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Crossflow heat transfer for tubes with periodically interrupted annular fins

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NOMENCLATURE

A_{tot}	surface area of fins and exposed tube
A_{tube}	surface area of unfinned tube
D	outside diameter of tube
D_f	fin tip diameter
k	thermal conductivity
p	axial pitch of fins
Q	heat transfer rate
Re	Reynolds number, $U_\infty D/\nu$
T_w	tube wall temperature
T_∞	freestream temperature
t	fin thickness
U_∞	freestream velocity
W	width of radial interruption gap.

Greek symbols

θ	circumferential coordinate
ν	kinematic viscosity.

Subscript

av	circumferential average.

Superscript

*	continuous-fin tube.

INTRODUCTION

THE USE of periodically interrupted surfaces to bound a flow passage is a well-established technique for attaining heat transfer coefficients higher than those for a flow passage bounded by continuous walls. This concept is frequently employed when the bounding surfaces are plate-type fins, and a substantial body of information on interrupted plate-fin heat transfer and pressure drop is available in the open literature [1–6].

The interrupted-surface concept is also employed for annular fins attached to the outer surface of a circular tube. In essence, each annular fin in the array is periodically interrupted by radial cuts which are separated by equal angular intervals. If the annular fins are individual disks, the successive radial cuts in a given fin create fin segments which are in-line with respect to each other. However, for annular fins which form a helical winding around the tube, the fin segments created by the successive cuts are offset with respect to their immediate neighbors.

In practice, the fabrication process for interrupted annular fins may impart a twist to each fin segment and/or may bend the segment at its leading and trailing edges. These irregularities will, in all likelihood, have a marked effect on the heat transfer and pressure drop characteristics of the finned tube. However, owing to the highly specific geometry of such twisted or bent fin segments, the resulting heat transfer and pressure drop characteristics, if available, would be of limited applicability.

Despite their widespread use in heat exchangers, it appears that there is no information in the open literature about tubes with interrupted annular fins. In the experimental investigation to be described here, crossflow heat transfer results are measured for such a finned tube and are compared with corresponding results for the same tube having continuous (i.e. uninterrupted) annular fins.

In view of the highly specific and often irregular geometries of commercial interrupted-annular-finned tubes, it does not appear to be appropriate to use such tubes as the initial focus of a fundamentals-level investigation. Rather, an annular-finned tube of regular and unambiguous geometry was fabricated for the experiments. The finned tube was first used without radial interruptions to determine a set of baseline heat transfer coefficients. Subsequently, radial interruptions were cut into the fins, and the effect of the interruptions on the heat transfer was quantified. Experiments were carried out for interruptions of two different widths. Data were collected for Reynolds numbers (based on the diameter of the unfinned tube) ranging from about 7500 to 44 000.

THE EXPERIMENTS

The finned tube used in the experiments was a composite structure consisting of an electrically heated test section flanked at either end by a heater-equipped guard section. The temperatures of the adjacent portions of the test section and guard sections were matched, thereby eliminating extraneous axial heat losses from the test section. To increase the sensitivity of the temperature matching procedure, a Teflon spacer was interposed between the adjacent faces of the respective sections. All dimensions of the test and guard sections were identical (except for their lengths), so that the assembled tube was hydrodynamically continuous along its axis. Its overall length was 30.48 cm (20.32 cm test section length), and it was mounted vertically in the 30.48 × 60.96 cm cross-section (height by width) of a low-turbulence, open-circuit wind tunnel.

Further details about the geometry of the finned tube are conveyed by Figs. 1(a) and (b). Figure 1(a) is a longitudinal sectional view which shows the fins to be integral with the tube. This was accomplished by fabricating the finned tube from a solid rod (of aluminum) and circumferentially grooving its outer periphery in order to create the interfin spaces. The rod was also bored along its axis to provide space for the insertion of a heater core and for the passage of power leads and thermocouple wires. Individual, separately controlled, uniformly wound heater cores were used for the test section and for the guard sections.

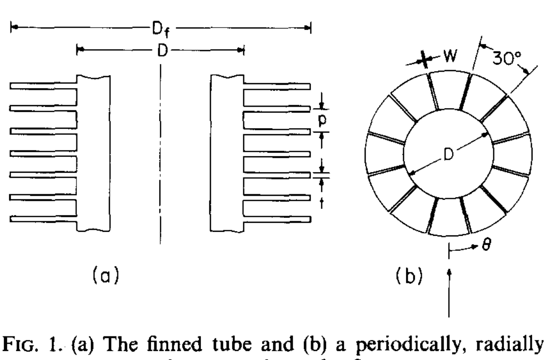

FIG. 1. (a) The finned tube and (b) a periodically, radially interrupted annular fin.

Figure 1(a) also shows the key dimensions of the finned tube. To reflect practice, the tube was designed with six fins per inch, yielding a center-to-center pitch p of 0.423 cm. The other indicated dimensions are t (fin thickness) = 0.106 cm, D (tube diameter) = 3.175 cm, and D_f (fin tip diameter) = 5.715 cm. The tube was fabricated with a relatively thick wall (0.635 cm). This thickness, as well as the use of aluminum as the finned tube material, was chosen to minimize temperature variations.

As noted earlier, the fins were initially fabricated without interruptions. Subsequently, radial cuts were machined into the fins by means of a slitting saw. The appearance of a typical interrupted fin is illustrated in Fig. 1(b). As seen there, the radial interruptions were periodically positioned at 30° intervals. Two values of the interruption gap width W were employed, namely W = 0.066 and 0.142 cm. The experiments for the smaller gap width were completed prior to the machining of the larger gap width. The orientation of the interruptions relative to the freestream direction is indicated by the freestream velocity vector shown in the figure.

The test section portion of the cylinder was equipped with eight thermocouples, while there were three thermocouples in each of the guard sections. The thermocouples were installed in the tube wall, with the junctions situated about 0.076 cm from the surface of the tube exposed in the interfin space. By design, the finned tube could be rotated about its axis, thereby enabling the thermocouples to be positioned at any circumferential location relative to the direction of the freestream flow. By this means, temperature measurements were made at 30° intervals around the circumference.

The freestream temperature was measured by a pair of thermocouples situated just forward and to the side of the finned tube, while the freestream velocity was determined by an impact tube/wall tap combination, also forward and to the side.

RESULTS AND DISCUSSION

The main objective of this investigation is to compare the rate of heat transfer per unit temperature difference for a tube having interrupted annular fins with that for an otherwise identical finned tube having continuous (i.e. uninterrupted) fins. The comparison is to be made at the same value of the freestream velocity U_∞ for the two configurations. In dimensionless terms, the freestream velocity may be represented by the Reynolds number

$$Re = U_\infty D/\nu. \tag{1}$$

The Reynolds number will serve as one of the parameters of the comparison. All the experiments were performed with the finned tube situated in crossflow.

The heat transfer rates Q to be compared were obtained directly from the electric power input to the test section heating element (end losses having been eliminated by the guard sections). The characteristic temperature difference between the tube wall and the freestream was evaluated as the average of the local values measured at $\theta = 0°, 30°, \ldots$ (θ is the angular coordinate illustrated in Fig. 1(b)). Prior to this averaging, any slight deviations in the temperature differences at geometrically symmetric positions (e.g. $\theta = 30°$ and $-30°$) were averaged out. Then

$$(T_w - T_\infty)_{av} = \left[\sum_{i=1}^{7} (T_w - T_\infty)_i\right]/7 \tag{2}$$

with $\theta_1 = 0°, \ldots, \theta_7 = 180°$.

As will be demonstrated shortly, the circumferential variation of $(T_w - T_\infty)_i$ was small (of the order of a few percent about the average). Also, since $(T_w - T_\infty)_{av}$ was limited to the neighborhood of 8°C, thermophysical property variations were not a factor in the experiments. The properties were evaluated at the mean of T_w and T_∞.

In terms of the foregoing quantities, the comparison of interrupted-fin and continuous-fin tubes will be made via the ratio

$$[Q/(T_w - T_\infty)_{av}]/[Q/(T_w - T_\infty)_{av}]^* \tag{3}$$

where the asterisk is used to identify the continuous-fin case, which serves as the baseline for the comparison. This ratio is plotted as a function of the Reynolds number in Fig. 2, where data are presented for two values of the interruption gap width W. Smooth curves have been passed through the data to provide continuity.

Inspection of Fig. 2 shows that the interruption of the fins is enhancing. That is, for the same wall-to-freestream temperature difference, the heat transfer rate for the interrupted-fin tube is greater than that for the continuous-fin tube. The enhancements are, however, moderate. In particular, for the smaller of the investigated gaps, the largest enhancement is about 6%. For the larger interruption gap, enhancements of up to 14% were encountered. The extent of the enhancement depends on the Reynolds number, especially for the larger interruption gap, for which the greatest enhancements occur in the higher range of the Reynolds number.

To rationalize these findings, it is relevant to take note of the gap-induced fluid flow processes. The basic mission of the gap is to enable the velocity (and temperature) profile that developed in the boundary layer upstream of the gap to relax (i.e. become more homogeneous), so that a new, thinner boundary layer can be initiated at the fin segment downstream of the gap. In addition to the molecular and turbulent diffusion active in the homogenization process, the presence of the gap opens the way for trailing-edge vortices to be spawned at the fin segment upstream of the gap and leading edge separation to occur at the segment downstream of the gap. It can be expected that the vigor of these processes depends on the gap width and on the Reynolds number, with the resulting heat transfer enhancement being that displayed in Fig. 2.

It is also worth noting that increases in the gap width must be made cautiously, since the fin surface area is reduced when the gap width is increased.

The relatively modest enhancements in evidence in Fig. 2 are believed due to the in-line positioning of the successive fin

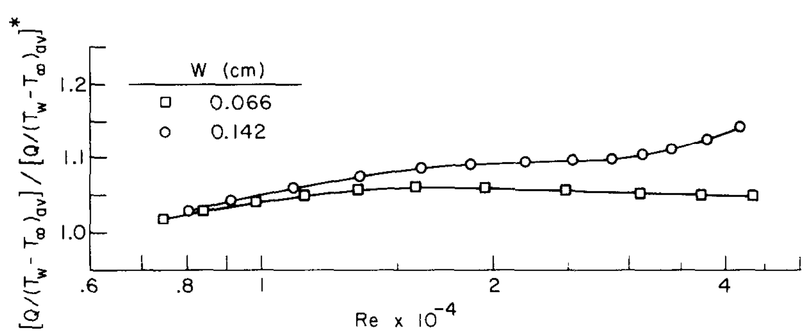

FIG. 2. Comparison of heat transfer rates for interrupted-fin and continuous-fin tubes.

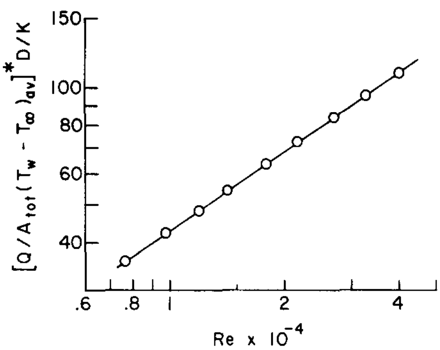

FIG. 3. Heat transfer results for a tube with continuous (uninterrupted) annular fins.

segments. It is likely that offset fin segments would yield greater enhancements, but with a corresponding pressure drop penalty.

For completeness, the Q^* results (continuous-fin case) are presented in Fig. 3. The ordinate group is a Nusselt number based on the total heat transfer area A_{tot} which includes the areas of the fin surfaces and of the exposed surfaces of the tube. The data lie on a straight line whose equation is

$$[Q/A_{tot}(T_w - T_\infty)_{av}]^* D/k = 0.0854\, Re^{0.675}. \tag{4}$$

The Nusselt number given by this equation is lower than that for an *unfinned* tube in crossflow [7, 8], typically by about 20%. If A_{tot} in equation (4) were replaced by the surface area A_{tube} of an unfinned tube with diameter D, then

$$[Q/A_{tube}(T_w - T_\infty)_{av}]^* D/k = 0.0854(A_{tot}/A_{tube}) Re^{0.675} \tag{5}$$

where $A_{tot}/A_{tube} = 9.6$ for the finned tube under investigation. Note that the Nusselt number on the LHS of equation (5) is identical in definition to that for an unfinned tube. Therefore, a comparison of equation (5) with literature correlations for the unfinned tube in crossflow yields a true measure of the enhancement due to finning (a factor of about $7\frac{1}{2}$ in heat transfer for a given temperature difference).

The circumferential variation of the tube wall temperature will now be examined, and Figs. 4 and 5 convey this information for tubes with interrupted and continuous fins, respectively. In each figure, the ratio of the local to average wall-to-freestream temperature difference is plotted as a function of the angular coordinate θ for each of five Reynolds numbers. The circumferential distribution of $(T_w - T_\infty)$ may also be regarded as an index of the circumferential distribution of the local heat transfer coefficient, with low values of the temperature difference corresponding to high values of the transfer coefficient and vice versa.

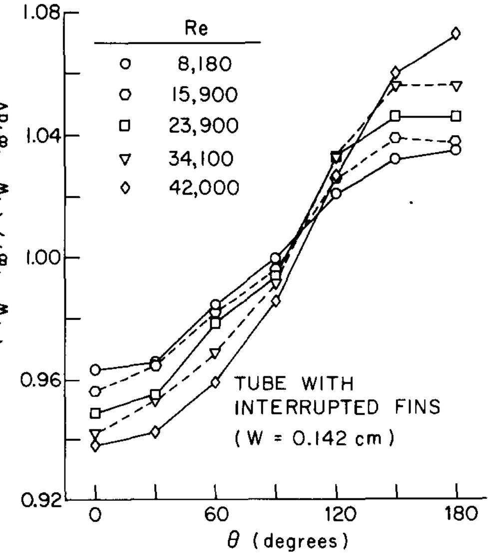

FIG. 4. Circumferential variation of the wall-to-freestream temperature difference for a tube with interrupted annular fins.

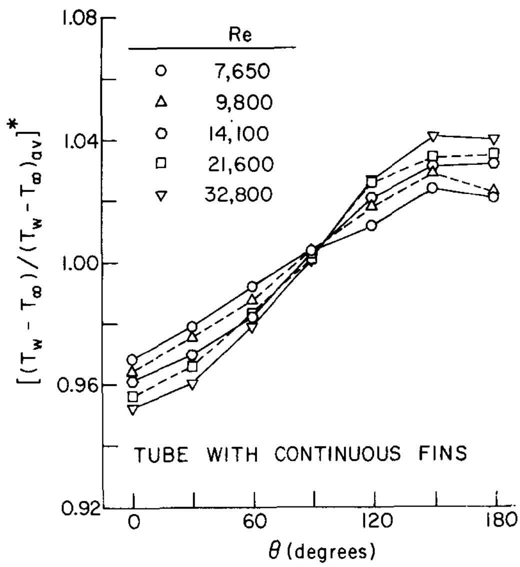

FIG. 5. Circumferential variation of the wall-to-freestream temperature difference for a tube with continuous (uninterrupted) annular fins.

Both figures indicate that the highest heat transfer coefficients occur at the forward stagnation point and that the lowest coefficients are at the rear stagnation point, with a more or less monotonic variation between. Also, both figures show that the extent of the circumferential variation increases with increasing Reynolds number.

The circumferential variations portrayed in these figures are modest. The smallest variation ($Re = 7650$ in Fig. 5) extends from $(T_w - T_\infty)/(T_w - T_\infty)_{av} \simeq 0.97$ at $\theta = 0°$ to 1.02 at $\theta = 180°$, while the largest variation ($Re = 42\,000$ in Fig. 4) extends from about 0.94 to about 1.07.

The extent of the circumferential variation is slightly but consistently affected by the presence of interruptions in the fins. In general, the overall variation (from $\theta = 0°$ to 180°) is about 2% greater when the fins are interrupted. This behavior is, in all likelihood, caused by the blockage of circumferential heat conduction in the fins by the interruption gaps.

CONCLUDING REMARKS

The present experiments have shown that periodic radial interruptions in annular fins affixed to a circular tube in crossflow serve to increase the rate of heat transfer relative to that for an otherwise identical tube with uninterrupted fins. The extent of the enhancement was more marked at larger interruption gap widths, especially at higher Reynolds numbers. The measured enhancements were moderate (in the 10% range) because the fin segments created by the successive radial interruptions were in-line with respect to each other.

For both the interrupted-fin and continuous-fin tubes, the local heat transfer was highest at the forward stagnation point and decreased more or less monotonically to a minimum at the rear stagnation point. The extent of the circumferential variations increased with increases in the Reynolds number. At a fixed Reynolds number, the presence of radial interruptions tended to slightly increase the circumferential variations.

The finning itself was found to be highly enhancing compared to an unfinned tube, yielding a $7\frac{1}{2}$-fold increase in the heat transfer rate.

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Effect of cross-flow on boiling heat transfer of refrigerant-12

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NOMENCLATURE

b	coefficient used in equation (5)
c_{sf}	constant in Rohsenow equation (2)
D	outside diameter of tube [m]
D_b	bubble diameter [m]
G	mass velocity [kg s^{-1} m^{-2}]
h	heat transfer coefficient [W m^{-2} K^{-1}]
h_{fg}	latent heat of vaporisation [J kg^{-1}]
k	thermal conductivity [W m^{-1} K^{-1}]
Nu	Nusselt number
n	index in the Kutateladze equation (11)
Pr	Prandtl number
q	heat flux [W m^{-2}]
Re	Reynolds number
T	temperature [K]
ΔT	wall superheat [K]
V	velocity [m s^{-1}].

Greek symbols

μ	dynamic viscosity of fluid [kg s^{-1} m^{-1}]
ρ	density [kg m^{-3}].

Subscripts

b	pool boiling, bubble
F	condition at mean film temperature
f	forced convection
fb	forced convection boiling
l	liquid
s	saturation condition
su	superficial vapour
w	heated surface condition.

INTRODUCTION

VERY little published work exists in the literature for cross-flow boiling of refrigerants. With R-12 in particular, such studies are almost non-existent. The purpose of this investigation was to determine experimentally the cross-flow nucleate boiling characteristics of R-12 at small velocities that are important for the design of flooded evaporators. This work is an extension of a previous study conducted by the authors on water to investigate the effect of cross-flow on boiling heat transfer [1].

Following the method of Rohsenow [2], a semi-empirical model has been proposed on the basis of superimposition of cross-flow effects over pool boiling. The experimental data of R-12 is seen to be successfully predicted by this model.

EXPERIMENTAL WORK

Figure 1 shows the schematic of the experimental apparatus used in the investigation. The test vessel was a rectangular steel container of 124 × 80 × 265 mm size, to hold the pool of boiling R-12. Stainless steel tubes with outside diameters of 16.0, 12.5 and 9.6 mm and each 11.4 mm long were successively employed as three test sections. One tube at a time was held horizontally in the test vessel by means of two copper strips. The tube was heated electrically; the power was supplied to it through the copper strips.

Multiple entry of the liquid was provided at the bottom of the test vessel through a distributor and two 100 sieve in^{-2} (16 sieve cm^{-2}) wire mesh baffles were placed in the flow path. Multiple outlets of the unevaporated liquid through a header, 50 mm above the test section level, were also provided in order to reduce the non-uniformity in the liquid flow across the test section. The vapour separator was provided to allow only liquid flow to the test vessel and the liquid level indicator showed the liquid level in the vessel. A bypass line was used to bypass most of the liquid and allow only the desired amount of refrigerant for make-up in the test vessel during the pool boiling runs. The vapour from the test vessel passed through a back pressure regulating valve to the compressor. The bypass liquid was sent to the auxiliary evaporator to ensure its complete evaporation before reaching the compressor. The compressed refrigerant then flowed through an oil separator, water- and air-cooled condensers, filter and a subcooler and finally through the rotameter to the expansion valve.

Test surface temperatures were measured by means of calibrated 32 gauge copper–constantan thermocouples carefully spot-welded on the outside surface at the top side and the bottom at two sections on each tube located 29 mm from the ends. The bulk temperature was measured by a thermocouple dipped in the liquid. All the tests were performed at a boiling temperature of 5°C. The stabilized power supply to the test section was made through an autotransformer and measured by means of a precision grade voltmeter and ammeter.