Int. J. Heat Mass Transfer. Vol. 25, No. 11, p. 1770, 1982
Printed in Great Britain

0017-9310/82/111770-01 $03.00/0

PERFORMANCE EVALUATION CRITERIA FOR AIR-COOLED FINNED-TUBE HEAT-EXCHANGER SURFACE GEOMETRIES

RALPH L. WEBB

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

(*Received* 26 *January* 1982)

NOMENCLATURE

A_c,	minimum cross-sectional flow area in finned-tube bank;
A_f,	heat exchanger flow frontal area, width × height;
G_f,	mass velocity based on A_f, w/A_f;
L,	pass length for flow inside tubes, or exchanger depth for finned-tube exchanger;
S,	flow area per tube, $\pi D^2/4$;
V,	volume of finned-tube exchanger, A_{fr} L.

Greek symbols

β,	heat transfer surface area per unit volume, A/V;
σ,	flow contraction ratio, A_c/A_f.

THIS paper extends the information presented previously [1] describing the Performance Evaluation Criteria (PEC) used to calculate the benefits of enhanced heat transfer surfaces. Although these PEC are applicable to all types of heat exchangers, the equations were developed for the case of flow inside tubes. The development maintained constant tube diameter for the enhanced and smooth (reference) surface geometries. Although it is not necessary to maintain constant tube diameter, there is good reason to do so; this yields the performance improvement due only to the 'geometry' of the enhanced surface.

The equations in ref. [1] are equally applicable to performance evaluation of finned-tube surface geometries used in air-cooled heat exchangers, if the variables are properly interpreted. Figure 1 illustrates the heat exchanger geometries of present interest. Table 1 defines how the variables of ref. [1] should be interpreted for air cooled exchanger geometries.

The nomenclature table lists only those variables not defined previously [1].

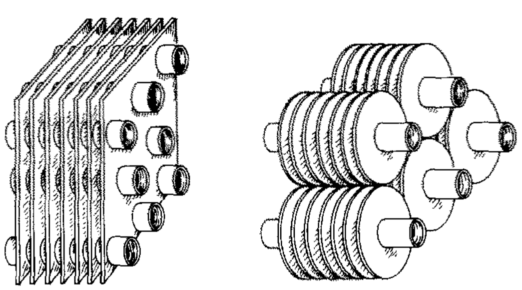

FIG. 1. Finned-tube heat exchanger geometries used for heat transfer to gases. (a) Plate-fin type. (b) Circular-fin type.

Table 1. Interpretation of geometric and flow variables ($D/D_s = 1$ for flow inside tubes)

Item	Flow inside tubes [1]	Fig. 1 geometries
Flow area	SN	Frontal Area (A_f)
Mass flow rate (w)	SNG	$A_c G = A_f G_f$
Surface area (A)	πDNL	βV
w/w_s	$\frac{N}{N_s}\frac{G}{G_s}$	$\frac{\sigma}{\sigma_s}\frac{A_f}{A_{f,s}}\frac{G}{G_s}$

Table 1 shows that the number of tubes per pass (N) is now interpreted as the finned-tube exchanger frontal area (A_f). The pass length (L) is now interpreted as the flow depth of the finned-tube exchanger.

When enhancement inside tubes is of interest, we have recommended that the tube diameter (D) be held constant. The corresponding specification for finned-tubes (Fig. 1) would require holding constant the geometric variables P_t, P_l, and D_0, 1 and fin pitch. Since the finned-tube problem involves five basic geometric variables rather than one (D) for flow inside tubes, it is usually impossible to satisfy the desired geometric constraints. Therefore the finned-tube performance will include the effects of the basic exchanger geometry variables, as well as those of the enhanced fin geometry. For example, consider two Fig. 1(a) heat exchangers, which are the same in all respects, except for the fin spacing. The PEC analysis will show that the geometry having the greater fin density is the 'best' design. Therefore, one should attempt to compare enhanced and plain fin geometries for the same fin pitch.

Also presented in this note are errata for ref. [1]. Equation (17) and (18) are incorrect, and should be the same as equations (5b) and (6), respectively. The exponent in equation (24a) should be 1/3 rather than 2/3. Equation (24b) should read

$$1 = \left(\frac{f}{f_s}\right)\left(\frac{A}{A_s}\right)\left(\frac{G}{G_s}\right)^3.$$

REFERENCES

1. R. L. Webb, Performance evaluation criteria for use of enhanced heat transfer surfaces in heat exchanger design, *Int. J. Heat Mass Transfer* **24**, 715–726 (1981).