A GUARD-HEATER METHOD FOR DETERMINING HEAT TRANSFER COEFFICIENTS OF FINNED TUBES

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A GUARD-HEATER METHOD FOR DETERMINING HEAT TRANSFER COEFFICIENTS OF FINNED TUBES

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LIST OF SYMBOLS

A	area, sq. ft.
C	a constant
c_p	specific heat, $B/lb_m\ ^oF$
D	diameter, ft.
h	heat transfer coefficient, $B/hr\ ft^2\ ^oF$
k	thermal conductivity, $B/hr\ ft\ ^oF$
L	length of tube test section, ft.
N_{nu}	Nusselt number, $\frac{h_o\ D_e}{k}$
N_{pr}	Prandtl number, $\frac{\mu\ c_p}{k}$
N_{re}	Reynolds number, $\frac{V\ D_e}{\nu}$
P	pressure, in. H_2O
q	heat transfer, B/hr
t	fin thickness, ft.
T	temperature, ^oF
V	velocity, ft/hr
w	fin height, ft.
θ	temperature difference, ^oF
μ	dynamic viscosity, $lb_m/hr\ ft$
ν	kinematic viscosity, ft^2/hr
ϕ	fin efficiency, per cent

Subscripts

a	air
b	base of fin on tube
d	downstream
e	effective or equivalent
f	fin
i	inside
m	mean value
o	overall fin and tube surface
r	fin edge
s	tube base surface
u	upstream
w	wall, tube base surface
1	upstream from orifice
2	across orifice

CHAPTER I

INTRODUCTION

The addition of extended surfaces, such as fins and splines, which are intended to increase the heat transfer through the original surface, is usually justified when the thermal resistance of the original surface is large, relative to other resistances in the system. In the case of tubes with liquids flowing through them, for instance, the thermal resistance due to the liquid may be small, relative to the resistance associated with a still or moving gas surrounding the tubes. The addition of fins to the outsides of the tubes, in this case, by providing additional surface area, should increase the heat transfer through the tubes.

Since the temperatures on the fin surface will be less than temperatures on the original surface because of the temperature drop in the fin material due to heat conduction, the heat transfer will not increase in the same proportion as the amount of added fin area. An expression, called the fin efficiency, can be calculated to account for the difference in temperature between the fins and the original base surface. The concept of fin efficiency is also useful in defining a coefficient of heat transfer of extended surface.

It is difficult to estimate the overall coefficient of heat transfer between two fluids through a finned tube because of the problem of predicting a heat transfer coefficient over the finned surface of the tube. The heat transfer coefficient over fins is determined from experimental tests, where, in most cases, this coefficient is determined indirectly, such as by subtraction of known thermal resistances from the measured overall thermal resistance.

The purpose of this research was to devise a test procedure and means of instrumentation so that the coefficient of heat transfer over finned tubes could be determined directly from the experimental measurements and from knowledge of the fin efficiency.

CHAPTER II

LITERATURE REVIEW

The earlier work on the conduction heat transfer through extended surfaces, especially that work of Murray (1) and Douglass (2), was extended by Gardner (3) in generalizing the analytical solution for conduction through surfaces of various geometries. In addition to assuming an isotropic, homogeneous fin material with constant thermal conductivity, Gardner assumed

(1) a uniform temperature of the surrounding fluid,

(2) no heat transfer through the fin edge,

(3) the coefficient of heat transfer to be uniform over all fin surface,

(4) a uniform temperature across the root of the fin,

(5) there was no temperature gradient in the fin in the axial direction of the tube.

Gardner's solutions for the fin efficiencies[1] of many different surfaces, in the form of Bessel functions, were also presented in a useful graphical form in his paper.

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[1] The efficiency of extended surfaces is defined as the ratio of the heat actually transferred through the fin surface to the heat transfer through the fin surface had the entire surface of the fin remained at the temperature of the fin root.

Figure 5 of that paper was used by the present author in later calculations.

Assumption (2) above, by Gardner, which neglects the heat transferred through the fin edge, was necessary only to simplify the mathematical expression for efficiency. Harper and Brown (4) suggest that the additional heat flux through a fictitious surface resulting from an extension of the fin height will account for neglecting the edge heat transfer.

Assumption (3) concerning a uniform coefficient over all fin surface is far from being true in practice, particularly for circumferentially finned tubes. Many studies, such as those of Thompson (5), Lemmon (6), and McAdams (7), indicate that there is usually a large variation of the forced convection coefficient with angle around the tube and fin and with radius along a fin. The distribution in the magnitude of the coefficient appears to be dependent on the Reynolds number of the flow and on the spacing between the fins along the tube.

Williams and Katz (8) determined the surface coefficient for copper and admiralty metal finned tube bundles from the value of the overall coefficient which was obtained by the Wilson plot procedure. Their tests included the use of water, glycerine and lube oil as the shell side fluids. The resulting data were correlated very well by the relationship

$$N_{nu} = C(N_{re})^{.65} (N_{pr})^{.375} \left(\frac{\mu}{\mu_w}\right)^{.14}$$

The values of the constant C ranged from 0.115 to 0.182 depending on the tube arrangement and fin material.

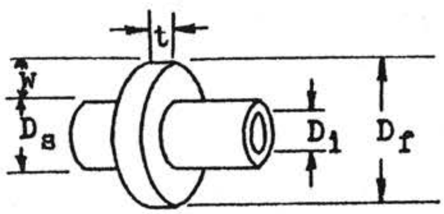
McCright (9) measured the surface temperature of a single finned tube while using an unguarded electric heating element to provide for heat fluxes. His data were reduced and compared to that obtained in this investigation.

A guarded electric heater similar to the one used in this investigation was used earlier by Snyder (10) in a somewhat similar fashion. Snyder's heater replaced, and thereby simulated, a tube in a bank of plain tubes in crossflow with air.

CHAPTER III

THE THESIS PROBLEM

The following discussion pertains particularly to the circumferentially finned tube which was used in this research. A simplified single fin and tube segment is represented in Figure 1. The complete finned tube consists of many equally spaced fins along its length.



$A_f = 2 \times nL \times \frac{\pi}{4}(D_f^2 - D_s^2)$

$A_s = \pi D_s L - nL\pi D_s t$

$A_o = A_f + A_s + A_r$

$A_r = nL\pi D_f t$

Fig. 1 Section of Circumferentially Finned Tube.

The significance of the fin efficiency expression is in providing a tool with which to form a logical definition of the surface coefficient of heat transfer on extended surfaces. Two ways of defining a coefficient are discussed here. The mean coefficient of heat transfer may be defined over all extended and base area by

$$q = h_o \int_0^{A_o} \theta \, da \qquad (1)$$

If equation (1) is written as

$$q = h_o A_o \theta_m \qquad (2)$$

then θ_m is interpreted as the mean temperature excess for all surface. The heat transfer may also be written as the sum of the heat flows through fin and tube surfaces

$$q = q_f + q_s = h_f\, A_f\, \theta_f + h_s\, A_s\, \theta_s \qquad (3)$$

Assuming that the heat transfer coefficient is the same for all tube and fin surfaces, $h_f = h_s = h_o$, and that the tube surface temperature is the same as the fin root temperature, $\theta_s = \theta_b$, the equation for fin efficiency is

$$\phi = \frac{\text{actual } q_f}{q_f \text{ for } \theta_f = \theta_b} = \frac{h_f\, A_f\, \theta_f}{h_f\, A_f\, \theta_b} = \frac{\theta_f}{\theta_b} = \frac{\theta_f}{\theta_s} \qquad (4)$$

and equation (3) becomes

$$q = h_o \theta_s\, (\phi A_f + A_s) \qquad (5)$$

If equation (5) is multiplied and divided by A_o,

$$q = h_o\, A_o \left[\theta_s\, \left(\phi\, \frac{A_f}{A_o} + \frac{A_s}{A_o}\right)\right] \qquad (6)$$

The quantity in brackets in (6) may be interpreted as the mean temperature excess, θ_m, over all outside tube and fin surface, as in equation (2).

An alternative procedure is to define the quantity in parenthesis in equation (5) as the effective area for heat transfer,

$$A_e = (\phi A_f + A_s) \qquad (7)$$

This method was used by Williams and Katz (8) and was followed by the present author. In other words, the outside coefficient of heat transfer was defined for the finned tube as

$$h_o = \frac{q}{\theta_s\, A_e} \qquad (8)$$

The problem in this research was to develop a procedure for obtaining the measurements needed to calculate the surface coefficient for the finned tube from equation (8). In equation (8), the total heat transfer q, the mean temperature excess of the tube surface, θ_s, and the effective tube area, A_e, are required.

An electric heater, in tubular form, guarded at the ends, was considered as a suitable means of providing for heat fluxes which could be readily controlled and measured.

Considering the relative ease of instrumentation, the mean tube surface temperature was defined in terms of the temperatures at several stations on the periphery of the tube and at the midpoints between the fins.

The effective area of the tube could be calculated when the fin efficiency was determined. The fin efficiency was available from Gardner's solutions, previously discussed.

CHAPTER IV

TEST EQUIPMENT

The finned tube employed in the experiment was installed in the 14 in. by 5 in. test section of a sheet metal duct system, shown in Figure 2. The test section was preceded by a bank of 1 in. straightening tubes of 1 ft length. Following the test section was a transition section and a 7 ft. length of straight 6 in. nominal diameter pipe in which a square-edged orifice, with flange taps, was installed. A stove pipe damper was used as a bleed valve for regulating the flow of air induced by the compressor through the test section.

Finned Tube Description

The finned tube was a model 6K16, supplied through the courtesy of the Griscom-Russel Company. The fins on the tube were helically wound at the rate of 8 fins per inch of tube length. The tube was 3/4 in. O.D. and 0.62 in. I.D. The fins were 13/32 in. high with a thickness of 0.014 in. The tube material was aluminum. The tube was tested as received from the manufacturer, in a clean condition, with no surface preparation of any sort. The tube was cut to a length of about 14 in. prior to installation.

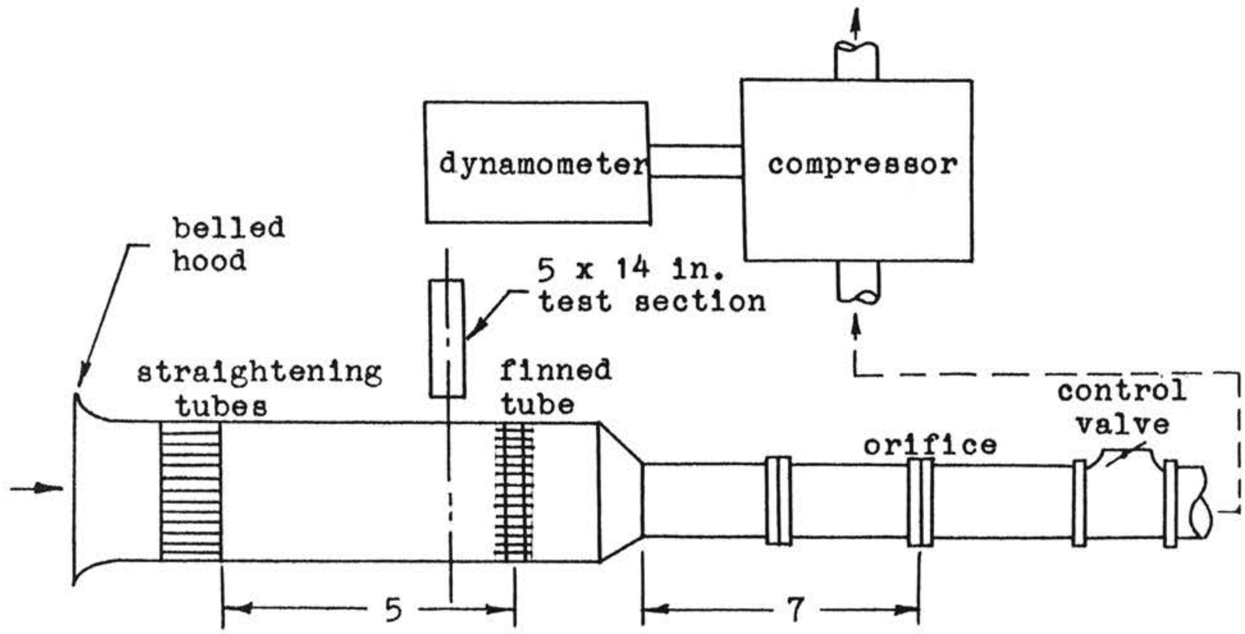


Fig. 2 Plan of Test Apparatus

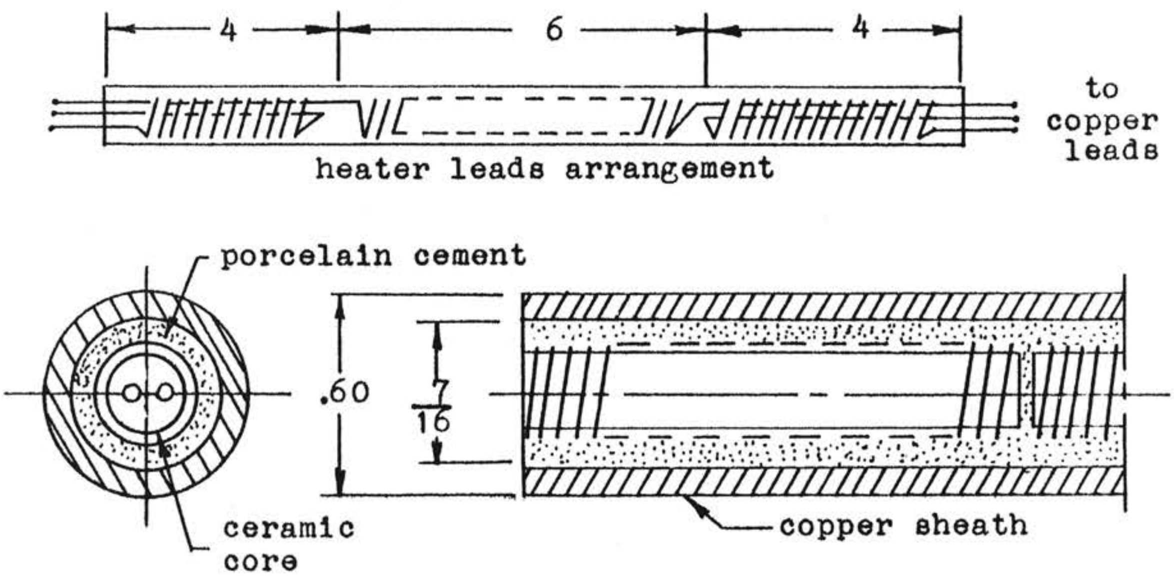


Fig. 3 Electric Guard-Heater Details

The Guard-Heater

Details of the electrical guarded heater are shown in Fig. 3. The heater includes three separately controlled heating circuits: two guard-heater coils, each 4 in. long, and one test heater coil, 6 in. long.

Each heating coil was lathe-wound from #26 gage B & S chromel wire stock. The coils were secured on short cylindrical sections of 7/32 in. O.D. high-temperature ceramic. The heater leads were brought out the end of the heater assembly through holes along the centers of the ceramic sections. Assembly was completed by positioning the heater core in a cylindrical copper sheath of 0.60 in. O.D. and 7/16 in. I.D. and filling the annular space between the core and sheath with a high-temperature plastic porcelain cement. A spacer of ceramic cement was built up on the center of the heater core before assembly into the sheath to provide for centering the core.

Preliminary tests with this heater showed that the resistance of the test heater coil was 36 ohms and of the guard heaters, 27 ohms each. The surface of the copper sheath oxidized badly at high temperatures, necessitating polishing of the heater surface after each test. Because of the close spacing of the heater coils, short circuit failures occurred with the first two heaters which were constructed. A third heater, which was used successfully in these tests, failed in a similar way when additional tests were attempted.

The effect of different degrees of contact of the heater with the inside finned tube walls was briefly investigated. The heater was rotated about 90° for successive tests without changing its axial position in the tube. The average surface temperature measurements recorded for the different heater positions differed from each other by less than 3°F. After a high temperature test of the heater when installed in the finned tube, it was often difficult to remove the heater from the tube, apparently due to oxidation of the heater surface. The build-up of oxide on the heater surface might have been sufficient to effectively "seal" the gap between the heater and tube surfaces, although this effect was not investigated beyond the cursory steps taken above.

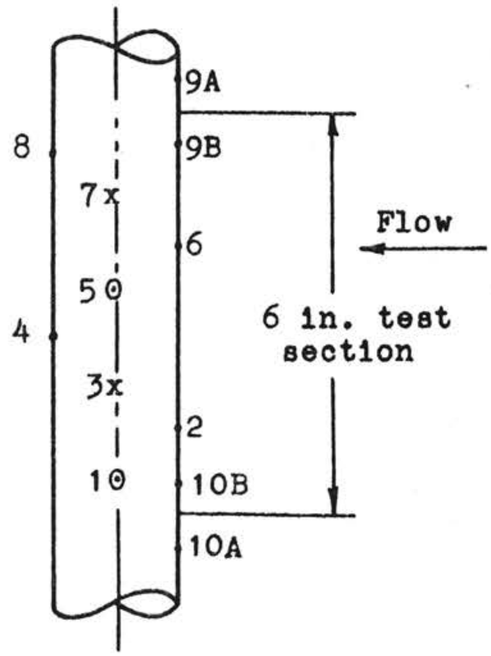
Instrumentation

The central 6 in. of the finned tube was considered to be the test section of the tube and the central coil of the heater was aligned accordingly. Thermocouples were affixed to the tube only on and near the test section, as illustrated in Fig. 4. Thermocouples 1 through 8 were attached, consecutively, about 5/8 in. apart along the tube axis and 90° apart around the tube circumference. Each Thermocouple junction was attached, as nearly as possible, midway between adjacent fins.

All thermocouples were made from iron-constantan #30 B. & S. gage wire manufactured by Leeds and Northrup Company. The thermocouple junctions were made by electro-welding under oil so as to form unoxidized beads.

The circuits of thermocouples 1 through 8 on the tube surface were connected so that measurements of the average emf could be made in parallel or the individual emfs could be measured separately. Therefore, the junctions of these thermocouples were electrically insulated from the tube. Insulation of the junctions from one another was achieved by applying two coats of high-temperature varnish to the junction beads and adjoining wire and then cementing the beads into 1/32 in. diameter drilled holes on the surface of the tube with plastic porcelain cement. The cement thickness on the thermocouple junction beads was on the order of 0.01 in. The thermocouple leads were wrapped at least one complete turn around the tube base and were brought away from the tube on the downstream side.

The circuit for thermocouple measurements is shown in Fig. 5. The circuit was made so that measurements of the average tube temperature were accomplished normally with the thermocouples in parallel. In order to compare the average of measurements from single thermocouples, the circuit could also be switched to a series circuit. The required common junctions for the parallel thermocouple circuit were formed by the use of bottles of mercury. All



Spacing

Thermocouples 1 through 8 are spaced 0.667 in.apart longitudinally.

Thermocouples 9A and 10A are 0.5 in. away from the test section; 9B and 10B are 0.5 in. inside the test section.

x - back

⊙ - front

Fig. 4 Thermocouple Locations on Test Section of Finned Tube.

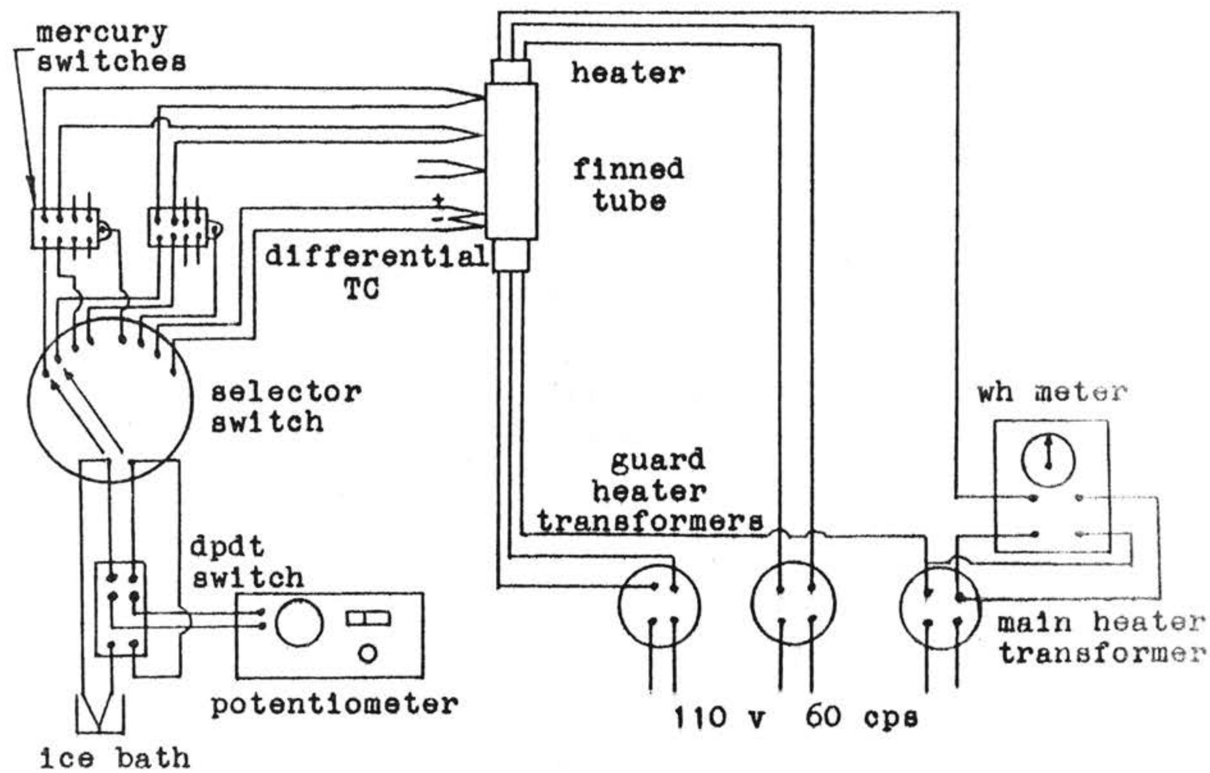


Fig. 5 Heater and Thermocouple Circuits

positive leads were inserted in one bottle of mercury, all negative leads in another. Removal of all leads from the mercury returned the circuit to a series circuit for measurements from the individual thermocouples. Once the leads were removed from the mercury, however, care had to be taken to remove the high resistance oxide film that formed on the portion of the thermocouple leads that had been in contact with the mercury before the leads were replaced in the mercury.

In preliminary tests, the average of tube temperatures measured singly was, within the accuracy of measurement, the same as the single measurement in parallel of the average tube temperature. For the actual tests, only the parallel circuit was used.

Thermocouples were also installed in the air stream at the entrance section of the duct and in the air stream at damper valve section for determining temperatures before and after the test section, respectively.

A Leeds and Northrup potentiometer, model no. 8662, a calibrated standard mercury thermometer, and an ice bath were used for all thermocouple measurements.

Variable power transformers of 0 to 140 volt range were used to control the power supplied to the test and guard heaters. The electrical power to the test heater was measured with a portable induction test watt-hour meter, type J-3, manufactured by the Sangamo Electric Company. Power circuits for the heater are shown in Fig. 5.

CHAPTER V

TEST PROCEDURE

With the desired flow rate of air established, as indicated by the orifice manometer, the voltages of the guard heaters were adjusted until differential thermocouple pairs, 9 and 10 (Fig. 4) indicated a temperature difference of 1°F between the guard and the test sections of the tube. Balancing of the guard heaters required about 10 to 15 minutes. When the above condition was satisfied and when spot checks on tube temperatures indicated a reasonably steady condition, the test was begun.

Temperature and manometer readings were recorded every 5 minutes. The count on the watt-hour meter was recorded at the beginning and end of each test, which was of 15 minutes duration, as timed with a stop watch.

The barometric pressure was constant at 29.15 in. Hg. and the ice bath temperature was constant at 32.0°F for all tests.

The time required to attain steadiness in the air flow and in temperatures between successive test runs was about 15 to 20 minutes, depending primarily on the speed with which the guard heaters could be balanced.

Test runs at thirteen different tube Reynolds numbers were conducted, the Reynolds numbers ranging from 1980 to 11,600.

Test measurements are shown in TABLE I.

TABLE I

TEST MEASUREMENTS FROM HEAT TRANSFER
STUDIES OF A SINGLE FINNED TUBE

Test No.	Time min.	w-h meter rev.	ΔP_2 in. H_2O	ΔP_1 in. H_2O	T_s mv	T_u mv	T_d mv
1	0	0.00	0.30	0.00	8.33	1.58	1.87
	5		0.30	0.00	8.41	1.56	1.88
	10		0.30	0.00	8.57	1.56	1.86
	15	98.12	0.30	0.00	8.44	1.56	1.85
2	0	0.00	0.70	-0.05	6.43	1.56	1.83
	5		0.70	-0.05	6.47	1.56	1.84
	10		0.70	-0.05	6.40	1.57	1.82
	15	96.88	0.70	-0.05	6.51	1.57	1.83
3	0	0.00	1.10	-0.10	5.61	1.55	1.75
	5		1.10	-0.10	5.61	1.57	1.80
	10		1.10	-0.10	5.70	1.58	1.78
	15	97.60	1.10	-0.10	5.72	1.57	1.78
4	0	0.00	1.62	-0.15	5.14	1.56	1.62
	5		1.62	-0.15	5.12	1.49	1.57
	10		1.62	-0.15	5.13	1.49	1.57
	15	98.25	1.62	-0.15	5.17	1.45	1.64

TABLE I (Continued)

Test No.	Time min.	w-h meter rev.	ΔP_2 in. H_2O	ΔP_1 in. H_2O	T_s mv	T_u mv	T_d mv
5	0		2.13	-0.15	4.68	1.27	1.45
	5		2.13	-0.15	4.72	1.28	1.43
	10		2.13	-0.15	4.67	1.28	1.45
	15	98.38	2.13	-0.15	4.67	1.28	1.45
6	0	0.00	2.87	-0.25	4.38	1.28	1.42
	5		2.87	-0.25	4.40	1.30	1.44
	10		2.87	-0.25	4.42	1.30	1.44
	15	97.88	2.87	-0.25	4.40	1.31	1.46
7	0	0.00	3.52	-0.35	4.24	1.32	1.40
	5		3.52	-0.35	4.25	1.32	1.42
	10		3.52	-0.35	4.27	1.33	1.40
	15	98.00	3.52	-0.35	4.28	1.33	1.43
8	0	0.00	4.35	-0.40	4.09	1.36	1.43
	5		4.35	-0.40	4.10	1.34	1.44
	10		4.35	-0.40	4.11	1.35	1.43
	15	98.32	4.35	-0.40	4.12	1.36	1.46
9	0	0.00	5.30	-0.50	4.13	1.51	1.56
	5		5.30	-0.50	4.14	1.50	1.59
	10		5.30	-0.50	4.15	1.49	1.57
	15	96.89	5.30	-0.50	4.15	1.52	1.58

TABLE I (Continued)

Test No.	Time min.	w-h meter rev.	ΔP_2 in. H_2O	ΔP_1 in. H_2O	T_s mv	T_u mv	T_d mv
10	0	00.00	6.45	-0.60	3.94	1.52	1.60
	5		6.45	-0.60	3.92	1.53	1.60
	10		6.45	-0.60	3.91	1.51	1.58
	15	95.70	6.45	-0.60	3.92	1.51	1.59
11	0	0.00	7.45	-0.75	3.80	1.49	1.52
	5		7.45	-0.75	3.80	1.50	1.55
	10		7.45	-0.75	3.79	1.50	1.58
	15	95.32	7.45	-0.75	3.75	1.51	1.55
12	0	0.00	8.65	-0.85	3.63	1.49	1.54
	5		8.65	-0.85	3.56	1.49	1.54
	10		8.65	-0.85	3.59	1.50	1.58
	15	95.20	8.65	-0.85	3.63	1.50	1.52
13	0		10.20	-1.00	3.54	1.50	1.52
	5		10.20	-1.00	3.54	1.50	1.55
	10		10.20	-1.00	3.53	1.50	1.55
	15	95.38	10.20	-1.00	3.53	1.50	1.56

CHAPTER VI

SAMPLE CALCULATIONS

The following sample calculations are from the data for Test 1, Table I.

Temperatures

The emf-temperature iron constantan thermocouple tables in the National Bureau of Standards Circular 561 (12) were used to obtain all temperatures from emf readings or averages of emf readings, where appropriate.

Air Properties

All properties of air were obtained from Table II-2, Elements of Heat Transfer (13). At an average air temperature of 92.5°F,

$$k = 0.0153 \text{ B/hr ft F}$$

$$\nu = 0.633 \text{ ft}^2\text{/hr}$$

$$\mu = 0.0455 \text{ lb}_m\text{/hr ft}$$

$$N_{pr} = 0.712$$

and at the surface temperature of the tube, 316°F,

$$\mu_w = 0.586 \text{ lb}_m\text{/hr ft.}$$

Velocity

The volumetric flow of air through the orifice was determined by the method outlined in The Orifice Meter (14). The following terminology is defined on page 36.

$Q_h = C' \sqrt{h_w \times P_f}$

$P_f = 14.29$ psia

$h_w = 0.30$ in. H_2O

$C' = F_b \times F_r \times Y_1 \times F_{pb} \times F_{tb} \times F_{tf} \times F_g \times F_{pv}$

$C' = 3711.4 \times 1.0175 \times 0.9998 \times 1.0008 \times 1.0481 \times 0.9680 \times 1.000 \times 1.000 = 3845$

$Q_h = C' \sqrt{h_w \times P_f} = 3845 \sqrt{0.30 \times 14.29} = 7970$ cfh

Test section area, $A = 5 \times 14/144 = 0.486$ sq ft

$V = Q_h/60 \times A = 7960/60 \times 0.486 = 273$ fpm

Heat Transfer through Test Section

$$q = \frac{(\text{Meter constant, w-h/rev})\ (\text{rev. of meter})\ (3.413\ \text{B/w-h})}{\text{time of test, hr.}}$$

$$q = \frac{(0.6)\ (98.12)\ (3.413)}{0.25} = 804\ \text{B/hr}$$

Finned Tube Areas

Fin area, $A_f = \pi/4(D_f^2 - D_s^2) \times 2 \times n \times L$

$$A_f = \frac{\pi/4(1.75^2 - 0.75^2) \times 2 \times 8 \times 6}{144} = 1.31 \text{ sq ft}$$

Tube area, $A_s = \pi D_s L - nL\pi D_s t$

$$A_s = \frac{\pi(0.75)(6) - (8)(6)\pi(0.75)(0.014)}{144} = 0.0875 \text{ sq ft}$$

Fin edge area, $A_r = \pi D_f t L n$

$$A_r = \frac{\pi(1.75)(0.014)(6)(8)}{144} = 0.0257 \text{ sq ft}$$

Total finned tube area, $A_o = A_f + A_s + A_r$

$$A_o = 1.31 + 0.0875 + 0.0257 = 1.423 \text{ sq ft}$$

Fin Efficiency and Heat Transfer Coefficient

Fig. 5 of Reference 3, for a fin diameter to tube diameter ratio of 2.34 is shown in Fig. 6.

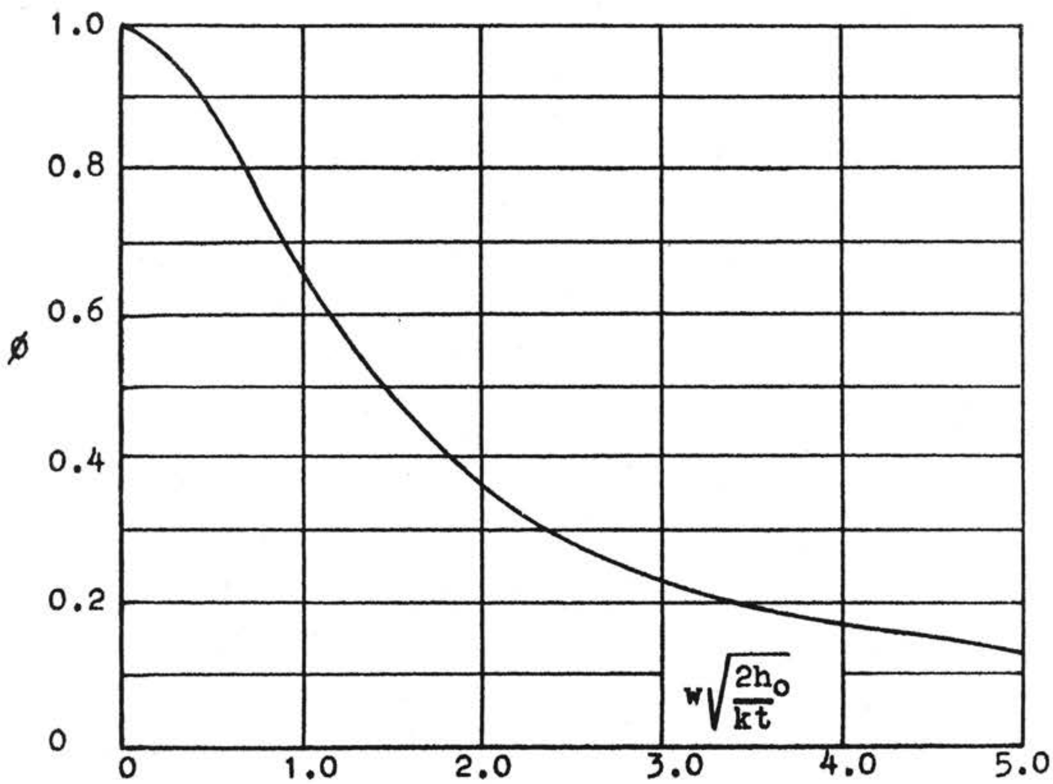


Fig. 6. Gardner's Solution for Fin Efficiency for a Fin to Tube Diameter Ratio of 2.34

The trial-and-error solution for fin efficiency, ϕ, and the heat transfer coefficient, h_o, is obtained by assuming a value for efficiency, calculating h_o from

$$h_o = \frac{q}{\theta_s\ (A_f\phi + A_s)}$$

and then calculating the parameter $w\sqrt{\frac{2h_o}{kt}}$ to obtain a new value for the fin efficiency from Fig. 6.

Assume $\phi = 0.90$

$$h_o = \frac{q}{\theta_s\ (A_f\ \phi + A_s)} = \frac{804}{229(1.334 \times 0.90 + 0.0875)} = 2.73\ \frac{B}{hr\ ft^2\ F}$$

Parameter $w\sqrt{\frac{2h_o}{kt}} = 0.5/12\sqrt{\frac{2 \times 2.73 \times 12}{124 \times 0.014}} = 0.256$

From Fig. 6, $\phi = 0.97$

$$h_o = \frac{804}{229(1.334 \times 0.97 + 0.0875)} = 2.54\ B/hr\ ft^2\ F$$

Parameter $w\sqrt{\frac{2h_o}{kt}} = 0.5/12\sqrt{\frac{2 / 2154 \times 12}{124 \times 0.014}} = 0.248$

From Fig. 6, $\phi = 0.97$, unchanged.

Equivalent Tube Diameter

The equivalent tube diameter is defined as the diameter of a plain tube having the same inside diameter and weight as that of the finned tube.

weight of plain tube of D_e = weight of finned tube of D_f

$$L \times \pi/4(D_e^2 - D_i^2) = nLt \times \pi/4(D_f^2 - D_s^2) + \pi/4L(D_s^2 - D_i^2)$$

$$D_e = \sqrt{8t(D_f^2 - D_s^2) + D_s^2}$$

$$D_e = \sqrt{(8)(0.014)(1.75^2 - 0.75^2) + 0.75^2}\ /12 = 0.0765 \text{ ft}$$

Dimensionless Groups for Equation (9)

Nusselt number, $N_{nu} = \dfrac{h_o\ D_e}{k} = \dfrac{(2.54)(0.0765)}{0.0153} = 12.7$

Reynolds number, $N_{re} = \dfrac{V\ D_e}{\nu} = \dfrac{(273 \times 60)(0.0765)}{0.633} = 1980$

Viscosity correction, $(\dfrac{\nu}{\nu_w})^{-0.14} = (\dfrac{0.0455}{0.0585})^{-0.14} = 1.036$

$(N_{nu})\ (N_{pr})^{-0.375}\ (\dfrac{\nu}{\nu_w})^{-0.14} = (12.7)(1.136)(1.036) = 14.9$

The calculated results for Tests 1 through 13 are shown in TABLE II.

TABLE II

CALCULATED RESULTS FROM HEAT TRANSFER STUDIES
ON A SINGLE FINNED TUBE

Test No.	T_a °F	T_s °F	θ_s °F	V fpm	h_o B/hr ft^2 F	ϕ dim	N_{re} dim	N_{nu} dim	$(\mu/\mu_w)^{-.14}$ dim	$(N_{nu})(N_{pr})^{-.375}(\mu/\mu_w)^{-.14}$ dim
1	92.5	316	229	273	2.54	.97	1980	12.7	1.036	14.9
2	91.5	251	164	415	3.60	.94	3020	18.0	1.027	21.0
3	90.5	225	138	520	4.30	.93	3780	21.5	1.023	25.0
4	86.0	208	124	633	4.88	.93	4680	24.6	1.021	28.5
5	80.0	192	115	718	5.29	.93	5320	26.8	1.020	31.0
6	80.5	183	105	833	5.71	.92	6270	28.9	1.019	33.4
7	80.5	178	99	923	6.03	.92	6950	30.6	1.018	35.3
8	81.5	173	93	1050	6.53	.91	7880	33.1	1.017	38.2
9	86.0	174	89	1138	6.75	.91	8420	34.0	1.016	39.2
10	87.0	167	81	1253	7.35	.90	9250	37.0	1.015	42.6
11	86.0	163	78	1342	7.65	.90	9950	38.5	1.014	44.3
12	85.5	156	71	1445	8.43	.89	10700	42.4	1.013	48.7
13	85.5	154	69	1565	8.69	.89	11600	43.7	1.012	50.2

CHAPTER VII

RESULTS AND COMPARISONS WITH OTHER INVESTIGATORS

The results were expressed and compared with results of other investigators by use of the equation

$$\left(\frac{h_o D_e}{k}\right) = C\left(\frac{V\ D_e}{\nu}\right)^{.65} \left(\frac{c_p \mu}{k}\right)^{.375} \left(\frac{\mu}{\mu_w}\right)^{.14} \qquad (9)$$

which was suggested by Williams and Katz (8). In equation (9) D_e is the equivalent diameter of the finned tube defined as the diameter of a plain tube having the same inside diameter and the same weight of metal as the finned tube. V is the face velocity at the tube. All properties of the shell side fluid, air, are taken at the mean temperature of the air, before and after the tube, with the exception of μ_w, taken at the surface temperature of the tube. The coefficient h_o is the same as that coefficient defined by equation (8).

The calculated data from this experiment are shown in Table II. The coordinates for the plotted data were the quantity $\left(\frac{h_o D_e}{k}\right)\left(\frac{c_p \mu}{k}\right)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-.14}$ as ordinate and the Reynolds number, $\frac{VD_e}{\nu}$, as abscissa, as shown in Fig. 7. The straight line which appears to best fit the plotted data has a slope of 0.65, which corresponds to the exponent of the Reynolds number in equation (9). The value of the coefficient C in equation (9) was determined from this straight line to be 0.115. The straight line fits the test results to within about 5 per cent, in the worst cases.

The results in Fig. 7 are also plotted in Fig. 8 with the results of Williams and Katz and of McCright (9). Some details concerning these tests are summarized in TABLE III.

TABLE III

FINNED TUBE MATERIALS

Investigator	Curve no. on Fig. 7	Value of C in eqn (7)	Tube(s)	Tube material	Est. Fin efficiency %
Williams, Katz	(a)	0.143	50 tubes 0.735 OD	admiralty	85
Williams, Katz	(b)	0.115	114 tubes 0.486 OD	copper	95
Williams, Katz	(c)	0.182	40 tubes 0.620 OD	admiralty	85
McCright	(d)	0.187	one tube 0.625 OD	cupro-nickel	65
Burley	(e)	0.115	one tube 0.750 OD	aluminum	90

Williams and Katz correlated measurements on bundles of plain and finned tubes. The coefficient h_o in their investigations was determined from the outside thermal resistance found by the Wilson plot procedure.

The measurements of McCright were determined from a single finned tube with the use of an unguarded electric heater and with measurements of the tube surface temperature.

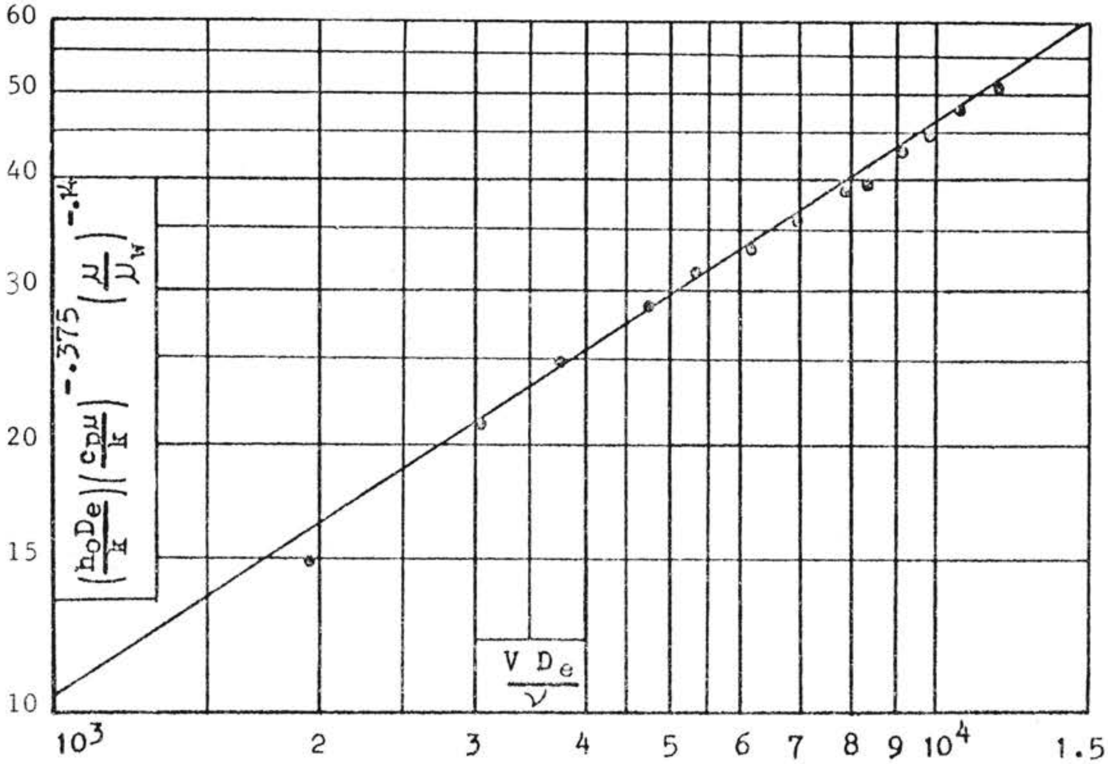


Fig. 7 Heat Transfer Characteristics for a Single Aluminum Finned Tube in Crossflow with Air.

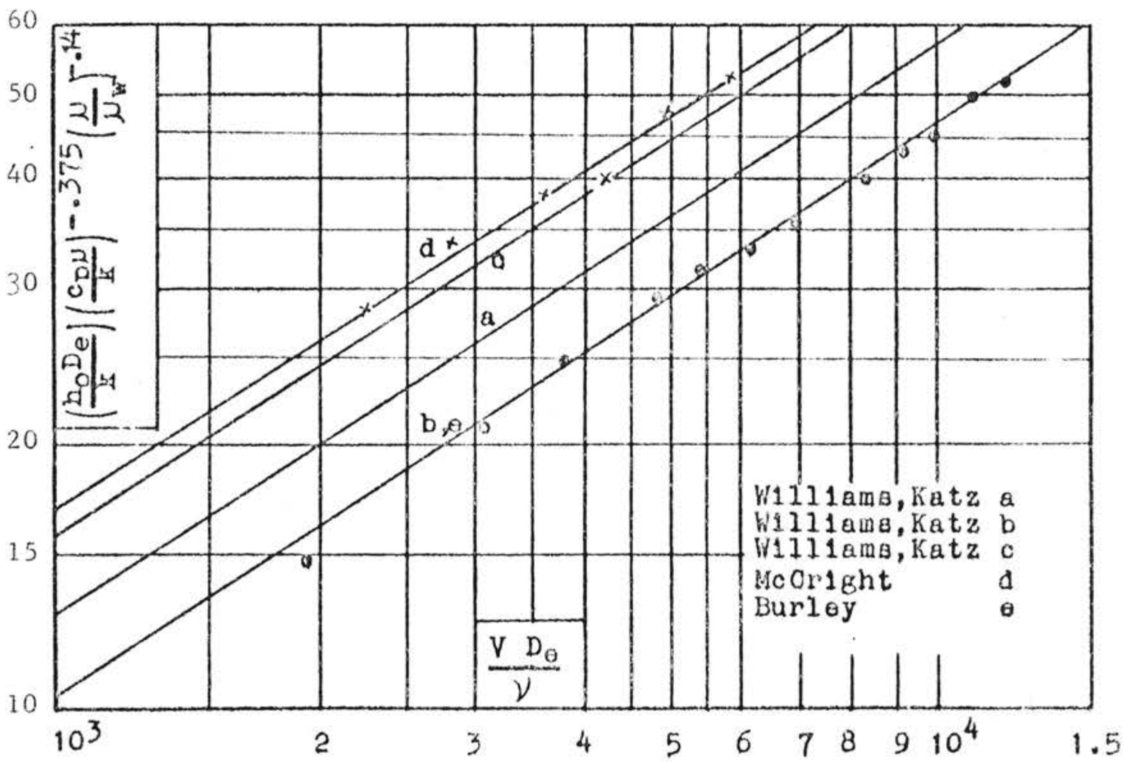


Fig. 8 Comparison of Heat Transfer Characteristics of Single Finned Tubes and Bundles of Finned Tubes.

McCright's original data have been put in the form of equation (9) by the present author.

Referring to Fig. 8, the author's data fit curve (b) for the copper tube bundle of Williams and Katz with the same value of C, 0.115, as a coefficient in equation (9). Also McCright's data compare favorably with curve (c) for the admiralty metal tube bundle of Williams and Katz.
One might suspect a correlation in the above cases on the basis of the fin efficiencies, which are near each other. However, curves (a) and (c) by Williams and Katz are each for admiralty tubes of about the same efficiency, but of slightly different tube bundle arrangements, and the curves differ from one another by about 30 per cent.

CHAPTER VIII

DISCUSSION

The results for the single finned tube studied established that the heat transfer coefficient is proportional to the velocity to the 0.65 power, which is in agreement with the results of other investigators cited. An agreement in the magnitude of the coefficients was not establised.

Fundamentally, there appear to be two reasons why results of finned tube investigations have not been correlated entirely successfully. First, a mathematically-derived fin efficiency expression based on highly simplifying assumptions, such as a uniform surface coefficient and surrounding temperature, is never entirely correct because the assumptions are not satisfied in practice. The degree of applicability of such an expression is always uncertain in a particular application. Secondly, simplified fin efficiency expressions do not lend themselves well to experimental applications because the temperature terms resulting in the efficiency expressions are not those temperatures which are readily obtainable experimentally.

Since, for some finned tube geometries, it would be desirable to instrument the tube on the base surface between the fins, the following concept of fin efficiency suggests itself. If, in developing the relationship for the heat transfer, discussed in CHAPTER II, the tube base and fin root temperatures had not been assumed to be equal, then the expression for the fin efficiency, equation (4), would remain

$$\phi = \frac{\theta_f}{\theta_b}$$

If this were combined with equation (3)

$$q = h_f \, A_f \, \theta_f + h_s \, A_s \, \theta_s$$

then the heat transfer coefficient could be defined as

$$h_o = \frac{q}{\theta_s \, (\phi \, \frac{\theta_b}{\theta_s} \, A_f + A_s)}$$

The quantity θ_b/θ_s times ϕ suggests that such a modified "efficiency" term might be useful to simplify the experimental determination of heat transfer coefficients on finned tubes. Such a solution for efficiency would necessitate a three-dimensional analysis for the heat conduction of the fin and tube system together.

For the tests in this investigation, the accuracy of the measured data depends on the accuracy of determining the heat transfer through the test section, and on the accuracy of the temperature measurements of the tube surface. That the power loss from the main heater coil was indeed the radial heat flux through the test section of the tube was assumed primarily because this heater coil was aligned with the test section. The temperature difference measured by the differential thermocouples at the ends of the test section on the tube was held, by controlling the guardheaters, to $1^{\circ}F$ or less during the experiments, but this temperature difference is of the same order of magnitude as the estimated temperature drop radially through the base of the finned tube.

The accuracy of the temperature measurements was believed to be on the order of about 1 per cent. However, the temperatures which were measured were assumed to be the same as temperatures at the fin base, which is not true. The fins were spaced at 8 fins per inch which corresponds to a distance of 0.125 in. between successive fins. Thermocouples were installed approximately midway between fins, or 0.0625 in. from the fin base. For the aluminum tube of high thermal conductivity used in this investigation the difference in temperatures at the station measured and at the fin base was probably quite small. For poorer conducting tubes, of wider fin spacing, this temperature difference might be considerable.

The tube surface thermocouple lead wires, which were wound once or more around the tube and trailed in the air stream, interfered to some unknown extent with the normal flow around the tube.

The correction in Gardner's efficiency term due to neglecting the fin edge area was investigated. The correction was on the order of one to two per cent, but the correction was not made (11). The correction to the fin area of the tube due to the fins being helically wound was not made, since this correction is less than one tenth of one per cent (9).

CHAPTER IX

SUMMARY

A heat transfer coefficient for a finned tube was defined in terms of the radial heat flux through a test section of the tube, the average temperature of the base surface, and the effective area of the tube which depended on the fin efficiency. An electric heater was manufactured and, in use, was aligned inside the finned tube so that guard heating elements would reduce or eliminate the heat fluxes away from the test section of the tube.

A series of thirteen tests was conducted with the heated tube in an induced flow of air, with the tube Reynolds numbers ranging from 1980 to 11,600. Results compared satisfactorily to those of other investigators with the relationship

$$N_{nu} = 0.115\, N_{re}^{.65}\; N_{pr}^{.375} \left(\frac{\mu}{\mu_w}\right)^{.14}$$

Additional studies of the construction and use of a guard-heater should be performed, especially regarding the effects of the lack of uniform contact between the heater and tube surfaces. A refinement of the surface temperature measurements, in which thermocouple lead wires interfere with the flow around the tube, should be considered. A definition of the heat transfer coefficient in terms of a modified notion of the fin efficiency is suggested, so that the measured tube base temperature could be employed correctly in this definition.

Advantages of the test equipment are simplicity and low cost, and ease and speed of use in testing. The results of this investigation, while inconclusive because of the limited number of tests, appear to be good. The principal disadvantage is the requirement for instrumentation for surface temperature measurements on individual tubes.

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