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Heat transfer to fuel oil in viscous flow-characteristics of a fin-type heat exchanger.

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HEAT TRANSFER TO FUEL OIL IN VISCOUS FLOW-
CHARACTERISTICS OF A FIN-TYPE HEAT EXCHANGER

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ABSTRACT

A unit was constructed for the study of film heat transfer coefficients of a heavy fuel oil in viscous or streamline flow. The heat exchanger was of the shell and tube type, having extended longitudinal steel fins welded to the tube. Auxiliary equipment consisted of an oil pump, oil storage reservoirs, piping, and means for measurement of temperatures and oil flow rates.

The unit was operated both as a fin type heat exchanger with the oil in contact with the finned surfaces, and as a straight tube and shell exchanger with the oil on the tube side. Oil flow rates and temperatures were varied over a wide range.

The data obtained was calculated by the means of existing empirical equations, and the results were compared with those of previous investigators and with the proposed correlations by plotting.

The experimentally determined film coefficients obtained on the exchanger when operated as a fin type were found to be in agreement with the best existing correlation. Coefficients determined on the straight tube and shell exchanger were found to be forty per cent higher than the values predicted by this correlation.

A comparison was drawn between the exchanger as a fin type and as a shell and tube type. The fin type was found to transfer approximately four hundred per cent as much heat per unit length of exchanger as did the straight shell and tube type. It is believed that a considerable saving would be effected by the use of a fin type exchanger in any application of heat transfer in which one fluid film definitely controlled the rate of heat transfer.

INTRODUCTION

This investigation was undertaken for the following purposes:

1. To determine film coefficients of heat transfer for a heavy fuel oil in viscous flow.
2. To determine film coefficients of the same oil when flowing through a fin-type heat exchanger of tube and shell construction.
3. To compare with existing correlations the data obtained in this investigation.
4. To draw a comparison between heat exchangers of tube and shell construction with and without finned surfaces.

The available data on heat transfer to fluids in viscous flow has been correlated by Colburn (1,2), Sieder and Tate (3), and McAdams (4); however, no data was available which had been obtained under conditions which gave a high ratio of the fluid viscosity at the main stream bulk temperature to the fluid viscosity at the tube wall temperature. Also, in recent investigations, Tepe (5), the data obtained were found to lie somewhat above the correlations of Sieder and Tate (3), and McAdams (4).

In this investigation it was attempted to obtain high values of the ratio of the main stream viscosity to the tube wall viscosity, and of the Graetz (8) number, Wc/kL. The coefficients of heat transfer which were determined using a tube and shell heat exchanger with and without finned surfaces[1] are compared with the theoretical values predicted by the correlations of McAdams, and of Sieder and Tate.

A comparison is drawn between a straight tube and shell exchanger and a tube and shell exchanger of the fin type, based on the rate of heat transfer per unit length of exchanger.

[1]The same exchanger was used in both cases; however, the extended fin area becomes effective only if used in contact with a fluid having relatively low film heat transfer coefficients.

HISTORICAL

A general correlation based on analysis of data on heat transfer to fluids inside round pipes is the Dittus-Boelter (6) equation:

$$\frac{hD}{k} = 0.023 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{n}$$

where n = 0.4 when the fluid is being heated, and 0.3 when the fluid is being cooled.

It has been found that, while this equation satisfactorily correlates data for high values of the Reynolds number, it fails to correlate data on the heating and cooling of hydrocarbon oils below values of the Reynolds number of 7000. Between values of 7000 and the critical value of 2100 the equation of Morris and Whitman (7) applies:

$$\frac{hD/k}{(c\mu/k)^{0.4}} = \psi \frac{DG}{\mu}$$

where ψ is a function obtained from a plot of $\frac{hD/k}{(c\mu/k)^{0.4}}$ vs. $\frac{DG}{\mu}$

Colburn and Hougen (2) presented a fundamental equation for fluids in general flowing vertically at low velocities:

$$h = 0.128k_f \left(\frac{c\mu_f}{k_f}\right)^{1/3} \left(\frac{k_f^2 \rho_f^2 \beta_f \Delta t g c}{\mu_f}\right)$$

The concept of "thermal turbulent" flow was introduced as the only effective motion of the fluid under conditions of viscous flow. Under such conditions the mean velocity of the fluid is held to be unimportant; the controlling motion being that set up by natural convection due to density differences caused by the temperature gradient. Since "thermal turbulence" is in reality natural convection, it is controlled by the same variables, and the Grashof number becomes effective. The significance of thermal turbulent flow is questionable when the main stream flow is horizontal, and the movement of the liquid due to convection is at right angles to the forced flow.

McAdams (4) demonstrated the effect of a viscosity gradient set up by the temperature gradient through a cross section of a fluid flowing in streamline motion[1]. He concluded that the effect of this viscosity gradient could not be ignored in any correlation for viscous flow, except under limited conditions of small temperature changes, etc.

Graetz (8) integrated the Fourier-Poisson equation (9) for radial conduction in a moving liquid, using simplifying assumptions[2] and obtained the relationship:

$$\frac{t_2 - t_1}{t_s - t_1} = 1 - 8\phi(n_1)$$

where $\phi(n_1)$ represents a convergent infinite series involving the relationship: $\frac{\pi k L}{4Wc}$

[1]Pages (16 & 17)

[2]Pages (17 - 20)

Introducing the definition of the individual average coefficient of heat transfer, the heat balance, and the arithmetic mean temperature difference, the equation of Graetz becomes[1]:

$$\frac{h_aD}{k} = \frac{2}{\pi}\left(\frac{Wc}{kL}\right)\left(\frac{1 - 8\phi(n_1)}{1 + 8\phi(n_1)}\right)$$

which represents the theoretical relation based on the parabolic distribution of mass velocity.

Drew and McAdams (10) proposed the empirical equation:

$$\frac{h_aD}{k} = 1.75\left(\frac{Wc}{kL}\right)^{1/3} = 1.62\left(\frac{4Wc}{\pi kL}\right)^{1/3}$$

which agrees with the theoretical equation of Graetz for values of $\frac{Wc}{kL}$ greater than 10.[2]

The experimental data on the heating and cooling of oils and glycerine, flowing in either horizontal or vertical pipes, run considerably above the Drew-McAdams equation when the fluid is being heated, and considerably below when being cooled. These discrepencies were attributed to the lack of a term to allow for the effect of the radial variation in viscosity.

In developing his method of correlating forced convection heat transfer data and fluid friction Colburn (1) concluded that there is no apparent correlation between heat transfer and fluid friction in the viscous region. However, he proposed a general

[1] See Pages

[2] See Fig. 2 Page

method of correlating heat transfer data which could be used for the entire range of turbulent and viscous flow, based on data obtained using water, air, and petroleum oil: (2)

$$\frac{h_a D}{k} = 1.62 \left(\left(\frac{\mu}{\mu_f}\right)^{1/3} (1 + 0.015Z^{1/3}) \right) \left(\frac{4Wc}{\pi kL}\right)^{1/3} \ *$$

where Z is the Grashof number: $Z = \frac{D^3 \rho_f^2 g \beta \Delta t}{\mu_f^2}$

Sieder and Tate (3) sought a correlation which would be as accurate as that of Colburn, but simpler to use. By employing fluid properties at the main stream temperatures, in contrast to the film temperature properties used by Colburn, they derived the simplified equation:

$$\left(\frac{h_a D}{k}\right)\left(\frac{\mu_a}{\mu_w}\right)^{0.14} = 1.86 \left(\frac{4Wc}{\pi kL}\right)^{1/3} = 1.86 \left(\left(\frac{DG}{\mu}\right)\left(\frac{c\mu}{k}\right)\left(\frac{D}{L}\right) \right)^{1/3},$$

noting that for viscous liquids in tubes of ordinary size the Grashof number is small, and: (4)

$$1.62 \left(\frac{\mu_a}{\mu_f}\right)^{1/3} (1 + 0.015Z^{1/3}) \text{ reduces to } 1.86 \left(\frac{\mu_a}{\mu_w}\right)^{0.14}$$

The data correlated by the equation of Sieder and Tate contains few values of the ratio μ_a/μ_w above 10. It is the purpose of this investigation to obtain higher viscosity ratios

*McAdams (4) states that the constant 1.62 was incorrectly given as 1.5 in reference (1).

than those obtained heretofore, and to compare this data with the foregoing correlations.

When the thermal resistance on the inside of a metal tube is much lower than that on the outside, as when air is being heated by steam condensing in a pipe, external finned surfaces are of great value in materially increasing the rate of heat transfer per unit length of tube. Considerable data has been published for air and gases flowing outside and normal to banks of finned tubes (4).

No data is available, however, on coefficients of heat transfer obtained when longitudinal fins are added to the outer surface of the tube in the conventional shell and tube heat exchanger.

It is the added purpose of this investigation to determine such coefficients and to draw a comparison between the two types of exchangers.

THEORETICAL

The basic form of the conduction equation, under steady state conditions, is written as:

$$\frac{q}{\theta} = \frac{kA\Delta t}{L} = Q \qquad \text{Eq. (1)}$$

where q = total heat transferred in Btu

Q = total heat transferred per unit time, Btu/hr.

θ = time in hours

A = heat transfer area in ft.^{2}

L = thickness of heat transfer wall in ft.

Δt = temperature difference across heat transfer wall, ^{o}F.

k = thermal conductivity of the material of which the wall is made, Btu/hr. x ft.^{2} x ^{o}F./ft.

The thermal conductivity k is variable with temperature for any given substance, and this variation is generally linear, corresponding to an equation of the type:

$$k = a + bt$$

where a and b are constants and t is the temperature.

Consider a quantity of heat Q, passing through a wall of area A and composed of several thicknesses of different materials. Let the thicknesses of the layers be denoted by L_1, L_2, and L_3, and their thermal conductivities by k_1, k_2, and k_3 respectively. Let the temperature drop across the whole wall thickness be denoted by Δt, and across each individual thickness by Δt_1, Δt_2, and

Δt_3, respectively. It is then apparent that

$$\Delta t = \Delta t_1 + \Delta t_2 + \Delta t_3$$

Equation (1) can then be written for each of the layers as follows:

$$\Delta t_1 = Q_1 \times L_1/k_1A \qquad \text{Eq. (2)}$$

$$\Delta t_2 = Q_2 \times L_2/k_2A \qquad \text{Eq. (3)}$$

$$\Delta t_3 = Q_3 \times L_3/k_3A \qquad \text{Eq. (4)}$$

Adding (2), (3), and (4):

$$\Delta t_1 + \Delta t_2 + \Delta t_3 = Q_1L_1/k_1A + Q_2L_2/k_2A + Q_3L_3/k_3A$$

$$= \Delta t \qquad \text{Eq. (5)}$$

Since all the heat which passes through the first layer must pass through the second and third layers also,

$$Q_1 = Q_2 = Q_3 = Q, \text{ and:}$$

$$Q = \frac{\Delta t}{\frac{L_1}{k_1A} + \frac{L_2}{k_2A} + \frac{L_3}{k_3A}} \qquad \text{Eq. (6)}$$

Denoting L_1/k_1A, L_2/k_2A, and L_3/k_3A as resistances R_1, R_2 and R_3 respectively, Eq. (6) becomes:

$$Q = \frac{\Delta t}{R_1 + R_2 + R_3} \qquad \text{Eq. (7)}$$

In the above derivation the area A perpendicular to the direction of the flow of heat remained constant, being a flat surface; however, it is obvious that in the case of heat flow through a curved surface, such as through the wall and lagging of an insulated steam pipe the area perpendicular to the direction of heat flow becomes increasingly larger as the diameter increases. In such a case Eq. (6) becomes:

$$Q = \frac{\Delta t}{\frac{L_1}{k_1A_1} + \frac{L_2}{k_2A_2} + \frac{L_3}{k_3A_3}} \qquad \text{Eq. (8)}$$

where A_1, A_2 and A_3 represent the areas of the various thicknesses respectively.

In any case of heat transfer to or from a fluid through a wall there is a thermal resistance to heat flow, and therefore a temperature drop, across a thin film of the fluid adhering to the wall. This resistance may be denoted by $R_1 = L_1/k_1A_1$; however, due to physical difficulties in the measurement of the thickness L_1 and the conductivity k_1, these variables are combined into the film coefficient:

$$h_1 = k_1/L_1$$

where h_1 has the units Btu/hr. x ft.^{2} x ^{o}F.

Considering a tube and shell heat exchanger, with fluid flowing through the tube and steam condensing outside the tube, Eq. (8) would become:

$$Q = \frac{\Delta t}{\frac{1}{h_1A_1} + \frac{L}{kA_{av}} + \frac{1}{h_sA_s}} \qquad \text{Eq. (9)}$$

where h_1 = film coefficient of fluid in tube, Btu/hr.x ft.^{2} x ^oF.

A_1 = inside area of tube, ft.^{2}

L = thickness of tube wall, ft.

k = thermal conductivity of tube wall, Btu/hr.xft.^{2}x^oF/ft.

A_{av} = mean wall area

= mean of inner and outer wall areas[1]

h_s = steam film coefficient, Btu/hr.xft.^{2}x^oF.

A_s = outside area of tube, ft.^{2}

Since the values of the film coefficients cannot be conveniently determined directly from experimental data, it is customary to define an overall heat transfer coefficient U, on the basis of a definite area. For example, if A_1 is chosen, Eq. (9) becomes (multiplying numerator and denominator of the right hand side by A_1):

$$Q = \frac{A_1 \Delta t}{\frac{1}{h_1} + \frac{A_1 L}{A_{av} k} + \frac{A_1}{A_s h_s}} \qquad \text{Eq. (10)}$$

Defining U_1 as:

$$U_1 = \frac{1}{\frac{1}{h_1} + \frac{A_1 L}{A_{av} k} + \frac{A_1}{A_s h_s}} \qquad \text{Eq. (11)}$$

it can be seen that

$$Q = U_1 A_1 \Delta t \qquad \text{Eq. (12)}$$

which is the general mathematical expression for the flow of heat from one medium to another.

[1]When the value of A_s/A_1 does not exceed 2, the arithmetic mean $A_{av} = (A_1 + A_s)/2$ may be used. For values of $A_s/A_1 > 2$ use the logarithmic mean:

$$A_m = \frac{A_s - A_1}{2.3 \log \frac{A_s}{A_1}} \qquad \text{McAdams (4)}$$

It is clear that other overall coefficients U_{av}, U_s, etc., could be obtained on the basis of other areas.

In the case of thin walled tubes of large diameter, where the inner area, outer area, and mean wall area are all very nearly equal, it is permissible to use a common value for A as this will introduce a negligible error into the result. In such a case the resistance equation becomes:

$$U_1 = \frac{1}{\frac{1}{h_1} + \frac{L}{k} + \frac{1}{h_s}}$$

In any case of heat transfer to a fluid there are several variables which must be included in an equation which would predict the values of the film coefficient. These variables are fluid velocity, its viscosity, thermal conductivity, specific heat, density, pipe diameter, and others in some cases. The only satisfactory means yet found of arranging these variables into useful form is that of dimensional analysis. The following dimensionless groups are of particular importance:

Nusselt Number	hD/k
Reynolds Number	DG/μ
Prandtl Number	$c\mu/k$
Grashof Number	$D^3\beta\rho^2\Delta tg/\mu^2$

where h = film coefficient of heat transfer, Btu/hr.x$ft.^2$x °F.

D = pipe diameter, ft.

k = thermal conductivity of fluid, Btu/hr.x$ft.^2$x°F./ft.

G = mass velocity of fluid, lb./$ft.^2$xhr.

μ = fluid viscosity, lb./hr.x ft.

c = specific heat of fluid at constant pressure, Btu/lb.x^{o}F.

ρ = fluid density, lb./ft.^{3}

β = coefficient of thermal expansion, 1/^{o}F.

Δt = temperature difference, ^{o}F.

In the correlation of heat transfer data the above dimensionless groups usually occur in the form: (11)

$$hD/k = K(DG/\mu)^{a} \; (c\mu/k)^{b} \; (D^{3}\beta\rho^{2}\Delta tg/\mu^{2})^{c}$$

where K, a, b, and c are experimentally determined constants. The correlation may then be established by plotting the experimental data in various ways to obtain the proper relationship between the groups.

In the case of a viscous fluid flowing through a long pipe, McAdams (4) demonstrates the effect of a viscosity gradient in the fluid cross section, corresponding to the temperature gradient across the fluid cross section.

When a fluid is flowing at a constant rate through a long pipe under isothermal conditions and in viscous or streamline flow, a parabolic velocity gradient is set up over any cross section, with maximum velocity at the axis and zero velocity at the wall. This condition is shown by curve AA in Fig. 1.

If the fluid now enters a section of pipe jacketed by steam condensing at constant temperature, a temperature gradient is set up, the temperature at the wall being high and that at the axis being low. Since the viscosity of a liquid falls with rise in temperature a viscosity gradient is established, with low viscosity at the wall and high viscosity at the axis. As a result, the layers of liquid near the wall will flow faster than they did in the unheated section of pipe. Since total flow remains the same some of the liquid from the center of the pipe must flow toward the wall to maintain the increased velocity of the layers near the wall. The heating of the liquid therefore develops a radial component of the velocity which distorts the parabola to curve BB in Fig. I.

If the liquid were cooled a radial flow in the opposite direction would be developed, again distorting the parabola to the shape of curve CC, Fig. I.

If density change is appreciable with temperature other disturbances may occur, although, as pointed out before[1], these disturbances would probably be negligible in horizontal flow.

It can be concluded from the above presentation that theoretical equations which ignore the distortion of the parabola cannot be expected to apply except in cases where temperature differences are small or fluid properties vary only slightly with temperature.

Graetz (8) integrated the Fourier-Poisson equation (9) for radial conduction in a moving liquid, using the following conditions and assumptions:

1. Fluid of specific heat c and thermal conductivity k enters at temperature t_1, and is heated or cooled without change in phase.

[1]Page 6

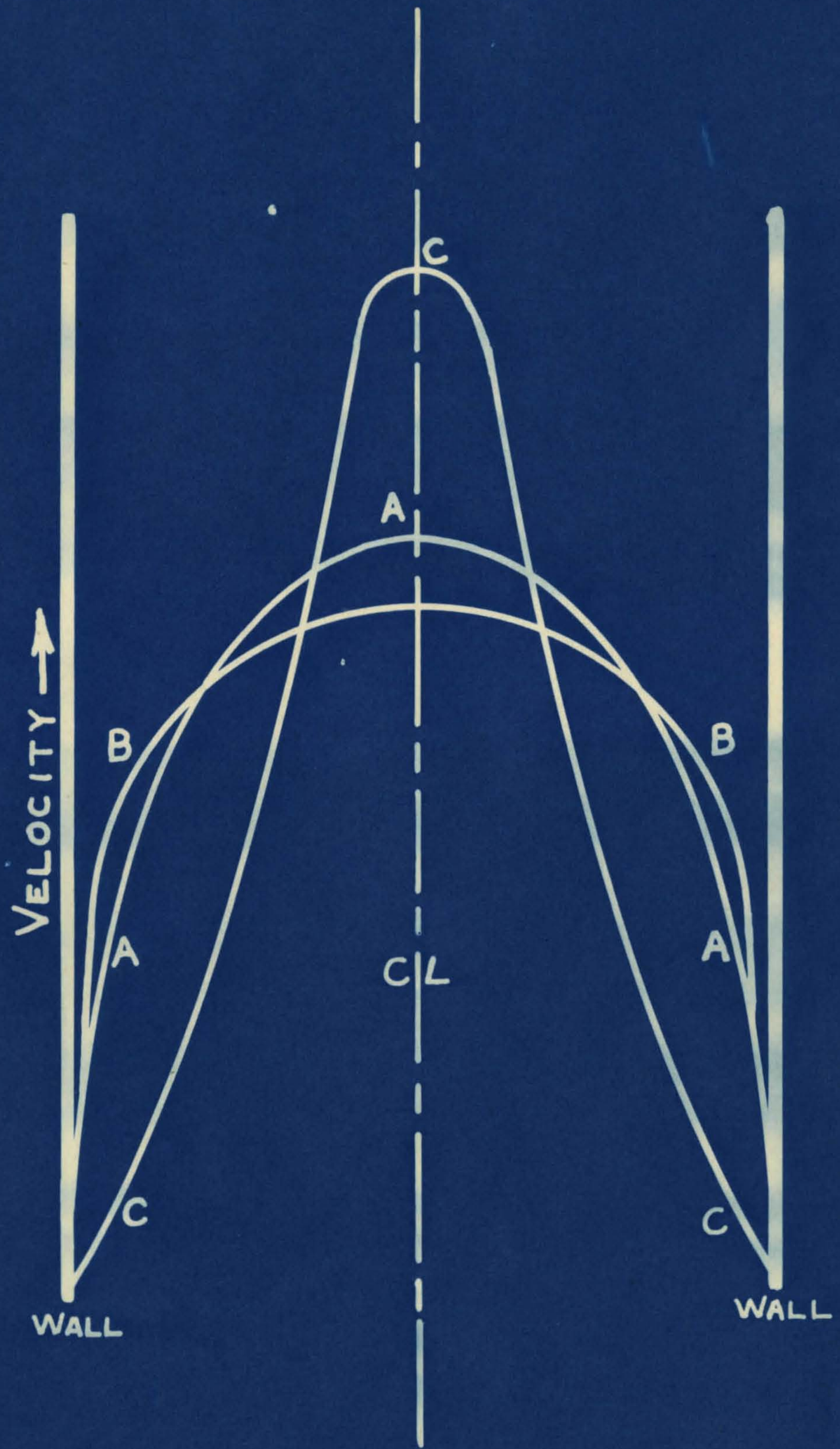


FIG. 1 EFFECT OF HEAT TRANSMISSION ON VELOCITY DISTRIBUTION IN STREAMLINE MOTION

2. Fluid is flowing inside a pipe having a heated or cooled length L, the flow being at constant mass rate in undistorted laminar motion.
3. Since the flow is assumed to be laminar in character, the distribution of local mass velocity over any cross section is parabolic, with zero wall velocity and maximum axis velocity (curve AA in Fig. 1).
4. Heat is assumed to be transferred by radial conduction only, with the thermal conductivity of the fluid remaining uniform. The temperature of the wall surface t_s is assumed to be uniform.

The relation obtained by the integration is:

$$\frac{t_2 - t_1}{t_s - t_1} = 1 - 8\phi(n_1) \qquad \text{Eq. (13)}$$

where $\phi(n_1) = 0.10238e^{-14.6272n_1} + 0.01220e^{-89.22n_1} + 0.00237e^{-212n_1} + \ldots\ldots\ldots\ldots$

and $n_1 = \pi kL/4Wc$

The individual average coefficient of heat transfer can be defined by:

$$h_a = A/A\,\Delta t$$
$$= (Wc)(t_2 - t_1)/(\pi DL)(t_s - t)_m$$

Multiplying through by 1/k and rearranging terms gives:

$$h_aD/k = 1/\pi \quad (Wc/kL) \quad \frac{(t_2 - t_1)}{(t_s - t)_m} \qquad \text{Eq. (14)}$$

The average h may be based upon any type of mean temperature desired. McAdams (4) recommends the use of the arithmetic mean of the terminal values for design purposes:

$$(t_s - t)_a = \frac{(t_s - t_1) + (t_s - t_2)}{2} \qquad \text{Eq. (15)}$$

Equations (13), (14), and (15) may be combined as follows:

$$t_2 - t_1 = (t_s - t_1)(1 - 8\phi(n_1))$$

$$h_aD/k = 1/\pi\,(Wc/kL)\,\frac{(t_s - t_1)(1 - 8\phi(n_1))}{\frac{(t_s - t_1) + (t_s - t_2)}{2}}$$

$$= 2/\pi\,(Wc/kL)\,\frac{(t_s - t_1)(1 - 8\phi(n_1))}{(t_s - t_1) + (t_s - t_2)}$$

$$t_2 - t_1 = (t_s - t_1) - 8\phi(n_1)(t_s - t_1)$$

$$t_2 - \not{t}_1 - t_s + \not{t}_1 = - 8\phi(n_1)(t_s - t_1)$$

$$(t_s - t_2) = 8\phi(n_1)(t_s - t_1)$$

$$h_aD/k = 2/\pi\,(Wc/kL)\,\frac{\cancel{(t_s - t_1)}(1 - 8\phi(n_1))}{\cancel{(t_s - t_1)}(1 + 8\phi(n_1))}$$

$$= 2/\pi\,(Wc/kL)\left(\frac{1 - 8\phi(n_1)}{1 + 8\phi(n_1)}\right) \qquad \text{Eq. (16)}$$

which represents the theoretical relation based on the parabolic distribution of the mass velocity. (curve AB in Fig. 2)

In the special limiting case when the fluid is heated nearly to the constant temperature of the wall, $t_2 = t_s$,

$$(t_s - t)_a = (t_s - t_1)/2 = (t_2 - t_1)/2$$

$$(t_s - t_1) = (t_2 - t_1)\text{ , and}$$

$$1 = 1 - 8\phi(n_1)$$

$$0 = - 8\phi(n_1)$$

Then Eq. (16) reduces to:

$$h_a D/k = 2Wc/\pi kL$$

which is the equation of the asymptote AE in Fig. 2. With constant surface temperature t_s, no reliable value of $h_a K/k$ can lie above this asymptote.

The empirical equation proposed by Drew and McAdams corresponds to the theoretical equation (Eq. 16) for values of Wc/kL above 10 (see Fig. 2):

$$h_a D/k = 1.75(Wc/kL)^{1/3} = 1.62(4Wc/\pi kL)^{1/3}$$

The development of the equation of Sieder and Tate (3) is covered in the previous section.[1]

The theory applied to the fin type heat exchanger is similar to that of the straight tube and shell exchanger with the following exceptions:

1. A hydraulic radius, based upon some method of evaluation, must be used in the determination of the equivalent pipe diameter.
2. The temperature of the fins is not equal to the tube wall temperature.
3. There is a large difference between the area of the steam side (inner tube area) and the fluid side (outer bare tube area plus fin area).

There are several methods of evaluating the equivalent diameter. The equivalent diameter is equal to four times the hydraulic radius, m. The hydraulic radius (11) is defined as the

[1]Pages (8-9)

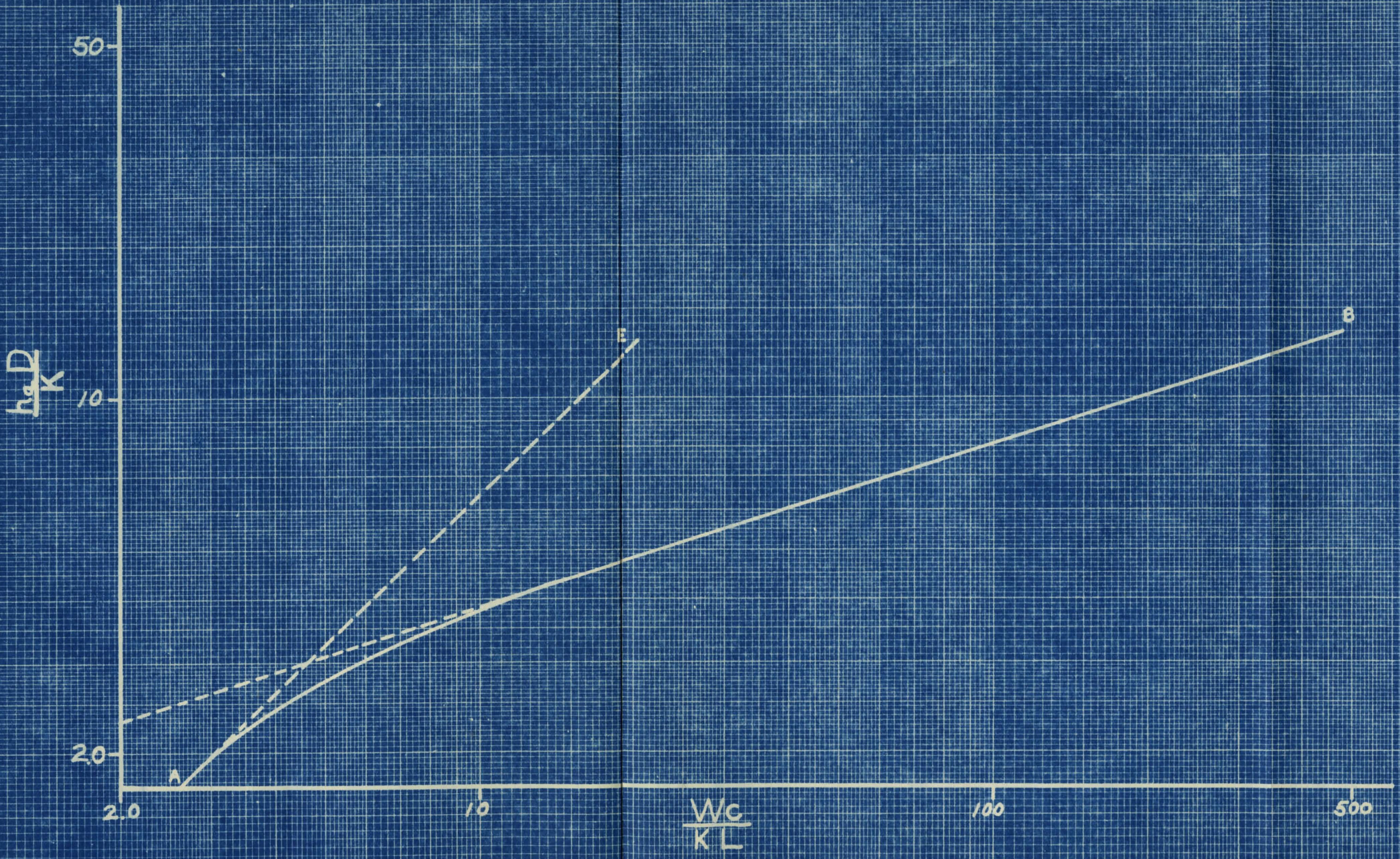


Fig. 2 McAdams Correlation

ratio of the cross sectional area to the wetted perimeter.

In this investigation the evaluation of m was made on two separate bases to determine which value was applicable under the conditions involved:

1. Considering one channel of the annulus alone (cross secional area bounded by the two walls of the annulus and two adjacent fins).

    a. Using total wetted perimeter.

    b. Using only that portion of the wetted perimeter which transfers heat.

2. Considering the total annulus and ignoring the fin area.

    a. Using total wetted perimeter.

    b. Using only that portion of the wetted perimeter which transfers heat.

McAdams (4) recommends the use of the total wetted perimeter in the calculation of fluid flow problems, and the use of only that portion of the wetted perimeter which transfers heat in the calculation of heat transfer data.

The values of the equivalent diameter obtained from calculation of the hydraulic radius by methods (1a) and (1b) were found to be too low as evidenced by abnormally low values of the Reynolds number DG/u, and of the Nusselt number h_aD/k. Evaluation of the hydraulic radius by method (2b) gave abnormally high values of the Reynolds number. The method of (2a) was used in the calculation of the results of this investigation as it gave reasonable values of both DG/u and h_aD/k.

McAdams (4) presents a method for predicting the temperature drop through bar fins from equations obtained by the integration of the conduction equation (4).

For finite fins of constant cross section S and perimeter b, having surface temperature t_x, exposed to surroundings at t_a, a heat balance gives:

$$-kd^2t_x/dx^2 = hbdx(t_x - t_a),$$

neglecting radial gradient in temperature. Integration gives:

$$(\Delta t)_x/(\Delta t)_0 = \cosh a(x_f - x)/\cosh ax_f$$

and

$$(\Delta t)_m/(\Delta t)_0 = \tanh ax_f/ax_f$$

where cosh and tanh represent the hyperbolic cosines and tangents, respectively:

$$\cosh y = (e^y + e^{-y})/2 \; ; \; \tanh y = (e^y - e^{-y})/(e^y + e^{-y})$$

and e = 2.718.

The term a is defined as: $a = (hb/kS)^{0.5}$;

b = exposed perimeter of the fin

h = film heat transfer coefficient of the fluid

k = thermal conductivity of the fin

S = cross sectional area of the fin

x_f = total length of the fin from its base

x = distance from base of fin

$(\Delta t)_0$ = temperature difference between the surrounding fluid and the base of the fin

$(\Delta t)_x$ = temperature difference between the surrounding fluid and the fin at distance x from the base

$(\Delta t)_m$ = mean temperature difference between the surrounding fluid and the entire fin.

In this investigation the calculated values of the heat transfer coefficient h are corrected for the drop in temperature along the fins by the above method.

The same heat exchanger was used throughout in obtaining the experimental data presented in this thesis; however, in those experimental runs which were made for a straight tube and shell heat exchanger the flows of steam and oil were interchanged, the steam being placed in contact with the finned surface. Since the controlling thermal resistance was, in all cases, the oil film, the presence of the extended fin area on the steam side had no effect upon the overall heat transfer coefficient, and the exchanger could be considered to be of the straight tube and shell type.

EXPERIMENTAL

APPARATUS

The experimental apparatus used in this investigation consisted of a shell and tube fin type heat exchanger, and auxiliary equipment. Two methods of operation were employed, using a heavy grade of industrial fuel oil as the experimental fluid:

1. Oil was passed through the tube and steam was introduced into the shell of the exchanger.

2. Oil was passed through the shell and steam was introduced into the tube of the exchanger.

In each of the methods of operation outlined above, the steam temperature, and the inlet and outlet oil temperatures were determined with thermometers, and the rate of oil flow was determined by weighing the amount of oil collected in a tared container over a given timed period.

The pressure of the steam was indicated by a gage, and used as a guide in maintaining constant pressure and therefore constant steam temperature. The pressure was not recorded, and steam quality was not determined, as it was not desired to run a check heat balance on the exchanger.

Two 550 gallon storage tanks were used as oil reservoirs with provisions for pumping to or from either tank. A rotary gear pump was used to provide oil circulation, and a by-pass across the pump discharge provided control of the rate of oil flow through the exchanger. Quick-opening valves on the return line to the reservoirs permitted instantaneous change of direction of flow from the return line into the tared weighing container at the beginning of a timed period, and from the container to the receiving tank at the end of the times period. Times were determined with a one-second interval timer.

The arrangement of the apparatus is illustrated in Fig. 3, with the individual parts being described in detail in the following section.

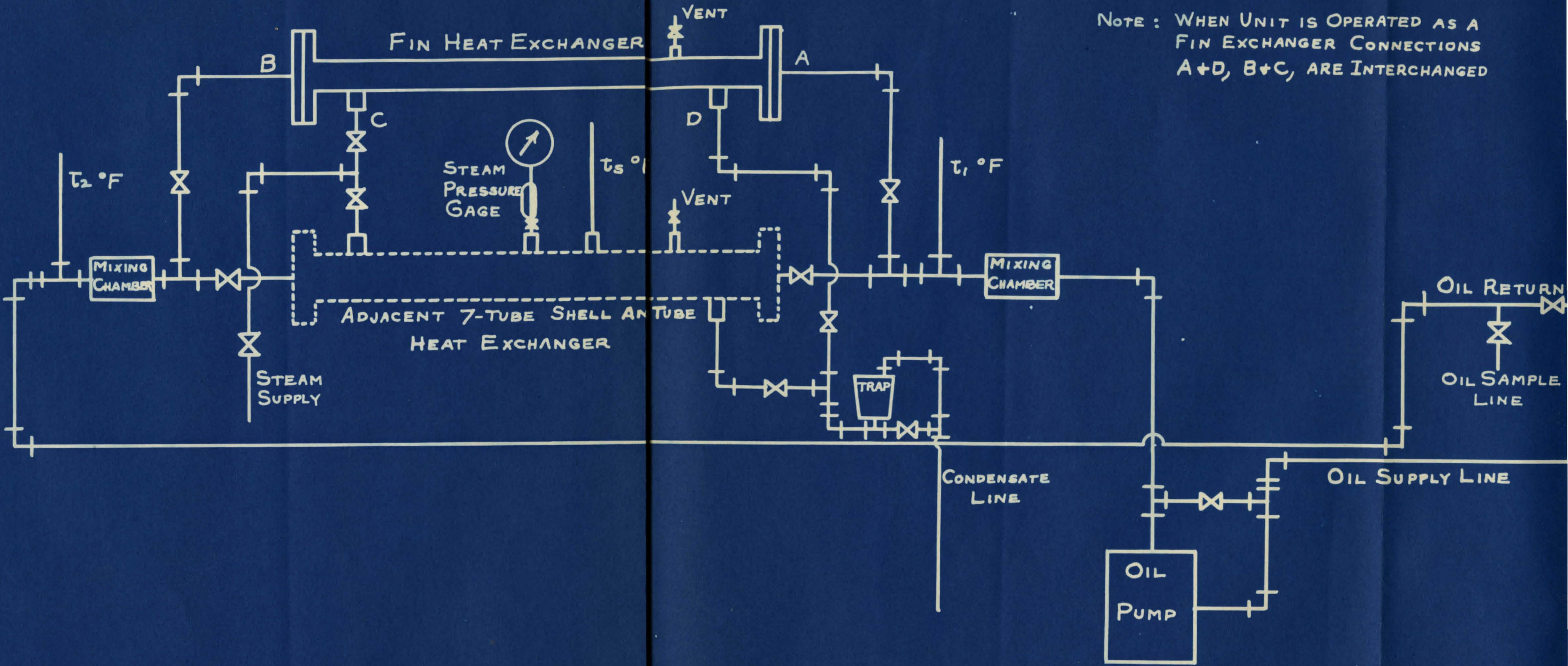
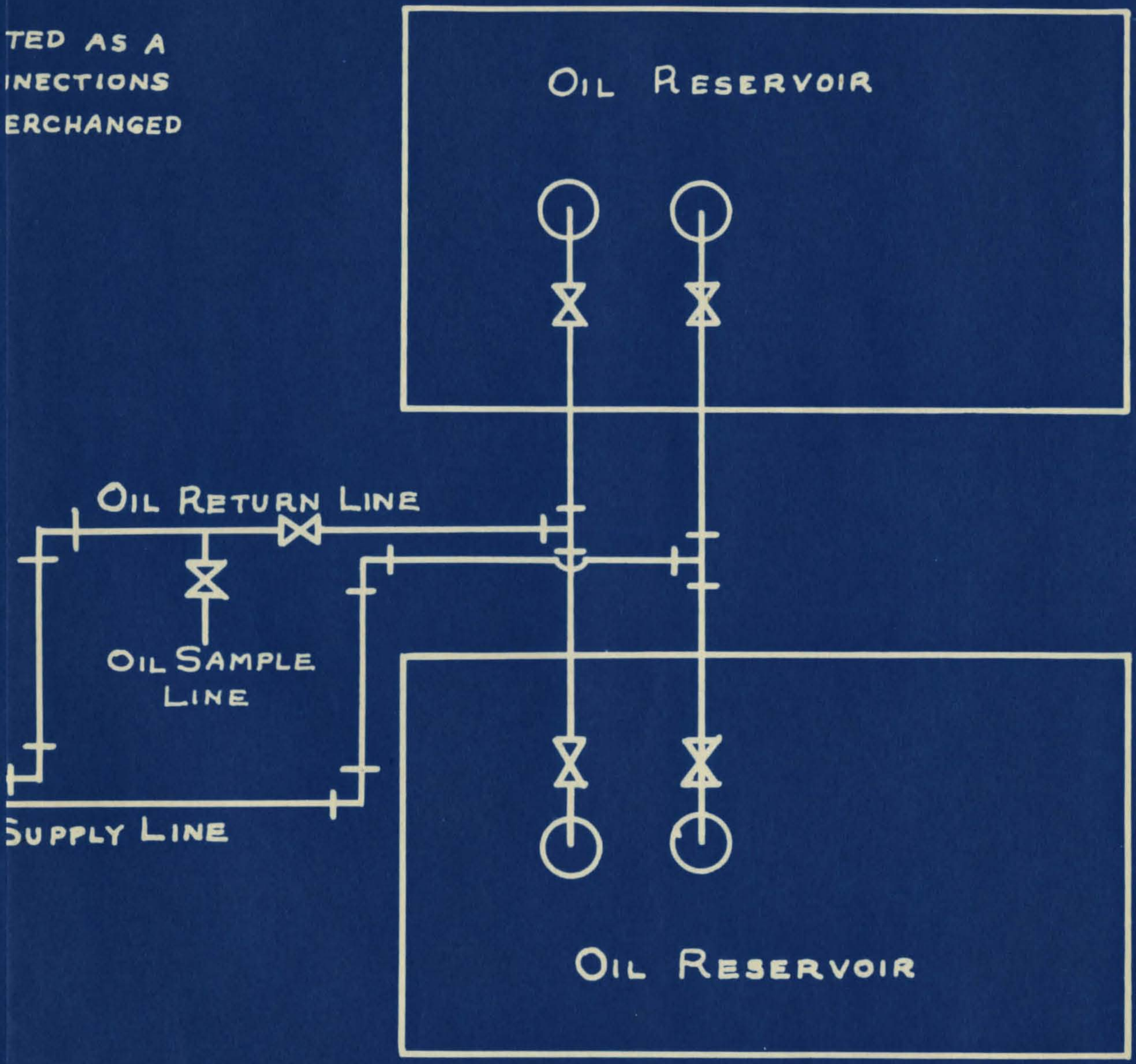


Fig. 3 Schematic Outline of Heat Transfer Unit and Auxiliary Equipment


TED AS A
INECTIONS
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OIL SAMPLE LINE
SUPPLY LINE
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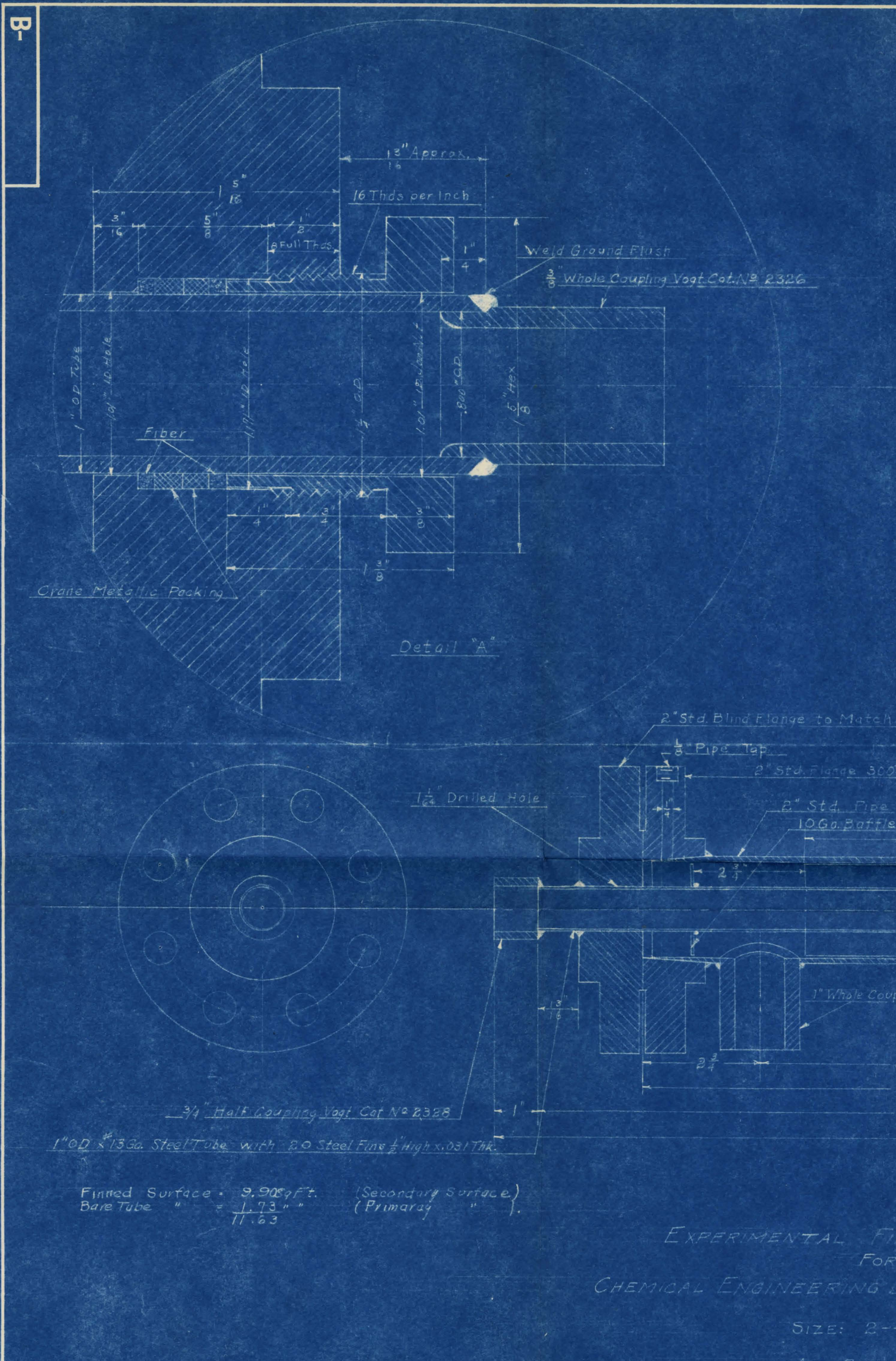
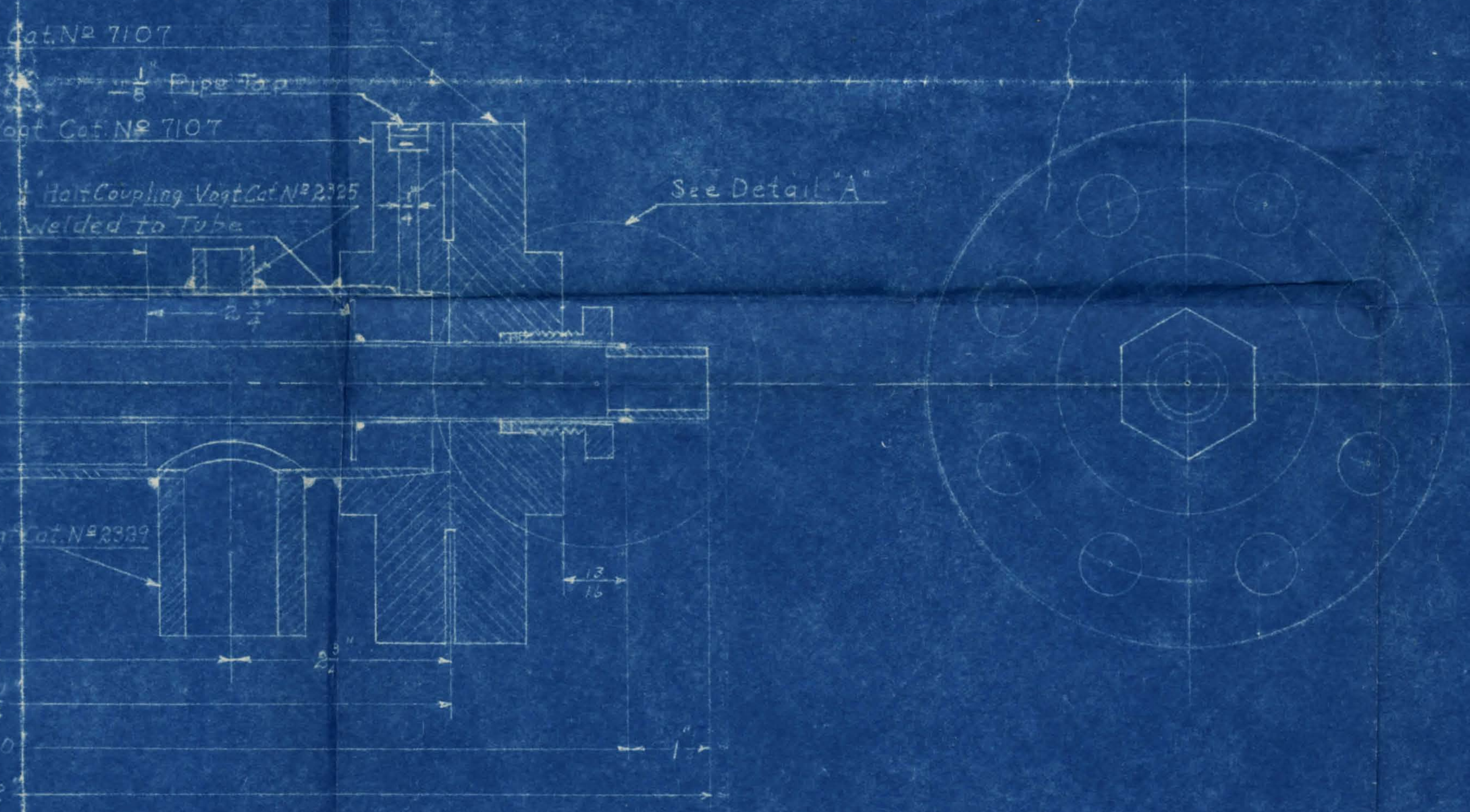
HEAT EXCHANGER

The heat exchanger employed in this investigation was supplied by the Henry Vogt Machine Co., of Louisville, Ky. (Fig. 4). It is of the shell and tube type, containing a single #13 gage steel tube with an outside diameter of one inch, and a length of approximately seven feet. The tube has twenty longitudinal steel fins spot welded to its outer surface, the fins being one-half inch high and thirteen one-hundredths of an inch in thickness, and approximately six feet in length. The outer extremity of each fin is in contact with the inner wall of the shell.

The shell is constructed of two-inch standard steel pipe, flanged at both ends, and fitted with standard one-inch couplings near each end at the bottom for introduction of steam and removal of condensate. A three-eighths inch standard coupling welded into the top of the shell near the exit end provides for the removal of air from the shell when starting the period of operation. The shell was not insulated.

There was no provision on the exchanger for the attachment of a steam gage or thermometer well for determination of steam temperatures and pressures. These values were measured on the shell of an adjacent seven-tube shell and tube exchanger which was connected in parallel with the steam and oil lines of the exchanger used in this investigation, the seven-tube exchanger being provided with a pressure gage and shell thermometer well. During the operation of the test exchanger the steam and condensate lines from the parallel exchanger were left open for the


B-
1 3/16" Approx.
1 5/16"
16 Thds per Inch
3/16"
5/8"
1/2"
8 Full Thds.
1/4"
Weld Ground Flush
3/8" Whole Coupling Vogt Cat. No. 2326
1" O.D. Tube
1.01" I.D. Hole
1.11" I.D. Hole
1 1/4" O.D.
.800" O.D.
1 5/8" Hex
Fiber
1/4"
3/4"
3/8"
1 3/8"
Crane Metallic Packing
Detail "A"
2" Std. Blind Flange to Match
1/8" Pipe Tap
2" Std. Flange 300
1 1/64" Drilled Hole
2" Std. Pipe
10 Ga. Baffle
2 3/4"
1" Whole Coup
1 3/16"
2 3/4"
3/4" Half Coupling Vogt Cat No 2328
1"
1" OD x #13 Ga. Steel Tube with 20 Steel Fins 1/2" High x .031 Thk.
Finned Surface = 9.90 Sq Ft. (Secondary Surface)
Bare Tube " = 1.73 " " (Primary ")
11.63
Experimental F
For
Chemical Engineering
Size: 2-
All Stee
Fig. 4



...BE HEAT EXCHANGER

...RTMENT, SPEED SCIENTIFIC SCHOOL

...72 TYPE: VF-1-1

...ONSTRUCTION.

JOB Nº 175119

APPROVED BY M.F.F.

HENRY VOGT MACHINE CO. LOUISVILLE, KY.	
EXPERIMENTAL HEAT EXCHANGER.	
DRAWN BY RPH CHECKED BY E.T. CHIEF ENG. MS	DATE: 9-24-40 SCALE: NONE

determination of steam temperature and pressure.

In the operation of the unit as a fin type exchanger, the oil and steam lines were interchanged, allowing the oil to pass through the shell in contact with the finned surfaces of the tube, and the steam to be introduced into the tube at one end, with provisisons for air venting and condensate removal at the exit end.

Mixing Chambers:

To assure thorough mixing of the oil before determination of the inlet and exit temperatures two cylindrical perforated plate mixing chambers were used. These chambers were constructed and installed by Tepe (5), and a complete description of their construction together with blueprints can be obtained from this thesis. Mixing was effected by turbulence caused by the oil flowing through staggered holes of various sizes in a series of perforated plates.

Thermometer Wells:

Thermometer Wells for the determination of inlet and outlet oil temperatures were constructed and installed by Tepe (5). Each well consisted of a one-quarter inch copper tube closed by sweating at one end and of sufficient length to reach into the main stream of the oil without touching the pipe wall. Each of these tubes was brazed into a one-inch standard iron pipe plug. The plugs were screwed into tees at the points at which it was desired to take the oil temperatures.

Thermometer wells were filled with cottenseed oil. One-fifth degree Fahrenheit thermometers were used for the determination of oil and steam temperatures.

AUXILIARY EQUIPMENT

Pump:

The pump employed for the circulation of the fuel oil through the heat exchanger is a rotary gear pump, manufactured by the Viking Pump Company, of Cedar Falls, Iowa. It is Type BL, with two-inch suction and discharge openings.

Motor:

The oil pump was driven by a Westinghouse two hundred and twenty volt three phase five horsepower seventeen hundred and fifty revolutions per minute squirrel cage induction motor. The speed of pump rotation was reduced to one hundred and seventy-five revolutions per minute using a line shaft with intermediate pulleys of twenty-five and five inch diameters. The motor pulley was six inches in diameter and the pump pulley twelve inches in diameter.

In order to reduce the pressure drop between the reservoirs and the suction side of the pump, the pump and motor were removed from their existing location on the operating floor and installed in the basement of the laboratory. Since the operating floor is one story above the reservoirs and the basement one-half story below them, this change provided a constant head upon the inlet side of the pump, thus increasing the capacity.

Reservoirs:

The reservoirs consisted of two five hundred and fifty gallon underground gasoline storage tanks supplied by the Standard Oil Company of Kentucky. During operation of the heat exchanger oil was pumped from one tank through the heat exchanger into the other tank. The feed and return lines were connected to both tanks so that direction of flow could be reversed at will.

EXPERIMENTAL PROCEDURE

The following procedure was used in making each experimental run on the heat exchanger:

1. Oil lines were checked
   a. To assure correct flow of oil from and to the reservoirs.
   b. To make sure that all valves on the discharge side of the pump, including the bypass across the pump discharge and suction lines, were wide open. If the pump were started with the discharge line closed the oil line would be broken as the pump is of the positive displacement type.
2. Steam lines were checked
   a. All vents were opened and the condensate drained from the lines.
   b. Steam was introduced into the shell of the exchanger and vented to the atmosphere for several minutes to assure removal of air from the shell of the exchanger.
   c. Vents were then closed and the steam pressure was adjusted to the desired value.

3. Pump was started and the oil flow rate was adjusted to the desired value by regulation of the pump bypass. Maximum oil flow was obtained by completely closing the bypass.

4. Thermometers were then inserted into the wells for measuring the steam temperature, and the inlet and outlet oil temperatures.

5. The exchanger was allowed to operate for approximately

one-half hour to attain equilibrium conditions. During this time temperatures and steam pressure were noted.

6. When inlet and outlet oil temperatures, steam temperature and steam pressure became constant, the timed test period was started. Readings of inlet and outlet oil temperatures, and of steam temperatures, were taken at five minute intervals. Steam pressure was maintained at a constant value. Oil rates of flow were determined in most of the runs ten minutes and thirty-five minutes after the start of the run. In a few cases where it became apparent that equilibrium had not been reached at the beginning of the run, the length of the run was extended to one hour's time and a third oil rate was taken fifty minutes after the starting time.

7. At the completion of the run the thermometers were removed from the wells, the pump was shut down, the steam was shut off, and all vents were opened to the atmosphere.

RESULTS

Before the experimental data obtained could be converted to useful form, it was necessary to determine the variation of the physical properties of the oil with change in temperature. Information on the variation of these properties was obtained from the thesis of Tepe (5), who determined them experimentally or from reliable sources.

In order to simplify the calculations of the experimental runs made on the heat exchanger, values of the oil viscosity in English units, lb/ft.x hr., were calculated and plotted vs. temperature in Fig. (6). The values in terms of Saybolt Seconds were read from Fig. 5 (5) at 20°F. intervals. Values of the specific gravity of the oil in gm./cc. were read at the same temperature intervals from Fig. (5), which was replotted from Fig. 8 (5). Conversion to the English units was effected by the method of McAdams (4):

$$\mu'/\rho = A\theta - B/\theta$$

where μ' = viscosity in poises

ρ = density in gm./c.c.

θ = time of efflux in Saybolt Seconds

A = constant 0.0022

B = constant 1.8

The value of μ' obtained in poises may be converted to μ, lb./ft.x hr., by multiplying by 100 to obtain centipoises and by the constant 2.42 to obtain lb./ft.x hr. (11)

The values of the oil viscosity in lb./ft. x hr. units were determined as in Table I and plotted vs. temperature in Fig. (6).

Values of the specific heats of the oil at various temperatures were replotted in Fig. (7) from Fig. 6 (5).

Values of the thermal conductivity of the oil at various temperatures were replotted in Fig. (8) from Fig. 7 (5).

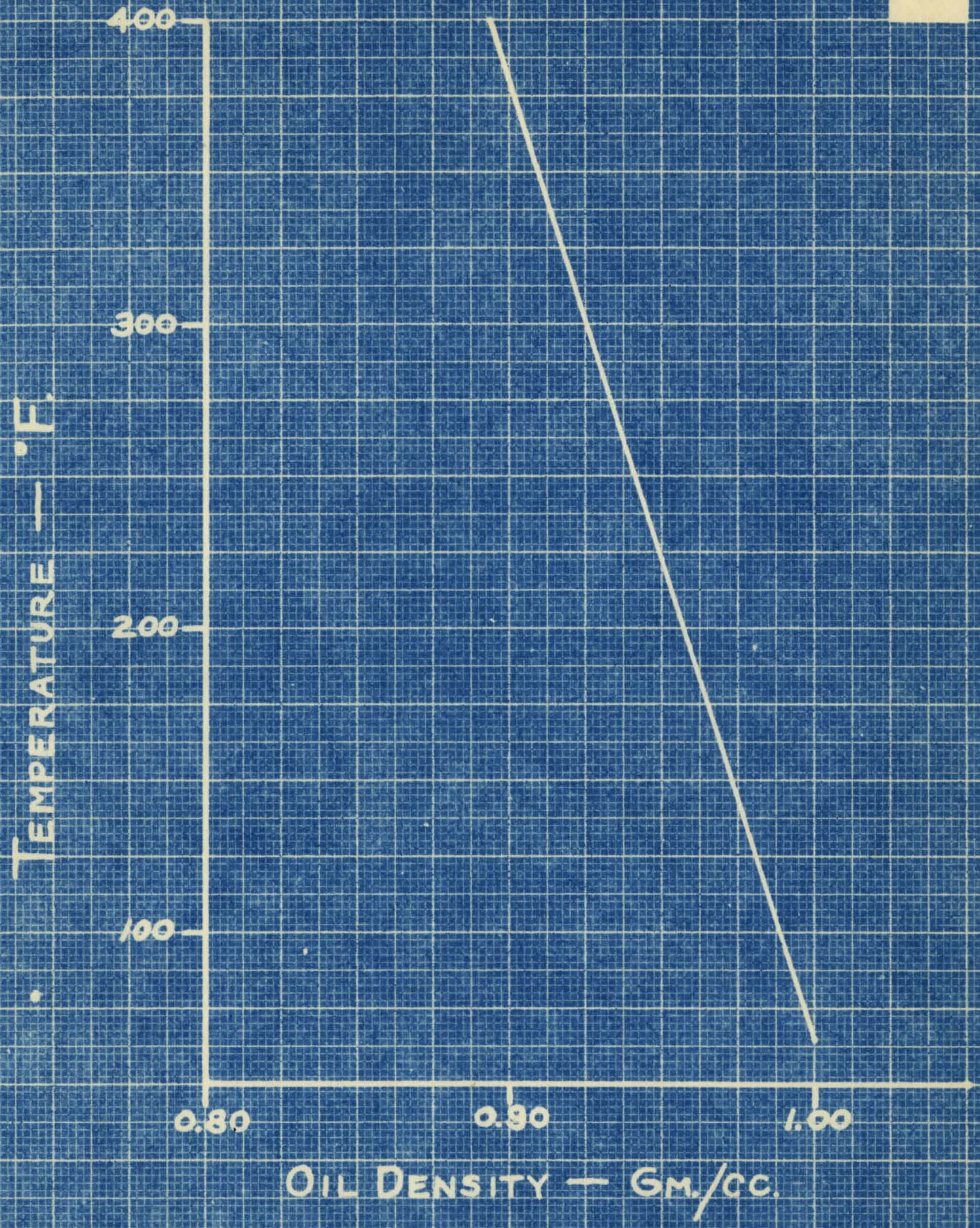


FIG. 5 OIL DENSITY VS. TEMPERATURE

TABLE I

COMPUTATION OF VALUES FOR VISCOSITY VS. TEMPERATURE CURVE

Assumed Oil Temp.	Oil Density	Oil Viscosity	Oil Viscosity
t	ρ	θ	$\mu = (0.00220-1.8/0) \times (242\rho)$
$^{o}F.$	Gm./cc.	Saybolt Seconds	lb./ft.xhr.
90	0.9925	1240	655
110	0.9860	580	304
130	0.9795	310	160.3
150	0.9730	182	91.7
170	0.9675	118	57.2
190	0.9610	84.0	38.0
210	0.9545	65.0	26.7
230	0.9480	54.2	19.7
250	0.9420	47.6	15.3
270	0.9355	43.0	11.9
290	0.9300	39.8	9.52
310	0.9235	37.6	7.76

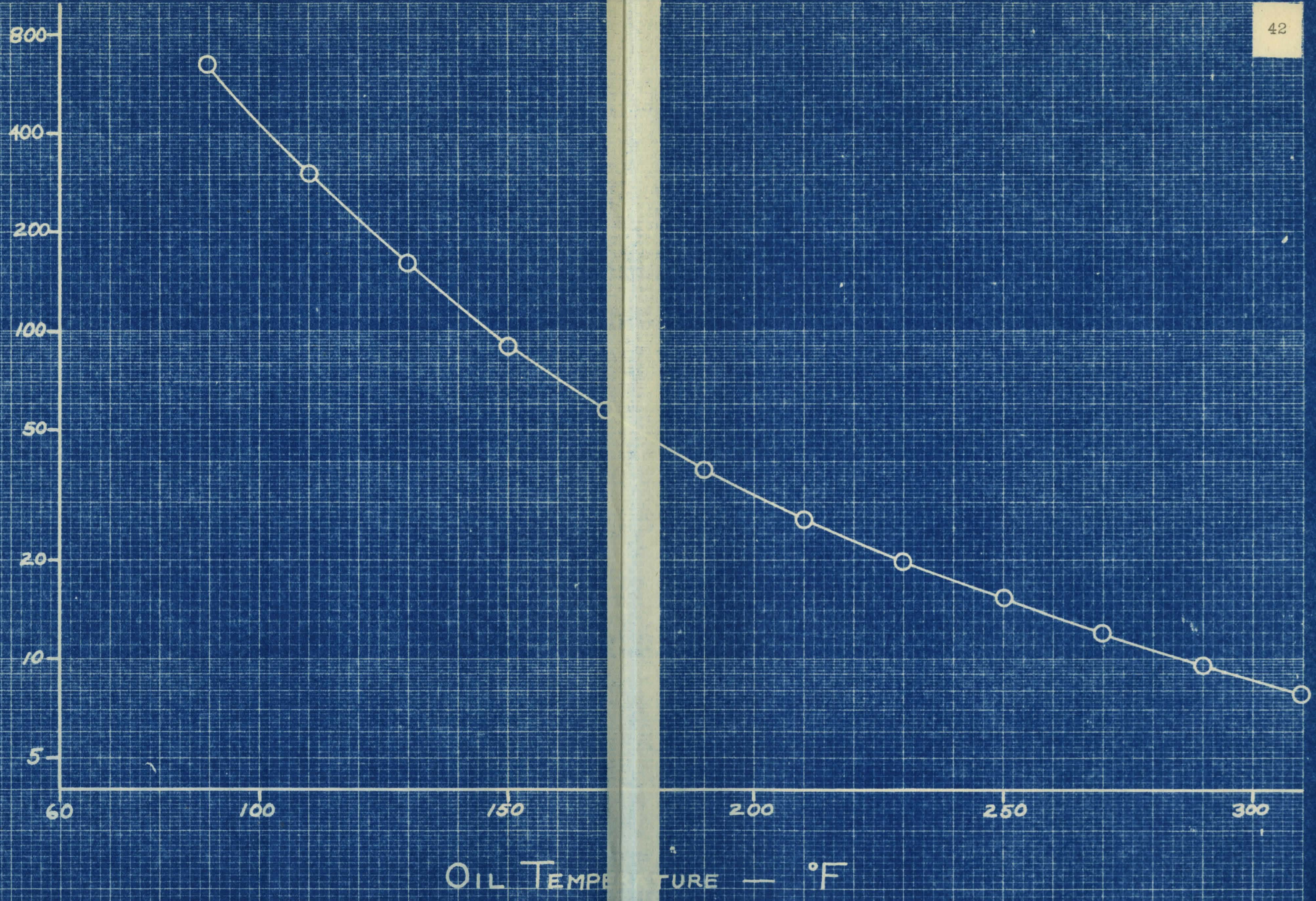


Fig. 6 Viscosity vs. Oil Temperature

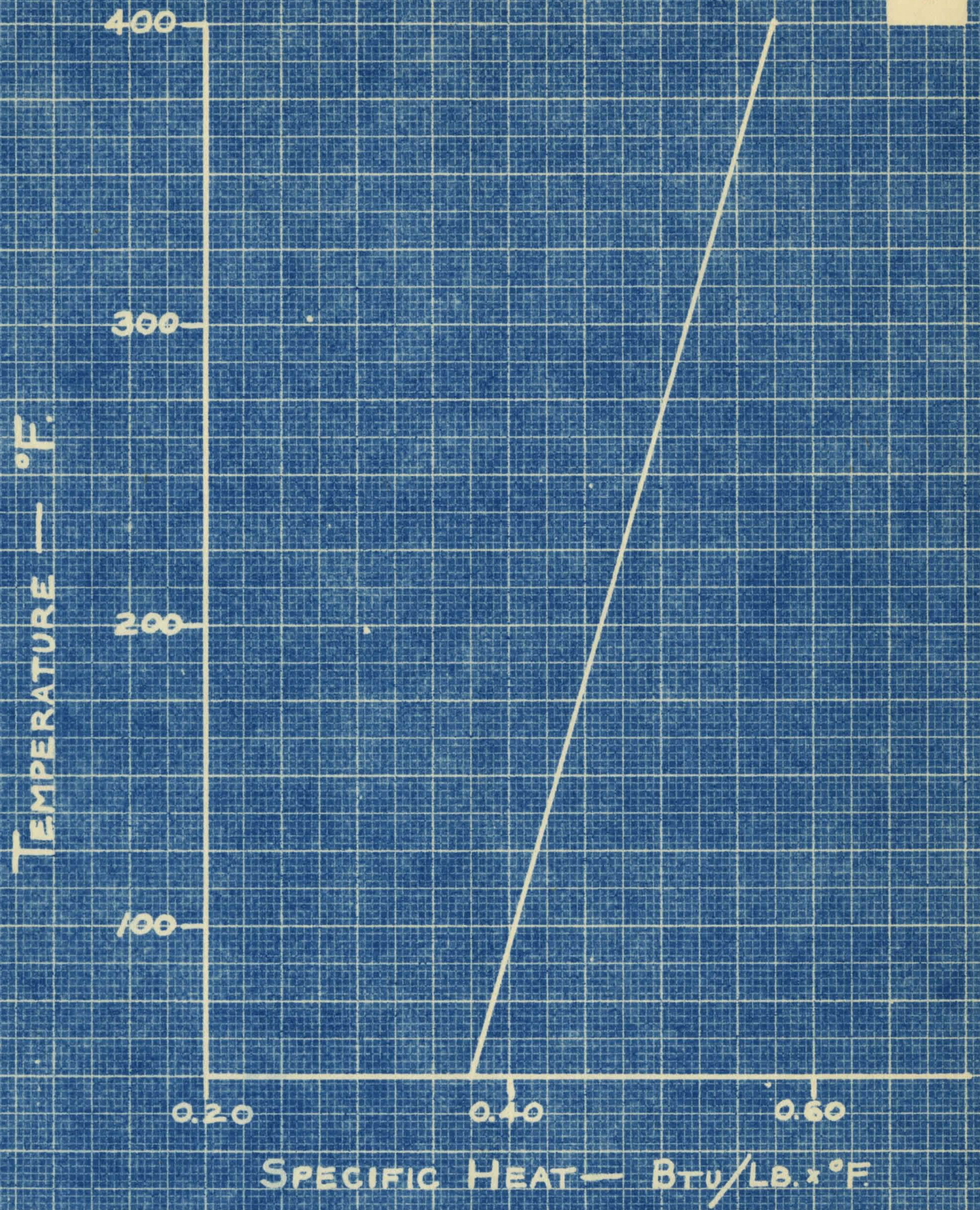


Fig. 7 Specific Heat vs. Temperature

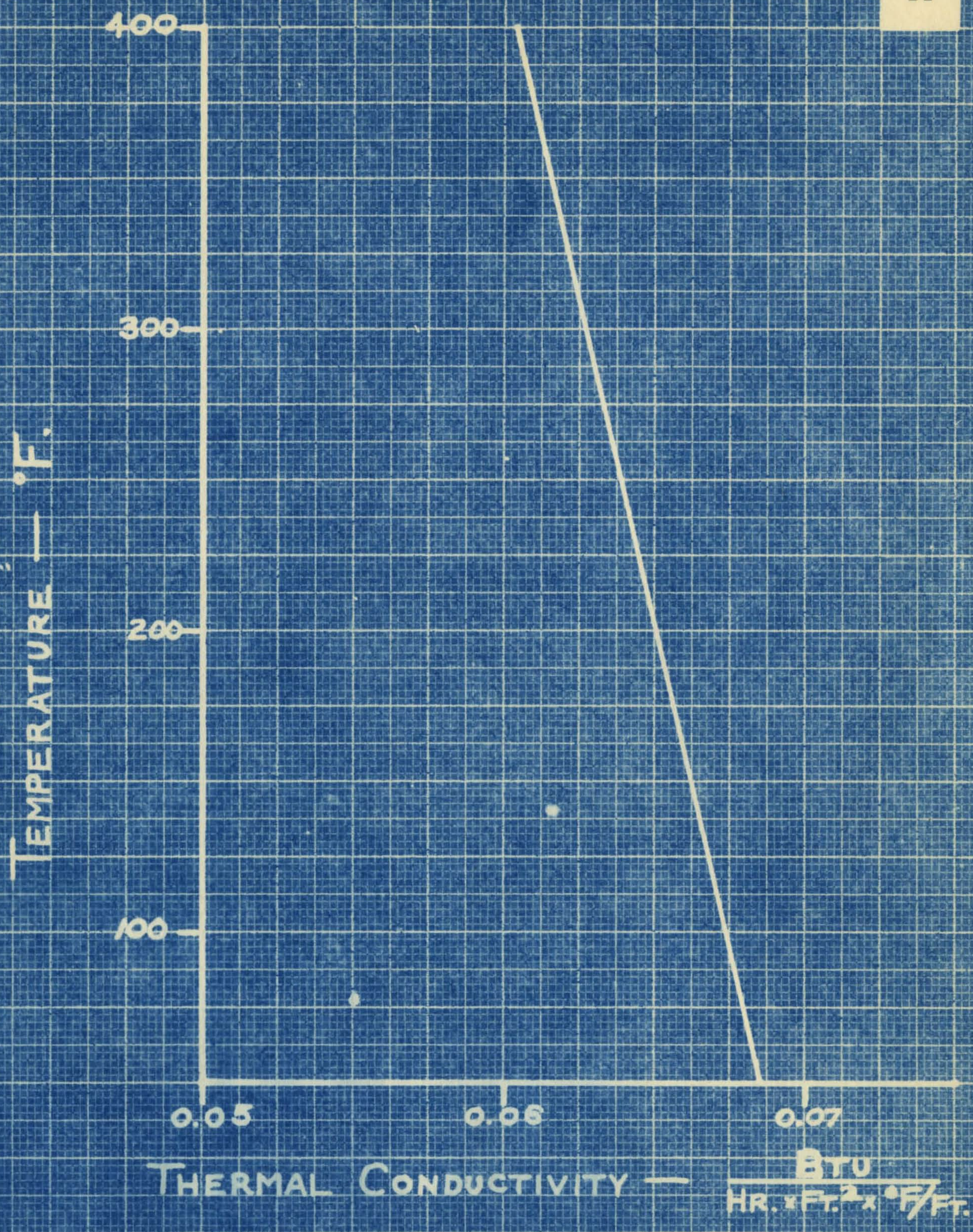


FIG. 8 THERMAL CONDUCTIVITY VS. TEMPERATURE

The observed and calculated data are tabulated in Table II, together with the indicated mathematical operations which enable the calculation of each succeeding step. Values of the variables and constants used in this table are derived in the Sample Calculations[1].

The data for each run are based on the average values of observations made at five minute intervals during the period of each run, during which time operating conditions were kept as nearly constant as possible. As described in the Experimental Procedure[2] the rate of oil flow was determined either two or three times during each run, depending on the length of the run. In those runs in which it became apparent that equilibrium had not been reached at the beginning of the run, the unreliable readings taken at the beginning of the run were discarded and the starting period of the run was advanced to a point at which it was apparent that equilibrium had been reached.

As an inspection of Table II will reveal, it is divided into two sections. Table IIA contains data and results for the experimental runs made with the oil on the tube side and the steam on the shell side of the exchanger. Table IIB contains data and results for the experimental runs made with the oil on the shell side, in contact with the fins, and with steam on the tube side.

Since the exchanger was not lagged there was a heat loss from the shell to the surrounding atmosphere due to conduction and convection. This loss was of no significance when the exchanger was operated with oil on the tube side and steam in the shell, as all

[1]Pages (80-91)
[2]Pages (36)

heat lost was furnished by the steam in the shell. However, with the exchanger operating with the oil flowing on the shell or fin side the heat lost to the atmosphere was given up by the oil, and this introduced an error into the heat balance which was calculated on the basis of oil temperature rise and rate of flow.

The quantity of heat lost under these conditions was estimated by the method of McAdams (4) and the total heat transferred per hour, Q, was corrected for each run. These corrections were not great, however, ranging from one to five per cent of the total heat transferred per hour.

TABLE II-A

EXPERIMENTAL RESULTS - OIL ON TUBE SIDE

	W	G	t_1	t_2	$t_2 - t_1$
		$\frac{W}{0.003418}$			
Run No.	Lb./Hr.	$\frac{Lb.}{Hr. \times Ft.^2}$	^oF	^oF	^oF
1	56.9	16,650	91.3	184.6	93.3
2	90.7	26,550	89.7	181.6	91.9
3	226.0	66,150	85.3	146.2	60.0
4	178.0	52,100	87.9	161.4	73.5
5	416.0	121,800	80.0	141.1	61.1
6	391.5	114,600	80.0	141.0	61.0
7	55.5	16,230	87.8	212.9	125.1

TABLE II-A (Cont.)

EXPERIMENTAL RESULTS - OIL ON TUBE SIDE

Run No.	t_a $\frac{t_1 + t_2}{2}$ °F	t_s °F	Δt_1 $t_s - t_2$ °F	Δt_2 $t_s - t_2$ °F	Δt_{lm} $\frac{\Delta t_1 - \Delta t_2}{2.3 \log \frac{\Delta t_1}{\Delta t_2}}$ °F
1	138.0	262.0	170.7	77.4	118.2
2	136.0	272.7	183.0	91.1	131.8
3	116.0	296.8	211.5	150.6	179.4
4	125.0	296.9	209.0	135.5	170.0
5	110.6	301.8	221.8	160.7	190.0
6	110.5	300.1	220.1	159.1	188.8
7	169.0	299.6	211.8	86.7	140.2

TABLE II-A (Cont.)

EXPERIMENTAL RESULTS - OIL ON TUBE SIDE

Run No.	Δt_a $\frac{\Delta t_1 + \Delta t_2}{2}$ ^oF	c At t_a $\frac{Btu}{Lb. x\ ^oF}$	k At t_a $\frac{Btu}{Hr. x Ft. x\ ^oF}$	μ_a At t_a $\frac{Lb.}{Ft. x Hr.}$	μ_w At t_s $\frac{Lb.}{Ft. x Hr.}$
1	124.0	0.4450	0.06570	128	13.1
2	135.7	0.4425	0.06575	136	11.7
3	181.0	0.4340	0.06620	250	8.95
4	172.2	0.4380	0.06600	187	8.95
5	191.3	0.4300	0.06626	307	8.45
6	189.6	0.4300	0.06626	307	8.60
7	149.3	0.4600	0.06510	58.5	8.65

TABLE II-A (Cont.)

EXPERIMENTAL RESULTS - OIL ON TUBE SIDE

Run No.	$\frac{DG}{\mu_a}$ $\frac{0.0675G}{\mu_a}$	Q $Wc(t_2-t_1)$ Btu/Hr.	U_a $\frac{Q}{1.370\Delta t_a}$ $\frac{Btu}{Hr. x Ft.^2 x {}^oF}$	U_{lm} $\frac{Q}{1.370\Delta t_{lm}}$ $\frac{Btu}{Hr. x Ft.^2 x {}^oF}$	$\frac{1}{U_a}$ $\frac{Hr. x Ft.^2 x {}^oF}{Btu}$
1	8.78	2,360	13.9	14.6	0.0719
2	13.17	3,685	19.9	20.4	0.0502
3	17.87	5,965	24.1	24.3	0.0415
4	18.81	5,740	24.3	24.6	0.0411
5	26.8	10,930	41.7	42.0	0.0240
6	25.2	10,510	40.5	40.6	0.0247
7	17.73	3,200	15.7	16.6	0.0636

TABLE II-A (Cont.)

EXPERIMENTAL RESULTS - OIL ON TUBE SIDE

Run No.	$\frac{1}{U_{lm}}$ $\frac{Hr.xFt.^2x^oF}{Btu}$	$\frac{1}{ha}$ $\frac{1}{U_a}$ -0.0008 $\frac{Hr.xFt.^2x^oF}{Btu}$	ha $\frac{Btu}{Hr.xFt.^2x^oF}$	$\frac{1}{h_{lm}}$ $\frac{1}{U_{lm}}$ -0.0008 $\frac{Hr.xFt.^2x^oF}{Btu}$
1	0.0685	0.0711	14.1	0.0677
2	0.0490	0.0494	20.2	0.0482
3	0.0411	0.0407	24.6	0.0403
4	0.0406	0.0403	24.8	0.0398
5	0.0238	0.0232	43.1	0.0230
6	0.0246	0.0239	41.8	0.0238
7	0.0602	0.0628	15.9	0.0594

TABLE II-A (Cont.)

EXPERIMENTAL RESULTS - OIL ON TUBE SIDE

Run No.	h_{lm} $\frac{Btu}{Hr. x Ft.^2 x\ ^oF}$	$\frac{haD}{k}$ $\frac{0.0675ha}{k}$	$\frac{h_{lm}D}{k}$ $\frac{0.0675h_{lm}}{k}$	$\frac{Wc}{kL}$ $\frac{Wc}{6.458k}$	$(\frac{4}{\pi}\frac{Wc}{kL})$
1	14.8	14.5	15.2	59.6	75.8
2	20.7	20.8	21.2	94.5	120.4
3	24.8	25.1	25.3	229.0	291.5
4	25.1	25.4	25.7	183.0	232.6
5	44.0	43.8	44.8	418.0	532.0
6	42.0	42.6	42.8	393.0	500.0
7	16.8	16.5	17.4	60.8	77.3

TABLE II - (Cont.)

EXPERIMENTAL RESULTS - OIL ON TUBE SIDE

Run No.	$\left(\frac{4}{\pi}\frac{Wc}{kL}\right)^{-1/3}$	$\left(\frac{haD}{k}\right)\left(\frac{4}{\pi}\frac{Wc}{kL}\right)^{-1/3}$	$\frac{\mu_a}{\mu_w}$
1	0.2362	3.422	9.77
2	0.2025	4.210	11.61
3	0.1509	3.785	27.95
4	0.1626	4.130	20.90
5	0.1234	5.405	36.35
6	0.1260	5.370	35.70
7	0.2347	3.875	6.76

TABLE II-B

EXPERIMENTAL RESULTS - OIL ON FIN SIDE

Run No.	W Lb./Hr.	G $\frac{W}{0.01546}$ $\frac{Lb.}{Hr. x Ft.^2}$	t_1 °F	t_2 °F	t_2-t_1 °F	t_a $\frac{t_1 + t_2}{2}$ °F
8	216.2	14,000	93.0	198.5	105.5	145.0
9	1018.0	65,850	86.0	148.7	62.7	117.8
10	868.0	56,150	84.3	156.4	72.1	120.8
11	902.0	58,400	89.6	146.6	57.0	118.1
12	1326.0	85,750	90.0	140.3	50.3	115.2
13	4220.0	273,200	119.2	139.7	20.5	129.5
14	1063.0	68,800	89.9	170.9	81.0	130.4
15	1193.0	77,200	90.5	163.7	73.2	127.1
16	1792.0	116,000	96.7	150.2	53.5	123.5
17	840.0	54,400	86.7	157.9	71.2	122.3
18	980.0	63,400	88.5	173.8	85.3	131.2
19	337.5	21,800	83.0	216.4	138.4	149.7
20	341.5	22,100	82.5	171.4	88.9	127.0
21	351.0	22,700	82.5	190.6	108.1	136.6
22	373.0	24,150	82.4	203.8	121.4	143.1
23	396.0	25,600	83.0	218.9	135.9	160.0
24	394.0	25,480	83.0	223.6	140.6	153.3

TABLE II-B (Cont.)

EXPERIMENTAL RESULTS - OIL ON FIN SIDE

Run No.	t_s °F	Δt_1 $t_s - t_1$ °F	Δt_2 $t_s - t_2$ °F	Δt_{1m} $\frac{\Delta t_1 - \Delta t_2}{2.3 \log \frac{\Delta t_1}{\Delta t_2}}$ °F	Δt_a $\frac{\Delta t_1 + \Delta t_2}{2}$ °F
8	261.3	168.3	62.8	107.0	115.6
9	258.4	172.4	109.7	138.8	141.0
10	262.0	177.7	105.6	138.9	141.7
11	257.6	168.0	111.0	140.3	139.5
12	256.7	166.7	116.4	140.7	141.6
13	256.9	137.7	117.2	128.6	127.4
14	294.5	204.6	123.6	160.3	164.1
15	286.1	195.6	122.4	156.7	159.0
16	286.8	190.1	136.6	162.1	163.4
17	260.8	174.1	102.9	135.2	138.5
18	307.3	218.8	133.5	172.7	176.2
19	284.3	201.3	67.9	122.9	134.6
20	235.2	152.7	63.8	102.0	108.3
21	256.1	173.6	65.5	111.1	119.6
22	271.8	189.4	68.0	118.8	128.7
23	294.7	211.7	75.8	132.2	143.8
24	305.3	222.3	81.7	140.6	152.0

TABLE II-B (Cont.)

EXPERIMENTAL RESULTS - OIL ON FIN SIDE

Run No.	c At t_a $\frac{Btu}{Lb. \times {}^oF}$	k At t_a $\frac{Btu}{Hr. \times Ft. \times {}^oF}$	μ_a At t_a $\frac{Lb.}{Ft. \times Hr.}$	μ_w At t_s $\frac{Lb.}{Ft. \times Hr.}$	$\frac{D_eG}{\mu_a}$ $\frac{0.0883\ G}{\mu_a}$
8	0.4480	0.06560	106.0	13.3	11.62
9	0.4350	0.06610	235.0	13.8	24.74
10	0.4360	0.06605	214.0	13.1	23.15
11	0.4350	0.06610	235.0	13.8	21.90
12	0.4330	0.06620	260.0	14.0	29.05
13	0.4410	0.06590	264.0	14.0	91.40
14	0.4415	0.06585	258.0	9.15	23.55
15	0.4390	0.06595	177.0	10.0	38.45
16	0.4380	0.06600	196.0	9.90	52.20
17	0.4370	0.06610	205.0	13.4	23.40
18	0.4420	0.06585	156.0	7.95	35.82
19	0.4510	0.06545	93.0	10.1	20.65
20	0.4390	0.06595	177.0	18.4	11.02
21	0.4440	0.06575	132.0	14.0	15.18
22	0.4470	0.06565	111.0	11.8	19.18
23	0.4560	0.06530	71.5	9.10	31.60
24	0.4525	0.06540	84.2	8.20	26.68

TABLE II-B (Cont.)

EXPERIMENTAL RESULTS - OIL ON FIN SIDE

Run No.	Q' $Wc(t_2-t_1)$ $\frac{Btu}{Hr.}$	Δt_{sa} Shell to Atmosphere t_a - 80 ^oF	$h_c + h_r$ $\frac{Btu}{Hr. x Ft.^2 x {}^oF}$	Q'' $(4.03 \Delta t_{sa}) \cdot (h_c + h_r)$ $\frac{Btu}{Hr.}$
8	10,220	65.0	2.22	581
9	27,780	37.8	2.11	322
10	27,320	40.8	2.12	346
11	22,390	38.1	2.11	324
12	28,880	35.2	2.10	298
13	40,050	49.5	2.16	431
14	38,100	50.4	2.16	439
15	38,350	47.1	2.15	409
16	42,000	43.5	2.13	374
17	26,200	42.3	2.13	364
18	36,950	51.2	2.16	446
19	20,350	69.7	2.24	630
20	13,320	47.0	2.15	407
21	16,860	56.6	2.19	500
22	20,240	63.1	2.21	562
23	24,540	80.0	2.28	735
24	25,060	73.3	2.26	668

TABLE II-B (Cont.)

EXPERIMENTAL RESULTS - OIL ON FIN SIDE

Run No.	Q $Q' + Q''$ $\frac{Btu}{Hr.}$	U_a $\frac{Q}{11.63\Delta t_a}$ $\frac{Btu}{Hr. x Ft.^2 x\ ^oF}$	U_{lm} $\frac{Q}{11.63\Delta t_{lm}}$ $\frac{Btu}{Hr. x Ft.^2 x\ ^oF}$	$\frac{1}{U_a}$ $\frac{Hr. x Ft.^2 x\ ^oF}{Btu}$
8	10,800	8.02	8.56	0.1247
9	28,100	17.11	17.40	0.0584
10	27,670	16.80	17.12	0.0595
11	22,710	13.98	13.89	0.0715
12	29,180	17.70	17.80	0.0565
13	40,480	27.25	27.00	0.0366
14	38,540	20.15	20.60	0.0496
15	38,760	20.92	21.22	0.0478
16	42,370	22.22	22.40	0.0449
17	26,560	16.46	16.84	0.0607
18	37,400	18.21	18.61	0.0549
19	20,980	13.40	14.68	0.0746
20	13,730	10.89	11.57	0.0918
21	17,360	12.47	13.40	0.0803
22	20,800	13.90	15.05	0.0719
23	25,280	15.10	16.40	0.0661
24	25,730	14.52	15.71	0.0688

TABLE II-B (Cont.)

EXPERIMENTAL RESULTS - OIL ON FIN SIDE

Run No.	$\frac{1}{U_{lm}}$ $\frac{Hr.xFt.^2x^oF}{Btu}$	$\frac{1}{ha}$ $\frac{1}{U_a} - 0.0126$ $\frac{Hr.xFt.^2x^oF}{Btu}$	$\frac{1}{h_{lm}}$ $\frac{1}{U_{lm}} - 0.0126$ $\frac{Hr.xFt.^2x^oF}{Btu}$	ha Uncorrected $\frac{Btu}{Hr.xFt.^2x^oF}$
8	0.1168	0.1121	0.1042	8.91
9	0.0575	0.0458	0.0449	21.82
10	0.0584	0.0469	0.0458	21.35
11	0.0720	0.0589	0.0594	17.00
12	0.0561	0.0439	0.0435	22.80
13	0.0370	0.0240	0.0244	41.60
14	0.0485	0.0370	0.0359	27.00
15	0.0470	0.0352	0.0344	28.40
16	0.0446	0.0323	0.0320	30.95
17	0.0593	0.0481	0.0467	20.80
18	0.0536	0.0423	0.0410	23.62
19	0.0681	0.0620	0.0555	16.10
20	0.0865	0.0792	0.0739	12.62
21	0.0746	0.0677	0.0620	14.77
22	0.0665	0.0593	0.0539	16.87
23	0.0610	0.0535	0.0484	18.68
24	0.0636	0.0562	0.0510	17.78

TABLE II-B (Cont.)

EXPERIMENTAL RESULTS - OIL ON FIN SIDE

Run No.	h_{lm} Uncorrected $\frac{Btu}{Hr.xFt.^2 x ^oF}$	log ax_f: log ha	log ax_f: 0.5 log ha	log ax_f: 0.5 log ha - 1.1763	ax_f
8	9.58	0.9499	0.4750	-0.7013	0.1989
9	22.26	1.3389	0.6694	-0.5069	0.3113
10	21.82	1.3294	0.6647	-0.5116	0.3079
11	16.83	1.2304	0.6152	-0.5611	0.2748
12	23.00	1.3579	0.6789	-0.4974	0.3181
13	41.00	1.6191	0.8096	-0.3667	0.4298
14	27.82	1.4314	0.7157	-0.4606	0.3462
15	29.05	1.4533	0.7266	-0.4497	0.3550
16	31.20	1.4907	0.7454	-0.4309	0.3708
17	21.40	1.3181	0.6590	-0.5173	0.3039
18	24.40	1.3733	0.6866	-0.4897	0.3238
19	18.00	1.2068	0.6034	-0.5729	0.2674
20	13.52	1.1011	0.5506	-0.6251	0.2371
21	16.10	1.1694	0.5847	-0.5916	0.2561
22	18.56	1.2271	0.6136	-0.5627	0.2737
23	20.65	1.2714	0.6357	-0.5406	0.2880
24	19.60	1.2500	0.6250	-0.5513	0.2809

TABLE II-B (Cont.)

EXPERIMENTAL RESULTS - OIL ON FIN SIDE

Run No.	$\tanh ax_f$	$\frac{(\Delta t)_m}{(\Delta t)_o}$ = $\frac{\tanh ax_f}{ax_f}$	h_a Corrected = $\frac{h_a}{(\Delta t)_m/(\Delta t)_o}$	h_{lm} Corrected = $\frac{h_{lm}}{(\Delta t)_m/(\Delta t)_o}$	$\frac{h_a D_e}{k}$ = $\frac{0.0883\ h_a}{k}$
8	0.1963	0.988	9.03	9.70	12.15
9	0.3016	0.969	22.55	23.00	30.10
10	0.2985	0.970	22.00	22.52	29.40
11	0.2681	0.975	17.42	17.28	23.25
12	0.3078	0.967	23.60	23.80	31.45
13	0.4051	0.944	44.10	43.50	59.10
14	0.3330	0.961	28.05	29.00	37.65
15	0.3408	0.960	29.60	30.25	39.60
16	0.3547	0.956	32.40	32.60	43.40
17	0.2949	0.971	21.40	22.05	28.60
18	0.3129	0.967	24.42	25.22	32.80
19	0.2612	0.976	16.50	18.42	22.25
20	0.2328	0.983	12.85	13.77	17.20
21	0.2506	0.979	15.08	16.44	20.24
22	0.2671	0.976	17.28	19.00	23.20
23	0.2803	0.974	19.18	21.20	25.94
24	0.2737	0.974	18.27	20.12	24.65

TABLE II-B (Cont.)

EXPERIMENTAL RESULTS - OIL ON FIN SIDE

Run No.	$\frac{h_{lm} D_e}{k}$ $\frac{0.0883\ h_{lm}}{k}$	$\frac{Wc}{kL}$ $\frac{Wc}{6.458\ k}$	$\left(\frac{4\ Wc}{\pi\ kL}\right)$	$\left(\frac{4\ Wc}{\pi\ kL}\right)^{-1/3}$	$\left(\frac{ha\ D_e}{k}\right)$ $\left(\frac{4\ Wc}{\pi\ kL}\right)^{-1/3}$	$\frac{\mu_a}{\mu_w}$
8	13.05	228.4	290.5	0.1510	1.83	8.0
9	30.70	1037.0	1320.0	0.0912	2.74	17.0
10	30.05	887.5	1130.0	0.0960	2.82	16.3
11	23.05	918.5	1170.0	0.0949	2.20	17.0
12	31.75	1342.0	1708.0	0.0837	2.64	18.6
13	58.30	4370.0	5560.0	0.0564	3.34	18.9
14	38.90	1105.0	1408.0	0.0892	3.36	28.2
15	40.50	1229.0	1562.0	0.0862	3.42	17.7
16	43.60	1842.0	2345.0	0.0753	3.26	19.6
17	29.45	860.0	1094.0	0.0970	2.78	15.3
18	33.85	1018.0	1297.0	0.0917	3.01	19.6
19	24.85	359.5	457.0	0.1298	2.89	9.2
20	18.42	351.0	446.5	0.1308	2.25	9.6
21	22.08	366.5	466.0	0.1290	2.61	9.4
22	25.55	392.5	500.0	0.1260	2.92	9.4
23	28.65	428.0	545.0	0.1225	3.18	7.9
24	27.16	422.0	537.0	0.1230	3.04	10.3

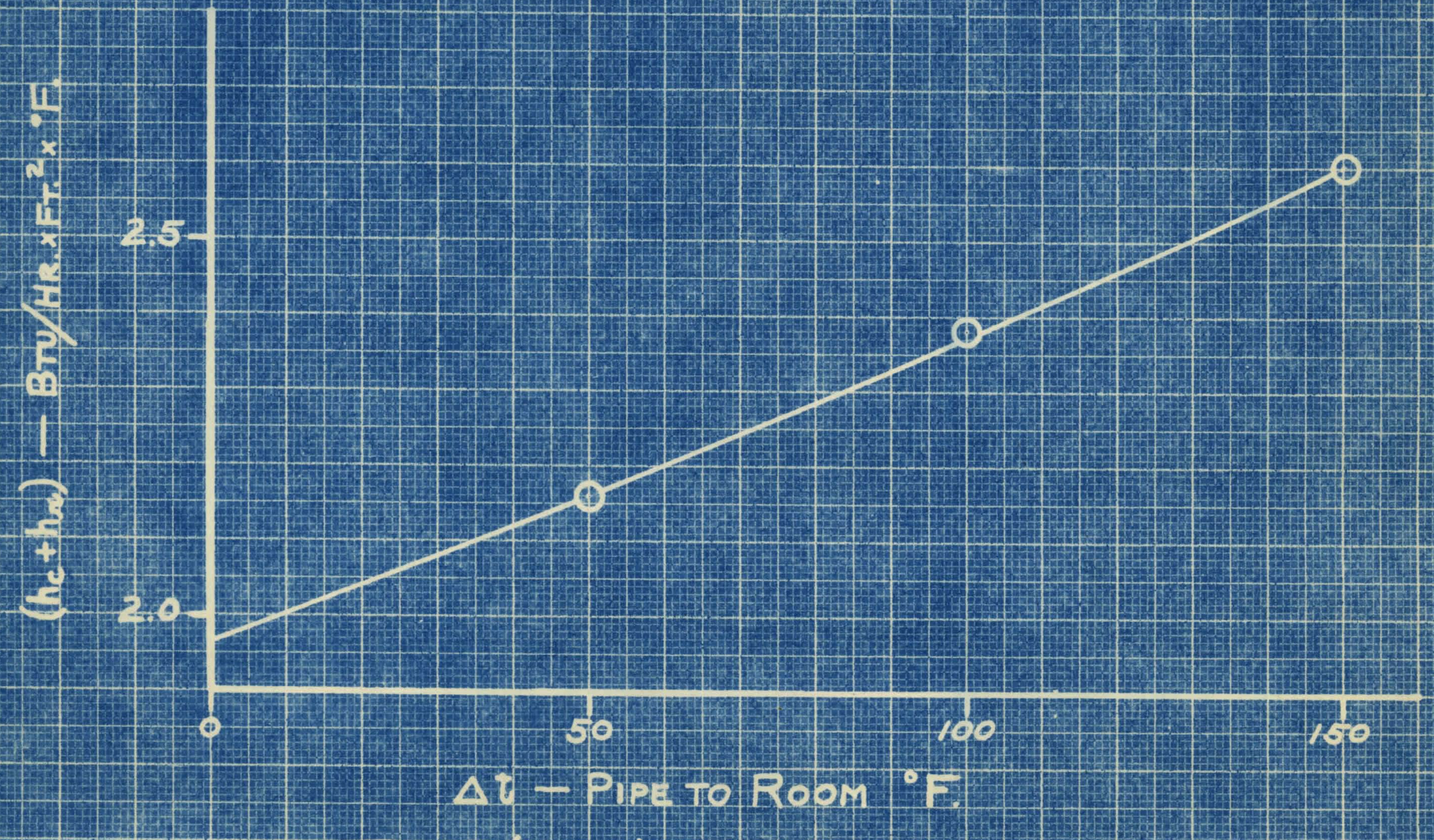


FIG. 9 $(h_c + h_r)$ VS. TEMPERATURE DIFFERENCE TO ROOM AT 80 °F.

DISCUSSION

The derived data obtained as a result of this investigation were plotted according to the existing methods of correlation, in order to determine whether the empirical equations proposed by these correlations would be substantiated by this data.

In Fig. (10) the proposed correlation of Sieder and Tate (3) is reproduced, and the values obtained in this investigation are plotted on this figure. Most of the data were obtained under conditions such that the values of the ratio μ_a/μ_w were larger than those of the data correlated by Sieder and Tate; the data of the latter including few values of μ_a/μ_w above 10, while the data derived in this investigation includes values of μ_a/μ_w from 7 to 37.

The data obtained when the heat exchanger was operated with the oil on the tube side was found to lie about forty per cent above the extension of the Sieder and Tate curve, which is in general agreement with the findings of Tepe (5) who reported data approximately fifty-five per cent above the curve.

The data obtained when the heat exchanger was operated with the oil on the fin side was found to lie close to and on both sides of the Sieder and Tate curve, the mean being almost identical with the curve.

The McAdams (4) correlation is reproduced in Fig. (11), and the data of this thesis plotted upon it. The data obtained in both methods of operation of the heat exchanger were found to lie considerably above this curve.

In order to draw a comparison between a fin type heat exchanger and a tube and shell exchanger of the same size, values of the total heat transferred per hour per unit length of exchanger, Q/L, were tabulated with the corresponding values of the mass velocity, G, in Table III. The values of Q/L were then plotted vs. G for each type of exchanger in Fig. (12).

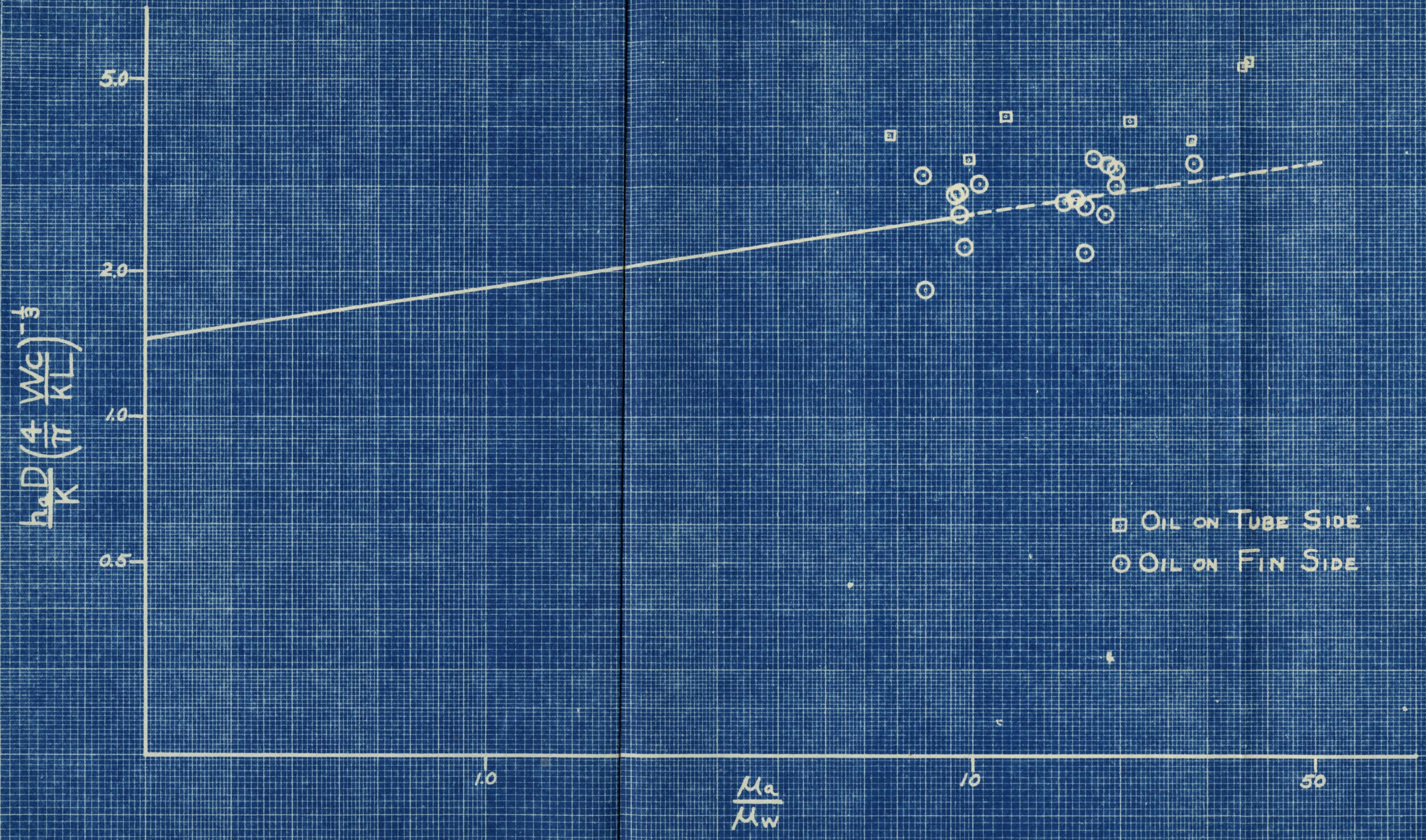


Fig. 10 Comparison with Sieder and Tate Correlation

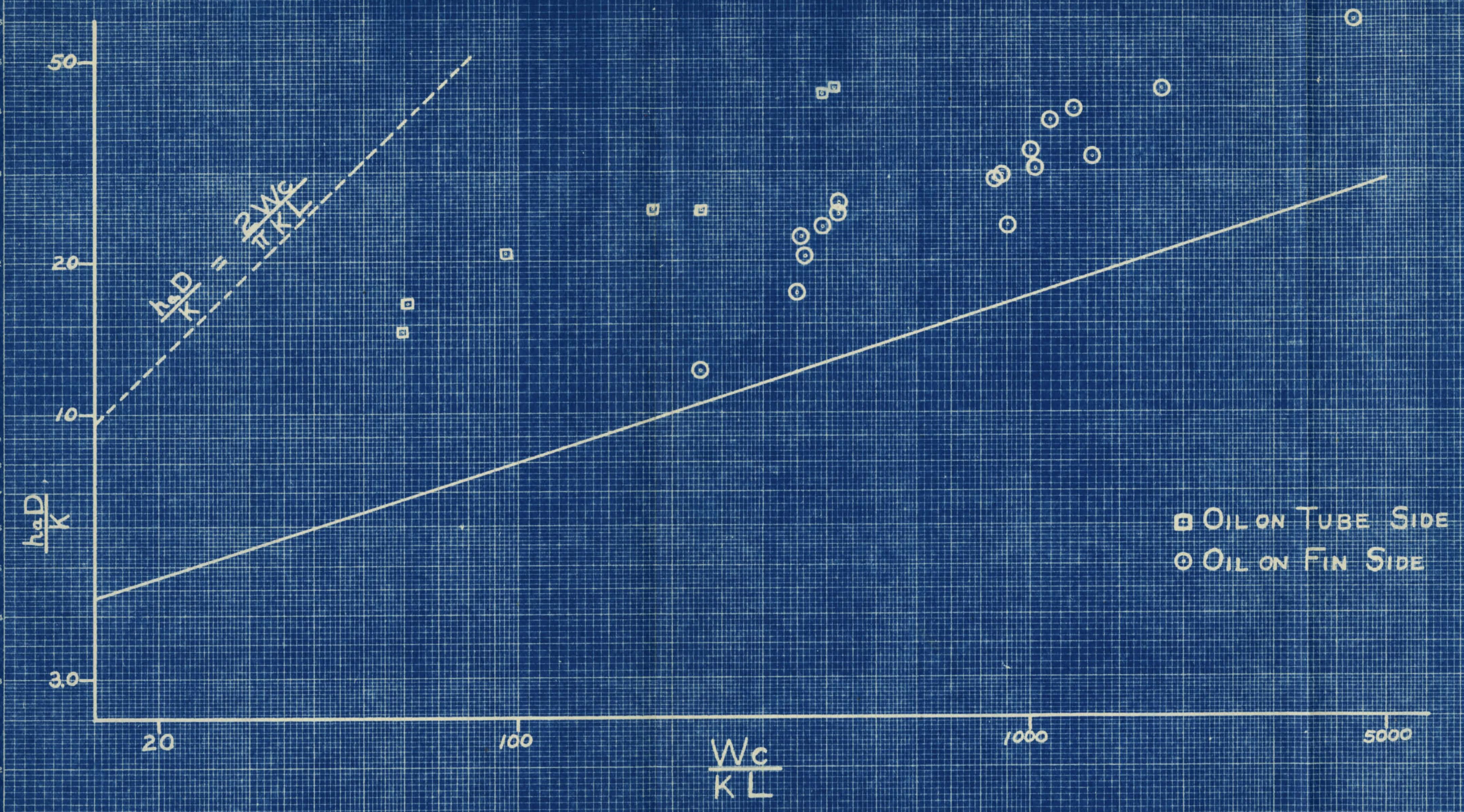


FIG. 11 COMPARISON WITH MCADAMS CORRELATION

TABLE III

COMPARISON BETWEEN SHELL AND TUBE HEAT EXCHANGERS WITH AND WITHOUT FINS

Run No.	Q/L Q/6.458 Btu/hr.x ft.	G lb./hr.x ft.^{2}
1	365	16,650
2	570	26,550
3	924	66,150
4	888	52,100
5	1691	121,800
6	1628	114,600
7	495	16,230
8	1670	14,000
9	4350	65,850
10	4285	56,150
11	3510	58,400
12	4510	85,750
13	6260	273,200
14	5960	68,800
15	5995	77,200
16	6550	116,000
17	4105	54,400
18	5790	63,400
19	3245	21,800
20	2120	22,100
21	2684	22,700
22	3230	24,150
23	3915	25,600
24	3980	25,480

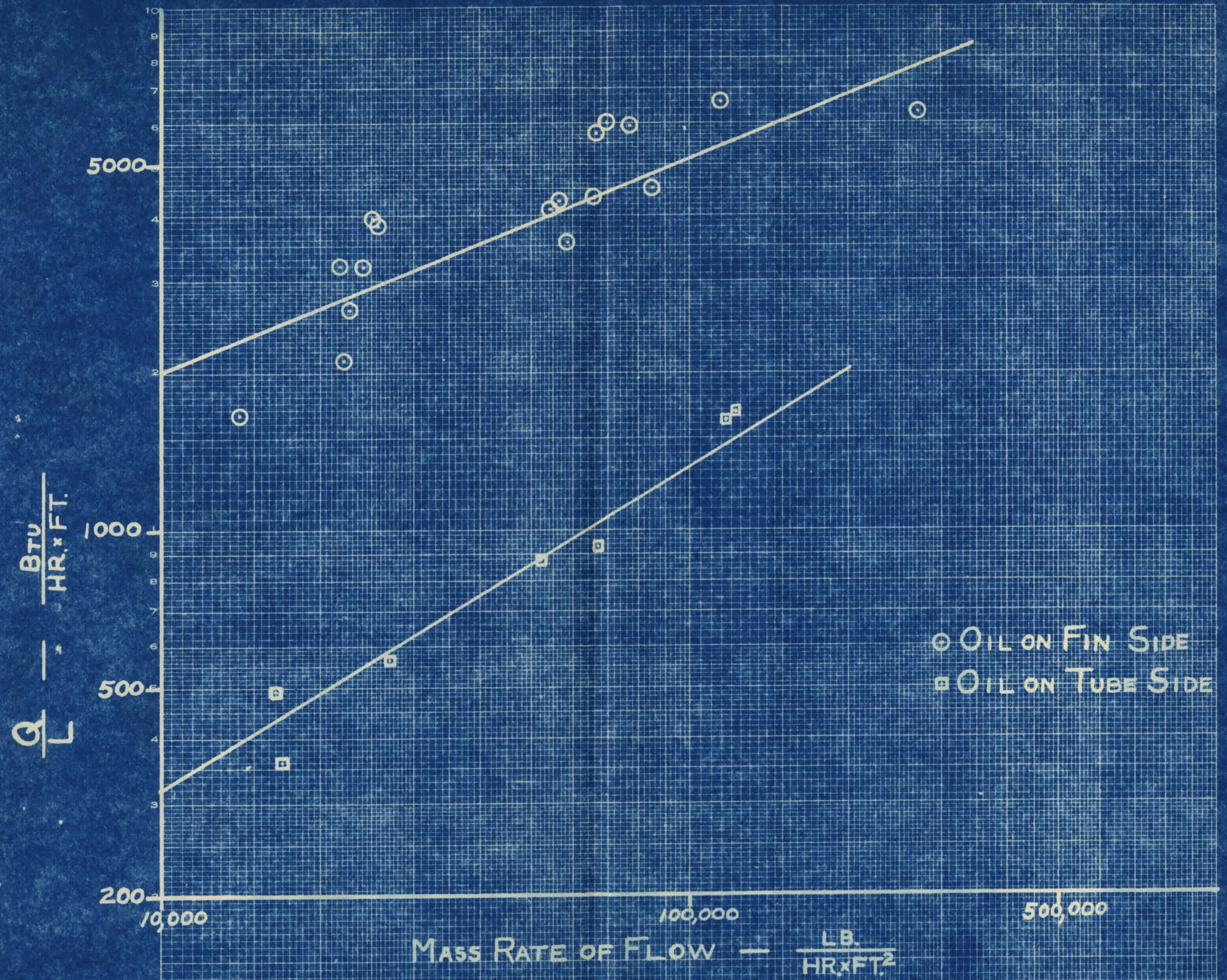


Fig. 12 Comparison Between Heat Exchangers

SUMMARY AND CONCLUSIONS

From the comparisons drawn between the data of this thesis and the correlations of McAdams (4) Fig. (11), and of Sieder and Tate (3) Fig. (10), it can be concluded that the use of a term such as μ_a/μ_w, the ratio of the oil viscosities at the bulk temperature and the wall temperature, is necessary to allow for the effect of radial variation in fluid viscosity caused by the temperature gradient through the fluid cross section. This is demonstrated particularly well by the data obtained on the fin type heat exchanger[1], which is correlated fairly well by the Sieder and Tate method, while falling one hundred per cent above

[1]It should be pointed out here that in order to obtain a correlation of the fin heat exchanger data with previously proposed methods of correlation, it was necessary to modify the previously proposed methods of evaluation of the equivalent pipe diameter. The data correlated by Sieder and Tate (3) was obtained on liquids flowing inside tubes, and therefore the problem of evaluating an equivalent diameter did not arise. However, in order to obtain an agreement between this data and the fin heat exchanger data the equivalent diameter had to be evaluated using the total wetted perimeter of the annulus alone, excluding the perimeter of the fins. This is in contrast with the proposed method for heat transfer (4), under which evaluation would be made using only that portion of the wetted perimeter which transfers heat.

the correlation of McAdams, the latter containing no term to allow for the radial variation in viscosity. The data obtained on the straight tube and shell exchanger, while not satisfactorily correlated by either method, falls closer to the curve of Sieder and Tate.

The failure of the Sieder and Tate correlation for values of the ratio μ_a/μ_w above 10 is indicated by the data of Tepe (5) and borne out by the data on the straight tube and shell exchanger which was obtained in this investigation. It is therefore evident that in order to obtain a closer correlation of the data having values of μ_a/μ_w above 10, an additional factor which takes this into account should be included in the correlation.

An examination of the plot of Q/L vs. G (Fig. 12) for the tube and shell exchanger with and without fins shows that the addition of fins permitted an average increase of approximately four hundred per cent in the heat transferred per foot of exchanger length over the same exchanger without fins[1]. While this comparison is not quantitative[2]; considering the results qualitatively it is apparent that in any case of heat transfer where one fluid film is definitely controlling the rate of heat transfer, the use of an exchanger of the fin type would be desirable in view of the savings effected in material and installation space.

[1]See footnote 1, page 3.

[2]In one case the oil was flowing inside a tube and in the other case in the annulus around the tube. To draw a strict comparison between tube and shell heat exchangers, with and without fins, the oil would have to flow outside the tube in both cases.

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APPENDIX

LIST OF SYMBOLS

h - film coefficient of heat transfer, Btu/hr.x ft.^{2} x ^oF

D - pipe diameter, ft.

k - thermal conductivity, Btu/hr. x ft.^{2}x ^oF./ft.

G - mass velocity, lb./ft.^{2}x hr.

μ - viscosity, lb./ft. x hr.

c - specific heat of fluid at constant pressure, Btu/lb. x ^oF.

g - acceleration due to gravity, ft./ sec.^{2}

Δt - temperature difference, ^oF

t_1 - inlet temperature of fluid, ^oF

t_2 - outlet temperature of fluid, ^oF

t_s - steam temperature, ^oF

β - coefficient of thermal expansion, 1/^oF

ρ - fluid density, lb.ft.^{3}

L - length of tube, ft.

W - weight of fluid flowing per unit time, lb./hr.

q - total heat transferred, Btu.

Q - total heat transferred per unit time, Btu/hr.

θ - time, hrs.

R - thermal resistance, hr. x ft.^{2}x ^oF./Btu

A - heat transfer area, ft.^{2}

U - overall coefficient of heat transfer, Btu/hr. x ft.^{2}x ^oF

x - wall thickness and fin length, ft.

a - dimensional factor, 1/ft.

b - perimeter of heat transfer surface, ft.

S - cross sectional area of heat transfer surface, ft.^{2}

Subscripts:

a, av., arithmetic mean

m, lm, logarithmic mean

w, value at the wall or wall temperature

f, value at the liquid film or film temperature

s, value at the steam temperature; value for steel

0, value at the fin base

sa, value from shell to atmosphere

e, equivalent

1, inlet value

2, exit value

Nomenclature is that approved by the American Institute of Chemical Engineers.

SAMPLE CALCULATIONS

A. OIL ON THE TUBE SIDE

1. Calculation of G, mass velocity of the oil, lb./hr.xft.^{2}

   outside diameter of tube = 1.0 in.
   (Fig. 4)
   Tube wall is of #13 BWG gage steel

   #13 gage = 0.095 in. Perry (12)

   Inside diameter of tube

   $$D = 1.0 - 2(0.095)$$
   $$= 0.810 \text{ in.}$$
   $$= 0.0675 \text{ ft.}$$

   Internal cross sectional area of tube

   $$= \pi D^2/4$$
   $$= (3.14)(0.0675)^{2/4}$$
   $$= 0.003418 \text{ ft.}^2$$
   $$G = W/0.003418 \text{ lb./hr. x ft.}^2$$

2. The calculations of temperatures and temperature differences are self-explanatory in Table IIA.

3. Evaluation of the physical properties of the oil

   a. Specific heat, c, Btu/lb.x $^\circ$F., evaluated at the bulk oil temperature t_a from Fig. 7.

   b. Thermal conductivity, k, Btu/hr. x ft.^{2}x $^\circ$F/ft. evaluated at the bulk oil temperature t_a from Fig. 8.

   c. Oil viscosity at the bulk oil temperature, μ_a, evaluated at the bulk oil temperature, t_a, from Fig. 6.

Oil viscosity at the tube wall temperature, μ_s, evaluated at the tube wall temperature, t_s, from Fig. 6.

4. The value of the Reynolds number, DG/μ_a, was calculated using the internal tube diameter D = 0.0675 ft.

5. The total heat transferred to the oil, Q, Btu/hr., was calculated from the heat balance on the oil:

$$Q = Wc(t_2 - t_1)$$

6. Values of U, the overall heat transfer coefficient, Btu/hr. x ft.^{2} x ^{o}F. were obtained on the basis of both arithmetic and logarithmic mean temperature differences, using the equation for heat transfer:

$$Q = UA\,\Delta t$$

The heated length of the tube, L = 6.458 ft. (Fig. 4)

Internal tube wall area

$$A = 0.0675 \times 3.14 \times 6.458$$
$$= 1.370 \text{ ft.}^2$$

Then:

$$U = Q/A\,\Delta t$$
$$= Q/1.370\,\Delta t$$

7. The values of h, the oil film heat transfer coefficient, Btu/hr. x ft.^{2} x ^{o}F., were calculated on the basis of U_a and U_{lm} by the resistance equation:

$$U = \frac{1}{\frac{1}{h} + \frac{1}{h_s} + \frac{T_w}{k_s}}$$

k_s = thermal conductivity of tube wall

= 26 Btu/hr. x ft.^{2} x^{o}F/ft. Perry (12)

T_w = tube wall thickness

= 0.095/12

= 0.00791 ft.

h_s = 2000 Btu/hr. x ft.^{2} x ^{o}F. (approximate value used for the steam film coefficient when the liquid film coefficient is relatively small.)

$1/U = 1/h + 1/h_s + T_w/k_s$

$= 1/h + 1/2000 + 0.00791/26$

$= 1/h + 0.0005 + 0.0003$

$1/h = 1/U - 0.0008$

8. The calculation of the dimensionless groups is self-explanatory in Table IIA.

B. OIL ON THE FIN SIDE

1. Calculation of G, mass velocity of the oil, lb./hr. x $ft.^2$

Outside diameter of tube = 1.0 in. = d_1 (Fig. 4)

Shell of exchanger is 2 in. std. pipe

Internal diameter of shell = 2.067 in. Perry (12)

$= d_2$

G will be calculated on the basis of the free cross sectional area between the shell and the tube. The free area of this annulus is reduced by the total cross sectional area of the fins welded to the outside of the tube.

The tube has 20 fins, each 0.5 in. high and 0.031 in. in thickness.

The fins are arranged in pairs, each pair being connected at the base by a strip of metal 0.031 in. thick, spot-welded to the tube.

The width of this strip is:

$$\pi d_1/20 - (2 \times 0.031/2) = (\pi)(1)/20 - 0.031$$
$$= 0.1572 - 0.031$$
$$= 0.1262 \text{ in.}$$

The cross sectional area of the fins

$$= 0.5 \times 0.031 \times 20$$
$$= 0.31 \text{ in.}^2$$

The cross sectional area of the connecting strips

$$= 0.1262 \times 0.031 \times 10$$

$$= 0.039 \text{ in.}^2$$

Total area to be subtracted from the annular space

$$= 0.31 + 0.039$$

$$= 0.349 \text{ in.}^2$$

Then, the free area of the annular space

$$= \frac{\pi d_2{}^{2/4} - \pi d_1{}^{2/4} - 0.349}{144}$$

$$= \frac{(3.14)(2.067)^{2/4} - (3.14)(1.0)^{2/4} - 0.349}{144}$$

$$= 0.01546 \text{ ft.}^2$$

$$G = W/0.01546 \text{ lb./hr.} \times \text{ft.}^2$$

2. The calculations of temperatures and temperature differences are self-explanatory in Table IIB.

3. Evaluation of the physical properties of the oil was made as under A (3).

4. Calculation of the equivalent diameter of the annulus. Since flow occurs in an annular and non-circular cross section an equivalent diameter, D_e, ft., must be calculated.

$$D_e = 4m$$

$$m = \frac{\text{area of stream cross section}}{\text{wetted perimeter}} \tag{11}$$

In the calculation of the hydraulic radius m, ft., the total wetted perimeter of the annulus, excluding the fins, was used.[1]

Wetted perimeter

$$= \pi(2.067) - (20)(0.031) + (1.0) - (20)(0.031)$$

$$= 8.40 \text{ in.}$$

$$= 0.70 \text{ ft.}$$

Total free cross sectional area of annulus

$$= 0.01546 \text{ ft.}^2$$

Then:

$$D_e = 4m$$

$$= (4)(0.01546)/0.70$$

$$= 0.0883 \text{ ft.}$$

This value of the equivalent diameter is used in the calculation of the Reynolds number and other dimensionless groups.

5. The total heat transferred to the oil, Q, Btu/hr. was calculated from the sum of Q' and Q", where Q' was obtained from a heat balance on the oil:

$$Q' = Wc(t_2 - t_1)$$

The average temperature of the air surrounding the heat exchanger was 80°F. The temperature difference from

[1]See Page (72)

the shell to the atmosphere was calculated as the difference

$$t_a - 80 = \Delta t_{sa}$$

The heat loss from the shell to the atmosphere, Q", Btu/hr., was calculated by the method of McAdams (4). Values of $h_c + h_r$, the film coefficients of conduction and radiation respectively, are given for various sizes of bare standard steel pipe, for various values of Δt_{sa}, the temperature difference between the heated pipe and a room at 80°F. By interpolation, the values of $h_c + h_r$ for 50°F. intervals of Δt_{sa} were obtained, and plotted vs. Δt_{sa} in Fig. 9.

The area of the shell exposed to the atmosphere and to the oil on the inner side:

$$= 3.14 \times 2.38/12 \times 6.458$$

$$= 4.03 \text{ ft.}^2$$

the outside diameter of 2 in. standard pipe being equal to 2.38 in. Perry (12)

Values of $h_c + h_r$ were read from Fig. 9 for the calculated value of Δt_{sa} for each run, and the heat loss obtained by the relation:

$$Q'' = 4.03 \times \Delta t_{sa} \times (h_c + h_r)$$

6. Values of U, the overall heat transfer coefficient, Btu/hr. x ft.^{2} x ^oF., were obtained as under A (6) using the area of the fins + bare tube

$$A = 9.90 + 1.73$$
$$= 11.63 \text{ ft.}^2 \qquad \text{(Fig. 4)}$$

$$U = Q/A\,\Delta t$$
$$= Q/11.63\,\Delta t$$

7. The values of h, the oil film heat transfer coefficient, Btu/hr. x ft.^{2} x ^oF., were calculated on the basis of U_a and U_{lm} by the resistance equation:

$$1/h = 1/U - A/h_sA_s - x_wA/k_sA_w$$

where A = heat transfer area on oil or fin side

= 11.63 ft.^{2}

A_s = heat transfer area on steam or tube side

= 1.370 ft.^{2}

A_w = arithmetic mean of the outer and inner tube wall surfaces.

= (1.370 + 1.730)/2

= 1.55 ft.^{2}

h_s = 2000 Btu/hr. x ft.^{2} x ^oF

x_w = equivalent wall thickness

= mean of tube wall thickness and one-half the fin length.

= 0.095 + 0.500/2

= 0.345 in.

= 0.02875 ft.

k_s = thermal conductivity of steel
= 26 Btu/hr.x ft.^{2} x ^oF./ft.

The equivalent wall thickness x_w was calculated assuming that the tube wall must transfer all of the heat either to the oil or to the base of the fins. The heat transferred by the fins must pass through the base of the fins; however, approximately one-half of the total heat transferred by the fins is transferred in the lower half of the fin, and the other half by the upper half. To approximately account for this condition the equivalent wall thickness was taken as the arithmetic mean of the mean tube wall thickness and one-half of the fin height. Then:

$$1/h = 1/U - 11.63/(2000)(1.370) - (0.02875)(11.63)/(26)(1.55)$$

$$= 1/U - 0.00425 - 0.00830$$

$$= 1/U - 0.0126$$

8. Correction of the values of the oil film heat transfer coefficient for the drop in temperature from the base of the fin to the tip.
The values of h may be corrected by dividing the uncorrected values by the ratio $(\Delta t)_m / (\Delta t)_0$, where:

$$(\Delta t)_m/(\Delta t)_0 = \tanh\ ax_f/ax_f \quad *$$

$$a = (hb/kS)^{0.5}$$

$$b = \frac{2(0.5) + 0.031}{12}$$

$$= 0.086 \text{ ft.}$$

$$k = 26 \text{ Btu/hr. x ft.}^2 \text{ x } {}^{o}\text{F./ft.}$$

$$S = (0.5 \times 0.031)/12$$

$$= 0.001292 \text{ ft.}^2$$

The following equation was derived for the calculation of the value of ax_f:

$$a = (hb/kS)^{0.5}$$

$$\log a = 0.5(\log h + 0.5 \log(b/kS)$$

$$= 0.5 \log h + 0.5 \log(b/kS)$$

$$= 0.5 \log h + 0.5 \log(0.086)/(26)(0.001292)$$

$$= 0.5 \log h + 0.5 \log 2.56$$

$$= 0.5 \log h + (0.5)(0.4082)$$

$$= 0.5 \log h + 0.2041$$

$$x_f = 0.5/12$$

$$= 0.04165 \text{ ft.}$$

$$\log x_f = \log 0.04165$$

$$= 8.6196 - 10$$

$$= -1.3804$$

* The source of this equation and the definitions of the terms used are covered in the Theoretical section, pp. (22-23).

$$\log ax_f = \log a + \log x_f$$
$$= 0.5 \log h + 0.2041 - 1.3804$$
$$= 0.5 \log h - 1.1763$$

Using the uncorrected value of h the value of ax_f was calculated from the above equation. From a table of hyperbolic functions (13) the value of tanh ax_f was obtained, and the ratio $(\Delta t)_m/(\Delta t)_0$ calculated from tanh ax_f/ax_f. The uncorrected value of h was then corrected by dividing by the ratio $(\Delta t)_m/(\Delta t)_0$. (4)

9. The calculation of the dimensionless groups is self-explanatory in Table IIB.

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