A TEST METHOD FOR DETERMINING OUTSIDE FILM COEFFICIENTS OF FINNED TUBES IN FORCED CONVECTION WITH AIR

By

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LIST OF SYMBOLS

A	area, sq. ft.
a, n, x	constants or exponents
B and S	Brown and Sharpe
BTU	British thermal unit
C_p	Specific heat at constant pressure, BTU per lb. - ^oF
cu. ft.	cubic feet
D	diameter, in.
E	Surface coefficient of fin, BTU per hr. - sq. ft. - ^oF
^oF	degree Farenheidt
ft.	foot
G	mass velocity, lb. per hr. - sq. ft.
h	film coefficient, BTU per hr. - sq. ft. - ^oF
H.P.	horsepower
hr.	hour
I.D.	inside diameter, in.
I, K	modified Bessel functions
in.	inch
k	thermal conductivity, BTU per hr. - sq. ft. - ^oF per ft.
L	length, ft.
lb.	pound
m	$\sqrt{\frac{E}{ky}}$, in^{-1}
min.	minute

mv	millivolts
O.D.	outside diameter, in.
psia	pressure, lb. per square in. absolute
q	heat flow, BTU per hr.
Q_h	air flow, cu. ft. per hr.
R	outside radius of the fin, in.
r_o	inside radius of the fin, in.
rpm	revolutions per min.
sq. ft.	square foot
T	temperature, $^{\circ}F$
u	a function of the fin thickness
V	velocity, ft. per min.
V	volts
x	fin thickness, in.
y	one-half the fin thickness, in.
β	a constant
λ	pitch angle of the fin, 2.015 degrees
η	fin effectiveness
π	ratio of circumference to diameter, 3.1416
ϕ	fin efficiency
μ	absolute viscosity, lb. per ft. - hr.

SUBSCRIPTS

1	average outside fluid
2	tube surface
b	fin base

e	fin edge
f	fin
i	inside
o	outside

Some symbols of other authors have been changed to comply with the notation used in this thesis.

CHAPTER I

INTRODUCTION

In the last twenty years man has made more advancements in the development of material goods than in any other like period of time. And there are no indications that man's rate of advancement will decline in the next twenty years.

All of this advancement has drawn continually upon the presumption that when a design is completed there will be power available to produce it, to operate it, or to propel it.

The production of power, be it steam-driven electric alternators, reciprocating engines, or gas turbines, has had to grow at a phenomenal rate in order to remain ahead of demands. In each of these types of power production, heat transfer is present. In many fields it is necessary that the heat transfer apparatus be relatively small. In aircraft both size and weight limit the heat exchanger design. Automotive gas turbines are dependent upon some form of a regenerator for economical operation and the available space is generally small. Particularly is this problem amplified when the heat transfer is from a gas to a gas since the film coefficient for a gas is comparatively small. Therefore, the heat exchangers must usually be more efficient than the normal straight-tube type of unit.

As a result, designers have been required to use extended surfaces in order to obtain the necessary heat transfer area in a more compact unit. But design data for extended surfaces is very limited and test procedures are even fewer in number. It is the purpose of this thesis to outline a test procedure and a test apparatus with possible variations and to present the results of one such test.

CHAPTER II

DISCUSSION OF PREVIOUS INVESTIGATORS

Three general classes of extended surfaces have been investigated to some degree; straight fins, spines, and annular fins. This discussion will concern only annular finned surfaces.

In 1922, D. R. Harper and W. B. Brown[1] submitted a report to the National Advisory Committee for Aeronautics concerning mathematical equations for the conduction of heat in the fins of air-cooled engines. The mathematical expressions were quite involved, including Fourier's Series, Bessel functions, etc. The results of the work were collected in graphical form in a series of charts which were corrections to a simple formula first developed.

E. W. Still[2] wrote in 1936 that much of the information on film heat flow for turbulent conditions of gases across pipes could be expressed by one equation:

$$hD/k = a(DG/\mu)^{n}(C_{p}\mu/k)^{x}(L/D)^{n1} \quad . \ . \ . \ . (1)$$

[1] D. R. Harper and W. B. Brown, "Mathematical Equations for Heat Conduction in the Fins of Air-Cooled Engines," National Advisory Committee for Aeronautics, Technical Report No. 158, (1922), pp. 679-708.

[2] E. W. Still, "Some Factors Affecting the Design of Heat Transfer Apparatus," Institution of Mechanical Engineers, CXXXIV (1936), pp. 363-411.

However, he qualified this by saying that for finned tubes the equation "would have to be modified so that (a) the air flow is considered at its maximum velocity; (b) for the effective rate, use may be made of the projected area where cylinders are set at an angle to the air stream: . . ."[3] He assigned n the value of 0.6 and used the maximum air velocity.

Another investigator, William M. Murray,[4] derived a mathematical equation for the heat flow through the fins on a tube which considers the size of the fin. The expression,

$$q = 4\pi T_{\supset} r_o \sqrt{(Eky)} \left[\frac{I_1(mR)K_1(mr_o) - K_1(mR)I_1(mr_o)}{I_1(mR)K_o(mr_o) + K_1(mR)I_o(mr_o)}\right] \quad (2)$$

being a rather complicated one involving Bessel functions of the second kind, is somewhat tedious and lengthy to solve. Therefore, he arrived at a quantity called "fin effectiveness"[5] whereby the solution of practical problems is greatly simplified.

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[3] Ibid., p. 369.

[4] William M. Murray, "Heat Dissipation Through an Annular Disk or Fin of Uniform Thickness," Transactions of the American Society of Mechanical Engineers, LX, (1938), pp. A-78--A-80.

[5] "Fin effectiveness" is the ratio of the heat transferred through the base of a fin to that which would be transferred through the same base area were the fin not there as distinguished from "fin efficiency" which is the ratio of the average temperature difference over the extended surface to that over the basic surface.

$$\text{Fin Effectiveness}\ \eta = \sqrt{\left(\frac{k}{E\ y}\right)}\left[\frac{I_1(mR)K_1(mr_o)\ -K_1(mR)I_1(mr_o)}{I_1(mR)K_o(mr_o)\ +K_1(mR)I_o(mr_o)}\right] \quad (3)$$

The bracketed portion of Eq. (3) is represented by $\mathcal{E}$ and

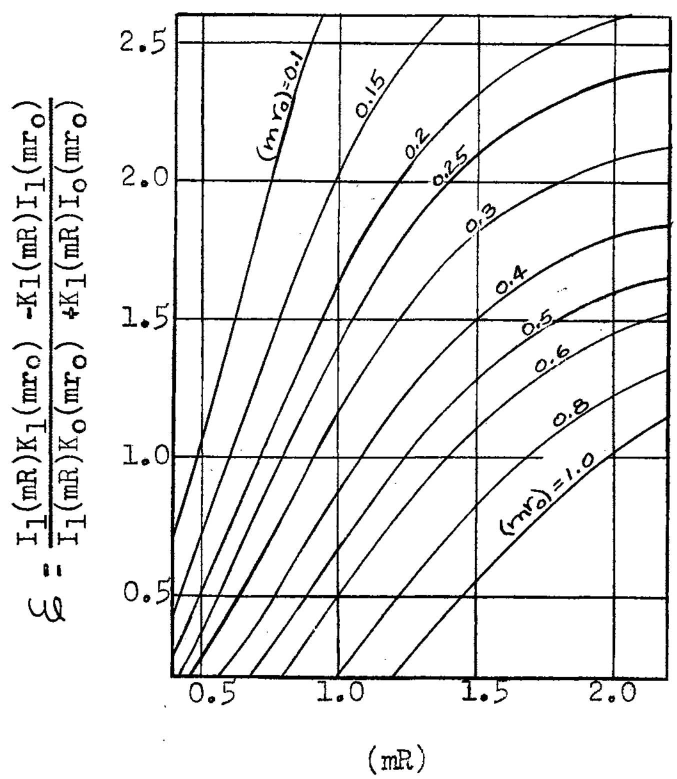

Figure II-1. Fin Effectiveness Functions

presented in graphical form. This chart is reproduced in Figure II-1 and a calculation based on this method will be made later (page 9).

Karl A. Gardner[6] gives an expression for the fin efficiency which also involves the use of Bessel functions;

$$\phi = \frac{2(1-n)}{u_b\left[1-\left(\frac{u_e}{u_b}\right)^{2(n-1)}\right]}\left[\frac{I_{n-1}(u_b)-\beta I_{n-1}(u_b)}{I_n(u_b)+\beta I_{-n}(u_b)}\right] \quad . \; . \; . (4)$$

with $n = 0$ for annular fins of constant thickness. A chart is given to simplify the solution of the equation. For the fin effectiveness Gardner uses the equation

$$\eta = \left(\frac{q_f}{q_b}\right)_{T_b = \text{constant}} = \frac{A_f}{a_b}\,\phi \quad . \; . \; . \; . \; . \; . \; . (5)$$

However, he feels that fin effectiveness is a misleading indication of the value of extended surface because "the addition of extended surface to a metal wall changes the base temperature to an extent depending on the heat-transfer coefficients on both sides of the wall."[7]

A series of tests were conducted by Katz, Beatty, and Foust[8] on tubes with integral spiral fins. One group of tests were made on single horizontal tubes with air in

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[6]Karl A. Gardner, "Efficiency of Extended Surface," Transactions of the American Society of Mechanical Engineers, LXVIII, (1945), pp. 621-628.

[7]Ibid., p. 623.

[8]D. L. Katz, K. O. Beatty, and A. S. Foust, "Heat Transfer Through Tubes with Integral Spiral Fins," Transactions of the American Society of Mechanical Engineers, LXVIII, (1945), pp. 665-674.

forced convection on the outside and condensing steam on the inside. The overall coefficient of heat transfer was calculated on the basis of the total outside area and the data plotted. An equation was found from the curve drawn through the average of the plotted data,

$$U_o = 0.236\ V_{max}^{0.53} \quad . \ . \ . \ . \ . \ . \ . \ . \quad (6)$$

However, no accurate method of measuring the mass flow of air was used and it is felt that using this equation would yield only a rough estimate of the true coefficient.

CHAPTER III

DISCUSSION OF THE THESIS PROBLEM

The basic equation for the flow of heat by conduction and convection from one fluid to another through a tube wall is given by the well-known equation

$$q = U_o A (T_i - T_o) \quad . \; . \; . \; . \; . \; . \; . \; . \quad (7)$$

where

$$U_o = \frac{1}{\frac{1}{h_i} + \frac{x}{k} + \frac{1}{h_o}} \quad . \; . \; . \; . \; . \; . \; . \quad (8)$$

Since it is a relatively simple matter to calculate the inside film coefficient for various fluids flowing inside a straight tube and the thermal conductivity of tube metals is known accurately, it was the feeling of the author that a more useful outside film coefficient for finned tubes would result from basing it on the temperature of the outside surface of the tube. This would particularly be useful when the inside fluid is a liquid and the outside fluid is a gas because then the thermal resistance of the gas film is by far the greatest of the three and, therefore, the determining factor affecting the flow of heat.

The primary interest in this paper, then, is the determination of the outside film coefficient. For the outside film one may write

$$q = h_o A_o (T_2 - T_1) \quad . \; . \; . \; . \; . \; . \; . \; . \quad (9)$$

If the amount of heat flow, q, is known or can be determined and the difference in temperature between the surface of the tube, T_2, and the flowing fluid, T_1, can be measured accurately, then the only unknown in the equation is the film coefficient, h_o, since the outside area, A_o, can be readily measured or calculated.

Marks' Handbook[1] gives for gas flow normal to a single smooth tube and the Reynolds Number greater than 1000

$$h_m = 0.37 \; C_p G^{0.56}/(D'_o)^{0.44} \quad . \; . \; . \; . \; . \; . \quad (10)$$

where h_m = mean film coefficient, BTU per hr.-sq. ft.-°F.

C_p = specific heat at constant pressure, BTU per lb.-°F.

G = mass velocity, lb. per hr.-sq. ft.

D'_o = outside diameter of tube, inches.

Using this equation for a mass velocity of 1350 lb. per hr.-sq. ft. (approximately the lowest flow tested) a value of h_m = 6.38 BTU per hr.-sq. ft.-°F is obtained. The Handbook states that this is an approximation.

To predict how much the transfer of heat will be increased by the addition of fins to the tube the author

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[1] Lionel S. Marks, Mechanical Engineers' Handbook, (New York, 1941), p. 398.

chose to use the method described by Murray.[2] The tube tested was a 5/8 in. O.D. cupro-nickel tube having nine spirally wound fins per inch approximately 0.013 in. in thickness with an outside diameter of 1 3/8 in. Then

$E = 6.38$ BTU per hr.-sq. ft.-^{o}F

$k = 15$ BTU per hr.-sq. ft.-^{o}F per ft.[3]

$r_o = 0.3125$ in. ($\frac{1}{2}$ tube diameter)

$R = 0.6875$ in. ($\frac{1}{2}$ fin outside diameter)

$y = 0.0065$ in. ($\frac{1}{2}$ thickness of fin)

$$\frac{E}{k} = \frac{6.38}{15 \times 12 \text{ in.}} = 0.0355 \text{ in.}^{-1}$$

$$m = \sqrt{\frac{E}{ky}} = \sqrt{\frac{0.0355}{0.0065}} = 2.34 \text{ in.}^{-1}$$

$$mr_o = 0.73 \qquad\qquad mR = 1.605$$

From Figure II-1, p. 5, and using these values for (mr_o) and (mR)

$$\mathcal{E} = 1.02$$

and the fin effectiveness will be

$$\eta = \frac{k}{E} m\mathcal{E} = \frac{1}{0.0355} \times 2.34 \times 1.02 = 67.3$$

That is, the effect of the fin is to increase the heat transfer through that area upon which it stands 67.3 times.

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[2]William M. Murray, p. A-80.
[3]Lionel S. Marks, p. 392.

There are nine fins per inch or 108 fins per foot of tube. The fins cover 0.013 inches per fin or 1.404 inches per foot of tube. Therefore, the total heat transfer per foot of length will be increased

$$\left[\frac{1.404}{12} \times 67.3\right] + \frac{10.596}{12} = 8.75 \text{ times}$$

and the film coefficient would then become

$$8.75 \times 6.38 = 55.8 \text{ BTU per hr.-sq. ft. - } {}^{o}F$$

Murray points out here that there are two possible reasons for this value of the coefficient not being realized in practice. First, the addition of fins might seriously change the temperature distribution around the tube which is quite likely if the temperature difference is very great. Second, the fin spacing may affect the fluid flow around the tube and thereby lowering the film coefficient. "For tubes in slowly moving air, investigations carried on in Germany lead one to believe that the addition of the fins has very little influence on the temperature distribution but that the fin spacing is of great importance."[4] Exactly what is meant by "slowly moving air" is not explained but it is likely that as the air velocity is increased the effect of fin spacing would

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[4] William M. Murray, p. A-80.

become less because the thickness of the boundary layer would decrease allowing less interference of the air flowing over adjacent fins. The comparison between theory and experimental data is in fair agreement until the distance between the fins is about one third the diameter of the tube where the difference between measured and computed heat transfer is about 12 per cent. The finned tube tested had a fin spacing of slightly less than one sixth the diameter of the tube. Therefore, it was expected that the measured film coefficient would depart from the calculated value by a considerable amount.

CHAPTER IV

DESCRIPTION OF TEST APPARATUS AND EQUIPMENT

The various parts of the test apparatus are shown diagramatically in Figures IV-1, IV-2 and IV-3.

The air duct consists of two principal sections - one of rectangular cross section measuring 5 in. x 14 in. which includes the test section and the other a circular pipe 6 in. nominal diameter which includes the air metering orifice.

Air is forced through the duct by a centrifugal fan. The rating of the fan is unknown, but the desired rate of flow is obtained by changing the pulley arrangement which increased the speed of the fan from 750 rpm to 1100 rpm. The fan is powered by a Lincoln Line-Weld induction motor rated at 2 H.P. at 3460 rpm.

Air enters the duct through a converging transition section. Following the entrance, the air passes through a straightening section formed by filling the section with 1 in. tubes. Small sheet metal trim tabs are attached to the ends of some of the tubes to allow for adjustment of the flow to obtain an essentially uniform velocity distribution across the test section.

Next, the air passes into a turbulence section. Eight inches from the end of this section are located seven holes for checking the velocity distribution. These holes are spaced 2 in. on centers and 1 in. from each edge in a straight line across the duct. The velocity check was made with an Anemotherm probe, Serial Number 1987.

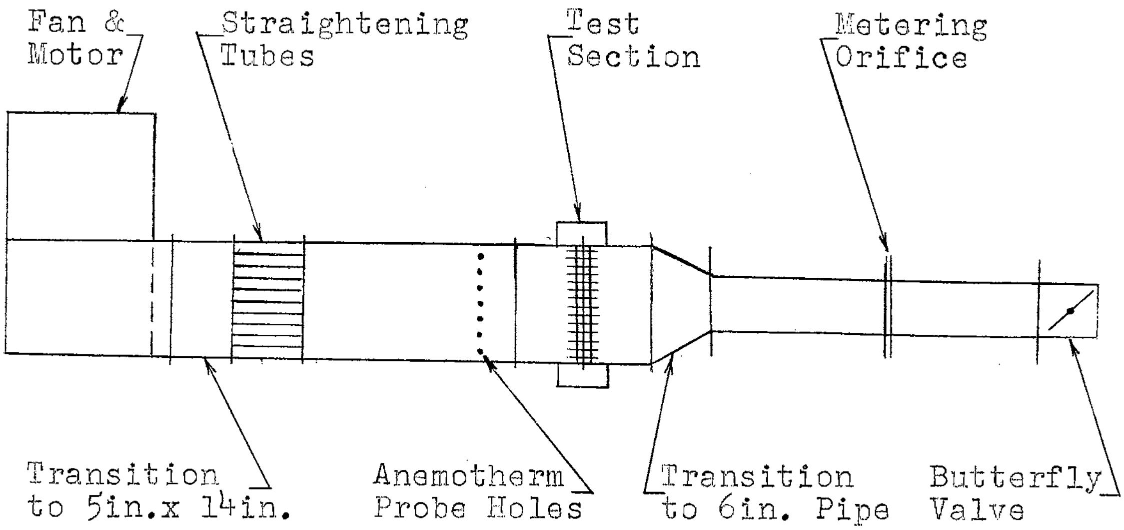

Figure IV-1. Plan of Test Duct

After the test section the air passes through a transition piece from a rectangular cross section to a 6 in. diameter pipe. In this pipe is located the metering orifice with flange taps. The orifice was made and installed in accordance with the specifications of The Orifice Meter.[1] Inclined water manometers are used to measure the static and differential pressures at the orifice. Following the metering section is a butterfly valve used to vary the flow rate.

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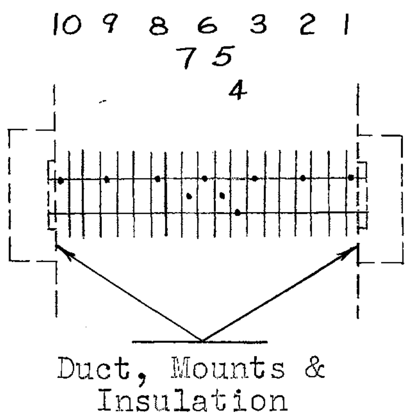
[1]The Orifice Meter, (Pittsburg, Pennsylvania, 1946).

The tube tested was a 5/8 in. O.D., 1/2 in. I.D., cupro-nickel tube with 9 helically-wound fins per in. The fins are 3/8 in. high and approximately 0.013 in. thick. The tube was manufactured by and supplied through the courtesy of the Condenser Service and Engineering Company.



5 on back, 7 on front.

Longitudinal spacing:

1-2, 2-3, 3-6, 6-8, 8-9, 9-10: 2-5/16 in.

4-5, 5-6, 6-7: 3/8 in.

Figure IV-2. Thermocouple Locations

The tube is mounted straight across the width of the test section. Referring to Figure IV-2, 10 thermocouples are placed along the surface of the tube. Seven of these are located on the top of the tube. The other 3 are placed 1 each on the front, back, and bottom of the tube. It is intended for this to provide a means of obtaining an accurate average tube-surface temperature. The exposed ends of the tube are covered with 5 in. squares of 2 in. thick glass wool insulation.

The thermocouples are single junction type made from Leeds and Northrup, No. 30 B. and S. gage, iron-constantan wire. The junctions are made by an electrowelding process under oil which forms a clean, unoxidized bead approximately 1/32 in. in diameter. The thermocouples are attached to the tube by making a slight saw-cut on the tube surface and placing the junction bead down in the cut. The junction is then covered with an iron compound cement. The wire is wrapped around the tube about a turn in order to prevent a temperature gradient away from the junction.

The cold junction temperature is maintained at 32°F by submerging it in an ice and water filled thermos bottle along with a calibrated standard thermometer.

A schematic diagram of the electric heater circuit and the thermocouple circuit is shown in Figure IV-3. All thermocouple leads are brought into an insulated selector switch box and are connected from there by copper leads to the potentiometer. The potentiometer is a Rubicon, Type B, Serial Number 77183, with a range of 16 millivolts. The smallest division is 5 microvolts and readings may be estimated to $\pm$ 1 microvolt (0.0285°F). The standard cell used to balance the circuit is an Epply Standard Cell, Serial Number 563748, and the galvanometer is a Leeds and Northrup, Serial Number 1108927, with a sensitivity of 0.45 microvolts per millimeter.

The heater in the tube is a Chromalox TI 2045, 635 watt, 120 volt, element 13½ in. long. A Varitrans Model

V-1 transformer is used to hold the line voltage constant at 110 V. A General Electric Portable Induction Test Meter, Type IB-6, Number 9290225, is used to measure the power input to the heater.

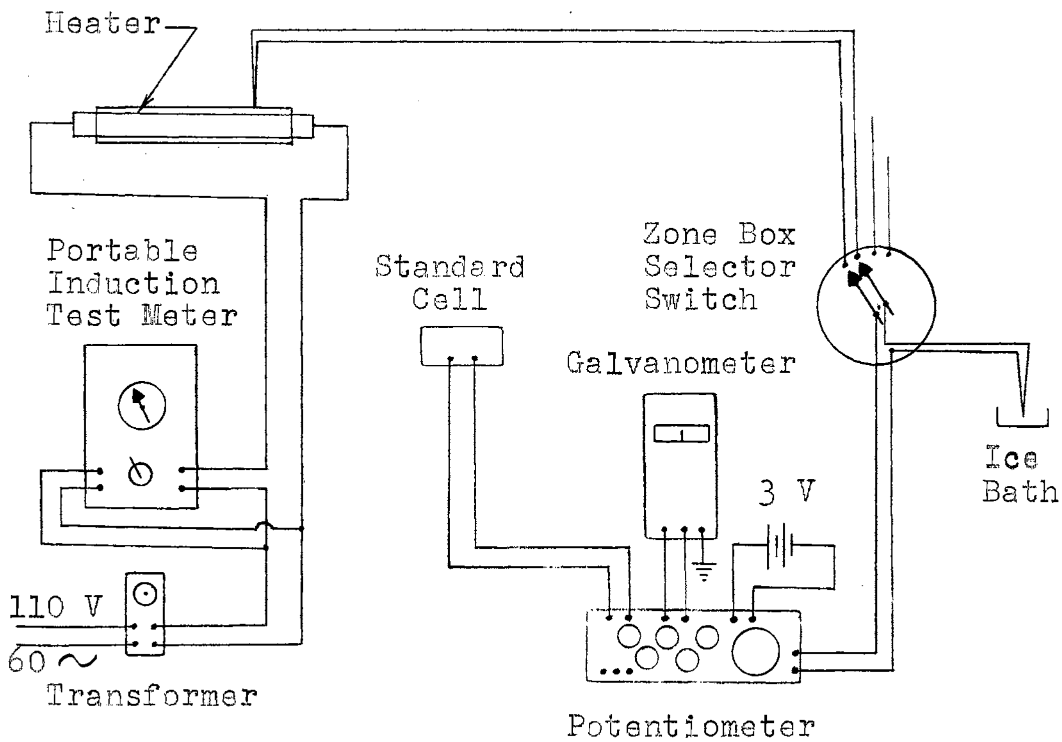


Figure IV-3. Heater and Thermocouple Circuits

CHAPTER V

TEST PROCEDURE

The transformer and portable test meter were placed in the heater circuit and the potentiometer circuit was completed as shown in Figure IV-3. The thermocouple cold junction was checked to insure that the temperature of the ice point was 32°F. The centrifugal fan was then turned on and the butterfly valve adjusted to obtain the desired flow. The accuracy of the watt-hour test meter was assured only if the input voltage did not vary from 110 volts by more than $\pm$ 5 volts. Therefore, the transformer was adjusted to 110 volts in all tests. A check was made on two or three thermocouples until equilibrium conditions were reached. The test was then begun. The duration of each test was arbitrarily set at 20 minutes. Readings of the manometers were taken at 0, 10, and 20 minutes. Since it was impossible to read all thermocouples at the same time, the readings were taken in numbered order as quickly as possible at 0, 10, and 20 minutes. The watt-hour meter was read at the start and at the end of 20 minutes as timed by a stop watch. The range of the test meter and the multiplying constant for that range were recorded. The wet and dry bulb temperatures were taken from thermometers located near the inlet to the fan and at the same elevation.

In all, 1 test was made at each of 7 rates of test section face velocity which were approximately 100 feet per minute intervals from 300 to 900 feet per minute inclusive.

CHAPTER VI

SAMPLE CALCULATIONS AND RESULTS

All calculations are made using data from Test 1, Table 1.

Face Velocity

All orifice coefficients and factors are taken from <u>The Orifice Meter</u>.[1]

$D_1 = 6$ in. = Line size = 6.065 in. actual I.D.

$D_2 = 4$ in. = Orifice size

Meter equipped with flange taps using upstream static pressure connection.

$T_f = 90^{\circ}F$ approximately

$P_f = 0.645 \text{ ft.} \angle\frac{6}{12} + 14.7 = 3.46 \text{ in. water} + 14.7 =$ 14.825 psia.

$P_b = 14.825$ psia

$T_b = 85^{\circ}F$ (Ambient air temperature)

$h_w = 0.14$ ft. at $\angle\frac{3}{12} = 0.407$ in. water

$$\beta = \frac{D_2}{D_1} = 0.66$$

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

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

$F_b = 3711.4$

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[1]<u>The Orifice Meter</u>, pp. 44-80.

For $\sqrt{h_w P_f} = \sqrt{6.03} = 2.46$

$F_r = 1.0217$ (extrapolated)

For $\frac{h_w}{P_f} = \frac{0.407}{14.825} = 0.0275$

$Y_1 = 0.9998$

$F_{pb} = 0.972$

$F_{tb} = 1.0481$

$F_{tf} = 0.9723$

$F_g = 1$

$F_{pv} = 1$

$C' = 3711.4 \times 1.0217 \times 0.9998 \times 0.972 \times 1.0481 \times$
$0.9723 \times 1 \times 1$

$C' = 3755$

$Q_h = 3755 \times 2.46 = 9240$ cu. ft. per hr.

Face Velocity, $V = \frac{9240}{60 \times \frac{70}{144}} = 317.5$ ft. per min.

Heat Transfer Area

The total air-side heat transfer area is the fin area (including the fin edge) plus the area of the tube minus the area of the fin base

$$A_t = \pi(\text{O.D.})L = \frac{\pi(0.625)}{12} \times \frac{14}{12} = 0.191 \text{ sq. ft.}$$

$$A_f = \pi(R^2 - r_o^2)\left(1 + \frac{1}{\cos\lambda}\right)(\text{no. of fins})$$

However, the correction for the fins being helically wound is so small (0.07 per cent) that it can be neglected. The fin area then becomes

$$A_f = \frac{0.7854}{144}\left[(1.375)^2 - (0.625)^2\right](14 \times 9) = 1.031 \text{ sq. ft.}$$

$$A_{fb} = \pi(\text{O.D.})L(\text{no. of fins}) = \frac{\pi \times 0.625}{12} \times \frac{0.013}{12} \times (14 \times 9) = 0.0224 \text{ sq. ft.}$$

$$A_{fe} = \frac{R}{r_o} A_{fb} = \frac{0.6875}{0.3125} \times 0.0224 = 0.0492$$

The total air-side area is

$$A_t = 0.191 + 1.031 + 0.0492 - 0.0224 = 1.2488 \text{ sq. ft.}$$

Average Flowing Air Temperature Across Tube

The maximum temperature rise will be approximately 4-5°F at the lowest flow rate. It is therefore felt that a sufficiently accurate average temperature would be obtained by adding 2°F to the dry bulb temperature.

$$T_1 = 85 + 2 = 87°F$$

Average Tube-Surface Temperature

It is assumed that any variation of the temperature around the circumference will be a linear variation and that the temperatures indicated by thermocouples 4, 5, 6, and 7 are typical of the variation at any point along the tube.

An average of the readings from thermocouples 4, 5, 6, and 7 is found.

$$\frac{7.2754 + 6.7055 + 7.5846 + 7.9737}{4} = 7.3848 \text{ mv. average}$$

This is a $\frac{7.5864 - 7.3848}{7.5864} = 2.63\%$ decrease from the emf indicated by thermocouple 6 located on top of the tube. Since this variation is typical, the readings taken from thermocouples 1, 2, 3, 6, 8, 9 and 10 are reduced by this per cent. An average of these 7 adjusted readings is found

$$\frac{3.5475+5.8147+6.7090+7.3848+6.8885+6.7396+3.7754}{7} =$$

5.8371 mv.

Finding the temperature corresponding to 5.8371 mv. in the Leeds and Northrup Temperature - emf tables for iron-constantan thermocouples yields an average tube-surface temperature of

5.8371 mv. $=$ 230.25°F $= T_2$

Heat Flow

Number of meter revolutions $=$ 48.7

Multiplying constant for range used $=$ 3

Time $=$ 20 min. $=$ 1/3 hr.

$q =$ 48.7 x 3 $=$ 146 watt-hours, per 1/3 hr.

$q =$ 146 x 3.415 x 3 $=$ 1496 BTU per hr.

The heat loss through the insulated ends of the tube and wooden mounting blocks is estimated to be less than 3 BTU/hr. if the average temperature difference between the ambient air and the surface of the wood and exposed tube is chosen as 75°F. At the extreme this is less than 0.25 per cent and can safely be neglected.

Film Coefficient

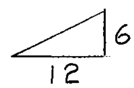
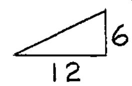
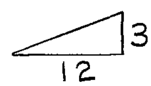
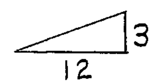
$$h_o = \frac{q}{A_o(T_2 - T_1)}$$

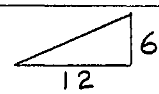
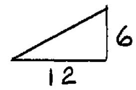
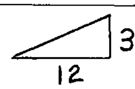
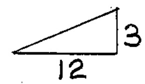
$$h_o = \frac{1496}{1.2488\ (230.25 - 87)} = \frac{1496}{1.2488 \times 143.25}$$

$h_o = 8.47$ BTU per hr. - sq. ft. - ^oF.

The results of all 7 tests are tabulated in Table 2 and graphically in Figure VI-1. The test data is tabulated in Table 1.

Table 1. Test data

			Test 1	Test 2		
Date:			4/22/55	4/22/55		
Length of Test			20 min	20 min		
Dry Bulb Temp.			85^{o}F	85^{o}F		
Wet Bulb Temp.			62^{o}F	72^{o}F		
Angle of Static Manometer						
Static Pressure			0.645 ft.	0.645 ft.		
Angle of Differential manometer						
Differential Pres.			0.14 ft.	0.230 ft.		
Meter Range			50	50		
Constant			3	3		
Revolutions			48.7	55.1		
Temperature Measurements						
Thermo-couple	start mv.	10 min mv.	20 min mv.	start mv.	10 min mv.	20 min mv.
1	3.6565	3.6383	3.6272	3.6897	3.7120	3.7041
2	5.9866	5.9726	5.9558	5.9314	5.9869	5.9744
3	6.9110	6.8727	6.8868	6.8503	6.9187	6.9094
4	7.2968	7.2574	7.2730	7.2739	7.3245	7.3256
5	6.7508	6.6865	6.6791	6.5972	6.6270	6.6632
6	7.6320	7.5600	7.5618	7.6063	7.6552	7.7271
7	8.0195	7.9512	7.9505	8.0025	8.0075	8.1221
8	7.1187	7.0568	7.0486	7.0158	7.0114	7.1137
9	6.9668	6.9037	6.8942	6.8630	6.8776	6.9752
10	3.9084	3.8948	3.8291	3.8602	3.8752	3.9002

	Test 3	Test 4
Date	4/23/55	4/23/55
Length of Test	20 min	20 min
Dry Bulb Temp.	83°F	85°F
Wet Bulb Temp.	61°F	64°F
Angle of Static Manometer	 6 12	 6 12
Static Pressure	0.65 ft.	0.63 ft.
Angle of Differential Manometer	 3 12	 3 12
Differential Pres	0.375 ft.	0.525 ft.
Meter Range	50	50
Constant	3	3
Revolutions	48.4	47.1

Temperature Measurements

Thermocouple	start mv.	10 min mv.	20 min mv.	start mv.	10 min mv.	20 min mv.
1	3.1433	3.1252	3.1552	3.0958	3.1174	3.1228
2	4.9018	4.8634	4.9168	4.5387	4.5897	4.5890
3	5.6740	5.6586	5.7046	5.2658	5.3263	5.3351
4	5.9650	5.9570	6.0170	5.5243	5.5827	5.5794
5	5.3822	5.3586	5.4226	4.9457	4.9991	4.9888
6	6.2644	6.2524	6.3177	5.8134	5.8967	5.8839
7	6.6165	6.6227	6.6473	6.1532	6.1800	6.2260
8	5.6913	5.7018	5.7113	5.2614	5.2218	5.2992
9	5.5743	5.5888	5.5902	5.1686	5.1651	5.2307
10	3.2280	3.2081	3.2132	3.1069	3.1029	3.1243

	Test 5	Test 6
Date	4/23/55	4/23/55
Length of Test	20 min	20 min
Dry Bulb Temp.	$85^{o}F$	$87^{o}F$
Wet Bulb Temp.	$64^{o}F$	$65^{o}F$
Angle of Static Manometer	6 12	6 12
Static Pressure	0.61 ft.	0.605 ft.
Angle of Differential Manometer	3 12	3 12
Differential Pres.	0.73 ft.	0.935 ft.
Meter Range	50	50
Constant	3	3
Revolutions	46.9	45.3

Temperature Measurements

Thermocouple	Test 5 start mv.	Test 5 10 min mv.	Test 5 20 min mv.	Test 6 start mv.	Test 6 10 min mv.	Test 6 20 min mv.
1	3.0050	3.0004	3.0024	2.8886	2.9082	2.9127
2	4.3068	4.3079	4.3054	4.0363	4.0567	4.0355
3	4.9947	4.9982	4.9970	4.6678	4.6956	4.6731
4	5.2452	5.2426	5.2362	4.8989	4.9155	4.8888
5	4.6653	4.6630	4.6608	4.3466	4.3698	4.3523
6	5.5170	5.5157	5.5183	5.1716	5.1830	5.1579
7	5.8504	5.8455	5.8364	5.4686	5.4976	5.4689
8	4.9376	4.9390	4.9359	4.6015	4.6252	4.6128
9	4.8371	4.8406	4.8286	4.5175	4.5484	4.5290
10	2.9672	2.9690	2.9710	2.8656	2.8695	2.8781

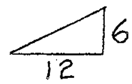
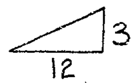
			Test 7
Date			4/23/55
Length of Test			20 min
Dry Bulb Temp.			88^{o}F
Wet Bulb Temp.			66^{o}F
Angle of Static Manometer			
Static Pressure			0.585 ft.
Angle of Differential Manometer			
Differential Pres.			1.220 ft.
Meter Range			50
Constant			3
Revolutions			45.3
Temperature Measurements			
Thermo-couple	start mv.	10 min mv.	20 min mv.
1	2.8300	2.8260	2.8325
2	3.8317	3.8308	3.8292
3	4.4364	4.4441	4.4202
4	4.6397	4.6512	4.5312
5	4.1151	4.1233	4.0917
6	4.8878	4.9051	4.8641
7	5.1806	5.2015	5.1535
8	4.3387	4.3570	4.3196
9	4.2722	4.3038	4.2515
10	2.7747	2.7900	2.7612

Table 2. Test Results

		Test 1	Test 2	Test 3	Test 4	Test 5	Test 6	Test 7
V,	Face Velocity, feet/min.	317.5	403	513	604	710	835	922
A_o,	Total Air side area, sq. ft.	1.2488	1.2488	1.2488	1.2488	1.2488	1.2488	1.2488
T_1,	Average Air Temp. °F	87	87	85	87	87	89	90
	Average indicated emf, mv.	5.8371	5.6792	4.7689	4.6243	4.2162	3.5989	3.6332
T_2,	Average tube-surface temp. °F	230.25	225	195	190.3	176.3	167.7	156.4
q,	Heat transferred BTU/hr.	1496	1693	1487	1448	1442	1390	1390
	h_o, outside film coefficient, BTU/hr.-sq.ft.-°F	8.47	9.84	10.83	11.23	12.94	14.14	16.81

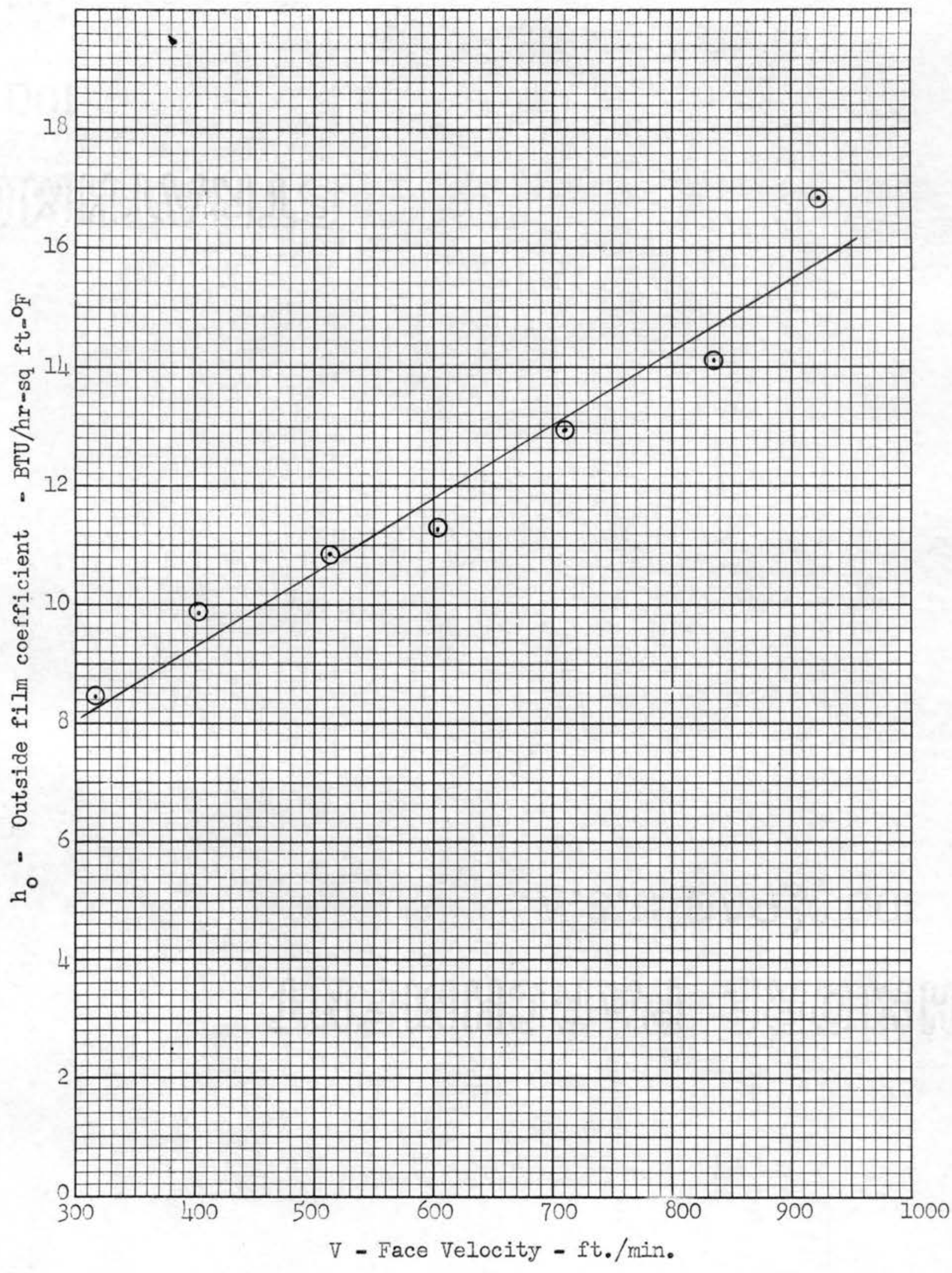


Figure VI-1 Graphical Plot of Test Results

CHAPTER VII

INTERPRETATION OF RESULTS

The results of this series of tests were largely as expected. There were no major deviations from the general pattern of plotted results.

The value of the film coefficient rose steadily with increasing face velocities. It should be observed, however, that the measured film coefficient differed markedly from that calculated using Murray's method.[1] This was as expected for three reasons: (1) the fin spacing was much less than 1/3 of the outside tube diameter at which point Murray states that the error is about 12 per cent; (2) the measured temperature was the tube-surface temperature between the fins and it is highly probable that the temperature at the base of the fin was lower than the temperature between the fins, possibly quite a bit lower; and (3) the fin arrangement used in Murray's derivations was for annular fins while those on the tube tested were helically wound which probably interfered somewhat with the air flow around the tube. All three of these would tend to reduce the film coefficient with the third probably having the least effect.

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[1] See page 11.

An examination of the test data indicates that the temperature varied considerably along the length of the tube. The two extreme low temperatures at the ends of the tube were low because the heater was only $13\frac{1}{2}$ in. long, therefore, leaving $\frac{1}{4}$ in. unheated tube at each end. The heat flow from the inside of the tube then would have a tendency to flow from the center of the tube toward the ends which would result in the higher temperatures being at the middle of the tube.

Again referring to the data, it can be seen that the temperatures varied quite a bit during the tests. Some varied around an average while some either constantly increased or decreased. It is believed that this was caused by the varying velocity distribution across the duct. The velocity distribution was checked with an Anemotherm previous to the tests. Although the velocity at each point was varying constantly, the distribution across the duct was fairly even, the maximum variance being about 30 fpm at the lowest flow.

It is felt that the method used for obtaining the average tube-surface temperature was as accurate as possible under the circumstances. The resistance to heat flow of the tube wall is a very small part of the total resistance. Therefore, even if the thermocouples were not located at the exact surface of the tube the error in the indicated temperature would be so small as to be negligible. The error was estimated to be ±5 per cent due to the probability that the temperature variation around the tube and along the tube was not linear as was assumed in the calculations. The accuracy of the potentiometer was guaranteed ±0.015 per cent for the range used which was negligible in the range of error being considered here.

Any error involved in the measurement of the energy input to the tube was due to heat flow through the insulated ends of the tube and any inaccuracy of the test meter. The energy loss through the insulation was estimated at less than 0.2 per cent which was considered a very liberal allowance. No calibration data was available for test meter, but since it was the standard test meter for the City of Stillwater the maximum error was assumed to be ±0.5 per cent. The total possible error in the measurement of heat flow was then ±0.7 per cent.

The accuracy of the flow measurement was dependent upon the temperature of the flowing air which was estimated. However, an error as large as 3°F in this estimate would produce an error in the flow measurement of only 0.27 per cent. The orifice plate was made and installed in accordance with established rules. To be on the safe side, the total error in the flow measurement was chosen to be ±1.0 per cent.

The estimated accuracy of the tests would be;

h_o, outside film coefficient	
Temperature measurement	= ±5.0
Heat flow measurement	= ±0.7
Total	= ±5.7 per cent
V, average face velocity	= ±1.0 per cent

This agreed very closely with the curve drawn through the plotted data in Figure VII-1.

CHAPTER VIII

SUMMARY AND CONCLUSIONS

The desirability of accurate design data on the performance of finned tubes has become increasingly important. The dependability of this heat transfer data is dependent upon the test method used. The cost of the data is dependent upon the simplicity of the test apparatus and test procedure.

These factors were kept in mind during the establishment of the test method outlined in this thesis. The method is adaptable to air flows above and below the range used in these tests. The apparatus is easily fabricated and the accessory equipment is usually available to institutional and industrial laboratories. The test procedure was very simple and the calculations were as direct as it is possible to make them.

However, several modifications should be suggested. It would be desirable to have an induced draft air system to reduce the difficulty in obtaining a uniform velocity distribution and a more constant velocity at each point of the test section. This would require a fan of higher rating than the one used. A centrifugal blower would be best since they are designed to operate against a higher static pressure.

The method of heating the tube is limited as the electrical load could become excessive if the tests were conducted on a bank of tubes. Condensing steam, if available, provides an adequate source of heat although its use would complicate the test procedure and calculations. Instrumentation would be more elaborate and costly.

Again, if a bank of tubes was to be tested, the temperature rise of the air would become significant and a means of mixing the air after the test section and measuring its temperature would be necessary. This would also offer a means of obtaining an energy balance which was not available to the test method described.

The test method outlined in this thesis was simple, direct, and inexpensive. The results obtained are believed to have been of sufficient accuracy to make this method practical and reliable.

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