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Advances in Enhanced Heat Transfer - 1981

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PREFACE

Heat transfer enhancement has developed into a major specialty area in heat transfer research and development. Today, we are finding more and more use of enhanced heat transfer surfaces in commercial equipment. The present volume contains the papers presented at two sessions of the 1981 National Heat Transfer Conference. One session "Enhanced Nucleate Boiling Heat Transfer" was sponsored by the AIChE, and the second session "Enhanced Heat Transfer" was sponsored by the ASME. Due to the high interest in enhanced heat transfer technology, it was decided to publish the papers from both sessions in a symposium volume.

This volume is the third symposium volume on the subject published since 1970. The volume "Advances in Heat Transfer" was published for the 1979 National Heat Transfer Conference, and contains both AIChE and ASME papers. The first of these three symposium volumes was published in 1970 by the ASME, "Augmentation of Convective Heat and Mass Transfer."

The large growth of publications in enhanced heat transfer has been documented by two bibliographic reports issued by Iowa State University, under the direction of Prof. A. E. Bergles. "A Bibliography on Augmentation of Convective Heat and Mass Transfer," by A. E. Bergles, R. L. Webb, G. H. Junkhan and M. K. Jensen ISU-HTL-19, May 1979 provides 1976 references from the open literature. A second report "Bibliography of U. S. Patents on Augmentation of Convective Heat and Mass Transfer," by R. L. Webb, G. H. Junkjan and A. E. Bergles, ISU-HTL-25, September 1980 provides references to 321 U. S. Patents on the art.

The serious present attention devoted to enhanced heat transfer is, in part, due to escalating energy costs. Enhanced surfaces may be employed to increase the thermodynamic efficiency of heat exchange equipment. And, manufacturing technology is rapidly developing to allow manufacture of these special surface geometries at economically competitive prices. Thus, it appears that there will continue to be high interest in the subject of enhanced heat transfer for some time to come.

The editors would like to express their appreciation to the authors and the many reviewers who contributed to the work of making this symposium volume possible. And, we are grateful for the cooperative spirit of the ASME and AIChE, which permitted the publication of a volume containing papers from both societies.

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ON THE PRESENTATION OF PERFORMANCE DATA FOR ENHANCED TUBES USED IN SHELL-AND-TUBE HEAT EXCHANGERS

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ABSTRACT

As the efforts to produce more efficient heat transfer equipment continue, an increasing number of augmented surfaces are being produced commercially. Consequently, the designer faces an almost overwhelming task in comparing and evaluating the performance of various surfaces because of the many different ways in which the test data are currently presented in the literature. Thus, a uniform format for presenting pressure drop and heat transfer data for enhanced surfaces has virtually become a necessity. This paper is concerned with one important aspect of this problem, namely that of tubular enhanced surfaces used in shell-and-tube heat exchangers. As an initial step, the subject is limited to single-phase pressure drop and heat transfer; however, both tubeside and shellside flow are taken into consideration. A comprehensive list of commercial augmented tubes which may be considered for use in shell-and-tube exchangers is given, along with a survey of the performance data which are available in the literature. A standardized data format which uses the inside and outside envelope diameters as the basis for presenting the various geometrical, flow, and heat transfer parameters for all tubular enhanced surfaces is proposed and discussed.

NOMENCLATURE

A_f	flow area
A_s	surface area
A_{shell}	flow area of empty shell
C_p	specific heat at constant pressure
D	diameter
D_b	bore diameter
F	LMTD correction factor
$F\{\ \}$	function of { }
g	acceleration due to gravity
G	mass velocity
h	heat transfer coefficient
k	thermal conductivity
L	tube length
LMTD	logarithmic-mean-temperature difference
N_r	number of tube rows
N_t	total number of tubes
p	actual wetted perimeter
P	tube pitch
Q	heat duty
R_w	tube wall resistance
R_{fi}	inside fouling resistance
R_{fo}	outside fouling resistance
t_w	tube wall thickness
T	absolute temperature
U_o	overall heat transfer coefficient
W	mass flow rate of fluid

Dimensionless Parameters

f	Fanning friction factor
Gr	Grashof number
Gz	Graetz number
j	Colburn j-factor
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
St	Stanton number

Greek Symbols

α	tubefield layout angle
β	coefficient of volumetric expansion of fluid
Δp	pressure drop
μ	viscosity of fluid
ρ	density of fluid

Subscripts

b	mean bulk temperature
c	cross flow
i	inside tube
o	outside tube
p	parallel flow
w	wall

Superscript

envelope diameter basis

INTRODUCTION

The shell-and-tube heat exchanger has been in use for many years and is the most widely used type of industrial heat transfer equipment. In order to carry out the thermal-hydraulic design of a shell-and-tube exchanger, pressure drop and heat transfer correlations (or tabulated data) must be available for both the tubeside and the shellside. In the early days only plain tubes were used in shell-and-tube exchangers. However, as increasing energy and material costs have provided significant incentives for more efficient heat exchangers, considerable emphasis has been placed on the development of various augmented, or enhanced, heat transfer surfaces. The use of enhanced surfaces allows the designer to increase the heat duty for a given exchanger, usually with a pressure drop penalty, or to reduce the size of the exchanger for a given heat duty.

The introduction of enhanced surfaces, and their inherently more complicated geometry, has resulted in the presentation of thermal-hydraulic data for these surfaces in many different ways. Therefore, the importance of standardization and compilation of performance data for enhanced surfaces was emphasized at the recent "Research Workshop on Energy Conservation through Enhanced Heat Transfer" in Chicago [1]. In particular, it was pointed out that such a procedure would save the designer time, confusion, and effort in evaluating potential enhanced surfaces.

The purpose of this paper is to present a unified, straightforward procedure for the presentation of performance data for various enhanced surfaces. Since the focus here is on shell-and-tube exchangers, the major emphasis is placed on tubes which are presently available commercially. In addition to the obvious benefits of a unified presentation format, it will be shown that such a procedure can be helpful in the initial screening of enhanced tubes for a specific application. It will also facilitate the introduction of enhanced performance data into digital computer programs which are ultimately used to design most shell-and-tube heat exchangers.

Although shell-and-tube heat exchangers are used in a broad range of applications--including those which involve condensation, boiling, and multi-phase flow--only single-phase Newtonian fluids will be considered in this paper. This somewhat arbitrary restriction was chosen to eliminate a number of complications which can arise in multi-phase flows. However, it is anticipated that the concepts outlined in this paper will provide the basis for a logical extension to more complex flow situations in the future.

The limitation to enhanced tubes used in shell-and-tube exchangers suggests several important restrictions. First, the maximum outside diameter of any tube is less than that of the tubeholes in the tubesheets. This constraint is a consequence of the common procedure in the fabrication and repair of shell-and-tube heat exchangers of inserting or pulling the tubes through the tube holes in the tubesheets. Enhanced tubes typically have plain-end sections so that they can be fastened securely to the tubesheets by rolling and/or welding. Second, the layouts considered here will be limited to those specified by TEMA, i.e., equilateral triangular, rotated triangular, square, and rotated square layouts, resulting in layout angles of $\alpha = 30^\circ$, 60°, 90°, and 45°. Third, in the presentation given here, it is assumed that compressibility and viscous dissipation effects are negligible on both sides of the exchanger. Thus, the Mach number and the Eckert number (or Brinkman number) will not appear as dimensionless parameters in the flows being considered. For industrial shell-and-tube exchanger applications, this assumption is generally warranted.

COMMERCIALLY AVAILABLE ENHANCED TUBES

Current classifications identify more than a dozen techniques to augment or enhance convective heat transfer [2]. Of interest to shell-and-tube heat exchangers with single-phase flows are the tubes with modified surfaces or inserts. While the origins of "enhanced" tubes are rather obscure, a "corrugated" tube was in use over 70 years ago [3]. For purposes of discussion, it is convenient to identify tubes with

- surface roughness
- extended surfaces
- inserts which create a swirl flow
- inserts which "mix" the flow

A representative list of commercially available tubes is given in Table 1. Within each designation,

Table 1 Representative Manufacturers and Sources of Data for Enhanced Tubes

Manufacturer	Trade Name or Designation	Inside or Outside	References for Data Sources
SURFACE ROUGHNESS			
Wolverine	Korodense	I	4-6
Wolverine	S/T Turbo-chil	I	7
Wolverine	S/T Trufin *	I	8
Turbotec	Turbotec	I	9-18
Yorkshire	enhanced	I	19
Yorkshire	Integron *	I	--
Wieland	GEWA *	I	--
Hitachi	circumfential groove	I	20
Marasei	embossed spiral	I	21
Spirance	spirally enhanced	I	22
EXTENDED SURFACE			
Hitachi	Thermofin	I	--
Noranda	Forge Fin	I	23-31
Noranda	multipassage	I	32,33
Turbotec	Turbotec	O	18
Wolverine	Trufin	O	34-40
Wieland	GEWA	O	41
Yorkshire	Integron	O	42
Anaconda	integral low-fin	O	--
HPTI	Fine-Fin	O	--
Southwest Alloy Supply	integral low-fin	O	--
Unifin	LOFIN	O	--
Dunham Bush	Inner-Fin	I	--
Wieland	EWE	I	--
American Standard	Amaclean	I	--
American Standard	Amatran	I	--
INSERTS - Swirl Flow			
Bas-Tex	turbulator	I	--
custom made	twisted-tape	I	31,43-50
custom made	wire inserts	I	51-53
INSERTS - Mixer			
American Standard	Amaspher	I	--
Intersurface	Generator	I	54
Kenics	Static Mixer	I	31,55-58
Koch	DY	I	31
Ross	Static Mixer	I	59

* Roughness is a byproduct of rolling external fins. These tubes may also have smooth inside diameters.

the products are identified as relating to inside (I) or outside (O) enhancement and sources of heat transfer and pressure drop are given. It is interesting to note that five countries are represented among the manufacturers.

In addition to the commercially available tubes mentioned in Table 1, there is another type of enhancement which is of considerable interest for shell-and-tube exchangers, i.e., twisted-tape inserts. These devices are used inside tubes, frequently as a fix to exchangers which have been underdesigned. However, since they are so easy to make, most shops manufacture their own. Thus, those studies carried out with non-commercial twisted-tape inserts are also listed in Table 1. Three studies dealing with custom-made wire-coil inserts to enhance tubeside heat transfer may be of interest, and they are included in Table 1 as well.

Cross-sections of eight commercially available tubes with internal enhancement are shown in Fig. 1 and two with external enhancement are shown in Fig. 2. These figures illustrate a few of the many enhanced tubes which are available commercially. It should be noted that some tubes, such as the Korodense tube shown in Fig. 2, have enhancement on both sides. In Figs. 1 and 2 each tube is identified by name and the inside and outside "smooth tube envelope" diameters are shown for each cross-section. These diameters are defined as follows:

1. $\tilde{D}_i$, maximum inside diameter for inside enhancement
2. $\tilde{D}_o$, maximum outside diamter for outside enhancement

In this paper the superscript ~ is used to designate quantities based on the envelope diameter, and the subscripts i and o denote inside and outside, respectively. In case there is no enhancement on one side of the tube, the envelope diameter is simply the plain tube diameter. The envelope diameter concept will be discussed more fully in the following section.

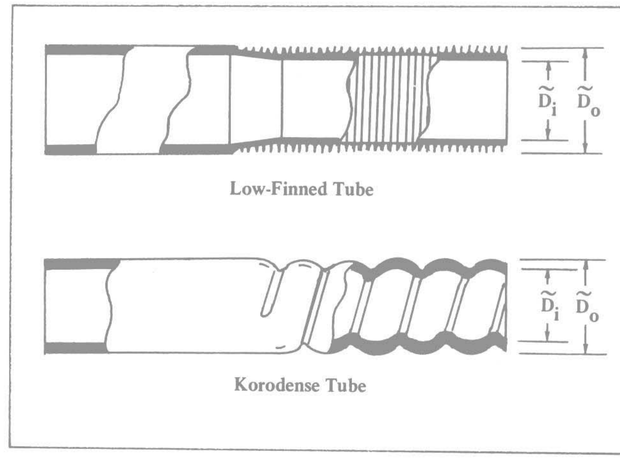

Fig. 2 Representative commercially available tubes with external enhancement.

PRESENTATION FORMAT

For a shell-and-tube heat exchanger, the heat duty may be calculated by

$$Q = A_o U_o F \text{ LMTD} \tag{1}$$

where the overall heat transfer coefficient U_o, based on the outside area A_o, is given by

$$1/U_o A_o = 1/h_i A_i + R_w + 1/h_o A_o + R_{fi}/A_i + R_{fo}/A_o \tag{2}$$

Although the designer must have a knowledge of each term in Eq. 2, the inside and outside heat transfer coefficients, h_i and h_o, are of primary interest here.

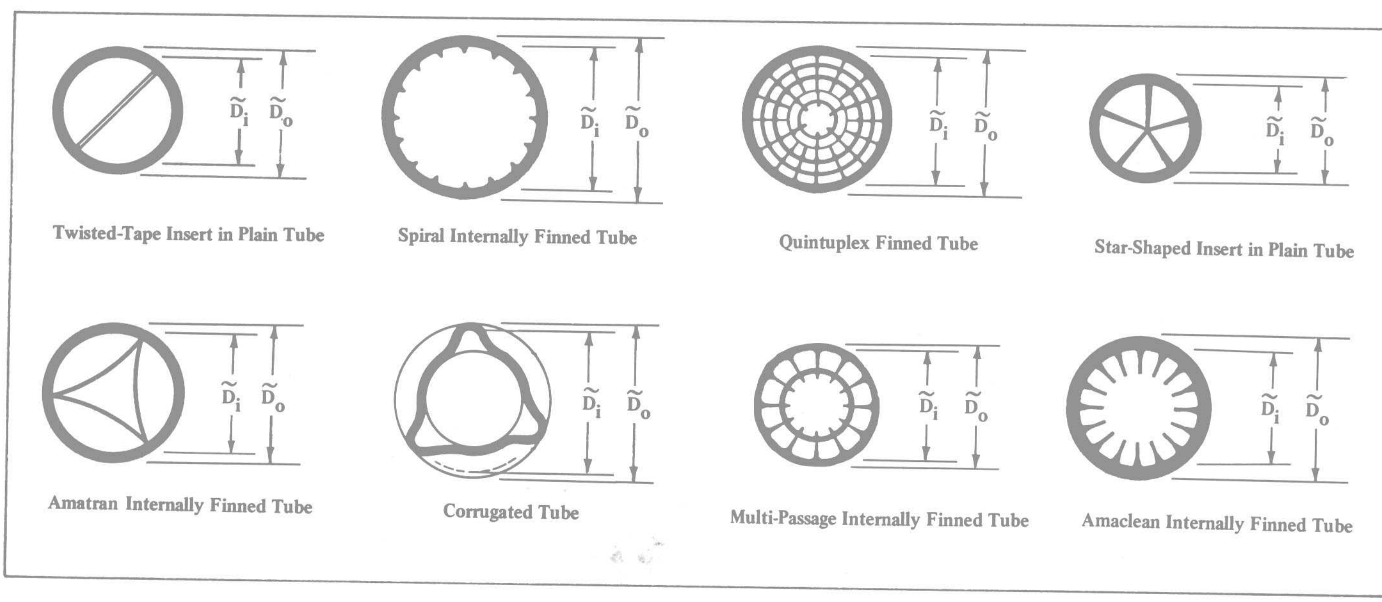

Fig. 1 Representative commercially available tubes with internal enhancement.

Unless fouling is being studied, the fouling resistances R_{fi} and R_{fo}, given here in units of m^2K/W, are generally assumed to be negligible when obtaining performance data. The wall resistance R_w is usually quite small in comparison to the convective resistances but some attention needs to be devoted to this term. In addition to h_i and h_o, the pressure drop performance is also of considerable importance in this study.

The basic performance data--for the sources cited in Table 1 as well as for non-commercial tube references in the literature--are presented in many different ways. Consider Table 2, where six tubeside enhancement references have been selected to illustrate this point. In Table 2 the pressure drop data are presented in three different ways and the heat transfer data in six different ways. Both dimensional and dimensionless formats are used. The Reynolds and Nusselt numbers are calculated using five different diameters as the characteristic length. The flow area, which appears in the mass velocity and hence the friction factor and Reynolds number, is computed using either the actual value or on the basis of the envelope, bore (minimum), or volumetric diameter. Finally, the surface area in Table 2, which is used as the basis for evaluating the heat transfer coefficient, is computed in five different ways. Thus, from just these limited references, it is clear that performance data for enhanced tubes are presented in many different ways in the literature. In general, the designer is faced with the unnecessary task of recasting the data from several different sources to a common form, prior to assessing the potential of the augmented tubes for a given application. Unfortunately, this job is frequently compounded by undefined parameters and incomplete information in the literature references.

Thus, it has been shown that standardization of the presentation format is highly desirable. While such a procedure is not absolutely essential, as long as the data treatment is complete and well defined, much confusion could be avoided and time saved if a uniform format were established and used. Although there are a number of logical choices available, the envelope diameter format is proposed here. This approach is based on simple, well-defined inside and outside characteristic lengths, $\tilde{D}_i$ and $\tilde{D}_o$, which are used to calculate all necessary parameters and dimensionless groups. In addition, this format permits a direct comparison of the performance of the augmented tube with that of the plain tube occupying the same space in the exchanger.

As a typical example, consider the Turbotec tube which is one of the commercially available corrugated tubes. The cross-sectional sketch shown in Fig. 3

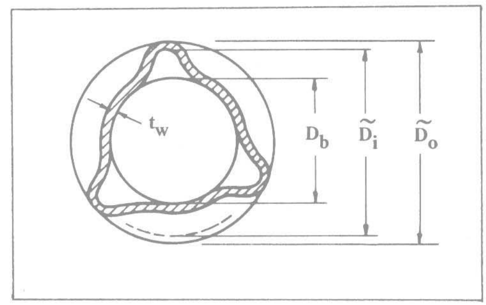

Fig. 3 Sketch of the Turbotec corrugated tube showing reference diameters.

defines the inside envelope diameter, $\tilde{D}_i$, the outside envelope diameter, $\tilde{D}_o$, and the bore diameter, D_b. Also shown in Fig. 3 is the tube wall thickness, t_w. The actual space occupied by the Turbotec tube is, of course, determined by the maximum outside diameter; however, only an internal dimension can characterize the internal thermal-hydraulics. In any event, the wall thicknesses of enhanced and smooth tubes are usually quite similar so that the true envelopes are essentially the same.

Tubeside Flow

Tubeside data for enhanced tubes are almost always obtained in single tubes. The most commonly used thermal boundary conditions are constant wall heat flux

TABLE 2 Format for Presentation of Tubeside Performance Data for Selected Enhanced Tubes

Reference	Type of Enhancement	Presentation Format	Diameter	Flow Area	Surface Area
Watkinson, Miletti, and Kubanek [28]	Internally Finned Tubes	f versus Re F{Nu} versus Re	Envelope and Effective	Actual	Envelope and Effective
Solimon and Feingold [33]	Internally Finned Tube Quintuplex Tube	Δp versus W Q versus W	---	---	---
Carnavos [29]	Internally Finned Tubes	f versus Re F{Nu} versus Re	Hydraulic	Actual	Actual
Marner and Bergles [31]	Internally Finned Tubes Static-Mixer Inserts Twisted-Tape Inserts	f versus Re Nu versus Re Nu versus F{Gz}	Envelope	Envelope and Actual	Envelope
Rozalowski and Gater [60]	Corrugated Tube	F{f} versus Re F{Nu} versus F{Gz}	Bore	Bore	Bore
Dipprey and Sabersky [61]	Artificial Roughness	f versus Re St versus Re	Volumetric	Volumetric	Volumetric

and constant wall temperature. In general, laminar flow heat transfer coefficients are highly sensitive to the thermal boundary conditions and to the thermal and hydrodynamic entry lengths. Consequently, laminar flow Nusselt numbers are characterized by the L/D_i ratio, either explicitly in the parameter $(L/D_i)/Re_i Pr_i$ or implicitly in the Graetz number, $Gz_i = W_i (C_p)_i / k_i L$. For design purposes, it is customary to present the mean Nusselt number versus Gz_i or a function of the Graetz number, $F\{Gz_i\}$. On the other hand, tubeside turbulent flow heat transfer coefficients have very short entry lengths and, except for liquid metals, are insensitive to the type of thermal boundary condition. Therefore, local heat transfer coefficients are usually measured for turbulent flow and the results presented in the form of $Nu_i/Pr_i^{0.4}$ versus Re_i with Pr_i as a parameter. This format is, of course, based on the well-known and widely used empirical Dittus-Boelter equation. The transitional flow regime--generally taken to cover the Reynolds number range between about 2,000 and 10,000--is an ill-defined combination of laminar and turbulent flow which may be handled by proration procedures.

The sketches shown in Figs. 4 and 5 pertaining to the important area of internal, single-phase, forced convection flow, demonstrate the pressure drop and heat transfer considerations. Fig. 4 indicates that by

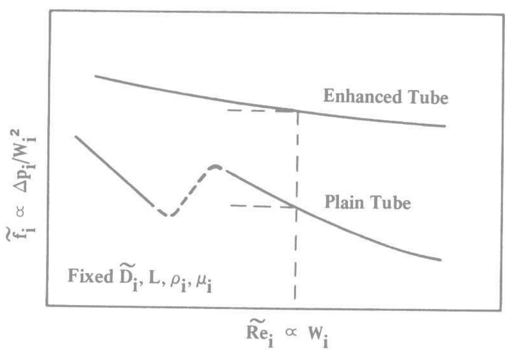

Fig. 4 **Interpretation of standard friction factor log-log plot according to envelope diameter – laminar and turbulent tubeside flow.**

fixing the envelope, both $\tilde{D}_i$ and L, a vertical line on the $\tilde{f}_i$ versus $\widetilde{Re}_i$ curve will give the augmented pressure drop relative to the plain tube pressure drop at the same flow rate.† Fluid properties and temperature-gradient effects, if any, are assumed to be the same for both tubes. Similarly, the relative $Q_i/\Delta T_i$ is given in Fig. 5. For laminar flow, ΔT_i may be the logarithmic or arithmetic mean over the entire tube length, while for turbulent flow the usual practice is to report "fully developed" values since the developing length is much shorter than in laminar flow. Two points should be made about this presentation scheme:

1. The intercepts at constant W_i do not, with the exception of Fig. 5, give the performance of one augmented tube relative to another unless the envelope diameters are identical.

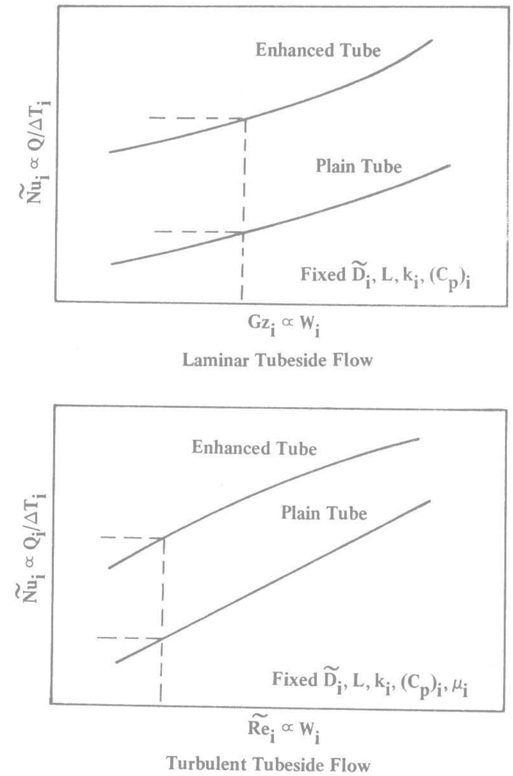

Fig. 5 **Interpretation of standard Nusselt number log-log plots according to the envelope diameter.**

2. The envelope diameter basis may not be the best way to correlate data; for example, bore diameter or hydraulic diameter may be a more significant dimension.

A complete description of the tubeside parameters for the proposed format is given in Table 3. This information is essentially self-explanatory, but a few points will be emphasized here. All parameters are based on the inside envelope diameter, $\tilde{D}_i$, unless otherwise noted. All properties are evaluated at the arithmetic mean bulk temperature between inlet and outlet, T_{bi}, unless specified otherwise. Corrections for the effect of variable properties may be then made in terms of the viscosity ratio for liquids and the absolute temperature ratio for gases. For example, it is expected that the pressure drop data will usually be taken under isothermal conditions. In general, the Grashof number will be important only for laminar flow conditions. Finally, it should be noted that the Graetz number, which is used in laminar flow, may be calculated as

$$Gz_i = W_i (C_p)_i / k_i L = (\pi/4) \widetilde{Re}_i Pr_i (\tilde{D}_i/L) \tag{3}$$

Note that although both $\widetilde{Re}_i$ and $\tilde{D}_i$ are based on the envelope diameter, the Graetz number Gz_i is based only on actual variables.

A very important aspect of the proposed format is that in addition to presenting all parameters in terms of $\tilde{D}_i$, it must also be possible for the individual investigator to recast these parameters in terms of any possible combination of characteristic length, heat transfer surface area, and flow area. Therefore, it is essential that each tube geometry be characterized completely. Thus, the actual inside surface area,

† Frictional pressure drop only is considered here. Entrance and exit losses and pressure variation due to momentum change must be considered in general.

TABLE 3 Parameters for Tubeside and Shellside Flow

Parameters	Tubeside Flow	Shellside Crossflow	Shellside Parallel Flow
Envelope Diameter	$\tilde{D}_i$	$\tilde{D}_o$	$\tilde{D}_o$
Tube Length	L	L	L
Total Number of Tubes	--	N_t	N_t
Number of Tube Rows	--	N_r	--
Tube Pitch	--	P	P
Tubefield Layout Angle	--	α	α
Flow Area	$\tilde{A}_{fi} = \pi\tilde{D}_i^2/4$	$(\tilde{A}_f)_{oc} = (N_t/N_r)L(P - \tilde{D}_o)$	$(\tilde{A}_f)_{op} = A_{shell} - N_t\pi\tilde{D}_o^2/4$
Mass Velocity	$\tilde{G}_i = W_i/\tilde{A}_{fi}$	$\tilde{G}_{oc} = W_o/(\tilde{A}_f)_{oc}$	$\tilde{G}_{op} = W_o/(\tilde{A}_f)_{op}$
Reynolds Number	$\widetilde{Re}_i = \tilde{G}_i\tilde{D}_i/\mu_i$	$\widetilde{Re}_{oc} = \tilde{G}_{oc}\tilde{D}_o/\mu_o$	$\widetilde{Re}_{op} = \tilde{G}_{op}\tilde{D}_o/\mu_o$
Fanning Friction Factor	$\tilde{f}_i = \rho_i\Delta p_i(\tilde{D}_i/L)/2\tilde{G}_i^2$	$\tilde{f}_{oc} = \rho_o(\Delta p_{oc}/N_r)/2\tilde{G}_{oc}^2$	$\tilde{f}_{op} = \rho_o\Delta p_{op}(\tilde{D}_o/L)/2\tilde{G}_{op}^2$
Prandtl Number	$Pr_i = (C_p)_i\mu_i/k_i$	$Pr_o = (C_p)_o\mu_o/k_o$	$Pr_o = (C_p)_o\mu_o/k_o$
Grashof Number	$\widetilde{Gr}_i = \rho_i^2 g\beta_i\lvert T_{wi} - T_{bi}\rvert\tilde{D}_i^3/\mu_i^2$	$\widetilde{Gr}_o = \rho_o^2 g\beta_o\lvert T_{wo} - T_{bo}\rvert\tilde{D}_o^3/\mu_o^2$	$\widetilde{Gr}_o = \rho_o^2 g\beta_o\lvert T_{wo} - T_{bo}\rvert\tilde{D}_o^3/\mu_o^2$
Graetz Number	$Gz_i = W_i(C_p)_i/k_iL$	--	--
Viscosity Ratio (Liquids)	μ_{bi}/μ_{wi}	μ_{bo}/μ_{wo}	μ_{bo}/μ_{wo}
Temperature Ratio (Gases)	T_{bi}/T_{wi}	T_{bo}/T_{wo}	T_{bo}/T_{wo}
Average Wall Temperature	T_{wi}	T_{wo}	T_{wo}
Surface Area	$\tilde{A}_{si} = \pi\tilde{D}_iL$	$\tilde{A}_{so} = N_t\pi\tilde{D}_oL$	$\tilde{A}_{so} = N_t\pi\tilde{D}_oL$
Nusselt Number	$\widetilde{Nu}_i = \tilde{h}_i\tilde{D}_i/k_i$	--	--
Colburn j-Factor	--	$\tilde{j}_{oc} = (\tilde{h}_{oc}/(C_p)_o\tilde{G}_{oc})Pr_o^{2/3}$	$\tilde{j}_{op} = (\tilde{h}_{op}/(C_p)_o\tilde{G}_{op})Pr_o^{2/3}$
Actual Flow Area	A_{fi}	$(A_f)_{oc}$	$(A_f)_{op} = (\tilde{A}_f)_{op}$
Actual Wetted Perimeter	p_i	--	--
Actual Surface Area	$A_{si} = p_iL$	A_{so}	A_{so}

actual inside flow area, and actual inside perimeter must also be given for each tube. Finally, a scaled drawing or a photograph of a typical cross-section will, in many cases, be a valuable aid in helping to interpret the tube geometry.

Shellside Flow

In contrast to tubeside flow, which in heat exchanger terminology is unmixed, the shellside flow is much more complicated. In a baffled exchanger, the flow consists of contributions from both crossflow and parallel, or longitudinal, flow. Although these two components will be handled separately here, it should be understood that in general a stream analysis method [62,63] must be applied to carry out a rigorous shellside thermal-hydraulic design. Such procedures almost always require the aid of a digital computer.

Low-finned tubes were among the first of the enhanced heat transfer devices to become popular in shell-and-tube heat exchangers. As might be expected, low-finned tube data have been presented in many different formats. For example, the following diameters have been used in the literature as characteristic lengths: root, fin, hydraulic, volumetric hydraulic, modified hydraulic, and plain-end diameters. Data for low-finned tubes have been reported in the literature for the following cases [34-42]:

1. specific baffled shell-and-tube exchangers
2. crossflow in ideal tubebanks
3. longitudinal flow over a single tube inside a smooth annulus.

Of the references cited above only that of Obermeier and Schaber [41] falls in Category 3, and there are apparently no data available in the literature for parallel flow over a bundle of low-finned tubes. Several sets of crossflow data have been obtained for ideal tubebanks using low-finned tubes. The recent paper by Rabas, Eckels, and Sabatino [36] provides an excellent summary of the work done in this area and presents correlations for both pressure drop and heat transfer over a broad range of parameters. Their treatment uses the root diameter as the characteristic length.

Very limited data have been obtained for longitudinal flow over bundles of fuel elements in nuclear reactors. However, in the nuclear industry, data for gas-cooled reactors, which have artificially roughened fuel element rods, are generally obtained using a single tube inside a smooth circular tube. Various transformations are available which then allow one to transform these single-tube results so that they are applicable to longitudinal flow in actual tube bundles. One of the earliest, and best known, of these transformations was formulated by Hall [64] in 1962. Since then, a number of additional transformations have been developed, and these methods are summarized and discussed in detail by Dalle Donne and Meyer [65]. In view of the difficulty in obtaining longitudinal pressure drop and heat transfer data in bundles of tubes, it is suggested that greater emphasis be placed on obtaining such data using individual enhanced tubes as described above. Such an approach should be especially useful for screening potential tubes for a specific application.

In Fig. 6 pressure drop and heat transfer data for low-finned tubes, taken from Ward [37], are used to illustrate the proposed presentation format for shell-side crossflow. The results are plotted in the form of

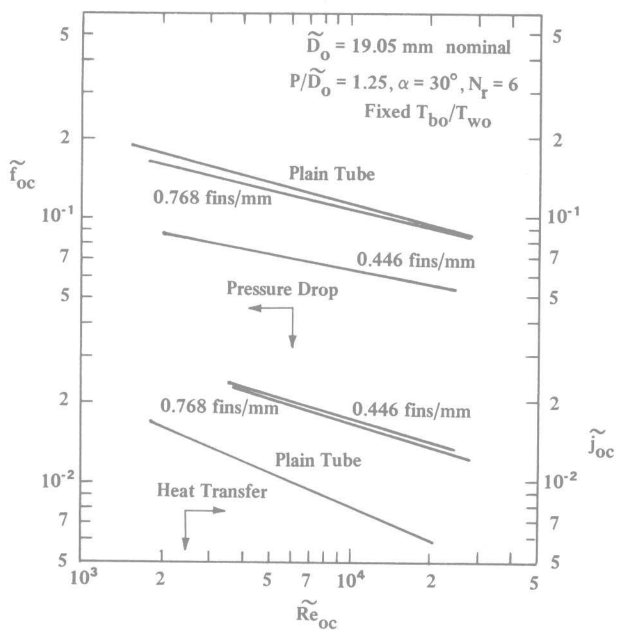

Fig. 6 Crossflow performance data from Ward [37] for low-finned tube bundles in envelope diameter format.

$\tilde{f}_{oc}$ versus $\widetilde{Re}_{oc}$ and $\tilde{j}_{oc}$ versus $\widetilde{Re}_{oc}$ with all parameters calculated on the basis of the outside envelope diameter, $\tilde{D}_o$. Data are presented for plain tubes and low-finned tubes with 0.446 fins/mm and 0.768 fins/mm, all with a nominal envelope diameter of $\tilde{D}_o$ = 19.05 mm, layout angle of $\alpha = 30^o$, and pitch ratio of $P/\tilde{D}_o = 1.25$. A quick inspection of Fig. 6 indicates that of the three tube bundles, the low-finned tubes with 0.446 fins/mm clearly have the superior heat transfer versus pressure drop performance.

The proposed shellside presentation format is summarized in Table 3 and consists of both crossflow and parallel flow. In the latter case, Table 3 is applicable to either a single tube or a tube bundle. In Table 3 all properties are evaluated at the arithmetic mean bulk temperature between inlet and outlet, T_{bo}, unless otherwise specified. In all cases the outside envelope diameter, $\tilde{D}_o$, is recommended as the basis for calculating the thermal-hydraulic characteristics. As was emphasized in the case of tubeside flow, the envelope diameter may not be the best characteristic length for correlational purposes. Therefore, it is essential that sufficient information be provided to calculate any characteristic length, minimum flow area, and heat transfer surface area desired as described in Table 3.

Tube Wall Parameters

Some mention must be made of the tube wall resistance, R_w, which appears in Eq. 2. If either the inside or outside wall has extended surfaces (fins) attached to it, further details must be provided in order to completely define the geometry. In particular, as the heat transfer coefficient increases, the fin height increases, or the fin thermal conductivity decreases, the fin efficiency decreases and must be taken into consideration. In those situations where the fin efficiency becomes important, it is recommended that the data be corrected and presented on the basis of 100 percent fin efficiency. The most comprehensive recent treatment of fin efficiency is given by Kern and Kraus [66].

The required tube wall parameters are specified in Table 4. By treating the extended surfaces through the fin efficiency, the calculation of the tube wall resistance, R_w, may be summarized as follows:

$$R_w = \ln(\tilde{D}_o/\tilde{D}_i)/2\pi k_w L \quad \text{(circular cross-sections)} \quad (4)$$

TABLE 4 Tube Wall Parameters

Parameter	Definition of Notation
Material	--
Thermal Conductivity	k_w
Actual Wall Thickness	t_w
Approximate Wall Resistance	$R_w = t_w/A_{sw}k_w$
Inside Dimensions required to completely characterize the tube	For example, for an internally finned tube with spiral fins the following parameters must be specified: * fin height * fin thickness * number of fins * fin pitch * fin efficiency * scale drawing or photograph of the fin contour
Outside Dimensions required to completely characterize the tube	For example, for a low-finned tube the following parameters must be specified: * fin height * fin thickness * fins per unit length * fin efficiency * scale drawing or a photograph of the fin contour

$$R_w = t_w/A_{sw}k_w \quad \text{(non-circular cross-sections)} \qquad (5)$$

where A_{sw} is the mean tube wall area. In general, the tube wall resistance will be quite small in comparison to the convective resistance.

CONCLUDING REMARKS

A systematic procedure for the presentation of performance data for enhanced tubes used in shell-and-tube heat exchangers has been proposed. This approach is based on the use of the inside and outside envelope diameters of the tube to evaluate all the required flow and heat transfer parameters. In addition, the procedure requires that sufficient geometric parameters be provided so that the envelope-based parameters can be recast into any form desired for purposes of comparison, correlation, and evaluation. Although the proposed procedure is based on single-phase flow, the concepts set forth in this paper should provide the basis for extension to multiphase systems in the future.

It is recommended that tubeside pressure drop data be presented in the form of $\tilde{f}_i$ versus $\widetilde{Re}_i$. Tubeside heat transfer data should be plotted as $\widetilde{Nu}_i$ versus Gz_i for laminar flow and $\widetilde{Nu}_i/Pr_i^{0.4}$ versus $\widetilde{Re}_i$ for turbulent flow. For outside enhancement it is proposed that the data be presented in the form of $\tilde{f}_o$ versus $\widetilde{Re}_o$ and $\tilde{j}_o$ versus $\widetilde{Re}_o$ for both crossflow and parallel flow.

The adoption of the proposed format would achieve at least four important objectives: (1) confusion could be avoided, (2) time could be saved, (3) initial screening of enhanced tubes could be carried out easily, and (4) new enhanced tube data could easily be input into existing computer programs.

Finally, although performance data for a number of commercially available tubes have been reported in the literature, additional data are badly needed in several areas. In particular, data for shellside flow and heat transfer are very limited, especially for parallel flow where data are virtually non-existent.

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TURBULENT FLOW CHARACTERISTICS IN AN INTERNALLY FINNED TUBE

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ABSTRACT

Fully developed isothermal air flows through an internally finned tube were investigated experimentally at Reynolds numbers of 50,000 and 71,000. The finning configuration consisted of six straight rectangular (38.1 x 4.9 mm) fins equi-spaced in a 114.3 mm I.D. tube. The reported measurements include friction factor, the cross-sectional distributions of mean axial velocity, secondary velocities and Reynolds normal stresses, and the distributions of local wall shear stress on the tube and fins. The average local wall shear stress on the tube was found to be almost the same as the average over the fin surface. For the mean velocity field, in addition to the global maximum in velocity at the tube centerline, an auxiliary peak flow region was present in each of the six bays formed by adjacent fins. Within each of the twelve primary flow cells, two counter-rotating cells of secondary flow were found to exist, with peak secondary velocities of about 4½% of the bulk average velocity. The cross-sectional distribution of turbulent kinetic energy showed distinct effects of the convection of turbulent kinetic energy by the secondary flow. Although the presented results were obtained under adiabatic conditions, the relevance of the experimental findings to internally finned tube heat transfer is discussed briefly.

NOMENCLATURE

b	average distance between fins
D_h	equivalent hydraulic diameter, 4 (cross-sectional area)/(actual wetted perimeter)
f	friction factor, $(2\ D_h/\rho U_{bt}^2)\ (dP/dx)$
H	dimensionless fin height, ℓ/R
ℓ	fin height
M	number of equi-spaced fins
P	pressure
$\bar{q}$	turbulent kinetic energy per unit mass, $\frac{1}{2}\,(\overline{u^2} + \overline{v^2} + \overline{w^2})$
r	radial distance from center of finned tube (radial coordinate)
R	inside radius of internally finned tube
Re	Reynolds number, $\rho U_{bt} D_h/\mu$
u,v,w	fluctuating components of the velocities in the axial, radial and peripheral directions respectively
$\overline{u^*}$	average friction velocity, $(\bar{\tau}/\rho)^{\frac{1}{2}}$
$\bar{U}$	axial mean velocity (time-average)
U_b	average mean axial velocity over primary flow cell (bulk velocity)
U_{bt}	overall bulk velocity for internally finned tube
$\bar{U}_{sec}$	resultant of $\bar{V}$ and $\bar{W}$, $(\bar{V}^2 + \bar{W}^2)^{\frac{1}{2}}$
$\bar{V}, \bar{W}$	radial and peripheral mean velocities (secondary velocities)
x	axial coordinate
y,z	directions (see Fig. 1)
y^*	distance along fin measured from fin tip (see Fig. 1)
θ	angular coordinate
μ	laminar dynamic viscosity
ρ	air density
τ_w	local wall shear stress
$\bar{\tau}_w$	average of local wall shear stresses over tube and fin
$\bar{\tau}$	average wall shear stress, $(dP/dx)\ (D_h/4)$

INTRODUCTION

The thermal effectiveness of tubular heat exchangers in which the convective resistance on the inner surface of the tube constitutes the main barrier to heat flow, can be substantially increased by a number of augmentative techniques such as twisted tape or wire coil inserts, internal surface roughness or fins, etc. Unfortunately for such modifications, the enhanced heat transfer over smooth tube conditions is typically accompanied by extra cost and maintenance and by increases in weight, pressure drop and pumping power. Nonetheless, in the case of internally finned tubes, Webb and Scott [1] have shown that finned tube heat exchangers are capable of superior performance vis-a-vis conventional heat exchangers under turbulent flow conditions, whereas our own study [2] shows that prospects are even better under laminar flow conditions. Thus there is considerable motivation for internally finned tube research since any information which improves our knowledge of such flows, will contribute to developing this technology for industrial applications.

According to Brouillette et al [3] experiments on internally finned tubes date back to 1923, but it is only during the last two decades that a concerted effort has been made to determine friction factors and heat transfer characteristics over a wide range of fin numbers (M) and fin heights (H) for longitudinal fins in both straight and helical patterns. For turbulent flows, important contributions to the experimental data base have been made by Hilding and Coogan [4], Lipets et al [5], Bergles et al [6], Watkinson et al [7,8] and Carnavos et al [9-11]. On the other hand, unlike the situation for laminar flow (e.g.[12]), theoretical work appears to have been limited to a single study [13,14]. Ivanovic [13] applied both a mixing length model and a low-Reynolds-number two-equation model to examine the local characteristics of flow and heat transfer for straight finned tubes in the combined range $6 \leq M \leq 18$ and $0.20 \leq H \leq 0.45$. However, these numerical predictions were for zero-thickness fins, and secondary flows were assumed to be negligible.

The present experimental results are for fully developed turbulent flows through a straight finned tube of M=6, H=0.667 under adiabatic flow conditions. The measurements include axial pressure drop, the cross-sectional distributions of mean axial velocity, secondary velocities and Reynolds stresses, and the distributions of local wall shear stress on the tube and fins. Compared to existing experimental data, the present results are novel, although it should be mentioned that Ornatskii et al [15] have reported Pitot tube measurements of the mean velocity distribution across the symmetry line of a channel which was purported to simulate a segment of an internally finned tube. The present isothermal results also have heat transfer overtones. For example, for the thermal boundary condition of constant heat input per unit axial length and uniform temperature circumferentially, the dimensionless temperature distribution (for constant fluid properties and high thermal conductivity fins) will be similar to the present mean axial velocity distribution, while the local heat flux is analogous to local wall shear stress.

The experimental study described in this paper involves Pitot tube and hot-wire anemometry measurements of the air flow through an internally finned tube for which the finning configuration consisted of six straight rectangular (38.1 x 4.9 mm) fins equi-spaced in a 114.3 mm I.D. tube. The flow cross-section for this finned tube is shown in Fig. 1. The 6.1 m long finned tube test section was installed on an existing wind tunnel which operated in the open circuit mode. The measurements were made from the discharge end of the test section. Since this measuring station was located approximately 135 equivalent hydraulic diameters from the inlet, the flow was assumed to be fully developed.

Taken in broad perspective, an internally finned tube is a non-circular duct which (under turbulent flow conditions) forms a secondary flow problem in the same class as rectangular and triangular ducts, eccentric annuli and infinite rod bundle arrays. As indicated in Fig. 1, symmetry permits the flow cross-section to be subdivided into twelve primary flow cells. Each of these cells is identical when viewed with respect to a rotated coordinate system (except for handedness of secondary circulation), and no net mass, momentum or energy is transferred across any boundary. A knowledge of the flow properties in any one cell is therefore sufficient to describe the entire flow field. Accordingly, following checks to confirm flow symmetry, the measurements were confined to Primary Flow Cell I as indicated in Fig. 1. Measurements were conducted at Reynolds numbers of 50,000 and 71,000. In addition, pressure drop measurements were made at several mass flow rates in order to generate an f versus Re curve for the finned tube. The main purpose of this paper is simply to describe the experiment and to communicate some of the experimental results. It was of course not known beforehand whether the secondary flows would be large enough to be measured. Fortunately, this part of the investigation turned out to be successful. On the other hand, the Reynolds shear stress measurements were not completely satisfactory. Measurements were made of both $\rho\overline{uv}$ and $\rho\overline{uw}$ (but not $\rho\overline{vw}$). Repeatability (at fixed Re) was found to be poor at several locations

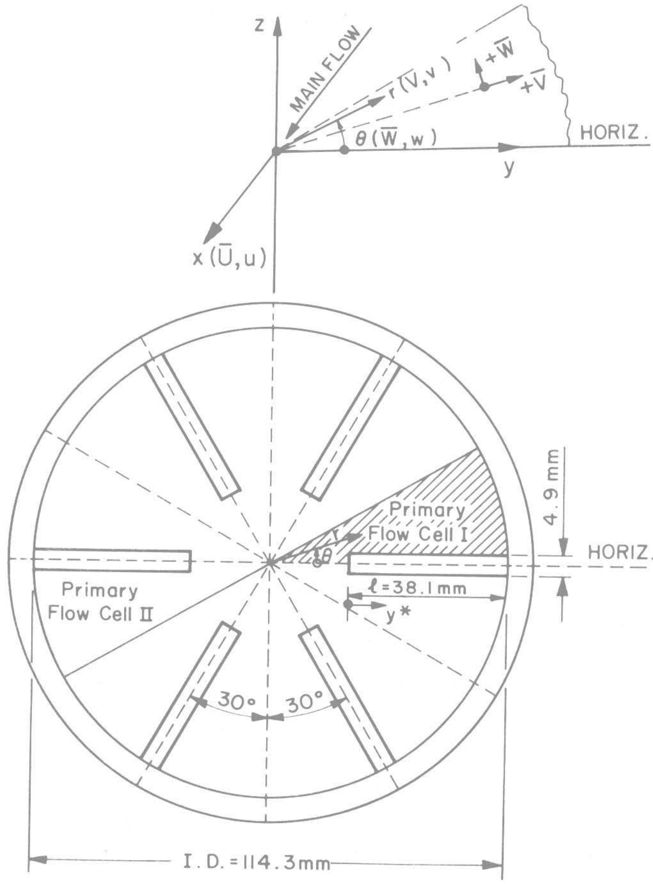

Fig. 1 Cross-section of Internally Finned Tube Test Section

in the primary flow cell cross-section. In addition, distributions (scaled by $\overline{u^{*2}}$) at the two Reynolds numbers showed excessive variability in certain regions of the flow cross-section. These shear stresses are being further investigated. Accordingly, no data on shear stresses (hence also eddy viscosities in the radial and peripheral directions) are offered at the present time.

Finally it is noted that an auxiliary purpose of the experiment was to generate experimental data to be used in tuning a turbulence model. In parallel with this work, a two-equation turbulence model was developed and is being extended to predict both fluid flow and heat transfer characteristics in internally finned tubes over a wide range of H and M for fully developed turbulent flow. The results of this study are expected to be available in the near future.

EXPERIMENTAL FACILITY AND EQUIPMENT

Wind Tunnel

The wind tunnel portion of the present facility was the same as that used previously by Gerrard [16] and by Aly et al [17] for the equilateral triangular duct. A photograph showing the internally finned tube test section installed on the wind tunnel is shown in Fig. 2. Atmospheric air was drawn through a contracting inlet by four counter-rotating axial aerofoil fans. Each of these fans was driven by a two-speed