 1$. If $P/P_s = 1$, the $W/W_s = 1$ exchanger will have a slightly higher U-value.

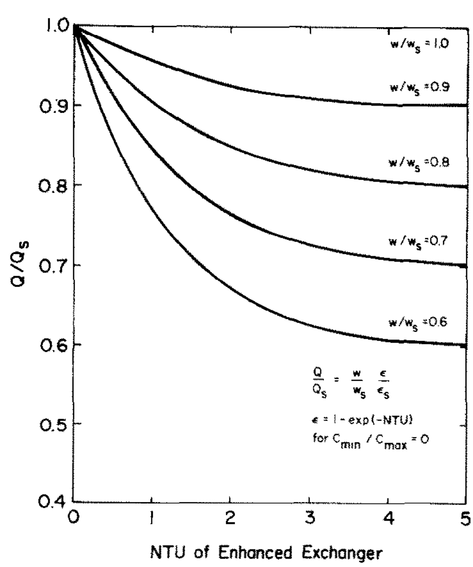

FIG. 2. Effect of reduced flow rate on Q/Q_s for $UA/U_sA_s = 1$.

To summarize the concept, assume that the smooth tube exchanger is designed for $\varepsilon_s = 0.632$ ($NTU_s = 1.0$) and the augmented exchanger provides a 50% larger U-value. For the VG-case ($W/W_s = 1$), $U/U_s = 1.5$ yields $A/A_s = 1/1.5$. If $W/W_s = 0.8$ for the FN-case, Fig. 2 shows $Q/Q_s = 0.903$. The FN-exchanger will require $A/A_s = 0.667/0.903 = 0.74$ to satisfy $Q/Q_s = 1$. Thus, the VG exchanger provides 10% greater surface area reduction than allowed by the FN-exchanger.

PEC FOR TWO-PHASE FLOW

The PEC used for single-phase flows may not be the most appropriate for two-phase exchangers. In two-phase exchangers, pressure drop of the phase change fluid may cause reduced LMTD. A good example is that of a direct-expansion refrigeration evaporator with tube-side evaporation. Because the compressor suction pressure is fixed, refrigerant pressure drop reduces the LMTD and affects reduced evaporator heat duty. An appropriate PEC should establish the refrigerant flow rate which yields maximum evaporator load. This criterion is employed in the analysis of test results on internally roughened tubes by Withers and Habdas [12], and by Lord, Bussjager and Geary [13]. Similar arguments may be stated for refrigeration condensers having fixed compressor discharge pressure. Alternatively, the condenser PEC may be based on fixed LMTD with consideration of the increased compressor power required to compensate for the increased pressure drop of the condensing vapor. These considerations may be more important if the two-phase pressure drop is greater for flow in tubes than on the shell side. Significant pressure loss increase has not been reported for use of shell-side enhanced nucleate boiling surfaces. Specific PEC have not been proposed for two-phase situations. However, Royal and Bergles [14] use the single-phase PEC to evaluate several surface geometries for in-tube condensation. By comparing the tube geometries for fixed pressure drop, they maintain constant LMTD and compressor power for a refrigeration application.

OTHER PERFORMANCE EVALUATION METHODS

In the introduction, we noted that Shah [7] has recently surveyed the subject. Table 1 of Shah's paper describes 19 different methods, which have been proposed by others. Further study of the literature will reveal additional proposed methods. For completeness, a comparison between the PEC given here and those presented by others is given. Due to space limitations, the comparison will be limited.

In 1957, Le Foll [15] developed a PEC which plots $(St/St_s)^3/(f/f_s)$ vs Re/Re_s. Cases FG-2a and VG-2a agrees with the Le Foll criteria if $\beta = \beta_s = 0$. In this case equation (20b) gives $(St/St_s)/(f/f_s)^{1/3} = hA/h_sA_s$. This form has been frequently used to evaluate the performance of externally roughened tubes used in a rod-bundle configuration for high temperature gas cooled reactors (HTGR), [16, 17]; Williams, Pirie and Warburton [16] define this as the "Thermal Performance Ratio". Similarly, White and Wilkie [17] use the form $(f/f_s)/(St/St_s)^3 = P/P_s$ to define the pumping power reduction afforded by rough surfaces in the HTGR application. This form is obtained from case FG-3 and VG-3 for $\beta = \beta_s = 1$, which defines the P/P_s ratio for $A/A_s = 1$.

Those interested in comparing the performance of surface geometries for compact heat exchangers (CHE) have developed comparison methods related to those in the present paper. This topic is the main focus of Shah's paper [7]. Shah uses the term "Goodness Factor", rather than the term PEC used here. A goodness factor defines the performance improvement (e.g. A/A_s or h/h_s) for fixed operating constraints (e.g. $P/P_s = 1$) and fixed geometry constraints (e.g. heat exchanger volume). Such analyses typically compare the performance of two surfaces and ignore the effect of the other resistances that would be present in an actual heat exchanger (the β and β_s terms of the present analysis). Typical analyses of this type are given by Kays and London (18), Le Haye, Neugbaur and Sakhuja [19], Cox and Jallouk [20] and Soland, Mack and Rohsenow [21]. Soland, Mack and Rohsenow compare the performance of different CHE surfaces, relative to that of a reference surface for several operating conditions and geometry constraints. Their analysis is similar to the cases presented here for $\beta = \beta_s = 0$.

The present analysis differs from the above approaches in two major respects: (1) we account for the effect of the composite thermal resistances (β and β_s) which occur in a two-fluid exchanger; (2) rather than comparing the relative performance of a given enhanced surface with a smooth surface, we show how to select the 'optimum' geometric parameters for any basic type of enhanced surface.

A different approach to PEC analysis is described by Bejan and Pfister [22] and Oulette and Bejan [23]. This approach compares the entropy generation rate (S') of the enhanced and smooth tubes. The entropy generation is the sum of the irreversibilities due to heat transfer and fluid friction. A preferred geometry yields $S'/S'_s < 1$. References [22] and [23] show calculated values of S/S'_s for several types of tube side enhancements. The analysis is performed for w/w_s and $Q/Q_s = 1$, which corresponds to case FG-1b of Table 1. Further, the analysis does not include the β, β_s terms in equation (1).

CONCLUSIONS

This paper provides a comprehensive treatise on Performance Evaluation Criteria (PEC) to assess the performance advantages offered by enhanced heat transfer surfaces. Four possible performance objectives are discussed and applied to eleven cases of interest. The eleven cases include three different geometric constraints on the heat exchanger geometry. The detailed equations necessary to quantify the performance benefits are developed, and step-by-step procedures are given for solution of the equations. The modified equations necessary to evaluate internally finned tubes are also included. The importance of using the PEC to select 'optimum' dimensions for the enhanced surface is stressed. Several of the PEC allow the flow rate of the enhanced exchangers to be reduced, which may severely penalize its performance benefits. A method is presented to evaluate the performance penalty due to reduced flow rate. Finally, modified PEC applicable to heat exchangers having two-phase flow are discussed.

Acknowledgements—I would like to express my appreciation to Professor A. E. Bergles for the many stimulating discussions we have had on the subject of Performance Evaluation Criteria. Table 1 includes a number of cases originally proposed by Professor Bergles, and it includes other information which he has suggested in our discussions. This study was performed as part of the work in Department of Energy Contract EG-78-S-02-4649.

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CRITERES D'EVALUATION DE PERFORMANCE POUR DES SURFACES A BONNE QUALITE D'ECHANGE THERMIQUE, DANS LA CONCEPTION DES ECHANGEURS DE CHALEUR

Résumé—Cette étude élargit un travail de Beigles et Webb pour établir des critères d'évaluation de performance (PEC) applicable à l'écoulement monophasique dans les tubes. Les équations tiennent en compte les effets d'accroissement du côté de la virole et l'encrassement et elles sont applicables aux tubes rugueux et à ailettes internes. Des procédures détaillées sont établies pour calculer l'accroissement de performances et pour choisir la géométrie optimale de surface. Des PEC sont présentés pour quatre cas : (1) réduction de matière, (2) accroissement de puissance thermique, (3) réduction de la moyenne logarithmique de température, (4) réduction de la puissance de pompage. Les onze cas discutés concernent une section de passage fixée ou au contraire variable. On discute aussi les PEC dans le cas des échangeurs diphasiques.

BEWERTUNGSKRITERIEN FÜR DEN EINSATZ VON VERBESSERTEN WÄRMEÜBERTRAGUNGSFLÄCHEN BEIM ENTWURF VON WÄRMEAUSTAUSCHERN

Zusammenfassung—Die Studie setzt frühere Arbeiten von Bergles und Webb fort und stellt in einem weiten Bereich Bewertungskriterien für die Einphasenströmung in Rohren auf. Die Gleichungen berücksichtigen Effekte der mantelseitigen Verbesserung und der Verschmutzung und sind anwendbar auf Rauhigkeiten und innenberippte Rohre. Ausführliche Verfahren zur Berechnung der Verbesserung der Wärmeübertragung und zur Auswahl der optimalen Oberflächengeometrie werden beschrieben. Die Kriterien werden für vier Auslegungsfälle dargestellt: (1) verminderter Materialaufwand des Wärmeaustauschers, (2) erhöhte Wärmeleistung, (3) reduzierte mittlere logarithmische Temperaturdifferenz und (4) reduzierte Pumpenleistung. Die 11 diskutierten Fälle behandeln konstanten und variablen Strömungsquerschnitt. Geeignete Kriterien für Zweiphasen-Wärmeaustauschflächen werden ebenfalls besprochen.

ИСПОЛЬЗОВАНИЕ КРИТЕРИЕВ ОЦЕНКИ ХАРАКТЕРИСТИК ПОВЕРХНОСТЕЙ ФОРСИРОВАННОГО ТЕПЛООБМЕНА ПРИ ПРОЕКТИРОВАНИИ ТЕПЛООБМЕННИКОВ

Аннотация — Предлагаемое исследование является продолжением работы Берглза и Вебба по определению критериев оценки характеристик однофазных течений в трубах. В уравнениях учитывается влияние внешнего корпуса теплообменника, шероховатости поверхности, загрязнения и внутреннего оребрения на интенсивность теплообмена. Подробно излагается методика расчета к.п.д. и выбора оптимальной геометрии поверхности. Приведены критерии для четырех случаев, встречающихся в практике проектирования: (1) минимальная металлоемкость теплообменника, (2) максимальная тепловая нагрузка, (3) минимальный среднелогарифмический напор и (4) минимальная мощность прокачки. В одиннадцати рассмотренных случаях используются фиксированная и переменная площади поперечного сечения. Рассматриваются также соответствующие критерии для определения поверхности нагрева двухфазных теплообменников.