DESIGN AND PERFORMANCE
OF
CONDENSING HEAT EXCHANGERS
(WITH SPECIAL APPLICATION TO REFRIGERATION)

A DISSERTATION SUBMITTED
IN
PARTIAL FULFILMENT OF THE REQUIREMENTS
FOR
THE DEGREE OF MASTER OF ENGINEERING
IN
APPLIED THERMODYNAMICS (REFRIGERATION & AIR-CONDITIONING)

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CERTIFICATE

CERTIFIED that the dissertation entitled "Design and Performance of Condensing Heat Exchangers (with special application to refrigeration)" , which is being submitted by Sri G.Aravinda Ghosh, in partial fulfilment for the award of the Degree of Master of Engineering in Applied Thermodynamics (Refrigeration and Air-Conditioning) of University of Roorkee is a record of student's own work carried out by him under my supervision and guidance. The matter embodied in this dissertation has not been submitted for the award of any other Degree or Diploma.

This is further to certify that he has worked for a period of six and a half months from 15th October, 1962 to 30th April, 1963 for preparing dissertation for Master of Engineering Degree at the University.

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CONTENTS

S Y N O P S I S

In the refrigerating plant, the condensing heat exchanger plays an important role in its performance and the capacity. The present work is intended for the study, design, and performance of condensing heat exchangers. Forced convection is the most important mode of heat transfer in the condensing heat exchangers.

To start with, the fundamentals of forced convection and Nusselt's analysis for condensation of pure vapours are dealth. Then, the correlations of numerous investigators for forced convection in inside tubes and in annular spaces are indicated. The effects of turbulent flow, vapour velocity, etc., on condensation of vapours are also discussed. A brief discussion on boundary layer treatment for condensation is also given.

Later, theeffect of the arrangement of tubes in a multi-tube condenser on condensation of vapours is discussed, and the thermal analysis, viz. LMTD approach, is dealt with.

For the experimental set up, the design of a double-pipe condensing heat exchanger for a capacity of 3/4 ton is made. The procedure is then discussed and the data obtained during the experiment are tabulated.

The results of the experimental work are discussed in the last. It is observed that the experimental condensing film and water film coefficients are in good agreement with the predictions made on the theoretical basis.

NOMENCLATURE

A	Area of heat transfer surface, square foot; A_i for inside, A_o for outside, A_w for wall, A_L for coolant side.
B	A function defined by Eq. 1.31, $ft.^4$
C	A dimensionless constant Eq. C_1, C_2 in Eq. 1.17.
c	A dimensional constant, $\left[g C_p (\rho - \rho_v) / 4 \nu K \right]^{1/4}$.
C_p	Specific heat at constant pressure, BTU/(lb fluid)(deg.F).
D	Diameter of tube, foot.
D_e	Equivalent diameter, $= D_3 - D_2$ for plain tubes, foot.
e	Base of natural logarithms, 2.718.
F_v	Total force owing to vapour friction, (lb.m/sq.ft.)(ft./hr²).
F_{vo}	Value of F_v calculated using dry values of f and average value of G_v.
F(η)	A dimensionless dependent variable defined as $\frac{\psi}{x^{3/4}} \cdot \frac{C_p \rho}{K c}$
	F', F'', F''' are first, second and third differentials of F.
f	A dimensionless friction factor.
G	Mass velocity of fluid, lb./(hr)(sq.ft. of cross section G_v for vapour, G_m for mean value of G_v as defined in Eq. 2.15.

g	Acceleration due to gravity, 4.17 x 10^8 ft./(hr.)^2
H	Height of a tube, feet.
h	Local coefficient of heat transfer, BTU/(hr)(sq.ft.)(deg.F) h_L is based on LMTD, h_w for water, h_∞ is at large L/D where the local h is constant.
$I(\eta,-1)$	A function defined as $\int_0^1 exp\left[3\eta_\delta^4\left(\frac{1}{6}\xi^3-\frac{1}{24}\xi^4\right)\right]d\xi$
J	A function as defined in Eq. 2.35
j	Product of dimension less groups; $j=\frac{h}{C_pG}\left(\frac{C_p\mu}{k}\right)_f^{2/3}$.
k	Thermal conductivity of fluid. B.T.U./(hr.)(sq.ft.)(deg.F/ ft.)
k_f	Thermal conductivity of condensate at T_f, BTU/(hr.)(sq.ft.)(deg.F.per ft.)
k_w	Thermal conductivity of tube or wall, BTU/(hr.)(sq.ft.)(deg.F. per ft.)
L	Length of a straight tube, feet.
m	Mass rate of flow of fluid, lb./hr.
N	Number of rows of tubes in a vertical tier.
N_{NU}	Nusselt's number, hD/k, dimensionless, N_{NUx} for section x.
N_{PR}	Prandtl number, $C_p\mu/k$, dimensionless.
N_{RE}	Reynold number, D_eG/μ or $\rho Dv/\mu$, dimensionless For a horizontal tube $N_{REf}=2\Gamma'/\mu$; for a vertical tube $N_{REf}=4\Gamma'/\mu$.

p	Pressure, lb./sq.ft.
Q	Rate of heat transfer through condensing film, BTU per hr. (Eq. 2.32)
Q'	Rate of heat transfer through the bottom condensate (Eq. 2.33), BTU per hr.
q	Rate of heat transfer, BTU/hr.
q'	Rate of heat transfer per unit length, BTU/hr. ft.
R	Thermal resistance, deg.F-hr. sq.ft. per BTU; R_c for coolant, R_v for vapour side, R_w for tube wall, ΣR for total resistance ($= \frac{1}{U}$.)
r	Radius of tube, ft.
s	Shearing stress, lb./sq.ft.
T	Temperature, degrees Fahrenheit, T_b for bulk temperature; $T_f = T_{sv} - \frac{3}{4}\Delta T$ for condensate film; T_s for condensing surface; T_{sv} for saturation temperature of vapour. T_1 for average cooling water temperature for runs with Freon-12 (Sect. 3-5); T_2 for average cooling water temperature for runs with other material.
U	Overall coefficient of heat transfer, BTU/(hr.)(sq.ft.)(deg.F.); U_i is based on A_i and U_o on A_o.
V	Average velocity, feet per hr.
W	Weight, lb; W' is weight per unit length, lb./ft.
w	Mass rate of flow of condensate from lowest point on condensing surface, lb./hr.
x	Coordinate measuring distance along plate from leading edge, foot.

Y	Thickness of condensate film, foot.
y	Distance from a solid boundary measured in a direction normal to surface, foot.
Z	A dimensionless functions defined by Eq. 1.32.

GREEK.

β	$= \frac{0.9036}{\Phi} \int_0^{\Phi} \frac{d\Phi}{Z^{1/4}}$ (Eq. 2.25)
$\theta(\eta)$	A function defined as $\frac{T - T_{sv}}{T_u - T_{sv}}$ θ', θ'' are first and second differentials of θ with respect to η
ϕ	Angular deviation from normal direction, radians.
ϕ	Film angle, angle making half-arc of a horizontal tube covered by a condensing film, radians. (Sec. 2-7)
Φ	As defined by Eq. 2.26, $= \left[\frac{K_f^3 \rho_f(\rho_f - \rho_{2v}) g^2}{\mu_f^2}\right]^{1/3}$, BTU/(hr.)(ft^2)(deg.F)
Ψ	Prefix, indicating function, also a value as defined by Eqn. 1.32.
Ω	A property, as defined in Eq. 2.21, $= \left[\frac{K_f^3 \rho_f(\rho_f - \rho_{2v}) g^2 \lambda}{\mu_f}\right]^{1/4}$, BTU per (hr) (ft.$^{7/4}$) (deg.F$^{3/4}$)
η	Similarity variable, $cy/x^{1/4}$; $\eta_\delta = c\delta/x^{1/4}$
Γ	Mass rate of flow of condensate from lowest point on condensing surface, divided by the breadth, lb./(hr)(ft.) for a vertical tube, $\Gamma_v = W/\pi D$, for a horizontal tube Γ_H or $\Gamma' = W/L$
λ	Latent heat of condensation, BTU/lb.

λ' = $\lambda + 0.6\, C_p\, \Delta t$.

Δt Temperature difference, degrees Fahrenheit; Δt_l for logarithmic mean. For condensing pure vapours, Δt equals $T_{cv} - T_s$; Δt_o for overall temperature difference.

μ Absolute viscosity, lb./(hr.)(ft^2); μ_f for condensate film at T_f; μ_v for gas-vapour stream.

ρ Density of liquid, slugs/cft; ρ_f for condensate film at T_f; ρ_v for gas-vapour mixture.

γ Specific weight, lb./cft.

<u>SUBSCRIPTS.</u>

b for bulk.

c for cold fluid.

f for surface film.

H for horizontal

h for hot fluid.

i for inside.

L for liquid.

m for mean

o for outside.

s for surface.

V for vertical

v for vapour.

w for wall.

CHAPTER 1

INTRODUCTION.

ABSTRACT.

This chapter deals with the fundamentals of convection. The dimensional analysis is discussed for forced convection. The Nusselt's analysis for condensation of vapour on horizontal tubes and the dimensionless form of this analysis for both horizontal and vertical surfaces are treated in detail. Correlation and prediction of heat transfer coefficients and application to design will be discussed in subsequent chapters.

1.1. Convection.

The exchange of heat from a wall to a fluid or from a fluid to a wall is a very important process in engineering heat transfer. It is involved in almost all types of heat exchangers; it is involved in various ways in the operation and performance of engines and other types of mechanical equipment. The transfer of heat may occur in one or more of three essentially different ways viz., conduction, radiation, and convection.

The most important mode of heat transfer in a condensing heat exchanger is by convection. Convection is the process of transfer of heat within a gas or a liquid by moving masses of the fluid. The motion of the fluid may be caused either by differences in density within the fluid or by some mechanical device. The first process is called natural convection; and the second forced convection.

In the heat exchangers the transfer of heat takes place between fluids that are separated by a layer of solid material. Heat flows from the hotter fluid to the surface of the solid by the process of convection and/or conduction. The heat then flows through the solid material by conduction and is finally delivered from the surface to the cooler fluid by convection and/or conduction due to the formation of stagnant film at the surface. The transfer of heat in such cases is simplified, however, by the use of so-called convection coefficients which include the combined effect of both conduction and convection.

The present discussion will be restricted to forced convection only. This is justified because not only does the most of the energy transfer take place by physical displacement of fluids, but also the identifying features of the process are due to the flow caused by mechanical means.

1.2. The Heat Transfer Coefficient.

The concept of a heat transfer or so-called "film" coefficient was introduced by Newton in 1701. He recommended the following equation to describe the convective cooling of objects:

$$q = hA(T_s - T_\infty) \qquad \text{........ 1.1}$$

Although this is often referred to as Newton's Law of Cooling, it has proved to be really a definition of h rather than a law. It states that the heat transfer per unit

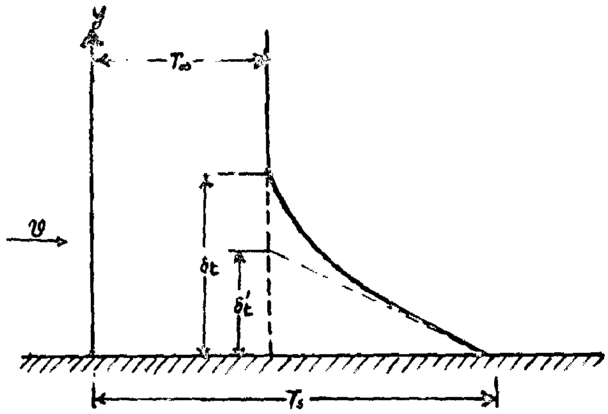


FIG. 1.1 TEMPERATURE DISTRIBUTION IN A TURBULENT BOUNDARY LAYER FOR A FLUID FLOWING OVER A HEATED PLATE

area and time from a surface to a fluid flowing along it is equal to the product of the temperature difference between the surface and the stream far from the surface and a quantity, h, which depends on the convective character of the flowing fluid, the thermal properties of the fluid, and the geometry of the system.

Although Eq. 1.1 is generally used to determine the rate of heat flow by convection between a surface and the fluid in contact with it, this relation is inadequate to explain the convective-heat-flow mechanism. A meaningful analysis which will eventually lead to a quantitative evaluation of the convective-heat-transfer coefficient must start with a study of the dynamics of fluid flow.

1.3 The Nusselt Modulus.

Considering the example of heat transfer from a solid wall to a fluid in a turbulent flow, we find that heat can only flow by conduction, in the immediate vicinity of the wall, because the fluid particles are stationary relative to the boundary. We naturally expect a large temperature drop in this layer. As we move further away from the wall, the movement of the fluid aids in the energy transport and the temperature gradient will be less steep, eventually, levelling out in the main stream. For air flowing turbulently over a flat plate, the temperature distribution shown in Fig. 1.1 illustrates these ideas qualitatively.

The foregoing discussion suggests a method for evaluating the rate of heat transfer between a solid wall and a fluid. Since at the interface (i.e. at $y = 0$) heat flows only by conduction, the rate of heat flow can be calculated from the equation

$$q = -k_f A \left.\frac{\partial T}{\partial y}\right|_{y=0} \qquad \ldots\ldots 1.2.$$

This approach has indeed been used, but for engineering purposes the concept of the convective-heat-transfer coefficient is much more convenient. In order not to lose sight of the physical picture, we shall relate the heat transfer coefficient defined by Equation 1.1 to the temperature gradient at the wall.

Equating Eqs. 1.1 and 1.2, we obtain

$$q = -k_f A \left.\frac{\partial T}{\partial y}\right|_{y=0} = h A (T_s - T_\infty) \qquad \ldots\ldots 1.3.$$

Since the magnitude of the temperature gradient in the fluid will be the same regardless of the reference temperature, we can write $\partial T = \partial(T - T_s)$. Introducing a significant length dimension of the system L to specify the geometry of the object from which heat flows, we can write Eq. 1.3 in dimensionless form as

$$\frac{hL}{k_f} = \frac{-\left.\frac{\partial T}{\partial y}\right|_{y=0}}{(T_s - T_\infty)/L} = \left.\frac{\partial\left(\frac{T_s - T}{T_s - T_\infty}\right)}{\partial\left(\frac{y}{L}\right)}\right|_{y=0} \qquad \ldots\ldots 1.4.$$

The combination of the convective-heat-transfer coefficient h, the significant length L, and the thermal conductivity of the fluid K_f in the form $\frac{hL}{K_f}$ is called the Nusselt number, N_{Nu}. The Nusselt number is a dimensionless quantity.

Inspection of Eq. 1.4 shows that the Nusselt number could be interpreted physically as the ratio of the temperature gradient in the fluid immediately in contact with the surface to a reference temperature gradient $(T_s - T_\infty)/L$. In practice the Nusselt number is a convenient measure of the convective-heat-transfer coefficient because, once its value is known, the convective-heat-transfer coefficient can be calculated from the relation

$$h = N_{Nu} \times \frac{K_f}{L} \qquad \ldots\ldots 1.5.$$

The temperature distribution for a fluid flowing past a hot wall, as sketched by the solid line in Fi. 1.1, shows that the temperature gradient in the fluid is confined to a relatively thin layer, δ_t, in the vicinity of the surface. We shall now simplify the true picture by replacing the actual temperature distribution by the dashed straight line shown in Fig. 1.1. The dashed straight line is tangent to the actual temperature curve at the wall and physically represents the temperature distribution in a hypothetical layer of fluid thickness δ_t' which, if completely stagnant, offers the same resistance to the flow of heat as the actual boundary layer. In this stagnant layer, heat can flow only by conduction and the rate of heat transfer per unit area is

$$\frac{q}{A} = k_f \frac{(T_s - T_\infty)}{\delta_t'} = h(T_s - T_\infty) \quad \text{.... 1.6}$$

An inspection of Eq. 1.6 shows that h may be expressed as

$$h = \frac{k_f}{\delta_t'} \quad \text{... 1.7}$$

and the Nusselt number as

$$N_{NU} = \frac{hL}{k_f} = \frac{L}{\delta_t'} \quad \text{.... 1.8}$$

While this picture is considerably oversimplified, it does illustrate the fact that the thinner the hypothetical boundary layer δ_t', the larger will be the convective conductance. To transfer large quantities of heat rapidly, one attempts to reduce the boundary layer thickness as much as possible. This can be accomplished by increasing the velocity and/or the turbulence of the fluid. If insulation of the surface is the desired aim, a thick stagnant layer is beneficial.

1.4 Evaluation of Convective-Heat-Transfer Coefficients.

There are four general methods available for the evaluation of convective-heat-transfer coefficients.

1. Dimensional analysis combined with experiments.
2. Exact mathematical solutions of the boundary-layer equations.
3. Approximate analysis of the boundary layer by integral methods.

4. The analogy between heat, mass, and momentum transfer

All four of these techniques have contributed to our understanding of convective heat transfer. Yet, no single method can solve all the problems because each one has limitations which restrict its scope of application. Only the first method is dealt with in the following section.

1.5 Dimensional Analysis.

Dimensional analysis is mathematically simple and has found the widest range of application. The chief limitation of this method is that results obtained by it are incomplete and quite useless without experimental data. It contributes little to our understanding of the transfer process, but facilitates the interpretation and extends the range of application of experimental data by correlating them in terms of dimensionless groups.

The most serious limitation of dimensional analysis is that it gives no information about the nature of the phenomenon. In fact, to apply dimensional analysis it is necessary to know beforehand what variables influence the phenomenon, and the success or failure of the method depends on the proper selection of these variables. It is therefore necessary to have atleast a preliminary theory or a thorough physical understanding of a phenomenon before a dimensional analysis can be performed. However, once the pertinent variables are known, dimensional analysis can be applied to most problems by a routine procedure which is outlined below.

PRIMARY DIMENSIONS AND DIMENSIONAL FORMULAE.

The first step is to select a system of primary dimensions. The choice of the primary dimensions is arbitrary, but the dimensional formulae of all pertinent variables must be expressible in terms of them. The primary dimensions of length L, time θ, temperature T, and mass M are used in this discussion.

The dimensional formula of a physical quantity follows from definitions or physical laws. The dimensional formulae and the symbols of physical quantities occurring in this discussion are given in table 1-1.

Table 1-1.

No.	Variable	Symbol	Dimensional Equation.
1.	Tube diameter	D	L
2.	Thermal conductivity of the fluid.	K_f	ML/θ^3T
3.	Velocity of the fluid.	v	L/θ
4.	Density of the fluid.	ρ	M/L^3
5.	Viscosity of the fluid.	μ	$M/L\theta$
6.	Specific heat at constant pressure.	C_p	L^2/θ^2T
7.	Heat-transfer-Coefficient.	h	M/θ^3T

BUCKINGHAM π -THEOREM.

To determine the number of independent dimensionless groups required to express relation describing a phenomenon, the Buckingham π- theorem may be used as a rule of thumb. According to this rule, the required number of independent dimensionless groups which can be formed by combining the physical variables of a problem is equal to the total number of these physical quantities n (for example, density, viscosity, heat-transfer coefficient, etc.) minus the number of primary dimensions m required to express the dimensional formulae of the n physical quantities. If we call these groups π_1, π_2, etc., the equation expressing the relationship among the variables has a solution of the form

$$F(\pi_1, \pi_2, \pi_3 \cdots\cdots) = 0 \qquad \ldots\ldots 1.0$$

For a phenomenon which can be described in terms of three dimensionless groups (i.e. if $n-m = 3$), Eq. 1.0 has the form

$$F(\pi_1, \pi_2, \pi_3) = 0 \qquad \ldots\ldots 1.20$$

but can also be written as

$$\pi_1 = f(\pi_2, \pi_3)$$

For such a case, experimental data can be correlated by plotting π_1, against π_2 for various values of π_3. Sometimes it is possible to combine two of the π's in some manner and to plot this parameter against the remaining π on a single curve.

DETERMINATION OF DIMENSIONLESS GROUPS (FORCED CONVECTION)

A simple method will now be illustrated for determining dimensionless groups by applying it to the problem of correlating experimental convective-heat-transfer data for a fluid in a turbulent flow through a heated tube.

From the description of the convective-heat-transfer process, it is reasonable to expect that the physical quantities listed in Table 1.1 are pertinent to the problem. There are seven physical quantities and four primary dimensions. We, therefore, expect that three dimensionless groups will be required to correlate the data. To find these dimensionless groups, we write h as a product of the variables, each raised to an unknown power,

$$h = C D^{a} K_f^{b} \rho^{d} \mu^{e} c_p^{f} v^{g} \qquad \text{.... 1.11}$$

and substitute the dimensional formulae.

$$[M/\theta^3 T] = C\,[L]^{a}[ML/\theta^3 T]^{b}[M/L^3]^{d}[M/L\theta]^{e}[L^2/\theta^2 T]^{f}[L/\theta]^{g} \qquad \text{.... 1.12}$$

Equation 1.12 can be true only if the exponent of each of the primary dimension on the right is equal to that on the left. This results in a system of four equations involving six unknowns. So, solving these four equations in terms of the two unknowns, d and f, and substitution of these values in equation 1.11 results in

$$h = C\, D^{-1+d}\, K^{1-f}\, v^{d}\, \rho^{d}\, \mu^{f-d}\, C_p^{\,f} \qquad \ldots\ldots\ 1.13$$

Rearrangement of the above equation gives

$$\frac{hD}{K} = C\left(\frac{\rho D v}{\mu}\right)^{d}\left(\frac{C_p \mu}{K}\right)^{f} \qquad \ldots\ldots\ 1.13a$$

where $\frac{hD}{K}$ = Nusselt's Number, N_{NU}

$\frac{\rho D v}{\mu}$ = Reynold's Number, N_{RE}

$\frac{C_p \mu}{K}$ = Prandtl Number, N_{PR}

which are the three dimensionless groups.

Equation 1.13a may also be written in the following form

$$\frac{hD}{K} = \psi\left(\frac{\rho D v}{\mu}, \frac{C_p \mu}{K}\right) \text{ or} \qquad \ldots\ldots\ 1.13b$$

$$\frac{h}{C_p \rho v} = \psi_1\left(\frac{\rho D v}{\mu}, \frac{C_p \mu}{K}\right) \qquad \ldots\ldots\ 1.13c$$

where $\frac{h}{C_p \rho v}$ = Stanton number = $\frac{N_{NU}}{N_{RE} \times N_{PR}}$

which is also a dimensionless group.

The values of c, a, and f are to be determined experimentally after which Eq. 1.19a can be used to determine the heat transfer coefficient in turbulent flow. The correlation of experimental data with the above dimensional analysis is dealt with in detail in the next chapter.

1.6. Condensation.

So far, the discussion on convection has assumed that the chemical composition and phase of the fluid did not change. As there is a change of phase in the present work on the heat exchangers, a brief discussion on the phenomenon associated with change of phase and the procedures used in determining energy transfers involved is made in the following sections, while correlation and the effects of turbulent flow are reserved for the subsequent chapters.

When either a saturated or super heated vapour comes in contact with a surface maintained at a temperature lower than the saturation temperature, heat flow results from the vapour releasing its latent heat and condensing on the surface. The condensation process appears to proceed in two, more or less, distinct ways. If there are no impurities in the vapour or on the surface, the condensate will form a continuous liquid film, which is called filmwise condensation. If, however, such contaminants as fatty acids are present, the vapour will condense in small droplets. These increase in size until their weight causes them to run down the surface, which is known as dropwise condensation. For the same temperature difference heat transfer with dropwise condensation may be fifteen to twenty times greater than with filmwise condensation. Although the dropwise type of condensation is very desirable, the conditions under which it will occur are not completely understood.

1.7. *Some particular features of Film and Drop Condensation.*

Film condensation has been theoretically analysed by Nusselt in 1916. If a vapour is condensing in a film on a vertical cooling surface, the film flows down, under the influence of gravity, but is retarded by shearing forces because of the viscosity of the fluid. The latent heat freed by condensation passes through the film from the condensing vapour to the wall. The film resists this heat flow because of the low conductivity of liquids, and, therefore, an appreciable temperature drop across the film occurs. Where the film is in contact with the cooling surface its temperature is equal to the surface temperature. Nusselt further assumed that the interface between film and vapour is at the saturation temperature. If this were exactly true no condensation could occur. Obviously, the temperature of the liquid surface in contact with condensing vapour must be slightly below saturation temperature and will decrease with decreasing wall temperature.

In any case, the condensation depends upon the thickness of the liquid film and the conductivity of the liquid and therefore will be comparatively small.

In dropwise condensation, on the other hand, the cooling surface is in direct contact with the vapour. Thus, the effective temperature difference is much larger, and formation of droplets is much more rapid than the corresponding formation of liquid on the film surface. In this type, the liquid is carried away more quickly than in the case of film condensation, as the drops roll down and take other droplets with them.

Not withstanding all the advantages of dropwise condensation, film condensation is of paramount importance, being established more easily and remaining much more stable in operation. With this little background, Nusselt's analysis on film-condensation will now be presented.

1.8. The Flow of a Liquid Film on A Solid Surface.

Nusselt[27] (1916) in his theory of film condensation has combined the laws of laminar flow of a liquid film and of heat conduction through it. First the equations for the flow of the film had to be developed. A liquid film is supposed to form and flow down a surface which may be inclined by the angle against a horizontal plane. Let v be the downward velocity parallel to the surface at a distance y from the surface. Then a shear stress occurs in the liquid which can be expressed by the well known equation.

$$S = \mu \frac{dv}{dy} \qquad \ldots\ldots \quad 1.16$$

where μ is the dynamic viscosity of the liquid. If, further, γ denotes the specific weight of the liquid, the equation of mechanical equilibrium will be

$$dS = -\gamma \sin\varphi \, dy \quad \text{OR} \quad \frac{dS}{dy} = -\gamma \sin\varphi \qquad \text{.... 1.25}$$

From Eqs. 1.24 and 1.25

$$\frac{d^2 v}{dy^2} = -\frac{\gamma}{\mu} \sin\varphi \qquad \text{.... 1.26}$$

and by integration

$$v = -\frac{\gamma \sin\varphi}{2\mu} y^2 + C_1 y + C_2 \qquad \text{.... 1.27}$$

Since the liquid adheres to the wall,

$$v = 0 \quad at \quad y = 0$$

and therefore $C_2 = 0$.

Herewith

$$v = \frac{-\gamma \sin\varphi}{2\mu} y^2 + C_1 y \qquad \text{.... 1.28}$$

The value of C_1 depends upon the flow of the condensing vapour, for, if the vapour streams along the film with the velocity v_v it exerts a shearing force upon the liquid.

For vapour which is virtually at rest no sensible friction occurs at its interface with the film. Let $y = Y$, the thickness of the film at any place; then, according to Eq. 1.24,

$$\left(\frac{dv}{dy}\right)_{y=Y} = 0 \qquad \cdots\cdots \quad 1\text{-}19$$

Herewith from Eq. 1.18,

$$C_1 = \frac{\gamma \sin\varphi}{\mu} Y \qquad \cdots\cdots \quad 1.20$$

so that for vapour at rest

$$v = -\frac{\gamma \sin\varphi}{2\mu}\left(y^2 - 2Yy\right) \qquad \cdots\cdots \quad 1.21$$

If the condensing vapour of density ρ_v flows along the surface with the velocity v_v, for instance, inside a pipe of width D, then the shearing stress occuring on the film surface can be calculated from the equation of pressure drop in tubes, which may be written as

$$\frac{dP}{dx} = -\frac{4f}{D}\,\frac{\rho_v v_v^2}{2} \qquad \cdots\cdots \quad 1.22$$

where f is a function of Reynolds number, and the coordinate x is in the direction of main flow of the condensate film.

On the other hand, equilibrium on the inner surface of the tube requires

$$S_{y=Y}\,\pi D = -\frac{\pi D^2}{4}\,\frac{dP}{dx} \qquad \cdots\cdots \quad 1.23$$

From Eqs. 1.22 and 23

$$S_{y=Y} = f\,\frac{\rho_v v_v^2}{2} \qquad \cdots\cdots \quad 1.24$$

Differentiating Eqn. 1.18 and substituting $y = Y$, using also Eqn. 1.14 and 1.24

$$S_{y=Y} = \pm \frac{f \rho_v v_v^2}{2} = \mu \left(\frac{dv}{dy}\right)_{y=Y} = -\gamma Y \sin\varphi + C_1 \mu$$

$$\text{OR} \quad C_1 = \pm \frac{f}{\mu} \frac{\rho_v v_v^2}{2} + \frac{\gamma}{\mu} Y \sin\varphi \qquad \ldots\ldots 1.25$$

The positive sign on the right holds for vapour flow downward, the negative sign for flow upward.

From Eqns. 1.23 and 1.25,

$$v = \left(\pm \frac{f}{\mu} \frac{\rho_v v_v^2}{2} + \frac{\gamma}{\mu} Y \sin\varphi\right) y - \frac{\gamma \sin\varphi}{2\mu} y^2 \qquad \ldots\ldots 1.26$$

By integration one obtains the mean velocity of the condensate film.

$$v_m = \frac{1}{Y} \int_0^Y v\, dy = \frac{\gamma \sin\varphi}{3\mu} Y^2 \pm \frac{f}{2\mu} \frac{\rho_v v_v^2}{2} Y \qquad \ldots\ldots 1.27$$

1.9. <u>Theory of Film Condensation of Vapour at Rest on Cylindric Surfaces.</u>

For the condensation of vapour at rest on the outside of a horizontal tube cooled from the inside, Nusselt performed the following derivation.

Let $D = 2r$ be the outer diameter, and in a cross section per-pendicular to the tube axis, φ the angle which any radius r forms with a vertical line. Then, according to Eq. 1.27, the mean velocity of the film at the end of the radius will be

$$v_m = \frac{\gamma}{3\mu} Y^2 \sin\varphi \qquad \ldots\ldots 1.28$$

In a tube section of unit length and arc element $r\,d\varphi$ the following amount of heat will be exchanged in unit time :-

$$d\left(\frac{q'}{2}\right) = \frac{k\,r\,(T_f - T_s)\,d\varphi}{y} = \gamma\lambda \cdot d(v_m y) \qquad \text{.... 1.29}$$

the last term being the amount of latent heat liberated by condensation.

From Eqns. 1.23 and 1.29

$$d\left(\frac{q'}{2}\right) = \frac{\gamma^2 \lambda}{3\mu}\, d\left(y^3 \sin\varphi\right) \qquad \text{.... 1.30}$$

From Eqs. 1.29 and 1.30

$$B\,d\varphi = y \cdot d\left(y^3 \sin\varphi\right) = \frac{3}{4}\sin\varphi\, dy^4 + y^4 \cos\varphi\, d\varphi \qquad \text{.... 1.31}$$

where

$$B = \frac{3\mu\, k\, r\,(T_f - T_s)}{\gamma^2 \lambda}$$

Introducing

$$z = \psi^4 = \frac{y^4}{B} \qquad \text{.... 1.32}$$

it follows that

$$\frac{3}{4}\sin\varphi\,\frac{dz}{d\varphi} + z\cos\varphi - 1 = 0 \qquad \text{.... 1.33}$$

The solution of this linear differential equation is

$$z = \frac{1}{\sin^{4/3}\varphi}\left(\frac{4}{3}\int \sin^{1/3}\varphi\, d\varphi + C\right) \qquad \text{.... 1.34}$$

If no liquid is dropping on the top of the tube, then for $\varphi = 0$ the film thickness y and therefore z will be finite. This requires $C = 0$

and therefore

$$z = \frac{4}{3 \sin^{4/3}\varphi} \int_0^{\varphi} \sqrt[3]{\sin\varphi}\; d\varphi \qquad \ldots 1.35$$

From Eqn. 1.32 and substituting the value of B in that,

$$y = \psi \sqrt[4]{B} = \psi \sqrt[4]{\frac{3 \mu K r (T_f - T_s)}{\gamma^2 \lambda}} \qquad \ldots 1.36$$

The local coefficient of heat transfer at the angle φ becomes

$$h = \frac{K}{y} = \frac{1}{\psi} \sqrt[4]{\frac{\gamma^2 \lambda K^3}{3 \mu r (T_f - T_s)}} = \frac{K}{\psi \sqrt[4]{B}} \qquad \ldots 1.37$$

By integrating over the range from φ_1 to φ_2 the following mean coefficient is obtained.

$$\Big[h_m\Big]_{\varphi_1}^{\varphi_2} = \frac{K}{\sqrt[4]{B}\,(\varphi_2 - \varphi_1)} \int_{\varphi_1}^{\varphi_2} \frac{d\varphi}{\psi} \qquad \ldots 1.38$$

Graphical integration of this expression showed that, under the assumed conditions, 59.4% of the total condensate was produced in the upper half of the tube and 40.6% in the lower one.

It may be mentioned that the equations derived above are also valid for outside cooling of tube and inside condensing of a vapour if r is used for the inside radius.

A mean coefficient of heat transfer h_m for film condensation of vapour at rest on a vertical surface is found to be

$$h_{m_{vert}} = \frac{4}{3}\sqrt[4]{\frac{\gamma^2 \lambda k^3}{4\mu H(T_f - T_s)}} = 0.943\sqrt[4]{\frac{\gamma^2 \lambda k^3}{\mu H(T_f - T_s)}} \quad \ldots 1.39$$

Comparison of Eqs. 1.30 and 1.38, taking $\varphi_2 - \varphi_1 = 180°$, reveals that the following relation holds between the mean film coefficient $h_{hor.}$ for a horizontal tube of diameter D and $h_{vert.}$ for a vertical tube or other surface of height H.

$$\frac{h_{hor.}}{h_{vert.}} = 0.770\sqrt[4]{\frac{H}{D}} \quad \ldots 1.40$$

Substituting the value of h_m from 1.39 for $h_{vert.}$ in Eq. 1.40,

$$h_{hor.} = 0.725\sqrt[4]{\frac{\gamma^2 \lambda k^3}{\mu D(T_f - T_s)}} \quad \ldots 1.41$$

If several horizontal tubes (say N) are located one above the other, then the condensate formed on any upper tube drops upon the next lower one, reducing the heat exchange on that tube.

The mean coefficient of heat transfer for such an arrangement will be found to be

$$h_m = 0.725\sqrt[4]{\frac{\gamma^2 \lambda k^3}{N\mu D(T_f - T_s)}} \quad \ldots 1.42$$

1.10. Dimensionless Relations for Film Condensation of Vapour at Rest.

The equations for h_m can be expressed in dimensionless form, as has been demonstrated by Kirkbridge (1933). Considering a horizontal surface, the local coefficient of heat transfer, h, may be expressed in different ways.

$$h = \frac{K}{Y} = \frac{dq'}{2(T_f - T_s)\, r\, d\varphi} = \frac{\lambda \cdot dW'}{2(T_f - T_s)\, r\, d\varphi} \qquad \ldots\ 1.43$$

where $W'/2$ and $q'/2$ are the weight flux and heat flux, respectively, for any point of the surface.

A mean coefficient of heat transfer h_m may be defined by

$$q' = h_m \times 2\pi r (T_f - T_s) = \lambda W_D'$$

or

$$h_m = \frac{\lambda W_D'}{2\pi r (T_f - T_s)} \qquad \ldots\ 1.44$$

where the subscript D refers to the point $\varphi = \pi$

From Eqs. 1.43 and 1.44

$$Y = \frac{K\, W_D'\, r\, d\varphi}{h_m (\pi r)\, dW'} \qquad \ldots\ 1.45$$

and from Eq. 1.27

$$Y = \sqrt[3]{\frac{3 \mu W'}{2 \gamma^2 \sin\varphi}} \qquad \ldots\ 1.46$$

From Eqs. 1.45 and 1.46

$$K \cdot \sqrt[3]{\frac{\gamma^2 \sin\varphi}{3\mu}} \cdot \frac{r\, d\varphi}{\pi r} = h_m \sqrt[3]{\frac{W_D'}{2}} \qquad \ldots\ 1.47$$

Integrating with in limits $\varphi = 0$ and $\varphi = \pi$,

$$\frac{K}{\pi} \sqrt[3]{\frac{\gamma^2}{3\mu}} \cdot \frac{4}{3} \int_0^{\pi} \sqrt[3]{\sin\varphi} \cdot d\varphi = h_m \sqrt[3]{\frac{W_D'}{2}} \qquad \ldots\ 1.48$$

By graphical integration

$$\frac{4}{3} \int_0^{\pi} \sqrt[3]{\sin\varphi} \cdot d\varphi = 3.480$$

0.925

Hence

$$h_m = 0.952\, K \sqrt[3]{\frac{\gamma^2}{\mu W_D'}} \qquad \ldots\ 1.49$$

Introducing

$$(N_{RE})_D = (v_m)_D \frac{4 y_0}{\nu}^{**} = \frac{4(W_0'/2)}{g\mu} = \frac{2\Gamma'}{\mu} \quad \ldots\ldots 1.50$$

which holds for $\varphi = \pi^{\neq\neq}$

From Eq. 1.49, after some rearrangement, we get

$$\left(\frac{g\mu^2 h_m^3}{\gamma^2 K^3}\right)^{1/3} = 1.2\left(\frac{2W_0'}{g\mu}\right)^{-\frac{1}{3}} = 1.2\,(N_{RE})_D^{-\frac{1}{3}}$$

$$= 1.51\left(\frac{4\Gamma'}{\mu}\right)^{-\frac{1}{3}} \quad \ldots\ldots 1.51$$

A similar expression as given below, may be derived for condensation of vapour on a vertical surface of height H. From Eq. 1.39 and after some rearrangement.

$$\left(\frac{g\mu^2 h_m^3}{\gamma^2 K^3}\right)^{1/3} = 1.47\left(\frac{4W_H'}{g\mu}\right)^{-\frac{1}{3}} = 1.47\,(N_{RE})_H^{-\frac{1}{3}}$$

$$= 1.47\left(\frac{4\Gamma}{\mu}\right)^{-\frac{1}{3}} \quad \ldots\ldots 1.52$$

where subscript H refers to the point which is at a height H.

$\neq\neq$ A second stream of liquid of the same strength comes down to $\theta = -\pi$

** $D_H = 4A/C$, where A is the area of a cross section through the film and C the part of its circumference on which friction is acting.

CHAPTER 2

THEORY OF FORCED CONVECTION IN TUBES AND FILM CONDENSATION OF VAPOURS

ABSTRACT.

The effects of temperature differences and L/D ratio on the heat transfer coefficient for turbulent flow in plain tubes and in annular spaces are considered along with the correlation of experimental data. The modifications of Nusselt analysis and condensation of vapours with laminar and turbulent flow are discussed. A boundary layer treatment of laminar condensation is introduced.

2.1. Heat Transfer Coefficients for Turbulent Flow in a Tube.

As shown in Chapter I, for turbulent flow in tubes, dimensional analysis gives

$$\frac{hD}{K} = \psi\left(\frac{\rho D v}{\mu}, \frac{C_p \mu}{K}\right) \text{ OR } \frac{h}{C_p \rho v} = \psi_1\left(\frac{\rho D v}{\mu}, \frac{C_p \mu}{K}\right)$$

where the functions ψ and ψ_1 are to be determined experimentally. If the tube length is important, as with short tubes, the ratio L/D must be included. For a given fluid whose physical properties are functions of temperature one could include a function of a ratio of two characteristic temperatures, such as surface and bulk stream.

Efforts to improve the correlation of data have resulted in slightly differing values for the constant and exponents, the use of one set of data for heating and another for cooling, and various procedures for evaluating the physical properties involved.

Moderate Δt

A number of observers report data for heating or cooling various fluids with Prandtl numbers ranging from 0.7 to 120 for Reynolds numbers from 10,000 to 120,000 and for L/D of 60 or more. These data for moderate Δt (i.e $T_b - T_w$) have been correlated by three types of equations. One evaluates all physical properties at the bulk temperature.

$$\frac{h_L D}{K_b} = 0.023\left(\frac{\rho D v}{\mu_b}\right)^{0.8}\left(\frac{C_p \mu}{K}\right)_b^{0.4} \qquad \ldots.\ 2.1$$

For viscous liquids where $d\mu/\mu\,dT$ is large there is some difference in opinion as to the temperature at which the viscosity should be evaluated.

Colburn evaluates all physical properties except C_p in the Stanton number, at the film temperature of the fluid T_f, being equal to $\frac{1}{2}(T_w + T_b)$.

$$\frac{h_L}{C_{p_b}\rho v}\left(\frac{C_p \mu}{K}\right)_f^{2/3} = \frac{0.023}{\left(\rho D v/\mu_f\right)^{0.2}} \qquad \ldots.\ 2.2$$

A third evaluates all physical properties at the bulk temperature, except μ_w in a viscosity ratio term.

$$\frac{h_L}{C_{p_b}\rho v}\left(\frac{C_p \mu}{K}\right)_b^{2/3}\left(\frac{\mu_w}{\mu_b}\right)^{0.14} = \frac{0.023}{\left(\rho D v/\mu_b\right)^{0.2}} \qquad \ldots.\ 2.3$$

Colburn defined a heat transfer factor j as

$$j = \frac{h}{C_p \rho v}\left(\frac{\mu C_p}{K}\right)^{2/3} = N_{St}\, N_{PR}^{2/3} \qquad \ldots.\ 2.4$$

He found for both laminar and turbulent flow in tubes and along flat plates that j was equal to one-half the friction coefficient. In the case of turbulent flow in pipe for Reynolds numbers from 5000 to 200,000 the friction coefficient (f) for smooth tubes is given by the relation

$$f = 0.046\left(\frac{\rho D v}{\mu_f}\right)^{-0.2} \qquad \ldots\ 2.5$$

Thus from Eqs. 2.2, 2.3 and 2.4

$$j = \frac{f}{2} \qquad \ldots\ 2.6$$

The Eq. 2.6 can be conveniently made use of in finding out the value of coefficient of heat transfer.

In addition to providing a direct comparison of heat transfer and friction, Colburn pointed out that the form of Eq. 2.2 has two other advantages over that of Eq. 2.1. First, data presented in terms of equation 2.1 were usually plotted as $\frac{hD}{k}\Big/\left(\frac{C_p\mu}{k}\right)^{1/3}$ versus $\left(\frac{\rho D v}{\mu}\right)\cdot\left(\frac{hD}{k}\right)\Big/\left(\frac{C_p\mu}{k}\right)^{1/3}$ will be found to be $j\left(\frac{\rho D v}{\mu}\right)$. Consequently the earlier procedure was equivalent to plotting $j\left(\frac{\rho D v}{\mu}\right)$ against $\left(\frac{\rho D v}{\mu}\right)$ which not only involved plotting a function against itself but required a much greater scale range for the ordinate. Secondly, it may be observed that $h(\pi D L)\Delta t_m = \frac{\pi D^2}{4}\rho v C_p (T_2 - T_1)$ so that the Stanton number

$$\frac{h}{C_p \rho v} = \frac{D}{4L}\,\frac{(T_2 - T_1)}{\Delta t_m} \qquad \ldots\ 2.7$$

may be evaluated directly from experimental data without determination of any physical properties.

The available data are correlated with in a maximum deviation of ± 40 percent by the three equations viz, 2.1, 2.2 and 2.3. For extrapolating results for liquids to high temperature difference between the surface and fluid, Eq. 2.3 contains the bulk Reynolds number and the ratio of the viscosities at the wall and bulk temperatures; Eq. 2.2 involves μ_f and $(c_p\mu/k)_f$ based on the arithmetic average of wall and bulk temperatures. The available data are inadequate to show whether or not one of these two equations, 2.2 and 2.3, is preferable to the other, and use of either is recommended by McAdams.[28]

Effect of Length-Diameter ratio.

Kroll finds that data of a number of observers for turbulent flow of gases and liquids for L/D from 1 to 68 indicate that

$$\frac{h_m}{h_\infty} = 1 + \left(\frac{D}{L}\right)^{0.7} \qquad \text{.... 2.8}$$

where the mean value of h from x of 0 to x of L, based on Δt_L, called h_m, is related to h_∞ at large L; when L/D reaches 60, the ratio h_m/h_∞ has fallen to 1.067 and decreases but little as L/D is further increased.

For turbulent flow at bulk Reynolds numbers, $\rho D V/\mu$, above 10,000, in tubes with square-edged entries, for Prandtl numbers from 0.7 to 120, the recommendation of McAdams[28] is to employ a combination of Eq. 2.8 and Eq. 2.3 or Eq. 2.2.

$$\frac{h_L}{c_p \rho V}\left(\frac{c_p\mu}{k}\right)_b^{2/3}\left(\frac{\mu_w}{\mu_b}\right)^{0.14} = \frac{0.023\left[1+(D/L)^{0.7}\right]}{(\rho D V/\mu_b)^{0.2}} \qquad \text{.... 2.9}$$

or

$$\frac{h_L}{c_p \rho V}\left(\frac{c_p\mu}{k}\right)_f^{2/3} = \frac{0.023\left[1+\left(\frac{D}{L}\right)^{0.7}\right]}{(\rho D V/\mu_f)^{0.2}} \qquad \text{.... 2.10}$$

2.8 Heat Transfer Coefficient for Turbulent Flow in Annuli.

Heat transfer to liquids flowing in annular spaces is frequently encountered in practice, particularly in double-pipe heat exchangers, jacketed pipes and similar equipment. Although a number of equations relating individual film coefficients to fluid properties and apparatus dimensions have been presented in the past many years, these, for the most part, have been based on experimental data covering limited ranges of test conditions. The expressions have taken many forms, primarily in that different functions of equivalent diameter or hydraulic diameter (as a shape factor) and of the diameter ratio, D_2/D_1 have been employed by various investigators.

The following are the correlations made by some investigators as quoted by McAdams.[28]

i) The value of heat transfer coefficient for turbulent flow in annuli at outer wall of diameter D_2 is usually correlated in terms of the dimensionless equation.

$$\frac{h_L}{C_p \rho v}\left(\frac{C_p \mu}{K}\right)_b^{2/3} = \frac{0.023\,\psi}{(D_e \rho v/\mu_b)^{0.2}} \qquad \ldots\ 2.12$$

or the equivalent.

ii) Monrad and Pelton's data for commercial sized annuli, D_2/D_1 of 1.86, with water flowing found

$$\psi = 1.17\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \qquad \ldots\ 2.12a$$

iii) Using data for air and water of several observers Davis[10] found

$$\psi = \left(\frac{D_2 - D_1}{D_2}\right)^m \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad \ldots\ldots 2.12b$$

where m lies between 0.05 and 0.16 approximately.

iv) Based on equations from various sources, for D2/D1, from 1 to 10, Wiegand proposed $\psi = 1.0$. For D_2/D_1 from 1 to 10, the relation recommended by McAdams[20] for the outer wall is

$$j' = \frac{h_L}{C_p \rho V}\left(\frac{C_p \mu}{k}\right)_b^{2/3}\left(\frac{\mu_w}{\mu_b}\right)^{0.14} = \frac{0.023}{\left(D_e \rho V/\mu_b\right)^{0.2}} \qquad \ldots\ldots 2.12c$$

v) For h at the inner wall of diameter D_1, Eq. 2.11 is again employed. For D2/D1 of 1.65, 2.45, and 17, Monrad and Pelton found

$$\psi = 0.87\left(\frac{D_2}{D_1}\right)^{0.53} \qquad \ldots\ldots 2.12d$$

vi) For various data, Davis[10] found

$$\psi = 1.35\left(\frac{D_2}{D_1}\right)^{0.15}\left(\frac{D_2}{D_1} - 1\right)^{0.2}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \qquad \ldots 2.12e$$

At $\frac{D_2}{D_1} = 1.2$, this ψ reduces to $\left(\mu_b/\mu_w\right)^{0.14}$, but for D_2 equal to D_1, an unreasonable result is predicted. For D_2/D_1 from 1 to 10, Wiegand proposed

$$\psi = \left(\frac{D_2}{D_1}\right)^{0.45} \qquad \ldots\ldots 2.12f$$

vii) Carpenter et[6]al made investigations to obtain heat transfer and friction data, which was used for finding out what equivalent diameter is to be used for annular spaces, for water flowing in the annulus of a vertical double-tube exchanger for D2/D1 of 1.333, over a wide range of Reynolds numbers.

The friction data indicated that the proper Reynolds number to employ is $D_e \rho v/\mu$, where D_e is the equivalent diameter of the annulus, D2 - D1. This data was compared for viscous and turbulent flow, with theoretical relations and with the corresponding flow inside commercial pipes, which was found to be a good agreement.

In the turbulent region, the heat transfer coefficient for annular spaces may be expressed by the relation

$$\frac{h_L}{C_p \rho v}\left[\frac{C_p \mu}{K}\right]_b^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = \frac{0.023}{\left(D_e \rho v/\mu_b\right)^{0.2}} \quad \ldots \; 2.12g$$

But the plot between the left side of Eq. 2.12g and Reynolds number has a slope nearer -0.1 than -0.2.

In the viscous region, the recommended equation for flow in the annular spaces is

$$\left(\frac{h_i D_e}{K}\right)\left(\frac{\mu_w}{\mu}\right)^{0.14} = 2.02\left[\frac{W C_p}{K H}\right]^{1/3} \quad \ldots \; 2.12h$$

viii) Prof. Mizushima[26] extended the concept of the analogy between heat transfer and fluid friction to apply it to the turbulent flow of air in annular space and obtained a semi-theoretical equation, which could correlate the data for air and was also in close agreement with Monrad and Pelton's equation, 2.12d, if applied for liquid. The heat transfer coefficient, h is given by

$$h = \frac{(f_0/2)\, C_p \rho \bar{v}}{N\left(\bar{v}_0/\bar{v} - v_b/\bar{v}\right) + N_{PR}\left(\frac{v_b}{\bar{v}}\right)} \qquad \ldots 2.12\ j$$

where

v = Local fluid velocity

$\bar{v}$ = Average fluid velocity

N = Turbulent core's resistance (max. temp. at r_m)

r = Radial distance from centre to a point in fluid stream.

Suffixes,

o = Inner tube wall or region inside radius of maximum velocity.

1 = Outer tube wall or region outside radius of maximum velocity.

b = Radius of boundary between laminar layer and turbulent core.

r_m = Radius of maximum velocity.

Finally, McAdams[28] recommends to use Eq. 2.12e for both the outer and inner walls of concentri-annuli.

2.3 The Effect of Vapour Velocity on Condensation Inside Tubes.

In the previous chapter, the fundamentals of condensation, and condensation of vapour at rest have been discussed. Since the present problem includes the flow of vapour inside the tube, a discussion on the effect of vapour velocity and turbulent flow on condensation inside vertical tubes will now be presented, followed by film condensation in horizontal tubes.

When the velocity of the uncondensed vapour is substantial compared with the velocity of condensate at the vapour -condensate interface, because of friction between vapour and the condensate film the vapour velocity influences the velocity and thickness of the condensate film and, consequently, the coefficient of heat transfer. Thus upward flow of vapour in a vertical tube tends to increase the thickness of the film, and with high vapour velocities condensate may be carried out at the top of the condenser. To avoid this, it may be necessary to use a number of short reflux condensers in parallel instead of one tall condenser of limited cross-section.

Carpenter and Colburn[3] conducted an experiment with downward flow of a condensing vapour at high velocity inside a tube, and measured the heat transfer coefficients which are as much as 10 times those predicted by Nusselt's equation (1.52) for film-type condensation neglecting the effect of vapour velocity. Even a comparison of the results with predictions using the Nusselt relation, which it includes the effect of vapour friction on the thickness of the viscous condensate layer, showed that the values were higher than the predictions, in some cases several-fold higher. The authors were, therefore, led to the hypothesis that in the presence of a high frictional force from the vapour on its outer surface, the condensate would become turbulent at much lower values of the Reynolds number than 2100. It was reasoned that when the major force acting on the condensate layer was vapour friction rather than gravity, the velocity distribution might follow that found for pipes filled with liquid, and the thickness of the laminar sublayer could be estimated by relations applying to this case.

They used the following relationship for vapour friction (F_v) on the condensate surface.

$$F_v = \frac{f G_v^2}{2 \rho_v} \qquad \ldots\ldots 2.13$$

(Note that F_v is related to the usual value of pressure drop as follows $F_v = \Delta P g_c D / 4L$). In the above equation the value of f is the friction factor for the vapour in the pipe containing condensate, and the value will be considerably higher than for a dry pipe. After conducting a series of tests, they made a simplified correlation of data for condensation of vapour flowing downward in a water-cooled tube at high velocity and obtained a dimensionless relation

$$\frac{h_c \mu}{K \rho^{1/2}} = 0.065 \left(\frac{C_p \mu}{K}\right)^{1/2} F_{vc}^{1/2} \qquad \ldots\ldots 2.14$$

The value of F_{vc} was calculated for the purposes of this correlation by utilising two simplifications. The first is that the friction factor is obtained from the usual graphs for one component flow (thus neglecting the presence of the condensate) and the second is that an "average" value of vapour velocity is employed. The proper average value to be used depends upon the condensing rates throughout the tube. If, for convenience, the assumption is made that the condensing rate is uniform and that, therefore, then the proper average value (i.e. the one which will give the same total friction) is derived to be

$$G_m = \left(\frac{G_1 + G_1 G_2 + G_2^2}{3} \right)^{1/2} \qquad \ldots\ldots 2.15$$

where G_1 is the value of G_v at the top of the tube, and G_2 is the value at the bottom of the tube. Note that when the vapour is totally condensed, $G_2 = 0$ and $G_m = 0.58 G_1$. Eq. 2.14 may be conveniently expressed in terms of average vapour velocity than in terms of friction as

$$h_c = 0.065 \left(\frac{C_p \rho K f}{2 \mu \rho_v} \right)^{1/2} G_m \qquad \ldots\ldots 2.16$$

It may be noted that in this equation, C_p, ρ, K, and μ are properties of the condensate, and f, ρ_v, and G_m refer to the vapour.

The vapours used in conducting the tests for the above correlation were steam, methanol, ethanol, toluene, and trichloroethylene.

2.4. Effect of Turbulence on Film Condensation.

With film condensation of vapour on a tall vertical tube, one can easily obtain condensation rates such that the Reynolds number $4\Gamma/\mu_f$ exceeds the critical value (1800) at which turbulence begins; this is not the case with horizontal tubes where the condensing height ($\pi D/2$) is inherently small. Because of turbulence in the layer of condensate on the lower part of the tube, the mean coefficient for the entire tube should lie above the line predicted from Eq. 1.62. Kirkbride[22] correlated the data for the condensation of diphenyl oxide and Dowtherm A on vertical tubes, giving the recommended

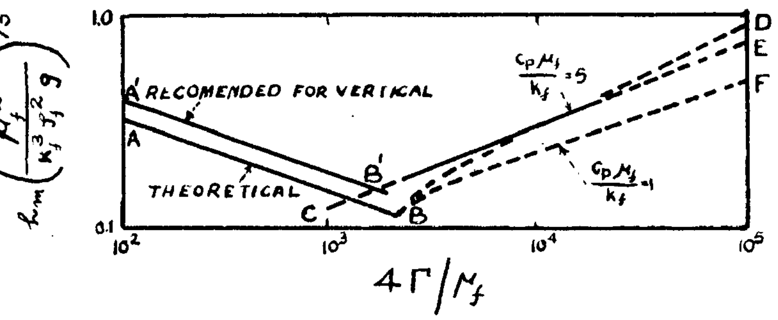


FIG. 2.1 McADAMS[25] RECOMENDED CURVES A'B' AND CE FOR FILM-TYPE CONDENSATION OF SINGLE VAPOURS ON VERTICAL TUBES OR PLATES

dimensionless equation for $4\Gamma/\mu_f$ exceeding 2000,

$$h_m\left(\frac{\mu_f^2}{k_f^3\rho_f^2 g}\right)^{1/3} = 0.0084\left(\frac{W}{\mu_f D}\right)_b^{0.4} = 0.0077\left(\frac{4\Gamma}{\mu_f}\right)^{0.4} \quad \ldots\ldots 2.17$$

Since $\dfrac{W}{\mu_f \pi D} = \dfrac{\Gamma}{\mu_f}$ or $\dfrac{W}{\mu_f D} = \dfrac{\pi\Gamma}{\mu_f}$

where W is weight rate of condensation.

A semi-theoretical relation of Colburn[7] is available for the condensate film for the case where $4\Gamma/\mu_f$ exceeding 2100 before reaching the bottom of a vertical tube. At the top of the tube the local coefficient for the stream line region is given by equation

$$h = \left(k_f^3\rho_f^2 g/3\mu_f\Gamma_z\right)^{1/3}, \quad \ldots\ldots 2.18$$

and that for the turbulent region is based on an equation similar to Eq. 2.8 with (ρu) replaced by Γ_z/y, $\rho D u/\mu_f$ replaced by $4\Gamma/\mu_f$, and omitting the term $(\mu/\mu_w)^{0.14}$, giving

$$\frac{hy}{C_p\Gamma_z}\left(\frac{C_p\mu_f}{k_f}\right)^{2/3} = 0.027\left(\frac{4\Gamma_z}{\mu_f}\right)^{-0.2} \quad \ldots\ldots 2.19$$

The resulting integrated relation and data on condensing diphenyl involving the mean coefficient for the entire tube and Γ, is shown in Fig. 2.1, curves ID and IF are for two values of $C_p\mu_f/k_f$ as predicted by the above equation, while the data scatter considerably, they show in general fair agreement with the predicted line for the turbulent region for a value of $C_p\mu_f/k_f = 5$. This plot is convenient to use where the number and diameter of tubes can be conveniently assumed, thus fixing the value of $4\Gamma/\mu_f$ and the length is left variant.

Grigull condensed several different vapours on the outside of vertical tubes having heights from 3.2 to 22 ft. The Reynolds numbers $4\Gamma_b/\mu$ ranged from 800 to 38,000, and the Prandtl numbers of the condensate film ranged from 1.8 to 5.0. The values of h_m ran progressively above the Nusselt equation for laminar flow as the condensation rates increased and the Prandtl number increased. The trends are similar to those shown in Fig. 2.1.

Seban[30] derived relations considering a vertical plane wall on film condensation with turbulent flow. In his analysis for turbulent flow in the layer, Nusselt's assumptions are retained and the method of analysis used is similar to that employed by Nusselt and by Colburn. The mean value of the heat transfer coefficients have been carried out for a range of Prandtl numbers, from 0.0 to 5.0 and for Reynolds numbers as high as 10^5. He concluded that the transition value was probably between 1200 and 1600 in an examination of available experimental results, which corresponds to Colburns assumption and Grigull.

The results of his tests are presented in terms of the group

$$\frac{h_m}{k}\left(\frac{\mu^2}{\rho^2 g}\right)^{1/3}$$

and the Reynolds number $4\Gamma/\mu$, for a number of values of the Prandtl number of the liquid. The same results are presented in a different way with the Reynolds number given in terms of the generalised length of the layer

$$\frac{x}{(C_p \mu / K)} \cdot C_p (T_1 - T_s) \left(\frac{g \rho^2}{\mu^2}\right)^{1/3}$$

Where T_1 is the temperature of the outer edge of layer.

This representation is more convenient than that of the above and presents the mean heat-transfer coefficient also, since

$$N_{RE} = \frac{4\Gamma}{\mu} = \frac{4 h_m (T_1 - T_s)}{\lambda \mu}$$

2.5. Subcooling of Condensate.

When it is desired to condense a vapour and cool the condensate in a single apparatus, one could employ a vertical tubular condenser with down flow of vapour and up flow of coolant, thus securing counterflow. With isothermal stream line flow of single phase of condensate under the influence of gravity, assuming a linear gradient in temperature through the film, and neglecting change in physical proper ties with change in temperature, Colburn et al[9] showed that the mean temperature T_m of the condensate, before subcooling, is

$$T_m = T_{sv} - \frac{3}{8}(T_{sv} - T_s) \qquad \text{....2.29}$$

The coefficient of heat transfer in the subcooling section, as given by Colburn, is

$$\frac{h_m}{\sqrt[3]{K_f^3 \rho_f^2 g / \mu_f^2}} = 0.01 \left(\frac{C_p \mu}{K}\right)^{1/3} \left(\frac{4\Gamma}{\mu_f}\right)^{1/3} \qquad \text{.... 2.31}$$

and that for condensing section by Eq. 1.39. It was shown by

Colburn et al[0] that this procedure is satisfactory for interpreting the data for condensing and subcooling several organic vapours; coefficients for subcooling ranged from 80 to 120.

2.6. Condensing or Cooling of pure superheated vapours.

Case-1. The mechanism of film- type of condensation of superheated vapour differs from that of saturated vapour. Nevertheless, with corresponding surface temperatures and vapour pressures, superheated vapour has been found to transfer heat at a rate only slightly higher than saturated vapour. For example, Merkel showed that, in condensing steam with 180°F superheat, the rate of heat-transfer q/A was only 3% more than for saturated steam at the same pressure and with the same wall temperature. Similar findings have been reported by others. Hence, for film-type condensation of superheated vapour, little error is made in computing the rate of heat-flow per unit area q/A by multiplying the value of h_m for a saturated vapour by the difference between the saturation temperature (T_{sv}) and the surface temperature (T_s):

$$q = h_m A (T_{sv} - T_s) \qquad .....2.22$$

Case-2. If the pipe wall is above the saturation temperature of the vapour, no condensation will take place; the vapour will merely loose some of its superheat. . The heat transfer mechanism is the same as that for cooling a non-condensable gas; Eq. 2.1 may be used for this type of cooling inside the tubes.

Case-3. It may happen that the temperature of the tube wall is above the saturation temperature of the pure vapour near the vapour inlet and below the saturation temperature at the outlet. Under these conditions, the vapour will be cooled without condensation in the first part of the apparatus and will condense in the latter part. It should be remembered that coefficient of heat transfer in the desuperheating region is very small compared with that in the condensing region and such problems should be divided into two stages- desuperheating and condensing.

2.7. Film Condensation of Vapour in a Horizontal Tube.

In sections 1.9 and 10, derivations for film-type condensation of vapour on a horizontal tube have been presented on the Nusselt's theory and also given the formula for a vertical surface.

Chaddock[5] indicated the formulae for film-condensation of vapour on a vertical tube with slight variations based on Nusselts theory, the resulting expression for the average coefficient of heat transfer is:

$$h_{mv} = 0.943 \left[\frac{K_f^3 \rho_f (\rho_f - \rho_v) g^2 \lambda'}{\mu_f \, H (T_f - T_s)} \right]^{\frac{1}{4}} = \frac{0.943 \, \Omega}{(H \Delta t)^{1/4}} \qquad \ldots\ldots 2.23$$

or, alternatively

$$h_{mv} = 1.47 \left[\frac{K_f^3 \rho_f (\rho_f - \rho_v) g^2}{\mu_f^2} \right]^{+\frac{1}{3}} \cdot \left[\frac{4 \Gamma_v}{\mu_f} \right]^{-\frac{1}{3}} = 1.47 \frac{\Phi}{(N_{RE})^{1/3}}$$

.... 2.24

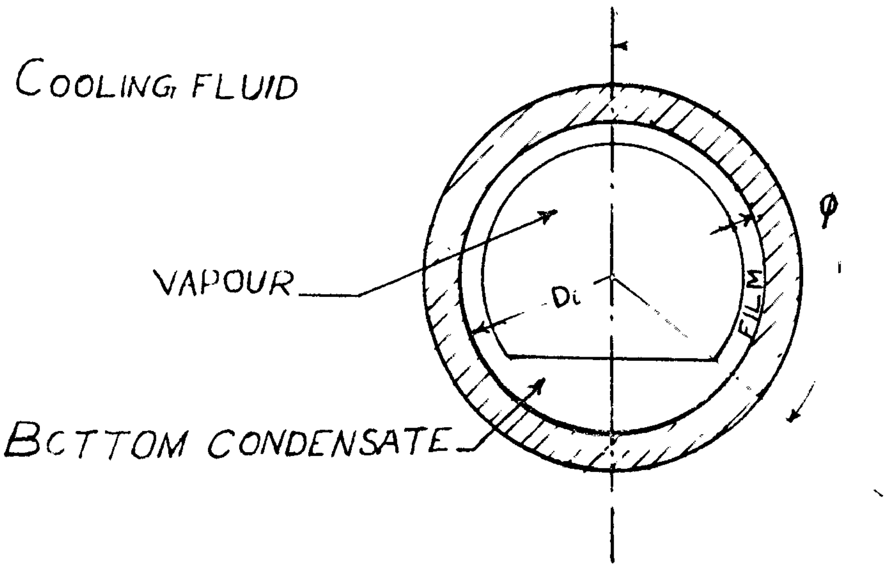


FIG. 2.2. FILM CONDENSATION OF VAPOUR INSIDE A HORIZONTAL TUBE

If vapour condenses on the inside of a horizontal tube, the condensate must collect in the bottom of the tube and flow off in the axial direction. The hot condensate filling the lower cross section of the tube acts to reduce almost completely the heat transfer through this surface. So, only that portion of the tube above the condensate, i.e. from angle 0 to φ as shown in the figure 2.8, for one half of the tube, will be effective for the heat transfer.

The Nusselt derivation may also be applied to condensation on the outside (or inside) of a horizontal tube, as discussed in Section 1.9. Proceeding in the same way as in Section 1.9 and integration over the surface covered by the condensing film (i.e. over the film angle 0 to φ only) gives the average heat transfer coefficient over this area

$$h_{m\varphi} = \left[\frac{0.9036}{\varphi}\int_0^{\varphi}\frac{d\varphi}{Z^{1/4}}\right]\left[\frac{K_f^3 \rho_f(\rho_f-\rho_v)g^2\lambda'}{\mu_f \cdot D(T_f - T_s)}\right]^{1/4} \qquad \ldots 2.25$$

$$= \beta\,\Omega/(D\,\Delta t)^{1/4}$$

or, alternatively

$$h_{m\varphi} = \frac{1.746}{\varphi}\left[\int_0^{\varphi}\frac{d\varphi}{Z^{1/4}}\right]^{4/3}\left[\frac{K_f^3 \rho_f(\rho_f-\rho_v)g^2}{\mu_f^2}\right]^{1/3}\left[\frac{4\Gamma_H}{\mu_f}\right]^{-\frac{1}{3}} \qquad \ldots 2.26$$

$$= \frac{B\,\Phi}{(N_{RE})^{1/3}}$$

Comparison of equations 2.23 and 2.26 shows that they are identical in form except that the tube length H for the vertical surface or tube is replaced by the tube diameter D in the case of the horizontal tube; also the constant 0.943 of Eq.2.23 is replaced in Eq. 2.26 by the quantity β, which is dependent only on the value of the film angle φ. The equations 2.24 and 2.30 also differ in that constant (1.47) of Eq. 2.24 is replaced in Eq. 2.29 by B, a function of the film angle φ. We note, however, in the comparison of these equations that both make use of the quantities Ω and Φ which are made up of physical properties of the fluid. In fact, the density of the vapour ρ_v may be evaluated at the same temperature as that of the liquid film, and the small refinement in the latent heat of vapourisation $\lambda' = \lambda + 0.6 C_p \Delta t$ may be neglected by dropping the second term, both with negligible error except when liquid subcooling is very high.

Comparing now the advantages of horizontal tube over vertical tube, we observe that as long as the condensing film on a vertical tube remains in laminar flow, the heat transfer coefficient will continually decrease along the tube length due to the growth of the film thickness. On the other hand, for the horizontal tube, at a particular angle φ measured from the top of the tube (Fig. 2.2), the film thickness is constant with length. Thus, as pointed out by McAdams, a single horizontal tube has a distinct advantage over a vertical tube for condensation on the outside surface whenever the tube length-to-diameter ratio is tube. If a tube had a ratio of length-to-diameter of 100:1, theoretically, h_m for the horizontal position would be 2.44 times that for the vertical position.

In practice, however, there are a number of factors which tend to reduce this advantage. First, experimental investigations have indicated that condensing heat transfer coefficients for vertical tubes are about 20% higher than those indicated by Eq. 2.23. Second, horizontal tubes are used in tiers rather than singly so that condensate falling from an upper row of tubes to a lower row tends to increase the thickness of the liquid film and, thereby, reduce the heat transfer from the lower tubes which is discussed in detail in the next chapter. Third, if the tube is sufficiently long a transition to turbulent flow of the condensing film occurs on the vertical tube with a subsequent increase in heat transfer rate. The first of these effects is easily taken into account by multiplying Eq. 2.23 by the factor 1.2; the second is accounted for by changing the diameter of Eq. 2.25 from $(D\Delta t)^{1/4}$ to $(ND\Delta t)^{1/4}$ where N is the number of horizontal tube tiers.

With these changes the ratio of heat transfer coefficients for condensation on a horizontal tube to that on a vertical tube becomes:

$$\frac{h_{mH}}{h_{mV}} = \frac{0.728\,\Omega/(ND\Delta t)^{1/4}}{(0.943)(1.2)\,\Omega/(H\Delta t)^{1/4}} = 0.644\left(\frac{H}{ND}\right)^{1/4}$$

.... 2.27

Considering now the third factor of transition to a turbulent condensing film, the Reynolds number at which turbulence begins in a vertical tube, assuming negligible vapour velocity, is generally accepted as 1800. Equating the right-hand members of Eq. 2.23 and 2.24, and solving for the Reynolds number gives:

$$(N_{RE})^{1/3} = 1.56 \left(\Phi / \Omega \right) (H \Delta t)^{1/4} \qquad \ldots \; 2.28$$

Substituting N_{RE} = 1800, and solving for the critical length, i.e., the continuous length upto the point of turbulence, one obtains:

$$H_{cr} = 3{,}700 \left[\frac{\lambda'}{K_f \Delta t} \right] \left[\frac{\mu_f^{5}}{\rho_f (\rho_f - \rho_v) g^{2}} \right]^{1/3} \qquad \ldots \; 2.29$$

Refrigeration condensers are normally designed for an overall temperature difference between 5° F and 20°F, so that Δt between refrigerant and tube wall is about 5°F to 10°F. It may be seen that the tube lengths calculated from the above equations, for the range of 2 to 10°F for Δt, are longer than normally employed in refrigerant condenser design, and, therefore, the transition to turbulent flow of the condensing film is not to be expected.

Let us draw a comparison now between vertical and horizontal arrangements when the vapour condenses inside the tubes. As discussed above, for the horizontal condenser this means that the entire lower cross section of the tube becomes filled with the hot condensate and, thereby, becomes ineffective as a heat transfer surface. If we neglect completely any heat transfer through this "bottom condensate", then from Eq. 2.26 the average heat transfer coefficient based on the total tube area becomes

$$h_{mH} = \frac{\varphi}{\pi} h_{m\varphi} = \frac{\varphi}{\pi} \left(\frac{\beta \Omega}{D \Delta t}\right)^{1/4} \qquad \ldots\ldots 2.30$$

where φ is the previously defined "film angle" measured from the top of the tube to the point where the condensing film ends at the surface of the bottom-condensate (Fig. 2.2.) Combining equations 2.23, and 2.30 for the ratio of horizontal-to-vertical tube performance (bringing in the recommended multiplying factor of 1.2 for the vertical tube) gives:

$$\frac{h_{mH}}{h_{mV}} = \frac{\varphi \beta}{1.13 \pi} \left(\frac{H}{D}\right)^{1/4} \qquad \ldots\ldots 2.31$$

It may be noted that for L/D of 100 and the tube half filled with condensate ($\varphi = 90^{0}$) , the horizontal tube has an advantage of 21%.

Considering the condensate filling the lower cross section of the horizontal tube of Fig. 2.2 is in laminar flow, Chaddock[8] derived an expression for the heat flow through this "bottom-condensate" given by

$$Q' = \left[\frac{2 K_f L \Delta t}{\varphi}\right] Log_e \left(\frac{D}{Y_\varphi} \sin^2 \varphi - 1\right) \qquad \ldots\ldots 2.32$$

where Y_φ is the thickness of the condensing film at the film angle φ. The heat flow through the upper portion of the tube surface covered by the condensing film will be given by

$$Q = h_{m\varphi} (\varphi_m D L) \Delta t \qquad \ldots 2.33$$

where $h_{m\varphi}$ is evaluated from Eq. (2.25), while the value of mean film angle (φ_m) is to be evaluated from

$$\varphi_m = \pi - \left[0.47 J \frac{L (\Delta t)^{3/4}}{d^{2.75}} \right]^{0.142} \qquad \ldots 2.31$$

Where

$$J = \left[\frac{k_f^3 (\rho_f - \rho_v)}{\mu_f \, \rho_f^3 \, \lambda^3} \right]^{1/4} \qquad \ldots 2.32$$

The complete solution for finding the mean heat transfer coefficient of the condensate lies in computing the value of mean film angle φ_m from Eq. 2.31 and substituting this value in Eq. 2.30 along with other values. To facilitate the calculations of this type, Chaddock tabulated the values depending on the physical properties of fluids such as β, B, Ω, J, etc. and other functions such as Z in his article[6].

2.6 A Boundary Layer Treatment of Laminar Film Condensation on Vertical Plates and Horizontal Cylinders.

Nusselt (1916)[27], in his analytical study of laminar film condensation, neglected the effects of both energy convection and fluid accelerations within the condensate layer. Over the years, there have been a number of improvements in Nusselt's analysis as discussed earlier. A characteristic of the previous analytical work has been a focussing of attention on the specific details of the condensation problem, with a consequent neglect of relationships with other heat transfer-fluid mechanics problems.

Condensation on Vertical Surfaces.

In the analysis of Sparrow and Gregg[33], film condensation on vertical surfaces has been approached as a member of the family

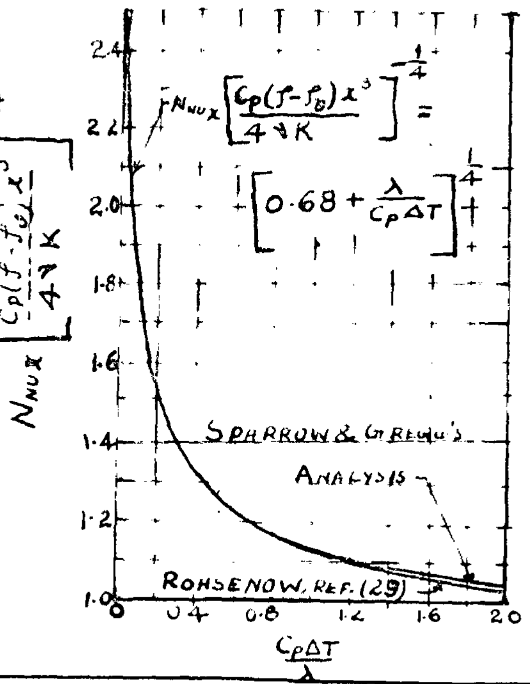
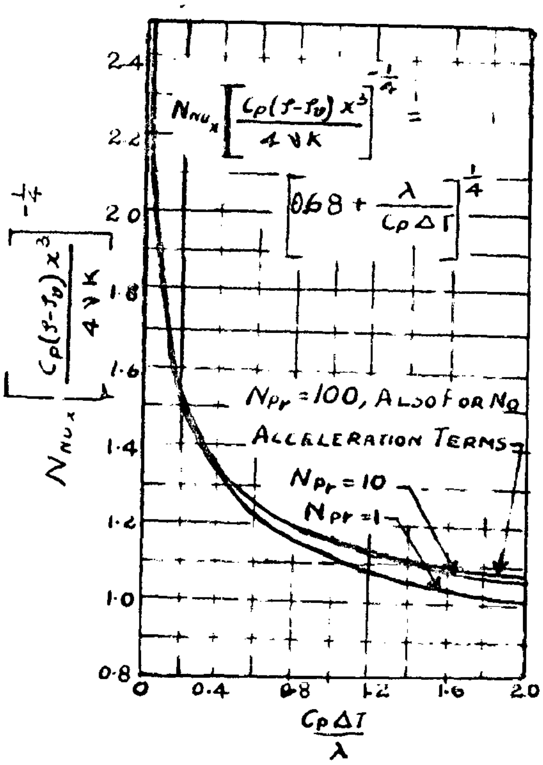


Fig. 2.3 Local Heat-Transfer Results Based On Neglect Of Acceleration Terms

$\left(h_m = \frac{4}{3} h_x\right)$

Fig. 2. 4 Local Heat-Transfer Results From Solutions Complete Boundary Layer Equations

$\left(h_m = \frac{4}{3} h_x\right)$



of boundary-layer phenomena. The starting point for their study is the boundary-layer equations, which includes fluid friction as well as energy convection. These partial differential equations are reduced to ordinary differential equations by means of a similarity transformation. The resulting ordinary differential equations are solved numerically by techniques which have been used previously in other boundary-layer problems. The solutions thus obtained join the ranks of the boundary-layer family.

Based on the above theory, the authors have arrived at the following relations for both heat transfer results i.e. with and without acceleration terms:

1. Solutions for No Acceleration Terms.

The local heat transfer coefficient as desired by the authors and expressed in terms of Nusselt number, may be written as

$$N_{Nux}\left[\frac{c_p(\rho-\rho_w)x^3}{4\nu k}\right]^{-\frac{1}{4}} = \frac{1}{\eta_s I(\eta_s,1)} \qquad 2.30$$

The right side of Eq. 2.30 depends on η_s, or alternatively $c_p \Delta t/\lambda$, since this is equal to $\eta_s^4 e^{3/8\,\eta_s^4} I(\eta_s,1)$. It may be easily shown that the average heat transfer coefficient h_m is 4/3 of the local value h_x. A plot of this Nusselt-number result as drawn by the authors is given in Fig.2.3 for values of $c_p \Delta t/\lambda$ between 0 and 2, this range extends well beyond most current practical applications.

The results of Rohsenow[20] are expressed, with excellent accuracy, by the following simple formula

$$N_{Nux}\left[\frac{c_p(\rho-\rho_v)x^3}{4\nu K}\right]^{-\frac{1}{4}} = \left[\frac{1+0.68\, c_p\,\Delta t/\lambda}{c_p\,\Delta t/\lambda}\right]^{1/4} \quad \ldots\ 2.37$$

The term (1 + .68 cp $\Delta t/\lambda$) on the right side is a refinement of his analysis which becomes more important at high values of liquid subcooling or more precisely high values of $c_p\,\Delta t/\lambda$. A plot of Eq. 2.37 is also given in Fig. 2.3. The agreement between the results of Sparrow and Gregg and Rohsenow is excellent for the range considered above.

2. Solutions with Acceleration Terms.

Sparrow and Gregg showed that it is reasonable to expect increasing effects due to retention of acceleration terms at the lower Prandtl numbers. Solutions were obtained by the authors numerically for Prandtl numbers 1, 10 and 100 for the range of $0 \leq \frac{c_p\Delta t}{\lambda} \leq 2$.

Solution of the equations in terms of the variables of the analysis for the local Nusselt-number is

$$N_{Nux}\left[\frac{c_p(\rho_f-\rho_v)x^3}{4\nu K}\right]^{-\frac{1}{4}} = \left(\frac{d\theta}{d\eta}\right)_{\eta=0} \quad \text{Eq. 2.38}$$

where $\left(\frac{d\theta}{d\eta}\right)_{\eta=0}$ is found from the solutions of equations of

$$F''' + \frac{1}{N_{PR}}\left[3F''F - 2(F')^2\right] + 1 \quad \text{Eq. 2.39}$$

and $\theta'' + 3F\theta' = 0$ Eq. 2.40

and is a function of Prandtl number and $C_p \Delta t/\lambda$

A plot of equation 2.38 is presented in Fig. 2.4. Also shown in the figure is a curve representing the results obtained from solutions with no acceleration terms. The N_{PR} = 100 results are seen to coincide (within the scale of the figure) with those for no acceleration terms. The results for N_{PR} = 10 and N_{PR} = 1 can be plotted as separate curves for at least part of the range; but their largest deviation from the curve for no acceleration terms is quite small. (The largest deviation is about 5% for N_{PR} = 1 at $C_p \Delta t/\lambda$ = 2).

So for practical purposes, the effect of Prandtl number on the heat transfer appears to be negligible for the range $N_{PR} \geq 1$. The heat transfer results are thus computable from Eq. 2.37, which was derived on the basis of no acceleration terms.

Condensation on a Horizontal Cylinder.

Sparrow and Gregg[31] derived the relations for heat transfer coefficients on horizontal cylinder also starting with boundary-layer equations appropriate to the horizontal cylinder and transforming the partial differential equation to ordinary differential equations which are valid over a major portion of the surface of the cylinder, as discussed earlier. The transformation has been carried out in such a way that the resulting ordinary differential equations coincide with those for condensation on a vertical plate. Utilizing the numerical solutions of the transformed equations, heat transfer results have been presented for the horizontal cylinder over the Prandtl number range from 0.003 to 100.

From the plot of $C_p \Delta t/\lambda$ and $N_{Nu} \Big/ 0.733\left[\frac{g\rho^2 \lambda D^3}{k\mu \Delta t}\right]^{1/4}$ drawn by the authors[24] for Prandtl numbers ranging from 0.003 to 100, the authors recommended Eq. 2.41 for small values of $C_p \Delta t/\lambda$, for the heat transfer results for fluids of any Prandtl number

$$N_{Nu} = 0.733\left[\frac{g\rho^2 \lambda D^3}{k\mu\,(T_f - T_s)}\right]^{1/4} \qquad \ldots\ldots 2.41$$

In the above equation, since values of $C_p \Delta t/\lambda$ are assumed to be small which correspond to thin films, the effects of both energy convection and inertia forces are very small. So, Eq. 2.41 describes the heat transfer process in the limiting situation where the energy convection and inertia play a negligible role. It is interesting to observe that Nusselt's analysis[27], Eq. 1.41, which deals with only this limiting situation, leads to a result identical to Eq. 2.41, except that his constant is 1 % lower.

To find the relationship between vertical plate height and cylinder diameter required to produce equal heat transfer coefficients for plate and cylinder, for the same temperature differences, and fluid properties, using Eqs. 2.41 and 1.30, it will be found that

$$H = 2.70\ D \qquad \ldots\ldots 2.42$$

as compared to a value of H = 2.87D as quoted by McAdams[25] based on Nusselt's Single theory. Using this equivalence, correlations of the heat transfer coefficient for cylinders may also be used for vertical plates and vice-versa.

If the diameter of a cylinder is small, the flow of vapour and liquid film on a vertical strip of the surface will interfere with the flow on neighbouring surfaces. Consequently, for vertical cylinders of extremely small diameters, the equations for vertical plates cannot be used.

CHAPTER 3

EXPERIMENTAL ANALYSIS OF CONDENSATION OF VAPOUR FOR SINGLE AND MULTITUBE CONDENSERS

ABSTRACT.

An analytical method along with correlation of experimental data are presented for laminar condensation inside horizontal and inclined tubes. Wilson's graphical method of determining the mean coefficient of heat transfer is indicated. A discussion of the effect of arrangement of tubes in multitube condensers, and the determination of heat transfer coefficients for such types are made. Finally, the other type of condensation, viz. dropwise condensation, is considered.

3.1 Laminar Condensation inside Horizontal and Inclined Tubes.

None of the investigations discussed in the previous chapter covers the situation where the condensate layer on the bottom of the tube is deep enough to influence the heat transfer seriously, and where the vapour shear is relatively low for most of a long condenser tube. For ordinary fluids with Prandtl numbers of the order of one or greater, the first approximation or Nusselt's analysis, yields accurate results. For fluids with low Prandtl numbers, such as liquid metals, the heat transfer coefficients are considerably below those predicted by the simplified solution.

Chato[6] investigated, both analytically and experimentally, fluid mechanics and heat transfer phenomena occuring during laminar condensation inside single pass horizontal and inclined tubes. He obtained a solution to the problem of condensation on an inclined surface by writing the momentum and energy equations for the condensate flow. By using the concept of similarity, these equations were then reduced to a single ordinary differential equation with but one dependent and one independent variable. He derived the formulae for both horizontal and inclined ~~d~~ tubes, for mean heat transfer coefficient, volumetric liquid condensate flow in the free portion of the tube and condensate flow on the bottom of the tube.

He conducted two experiments with refrigerant 113 to correlate the data and to simplify ~~the~~ the analytical values as the solution of these equations is tedious. The first was a fluid mechanical analogy apparatus using water. This set-up was used primarily for investigation of the behaviour of the liquid discharge at the outlet end. Flow depths were determined by measuring the the fraction of the tube circumference which was covered with liquid. Volumetric discharge rates were measured with a burette and a stop watch.

The second experimental set up was a natural circulation loop for condensation experiments. The condenser test section consisted of 28 .25 in. long, 0.572 in. mean I.D., standard finned copper tubing. Vapour was generated in a brass boiler set atop and a 2000-watt flat-plate heater. Electrical input was regulated by two variacs and measured by a watt meter. All thermocouples were made of 30-gauge copper-constantan wires, and had good reference junctions. Their potential was measured by a semi-precision potentiometer through a ten-point selector switch. Air velocities around the test section were controlled by two fans and were measured with an air meter. After equilibrium was reached in a test run, observations were made of the pressures, temperatures, flow conditions inside the tube and heater inputs.

For nearly horizontal tubes, with a slope of not more than 0.002, the mean condensate angles, $\theta = 2(\pi - \varphi)$, calculated by the numerical integration procedure and observed during the runs, were within 8 degrees of a mean vapour angle $\varphi = 120$ degrees. The heat transfer coefficients calculated by both the integration procedure and by assuming a mean angle of 120° showed good correlation with experimental data.

For the inclined positions a marked decrease in condensate depth and a corresponding increase in heat transfer rates was observed and calculated upto a slope of about 0.01. Further increase in slope decreased the depth rather slowly. Since for large inclinations the wall heat transfer begins to deteriorate, an optimum slope can be found for a given set of conditions which provides the maximum heat transfer rate. Depending on the parameters considered, this optimum inclination ranged from 10 to 20 degrees.

Chato recommended the following relations using a mean vapour angle of φ = 120 deg since the heat transfer results are relatively insensitive to the variation of the vapour angle.

a) Horizontal tubes. ($\sigma \leq 0.002$)

The mean heat transfer coefficient, based on the entire internal area is

$$h_m = 0.468\,A\left[\frac{g\,\rho_f(\rho_f-\rho_v)\,\lambda\left(1+\frac{0.68\,C_{pf}\,\Delta t}{\lambda}\right)k_f^3}{\mu_f\, r_i\,\Delta t}\right]^{1/4} \quad \ldots.\ 3.1$$

b) Inclined tubes. ($0.002 < \sigma < 0.6$)

The value of φ is to be found out by trial and error method for the required slope (σ) and this is to be substituted in the equation given below

With the vapour angle thus found, the mean heat transfer coefficient, based on the entire internal area becomes

$$h_m = 0.300\,A\left[\frac{g\,\rho_f(\rho_f-\rho_v)\,\lambda\left(1+\frac{0.68\,C_{pf}\,\Delta t}{\lambda}\right)k_f^3}{\mu_f\, r_i\,\Delta t}\right]^{1/4} \times$$

$$\left[\int_0^{\varphi} \sin^{1/3}\varphi\, d\varphi\right]^{3/4} \cos^{1/4}(\sin^{-1}\sigma) \quad \ldots.\ 3.2.$$

where A = Correction factor for low Prandtl numbers.

σ = Slope of Tube.

r_i = Inner radius of tube, ft.

$\Delta t = (T_f - T_s)$

3.2. Local Over-all Coefficient of Heat Transfer.

Before we proceed on to the other methods of

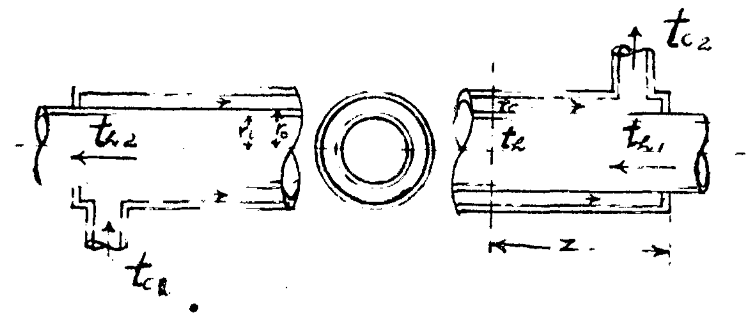


FIG. 3.1 DIAGRAM OF COUNTERFLOW HEAT EXCHANGER.

finding out the local coefficients of convective heat-transfer it is very useful to introduce the term local overall coefficient of heat transfer. In the majority of heat-transfer cases met in industrial practice, heat is being transferred from one fluid through a solid wall to another fluid.

In an apparatus such as is shown diagrammatically in Fig. 3.1, cold fluid flows through the jacket, and a hot fluid flows through the tube. All mass flow rates are constant, and the steady state has been attained; hence the temperature at each point in the apparatus is independent of time. Consider any cross section located at a distance Z from the entry of the cold fluid. If the cold stream were drawn off at this section and mixed, it would have a temperature t_c, called the bulk temperature; t_h is the corresponding bulk temperature of the hotter fluid at Z. For such conditions Newton suggested that the rate dq, of heat transfer per unit surface area was directly proportional to the overall difference between the temperatures of the warmer and colder fluids, $t_h - t_c$ or Δt_o, and to the heat transfer surface dA:

$$dq = U\,(dA)\,(\Delta t_o) \qquad \ldots\ 3.3.$$

and the proportionality factor U is called the local overall coefficient of heat transfer or merely the local overall coefficient.

In computing U, one may use the area of heating surface dA_h, the area of the cooling surface dA_c, or the logarithmic-mean surface dA_w. Since in a given case $dq/\Delta t_o$

in [illegible], one can obtain three values of local overall coeffi-cients U_h, U_w, U_c which are related by means of area ratios $U_h dA_h = U_w dA_w = U_c dA_c$. It is immaterial which heat transfer surface is chosen so long as it is specified. To avoid misunderstandings, the area basis of an overall coefficient should therefore always be stated.

As will be shown later, the numerical value of U in a given set of units may vary many fold, depending on the nature of the fluid, their velocities, and other factors. Eq. 3.8 is the basic relation for heat transfer between fluids separated by a retaining wall. This requires consideration of the nature of the thermal resistances met as the heat flows from the warmer fluid through the retaining wall to the colder fluid.

Relation between Local Overall and Local Individual Coefficients.

The heat transfer from the hot fluid to a short length of annular space at section Z would be given in terms of local heat transfer coefficient h_h and a local temperature difference Δt_h. For this local temperature difference it is convenient to use the fluid bulk temperature minus the inner wall tempera-ture; thus $dq = h_h dA_h (t_h - t_{wh})$. Similarly on the cold side at the location Z, $dq = h_c dA_c (t_{wc} - t_c)$ where t_{wc} denotes the temperature of the outer surface of the inner tube. Similarly the heat transfer rate through the wall,

$dq = K_w dA_w \left(\frac{t_{wh} - t_{wc}}{r_o - r_i}\right)$ where K_w is the thermal conductivity of the wall, A_w the mean area, and ($r_o - r_i$) the wall thickness. Solving for the individual temperature differences and addition of these differences will yield

$$\Delta t_o = t_h - t_c = dq\left(\frac{1}{h_h dA_h} + \frac{r_o - r_i}{K_w dA_w} + \frac{1}{h_c dA_c}\right) \quad \ldots\ 3.4$$

Comparison of Eq. 3.4 with 3.3 gives

$$\frac{1}{U} = \frac{dA}{h_h dA_h} + \frac{(r_o - r_i)\,dA}{K_w\, dA_w} + \frac{dA}{h_c\, dA_c} \quad \ldots\ 3.5a$$

and

$$\frac{1}{U_h} = \frac{1}{h_h} + \frac{(r_o - r_i)\,dA_h}{K_w\, dA_w} + \frac{dA_h}{h_c\, dA_c} \quad \ldots\ 3.5$$

which may be written in the integral form as

$$\frac{1}{U_h} = \frac{1}{h_h} + \frac{r_i \log_e \frac{r_o}{r_i}}{K_w} + \frac{1}{\frac{r_o}{r_i}\, h_c} \quad \ldots\ 3.6$$

By defining a total resistance, R_t, for unit area dA as $1/U$ and the various individual resistances, R_h as $dA/h_h dA_h$, R_w as $(r_o - r_i)dA/k_w dA_w$ and R_c as $dA/h_c dA_c$

Eq. 3.6 becomes $R_t = R_h + R_w + R_c$, showing that individual thermal resistances are additive for steady flow of heat through a series of resistances. Summarizing,

$$dq = \frac{\Delta t_o}{1/U dA} = \frac{\Delta t_o}{\frac{1}{h_h dA_h} + \frac{r_o - r_i}{K_w dA_w} + \frac{1}{h_c dA_c}} \quad \ldots\ 3.7$$

or $$U = \frac{1}{R_t} = \frac{1}{R_h + R_w + R_c} = \frac{1}{\Sigma R} \quad \ldots\ 3.7a$$

A further discussion on this is reserved for the next chapter.

3.3 Graphical Method of Wilson.

In surface condensers, a vapour is being condensed on the outer surface of a tube, while a liquid, usually cooling water, flows in turbulent motion inside the tube. Although in some cases tube temperatures have been measured, especially in testing single-tube experimental apparatus, in the majority of cases tube temperatures have not been measured; hence overall coefficients of heat transfer, U, are available. In attempting to correlate these overall coefficients, many engineers have failed to take advantage of the fact that the overall resistance to heat flow ($R = \frac{1}{U}$) is numerically equal to the sum of the individual series resistances, namely, the resistance on the vapour side R_V, that of the wall itself R_W, and that on the water side R_L. If there are dirt deposits on either side of the wall, they may also be added. In 1915, Wilson[33] employed a valuable graphical analysis of the overall coefficient of heat transfer. The following treatment presents the method and illustrates the application to several sets of data.

Consider a series of runs made in condensing substantially air-free vapour at a given temperature, employing different water velocities. From the concept of resistances it is clear that the total resistance is equal to the sum of the individual resistance $1/U = R_V + R_W + R_L$. According to the theoretical equation of Nusselt, the resistance on the vapour side depends upon the temperature difference and the temperature of the condensate film; and hence R_V should vary somewhat as water velocity is changed. Also, changes in tube-wall temperature, brought about by changes in water velocity, would cause minor variations in the thermal conductivity of the tube wall. However, except where very high water velocities are used, the water side resistance is usually the major resistance from condensing vapour to liquid, and, under these conditions, serious error would not be introduced by assuming that the sum of the first two individual resistances, $(R_V + R_W)$, is approximately constant. As is well known, the water-side resistance is an inverse function of the water velocity, V' , through the tubes, and, neglecting the effects of changes in water temperature, due to changes in water velocity, the water side resistance, R_L, could be taken as a function of the water velocity alone. As Wilson points out, a plot of $\frac{1}{U}$ vs $\frac{1}{\varphi(V')}$ should give a straight line when plotted to ordinary rectangular co-ordinates. For turbulent flow of the water in a given apparatus , $\varphi(V')$ may be taken as $C_1 (V')^{0.8}$ and hence

$$\frac{1}{U_i} = (R_V + R_W) + \frac{1}{C_1 (V')^{0.8}} \qquad \ldots\ldots 3.0$$

where C_1 is an empirical constant and may be considered as the apparent individual coefficient of heat transfer from tube to water, based on the inside surface for water velocity of 1 ft/sec.

From the above graph, the intercept on the vertical axis will be equal to the sum of the resistances of vapour and wall, and the reciprocal slope of the straight line will give the value of C_L. Knowing the resistance of the wall which can be easily calculated from the data of the tube, the resistance of the vapour will be equal to the value of the intercept minus the resistance of the wall. Thus, by the aid of this graphical method, the observed overall heat transfer coefficient has been subdivided into the three component resistances R_v, R_w, and R_L.

Even if one does not wish to evaluate the individual thermal resistances as outlined above, the graphical method gives a straight line, thereby facilitating interpolation and extrapolation. When the water velocity is too low, the assumption of an 0.8 exponent will be incorrect; hence this relation will break down at low water velocities.

In condensing a vapour whose condensate has a low thermal conductivity, theory shows that the thermal resistance on the vapour side should be larger than for condensing vapour, and hence some of the assumptions underlying this graphical method will appear somewhat questionable.

In modifying the Wilson method, Chu and others adjust the overall temperature potential to give the same average heat flux $U_i \Delta t_o$ at all water velocities, thus ensuring constant value of h_m on the vapour side. As the value of h_m on the water side is proportional to $(1 + 0.011t)(v')^{0.8}$, $\frac{1}{U_i}$ vs $1/(1+0.011t)(v')$

is plotted.

If refinements in the method be desired, it is possible to introduce factors for water viscosity and tube diameter into the abscissa.

3.4. Condensation on Single Horizontal and Finned Tubes.

A brief account of the condensation of vapour on horizontal finned tubes will now be given.

Donald et al[17] conducted experiments on six single, finned copper tubes and one plain copper tube, each 3 feet long to determine experimentally heat transfer coefficients which would serve as the basis for predicting the size of condensers for Freon-12 vapours. Superheated Freon vapours from a 3-ton capacity compressor or saturated vapours from a still were condensed on the outside of the tubes by passing water at a series of velocities 2,4,6,8 and 10 ft. per sec, through the inside.

The experimental unit consisted of a condenser, still and a compressor. When the latter was used, the refrigeration circuit was completed by having an expansion valve and an evaporator. The condenser consisted of two concentric tubes, the inner one being a finned tube or plain tube, as the case may be , on which the Freon vapours passing through the annular space were condensed. Cooling water was circulated in the inner tube. The still also consisted of two concentric tubes, just like the condenser. In this, Freon liquid was again circulated in the annular space and steam was circulated in the inner; finned tube to evaporate the Freon liquid. These vapours were circulated in the condenser and the condensate temperatures and weight rate and also water rate measurements were made for each type of tube to be tested. The difference between average temperatures of Freon and water were maintained constant at different water velocities so as to fulfil the conditions of Wilson plot.

The overall heat transfer coefficient was calculated from the equation

$$q = U A_o \Delta t \qquad \ldots 3.0$$

where Δt = average temperature difference between the saturation temperature of Freon in condenser and water.

From the Wilson plot, as explained in the previous section, the heat transfer coefficients of the condensate and that of water were calculated from the experimental data. The combined heat transfer coefficient of the condensate was also calculated analytically using the Eq. 1.29 for vertical surfaces and Eq. 1.41 for the horizontal cylinder portion taking these values proportional to their areas, plain tube surface being considered as horizontal portion and fin surface as vertical portion. The water-film coefficient was calculated analytically by using Eq. 2.1. The experimental values were compared with that calculated by analytical methods. It was found that experimental values of water-film coefficients exceed generally by 10% those obtained by the Eq. 2.1. But the experimental values of heat transfer coefficient of the condensate were less than those calculated analytically. The deviation of these results varied between 1.2 to 36.3%

The experimental values vary from 208 to 260 for plain tubes and 218 to 425 for finned tubes based on the outer area of the tubes, which varies from 0.898 to 4.528 sft. for finned tubes and 0.0870 for plain tube, for an average temperature difference of 40°F.

These variations of values are found to be within reasonable limits.

3.6 Laminar Condensation on Multiple Horizontal Tubes.

In the discussion made so far, various methods both theoretical and experimental, determining heat transfer coefficients of the condensation of vapour have been considered for a single horizontal or vertical tube with both inside and outside condensation. But commercial horizontal condensers are normally constructed with a bundle of plain or finned tubes in which the condensation from one tube drips into another tube. This dripping condensate from the tubes decreases the heat transfer coefficient for the lower tubes in a multitube condenser. So, a brief discussion of the effect of the arrangement in multitube condensers on the heat transfer coefficient will now be made.

For N horizontal tubes arranged in a vertical plane so that the condensate from one tube flows directly on to the top of the tube directly below without splashing, h_m in Eq. 1.42

theoretically depends on the inverse fourth of root of N. This effect of N is automatically allowed for in Eq. 1.51, which contains Γ', based on the total condensate from the lowest tube in the tier. Consider measured coefficients h_1, h_2 ....,h_N for each of N rows of horizontal pipes in a vertical tier or for successive sections of a vertical surface. The coefficients for the top tube or section may be compared with Eq. 1.42 or 1.39 for the coefficients on a surface having no condensate fed to the highest point. Since the second row or section receives condensate from the first, the coefficients for these sections should not be compared with Eq. 1.42 or 1.39. However, the mean coefficient for the two highest sections may correctly be compared with Eq. 1.51, involving Γ' leaving the second section. If the two sections are equal in size

$$h_m = \frac{(q_1 + q_2)/2A}{(\Delta t_1 + \Delta t_2)/2}$$

Hence $h_m \sqrt[3]{\mu_f^2/k_f^3 \rho_f^2 g}$ should be plotted vs $4\Gamma_2'/\mu_f$ and compared with Eq. 1.51, where $\Gamma_2' = (q_1 + q_2)/\lambda L$ for two horizontal tubes in a vertical tier and $\Gamma_2' = (q_1 + q_2)/\lambda \pi D$ for the two highest sections of a vertical tube. A similar procedure applies for any number of sections, so long as the Reynolds number is less than 2100 for the condensate leaving the bottom section.

The local coefficient h_N for any row N is theoretically related to that for the top row by the equation

$$\frac{h_N}{h_1} = N^{0.75} - (N-1)^{0.75} \qquad \ldots 3.19$$

Thus h_2/h_1 is 0.683, h_3/h_1 is 0.60, and $(h_1 + h_2 + h_3)/3h_1$ is 0.762 as is also predicted by $h_m/h_1 = 3^{0.25} = 0.762$. As shown later, the coefficients for the lower tubes run higher than predicted, owing to the effect of turbulence caused by dripping and splashing as the condensate falls intermittently from one tube to the next.

Numerous investigators have conducted experiments on a bank of both plain and finned horizontal tubes to compare the experimental results with that of theoretical values based on Nusselt's theory. Some of the investigations are reported here, in detail regarding the experimental set up, procedure and the correlation of the data with Nusselt's theory.

Young and Wohlenberg[30] conducted experiments on a horizontal condenser, consisting of 5 copper tubes, 3/4 in. O.D., 41 [illegible] in. long, with axes in a vertical plane. The shell was composed of a piece of 10 in. pipe with welded heads. The surface temperatures of the tube and that of gas and cooling water were measured by copper-constantan thermocouples. Freon-12 was used as the working fluid. The condenser pressure was controlled by regulating the steam flow to the evaporator. Surface temperature of the tube was controlled by varying both the initial temperature and velocity of the cooling water. A sufficient temperature rise of the cooling water was insured.

After the bank of five tubes had been tested, the consistency of the data was investigated by blanking off the top tubes, one at a time, and operating this smaller number of tubes over the range of test conditions. This process was carried on until banks of five, four, three, and two tubes had been tested. In this manner, each tube was tested in a number of relative positions in the tube bank.

The authors made observations of the tube through the sight glass which showed that, under all conditions and rate of condensation, a very quiet, streamline film formed on the first tube in the bank. At the bottom of the tube, the liquid formed drops which fell to the second tube. These did not always fall from the same point; at times the point of origin of each moved back and forth along the tubes. At high rates of condensation, these drops became small streams.

The films on all tubes below the first in the bank were definitely disturbed at all rates of condensation. A quiet, stream line film formed on all the tubes. A drop falling from above struck a tube and fell down over the top of the existing smooth film. This formed a rivulet approximately 1/8 in. wide which increased the film thickness around that part of the tube. As the rate of condensation increased, the number of rivulets became greater until all but the first tube exhibited a rapidly moving, rippled film. For all but the lowest rates of condensation, the liquid fell of the bottom of each of the lower tubes in steady streams. Under no conditions was any liquid seen to break away from the film and drop off the tube before reaching the bottom.

The experimental data was used to investigate the film-coefficient of heat transfer for condensing. Freon-12. In order to compare Freon-12 with other fluids, all test results were given in terms of heat-transfer coefficient, h, and $\frac{\lambda k^3 \rho^2}{\mu D \Delta t}$. This also simplified the analysis and permitted all data to be plotted against only two variables, since the term $\lambda k^3 \rho^2 / \mu D \Delta t$ takes into account the slight changes in physical properties of the fluid in this temperature range.

It was found from the experimental results, that the film coefficient was smaller for the top tube and greater for the lower tubes compared to the values based on Nusselt's theory.

From the plots made with film coefficient and $\lambda \rho^2 k^3 / \mu D \Delta t$ the following set of equations for the film coefficient, h, on the gas side of a horizontal multi-tubular condenser were given by Young and Wohlenberg[59].

The experimental values were also compared with the theoretical values predicted by Nusselt and McAdams. The equations for the coefficient of heat transfer were expressed as $h = C \sqrt[4]{\frac{\lambda \rho^2 k^3}{\mu D \Delta t}}$, where c is a constant which varies for each tube placed in a vertical tier. The values of c, as determined from the experimental results of Young et al[59] and as predicted by Nusselt and McAdams, are as follows:

	Young et al[20]	Nusselt (Eq. 3.10)	McAdams.
Table 1	c = 0.655	0.725	0.725
Table 2	c = 0.576	0.493	0.61
Table 3	c = 0.551	0.436	0.551
Table 4	c = 0.488	0.401	0.513
Table 5	c =.464	0.375	0.485

Katz et al[18] made experiments on a condenser which had 40 finned tubes, each three feet long, with 10 passes for the water, to obtain a comparison between the performance of single finned tubes and tubes in a multitube condenser. This commercial condenser was tested by installing it horizontally on a 25-ton air conditioning unit using Freon-12. The air-conditioning unit was operated with a heat load of 180,000 - 220,000 BTU/hr furnished by the building being cooled. The water rate for the condenser was varied from 2 to 12 ft. per sec. with a variation in Freon pressure in the condenser accompanying a change in the water rate. For calculation purposes, the temperature on the Freon side was used as the saturation temperature corresponding to the operating pressure in the condenser. The overall coefficients were based on the outside area of the finned tubes, which had 16 fins per inch a fin height of 0.068 inch and the root diameter ranged from 3/8" to 1".

Three different procedures, as outlined below, were used by the authors to correlate the data.

First Procedure.

The first procedure was to correlate the data with the theory of Nusselt for plain tubes. The average film coefficient for condensing vapour on N tubes in a vertical row was taken as a function of the condensing coefficient for a single tube by

$$h_{N\ tube} = \frac{h_{single\text{-}tube}}{N^{0.25}} \quad \text{.... 3.11}$$

where N = average number of tubes in a vertical row.

If the number of tubes on one vertical row are different from another row, the Eq. 3.12 based on Eq. 1.42 may be used to find out the average number of tubes per row.

$$N_{ave} = \left[\frac{N_1 + N_2 + N_3 + N_4 + \text{--- etc}}{N_1^{.75} + N_2^{.75} + N_3^{.75} + N_4^{.75} + \text{---- etc}}\right] \quad \text{.... 3.12}$$

Subscripts 1, 2, 3, etc., refer to individual rows, and N is the number of tubes in each row.

To obtain the film coefficient for condensing Freon, the overall coefficient was seperated into a water film coefficient and the condensing film coefficient by Eq. 3.13.

$$\frac{1}{h} = \frac{1}{U} - \frac{A_o}{A_i\, h_w} - R_m \quad \text{.... 3.13}$$

where R_m is the resistance of the metal which may be neglected.

The water-film coefficients were computed by the Eq.2.1 multiplied by a modifying factor 1.25 on the right hand side.

The condensing film coefficient computed by Eq. 3.13 is the average condensing coefficient for rows having $N_{ave.}$ tubes per row (3.2 tubes in their experiment) vertically one above another. The condensing film coefficients for single finned tubes of the type used in these condensers were taken from their previous paper.[17]

From the available data, the average value of the exponent N in Eq. 3.11 was calculated to be 0.223 as compared to 0.225 used for plain tubes. Since finned tubes represent a combination of horizontal and vertical surfaces it appears difficult to predict the exact power that should be used for the average number of tubes in a row as indicated in Eq. 3.11. The data showed considerable spread in the exponent for several runs, as compared with average value.

Second Procedure.

A second procedure employed by them for correlating the data was to plot the reciprocal of the overall coefficient as a function of the reciprocal of the 0.8 power of the water velocity , via Wilson plot. For this comparison, calculated lines were drawn for different values of Δt and the actual points also were plotted on the same figure. The calculated lines were determined by values of the overall coefficient calculated at infinite and a practical water velocity. The values at infinite water velocity were obtained by multiplying the value for a single tube under same conditions by the factor 0.728 instead of 0.725 in Eq. 1.41. The overall coefficient at practical water velocity was calculated by combining the water film and condensing film coefficients. The water film coefficient was calculated by using the Eq. 2.1 as modified above. The condensing film coefficient was calculated by Nussel -t's equation, and the value was so adjusted by a trial and error process until the temperature drop across the condensing film used in the equation was proportional to the resistance of the condensing film when compared to the overall temperature drop and overall resistance.

It was seen that the data were generally consistent with these curves, with good agreement at the water rates corresponding to commercial practice.

Third Procedure.

A third procedure employed was for checking the relationship between calculated and experimental coefficients, to predict the overall coefficient for the actual conditions of each run by using the water-film coefficient and condensing film coefficient for single tubes modified by the factor corresponding to the 0.25 power for N. By such comparison, the experimental overall coefficients were about 10 percent higher, on the average, than the calculated coefficients.

Design charts were also constructed to facilitate the prediction of the surface requirements when condensing F-12 on integral spiral finned tubes having 16 fins per inch and the method of using these charts for designing a multi-tube condenser for F-12 was also indicated.

Katz et al[20] made experiments on a condenser having six horizontal finned tubes in a vertical row. The tubes had 15 actual fins per inch, but are listed as nominal 16 fins per inch, and made up of copper with 5/8" root diameter. The condenser shell was constructed from a 10-inch pipe, 3 feet long with tube sheets of 1 1/2 inch thick welded on each end. The experiments was conducted in the same way as explained before and using Freon-12, n-butane, acetone, and water as test fluids to give a wide range of ratios of surface tension to density.

In translating data for condensing refrigerants on finned tubes into expected performance for petroleum or organic chemicals, differences in physical properties should be considered. Static tests to determine the liquid retention on finned tubes indicated that liquids of high surface tension and low density adhered to the bottom of the finned tubes and increased the film thickness. Any retention of liquid under dynamic conditions would be most significant for the lower tubes in a vertical row.

The overall heat transfer coefficients were calculated for each tube from the data taken and a modified Wilson plot was made, the slope of which depends upon the physical properties of the cooling water and the condenser tubing. One modification consisted of a correction factor to include the effect of temperature on the physical properties and cooling water. By the use of the correction factor $(1+0.011\ T_1)/(1+0.011\ T_2)$ on the velocity term, the same slope was used for a given tube, independent of the material. The experimentally determined condensing film coefficients were compared with those calculated from the Nusselt equation for each of test fluid. Rather than applying the equations for each the horizontal and vertical surfaces to relative areas represented, as discussed previously, Eq. 3-11 was used with the following definitions of the equivalent diameter based upon the fractional horizontal and the fractional vertical surface.

$$\frac{1}{De^{1/4}} = \frac{0.943}{0.725}\ \frac{Af}{Aeq\ L^{1/4}} + \frac{At}{Aeq\ D^{1/4}} \qquad \ldots.\ 3.14$$

where

Af = Area of fins, sft.

At = Area of horizontal tube having diameter equal to root diameter, sft.

Aeq = A_t x A_f x fin efficiency, sft.

D = Root diameter, ft.

L = Average height of finned area, ft.

The fin efficiency was greater than 0.96 for all conditions and was used as 100 percent. The coefficient calculated from Nusselt equation for tube 2 was obtained from the coefficient for tube 1 by the Eq. 3.10. A similar calculation was made for the lower tubes.

The deviation between the experimental and predicted coefficients for the top tube (tube 1) for organic liquid was less than 14 percent and is as close as that previously found[10]. The data indicated that the decrease in condensing coefficients for tubes below the top tube was much less than that predicted by Nusselt. This is due to the reason that the liquid dripping from one tube may not strike the tube below, nor does it necessarily increase the film thickness over the entire heat-transfer area. Also, the effect of turbulent flow of the liquid would tend to increase the coefficients over the calculated values for the lower tubes in the row.

For a bank of six finned tubes in a vertical row the average condensing heat transfer coefficient was only 10 % below that of the top tube. For design purposes Nusselt's theory, applied to a bank of horizontal finned tubes, may be expected to give condensing heat transfer coefficients below the experimental values.

Short et al[32] had made experiments on twenty horizontal plain brass tubes, 0.625 in OD and 0.496 in. I.D. for F-11 and 0.52 T in. for steam and having a vertical spacing of 1.5 in.

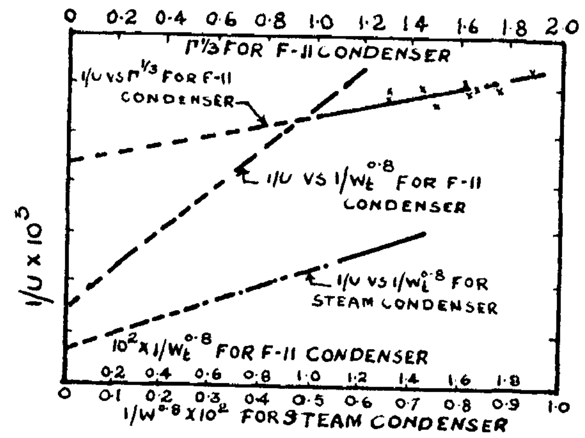


FIG.3.2. OVER ALL RESISTANCES AS FUNCTIONS OF FLUID FLOW RATES

between tube centres, mounted between tube sheets spaced 82.5 inches apart and determined the condensate-film heat transfer coefficients for individual horizontal tubes in a vertical bank of tubes and compared the experimental data with the values predicted by Nusselt's analysis for Freon-11 and steam.

The experimental procedure followed by the authors was the same as that described earlier. The mean temperature difference across the film ranged from 0.5 to 6.2°F for the Freon-11 tests and from 10 to 45°F for the steam tests. The condensate flow rates from the top tubes of the condenser, per foot of tube length, ranged from 1 to 7 lb./hr. for the Freon-11 tests and from 6.7 to 13 lb/hr. for the steam tests.

The analysis of the results followed by the authors was slightly different from the rest of the procedure discussed so far. The analysis of the results and correlation of the data will be briefly presented, now as follows:

Overall transfer rates were measured, but sufficient information was obtained to permit separation of the inside fluid resistance R_s from the outside condensate resistance R_h. Fig.3-2 is a plot of the overall thermal resistances for both condensers for three series of runs, two with variable water side (inside the tube) resistances and one with a variable condensate. In each case the fluid resistance on the opposite side of the tube was reasonably constant.

Consider the steam condenser performance represented by the lowest line of Fig. 3.2. An increase in the water rate through the tube caused a lower water film resistance R_c and, with the same overall temperature difference, an increase in the condensate rate would occur. The increased condensate rate would cause an increase in the thermal resistance R_h on the condensate side. R_h may increase as R_c decreases in such a manner as to cause R (the overall resistance) to be a linear function of $1/w^{0.8}$. To allow for some variation in the "reasonably constant" residual resistance, an equation of the form

$$\frac{1}{U} - R_w = R_t - R_w = R_h + R_c = C_1 \Gamma^{1/3} + C_2 / W^{0.8} \qquad \ldots 3.15$$

was used. C_2 is obtained from the slope of the line if the line is straight, or from the mean between two successive points if the line is curved. Overall resistance (R_t) and condensate rate (Γ) for either point are then used to obtain C_1. Such a procedure appears desirable when the resistances R_c and R_h are of the same order of magnitude and the procedure for maintaining one of the resistances constant is highly complicated.

Another plot was made, using the condensate film coefficients for all tubes determined in this fashion from the overall coefficients, between $\frac{h}{C_p \Gamma}\left(\frac{3 \mu \Gamma}{g \gamma^2}\right)^{1/3}\left(\frac{C_p \mu}{k}\right)^{2/3}$ and $4\Gamma/\mu$ for Freon-11 which is a good correlation with the line obtained from the predicted values based on the Nusselt's analysis for top tube.

A plot between $h\left(\frac{\mu^2}{K^3\gamma^2 g}\right)^{1/3}$ and $4\Gamma/\mu$ showed that the heat transfer coefficients with the Freon-11 to be higher for the top three tubes than indicated by Nusselt top tube line. It also showed, however, that the transfer rate for the middle tubes of the bank were less than that indicated by the Nusselt top tube line and that the lowest tubes of the bank had transfer rates greater than those indicated by this line.

For the steam condenser the improvement of the transfer rate by the spilling of the condensate onto a lower tube is not apparent untill the eighth to tenth tube of the bank. This may be a result of the higher surface tension of the water which would decrease the height of fall of drops or intermittent streams since the tube spacing was the same for each condenser.

3.6.Dropwise Condensation on a Metal Surface.

When a condensing surface is contaminated with a substance which prevents the condensate from wetting the surface, the vapour will condense in drops rather than as a continuous film. This is known as dropwise condensation. Dropwise condensation has been obtained with mixtures of steam and other vapours, but steam is the only pure vapour for which conclusive evidence of dropwise condensation is available. Numerous investigators have examined the phenomena and established that, under conditions of chemical cleanliness, water will condense on a metal surface as a continuous film. It is essential for dropwise condensation that some promoter, usually a fatty acid or certain other types of organic compound, be present on the surface, and these will give "lives" of purely dropwise condensation for limited lengths of time. Polishing of the metal surface, the presence of a small quantity of non-condensable gas, the temperature of the surface, and the rate of condensation also have appreciable effects on the life of a surface giving purely dropwise condensation, as discussed below.

Drew, Nagb and Smith[12] investigated the conditions under which dropwise condensation of steam would occur. They confirmed the reports in the literature that dropwise condensation is induced and maintained more easily on smooth than on rough surfaces. Generally, steam contains some contaminants, apparently inactive as drop-promoters for a given metal, which turn out to be highly effective after that metal has been polished or even slightly smoothed. For the easy promotion of dropwise condensation the condensation surfaces should be as smooth as possible, though film condensation may occur on highly polished surfaces in the absence of contaminants.

The effects of specific organic materials on the type of condensation were studied on several types of condensers, supplied some times with boiler steam and some times with pure steam. A very small amount of the organic compound under consideration was rubbed on the surface of the condenser tube or was injected into the steam chamber. Copper, brass, monel, nickel, chromium plate, chrome-nickel steel, mild steel, and aluminium were used as the testing surfaces for promotion of dropwise condensation. The general conclusions from the results of the above experiments for promotion of dropwise condensation may be condensed as follows:-

1. For all metals, mineral oils are ineffective or only slightly effective in promoting dropwise condensation.
2. For all metals, the higher fatty acids are very effective. Beeswax and the fatty binders in the commercial emery and Tripoli buffing compounds acted similarly.
3. Mercaptans, Xanthates, and dithiophosphates are very effective on copper and the copper alloys.

4. For all metals, the alcohols, organic Nitrogens compounds and organic halides tried are ineffective or only slightly effective.
5. Mild steel and aluminium could not be effectively treated to maintain dropwise condensation, although it could be procured temporarily on fresh surfaces by treatment with fatty acids.

In general, the quantity of a drop-promoter needed to maintain non-wettability of a condensing surface is minute. Presumably a mono-molecular layer of the active substance on the metal is sufficient.

Hampson et al[14] have investigated on the condensation of steam and found the following particular conditions necessary for long life of dropwise condensation: (1) initial cleanliness of the metal surface before applying the promoter, 2) a perfectly dry surface, and 3) application of the promoter in a solvent the solution strength used was solvent to promoter 10/1. It was found that some period of time was necessary -about half an hour-after introduction of steam to the apparatus before the surface assumed its 'ideal' appearance of dropwise condensation; but, once attained, this condition persisted for sometime before finally breaking up into the mixed type. The conditions favouring a long life of dropwise condensation were examined systematically, and it was found that a chromium-plated surface applied with a mixture of oleic acid and a little quantity of light lubricating oil, which had been buffed to a mirror polish gave a life of 40-60 hours, while an addition of 0.1 per cent of air to the incoming steam increased the life of this surface to something more than seven days, and the same effect on a polished brass surface was obtained with 0.15 per cent nitrogen in the steam.

Hampson et al[14] found that the values of the steam-side coefficients ranged from 12,000 to 27,000 BTU/F^2-hr-^{o}F, the lowest value for the horizontal position and the highest value for the vertical position.

In their investigations they found further that the effect of tube cleanliness is considerable depending on the time for which the condenser was used. The effect of water velocity on the overall heat transfer coefficient was very high, and it increases as the velocity of water increases, while a little change was observed in overall coefficient for filmwise condensation by increasing the water velocity beyond 15 fps. A comparision of steam side coefficients for the two types of condensation showed that dropwise condensation was 12 to 13 times more effective than film-wise condensation.

It was found that the steam side coefficient increased by 12 percent when the steam-chest pressure was increased from 2 to 10 inches on the water gauge, it was suspected that air diffused in some way onto the plate surface at the lower pressures. So, a further series of tests were undertaken to determine the magnitude of the effect of small quantities of inert non-condensable gas (Nitrogen was used) in the incoming steam. The experiments showed that the effect of the gas was much more for dropwise condensation compared to filmwise condensation. The reason for the decrease in the coefficient is due to the gas accumulation through which the vapour must diffuse to the surface of the condensate film.

CHAPTER -4

HEAT EXCHANGER ANALYSIS AND ITS APPLICATION TO PRACTICAL DESIGN.

ABSTRACT.

Thermal analysis, via LMTD approach, of heat exchanger is made. The correlation of experimental data with the thermal analysis is presented. The design, both qualitatively and quantitatively, of heat exchanger is discussed. Finally, the design of a double-pipe condensing heat exchanger is illustrated.

4.1. <u>Basic Types of Heat Exchangers.</u>

A heat exchanger is a device which effects the transfer of heat from one fluid to another. There are three types of exchanger systems as follows:

i) The conventional <u>direct-transfer types</u> in which the two fluids exchanging thermal energy are separated by the heat transfer area which are also named as recuperators.

ii) The <u>liquid-coupled indirect-transfer type</u> which consists essentially of two direct-transfer units coupled with a pumped heat transfer medium. The transfer fluid circulates between the hot fluid exchanger, where the thermal energy is picked up, and the cold fluid exchanger, where the thermal energy is used to heat the cold fluid.

iii) <u>The periodic-flow type</u>, like the common Ljungstrom air-preheater, consists of a matrix heat transfer surface which is rotated so that an element is periodically passed from the hot to the cold flow streams and back again. As the hot fluid passes through the matrix the fluid is cooled and the matrix is heated, and in the cold side part of the cycle the cold fluid is heated and the matrix cooled. ~~This~~ These types of exchangers are also called as regenerators.

In the following sections, the discussion will be limited to the first type, that is direct-transfer type or recuperators.

4.2. <u>Recuperators.</u>

There are many forms of recuperators ranging from a simple pipe-within-a-pipe with a few square feet of heat transfer surface upto a complex surface condensers and evaporators with many thousands of square feet of heat transfer surface. In between these extremes is a broad field of common shell-and-tube exchangers.

The simplest type of shell-and-tube heat exchanger is shown in Fig. 3.1. It consists of a tube or pipe located concentrically inside another tube which forms the shell for this arrangement. One of the fluids flows through the inner tube, the other through the annulus formed between the inner and the outer tube. Since both fluids streams transverse the exchanger only once, this arrangement is called a single-pass heat exchanger. If both fluids flow in the same direction, the exchanger is a parallel flow type; if the fluids move in opposite directions, the exchanger is of the counterflow type. The temperature difference between the hot and the cold fluid is, in general, not constant along the tube, and the rate of heat flow will vary from section to section.

When the two fluids flowing along the heat transfer surface move at right angles to each other, the heat exchanger is of the cross flow type. Three separate arrangements of this type of exchanger are possible. In the first case each of the fluids is unmixed as it passes through the exchanger and, therefore, the temperature of the fluids leaving the heater section are not uniform, being hotter on one side than the other. In the second case, one of the fluids is unmixed and the other is perfectly mixed as it flowsthrough the exchanger. The temperature of the mixed fluid will be uniform across any section and will vary only in the direction of flow. In the third case, both of the fluids are mixed as they flow through the exchanger.

In order to increase the effective heat transfer surface area per unit volume, most commercial heat exchangers provide for more than a single pass through the tubes, and the fluid flowing outside the tubes in the shell is routed back and forth by means of baffles.

4.3. Heat Exchanger Analysis.

Two approaches, are, generally followed in the analysis of heat exchangers,viz. i) Log-mean rate equation approach,

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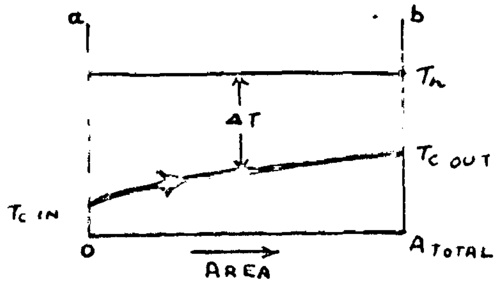
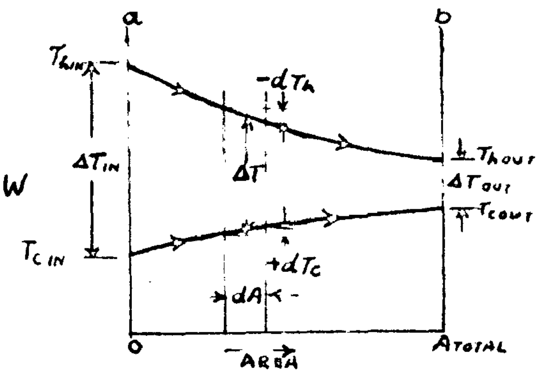


FIG. 4.1 SINGLE-PASS CONDENSER

FIG. 4.2 SINGLE-PASS PARALLEL-FLOW HEAT EXCHANGER



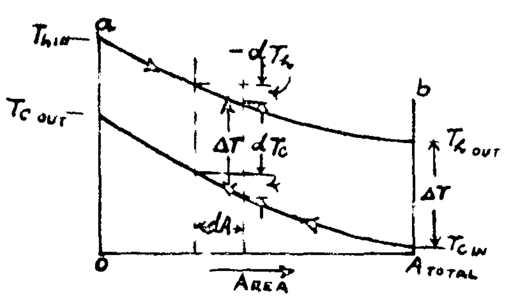


FIG. 4.3 SINGLE-PASS COUNTER-FLOW HEAT EXCHANGER

2) Effectiveness and Number of exchanger heat transfer units approach. Only the first approach is discussed now for a simple type of heat exchanger.

Log-mean Rate Equation Approach.

In section 3-2, the local over-all coefficient of heat transfer, U, has been defined and the local rate of heat transfer over a surface area of dA is given by the expression

$$dq = U\,dA\,(\Delta T) \quad \ldots\ 3.3.$$

The temperatures of fluids in a heat exchanger are generally not constant, but vary from point to point as heat flows from the hotter to the cold fluid. Even for a constant thermal resistance, $1/U$, the rate of heat flow will, therefore, vary along the path of the exchangers because its value depends on the temperature difference between the hot and the cold fluid at the section. Figures 4-1, 4-2, and 4-3 illustrate the changes in temperature that may occur in either or both fluids in a simple shell-and-tube exchanger (Fig. 3.1). The distances between the solid lines are proportional to the temperature differences ΔT between the two fluids.

Fig. 4-1 illustrates the case where a vapour is condensing at a constant temperature while the other fluid is being heated. Figs 4.2 and 4.3 represent the conditions in a parallel-flow and counterflow exchangers.

To determine the rate of heat transfer in any of the aforementioned cases the Eq. 3.3 must be integrated over the heat transfer area A along the length of the exchanger. If the overall unit conductance, U, is constant, it changes in kinetic energy are neglected, and if there are no heat losses, Eq. 3.3 can easily be integrated analytically for parallel and counterflow. An energy balance over a differential area dA yields

$$dq = m_h C_{ph} \, dT_h = \pm m_c C_{pc} \, dT_c = U dA (T_h - T_c) \qquad \ldots\ldots 4.1$$

where plus sign in the third term applies to parallel flow, and the minus sign to counterflow. If the specific heats of the fluids do not vary with temperature, we can write a heat balance from the inlet to an arbitrary cross section in the exchanger, or

$$-C_h (T_h - T_{h\,in}) = C_c (T_c - T_{c\,in}) \qquad \ldots\ldots 4.2$$

where $C_h = m_h C_{ph}$; $C_c = m_c C_{pc}$

Solving Eq. 4.2 for T_h gives

$$T_h = T_{h\,in} - \frac{C_c}{C_h} (T_c - T_{c\,in}) \qquad \ldots\ldots 4.3$$

from which we obtain

$$T_h - T_c = -\left[1 + \frac{C_c}{C_h}\right] T_c + \frac{C_c}{C_h} T_{c\,in} + T_{h\,in} \qquad \ldots\ldots 4.4$$

Substituting Eq. 4.4 for $(T_h - T_c)$ in Eq. 4-1 yields after some rearrangement

$$\frac{dT_c}{-\left[1 + \frac{C_c}{C_h}\right] T_c + \frac{C_c}{C_h} T_{c\,in} + T_{h\,in}} = \frac{U dA}{C_c} \qquad \ldots\ldots 4.5$$

Integrating Eq. 4.5 over the entire length of the exchanger (i.e. from A = 0 to A = A total) yields

$$\mathrm{Log}_e\left[\frac{-\left(1+\frac{C_c}{C_h}\right)T_{c\,out}+\frac{C_c}{C_h}T_{c\,in}+T_{h\,in}}{-\left(1+\frac{C_c}{C_h}\right)T_{c\,in}+\frac{C_c}{C_h}T_{c\,in}+T_{h\,in}}\right]=-\left(\frac{1}{C_c}+\frac{1}{C_h}\right)UA \quad \text{.... 4.6}$$

Equation 4.6 can be simplified to read

$$\mathrm{Log}_e\left[\frac{\left(1+\frac{C_c}{C_h}\right)\left(T_{c\,in}-T_{c\,out}\right)+\left(T_{h\,in}-T_{c\,in}\right)}{T_{h\,in}-T_{c\,in}}\right]=$$

$$-\left[\frac{1}{C_c}+\frac{1}{C_h}\right]UA \quad \text{.... 4.7}$$

From Eq. 4.2 we obtain for the total length of the exchanger

$$\frac{C_c}{C_h}=-\frac{T_{h\,out}-T_{h\,in}}{T_{c\,out}-T_{c\,in}} \quad \text{.... 4.8}$$

which can be used to eliminate the hourly heat capacities in Eq. 4.7. After some re-arrangement we get

$$\mathrm{Log}_e\left[\frac{T_{h\,out}-T_{c\,out}}{T_{h\,in}-T_{c\,in}}\right]=\left[\left(T_{h\,out}-T_{c\,out}\right)-\left(T_{h\,in}-T_{c\,in}\right)\right]\frac{UA}{q} \quad \text{.... 4.9}$$

since $q = C_c\left(T_{c\,out}-T_{c\,in}\right)=C_h\left(T_{h\,in}-T_{h\,out}\right)$

Letting $T_h - T_c = \Delta T$, Eq. 4.9 can be written

$$q = UA\,\frac{\Delta T_a-\Delta T_b}{\mathrm{Log}_e\left(\frac{\Delta T_a}{\Delta T_b}\right)} \quad \text{.... 4.10}$$

where the subscripts a and b refer to the respective ends of the exchanger. In practice it is convenient to use an average effective temperature difference $\overline{\Delta T}$ for the entire heat exchanger defined by

$$q = UA\,\overline{\Delta T} \qquad \ldots 4.11$$

Comparing Eq. 4.10 and 4.11, one finds that, for parallel or counterflow

$$\overline{\Delta T} = \frac{\Delta T_a - \Delta T_b}{\log_e\left(\frac{\Delta T_a}{\Delta T_b}\right)} \qquad \ldots 4.12$$

which is called the logarithmic mean over all temperature difference often designated as LMTD. The LMTD also applies when the temperature of one of the fluids is constant, as shown in Fig. 4.1.

The use of logarithmic mean temperature is only an approximation in practice because U is not constant. In general, U is treated as constant for design work. If U varies considerably, a numerical step-by-step integration may be necessary.

If the temperature difference ΔT_a is not more than 50 percent greater than ΔT_b, the arithmatic mean temperature difference will be within 1 percent of LMTD and may be used to simplify calculations.

For more complex heat exchangers such as the shell-and-tube arrangements with several tube or shell passes and with cross flow exchangers the mathematical derivation of an expression for the mean temperature difference becomes quite complex. The usual procedure is to modify the simple LMTD for counterflow by correction factors (F) which have been dealt in great detail for different types in reference 25.

4.4. <u>Heat Transfer in double-pipe Heat Exchangers.</u>

In the previous sections, a thermal analysis of heat exchangers has been discussed. Numerous investigators conducted experiments on different types of heat exchangers and the results were based, generally, on the methods of analysis as discussed earlier. In this and the following sections, the results of their data and their application to practical design of heat exchangers are presented.

In 1929, Stewart and Holland[35] conducted tests on double-pipe cooler and condenser using ammonia as the working fluid. The apparatus consisted of 8-1¼ in. pipes inside 2 in. pipes, 10 ft. long, arranged in the form of coils. The ammonia vapours entered through the annular space and cooling water entered through the inner tube, the fluids being arranged to have a counter flow. The cooling water was circulated with an inlet temperature of 60,70,80 and 90°F and with a consumption rate of 5,10,15 and 22 gallons per min. The ammonia to be condensed was then so adjusted for each of the above temperatures that a pressure corresponding to a saturation temperature of 2°, 4, and 6°F above the temperature of water leaving the condenser, resulted.

The condenser tonnage was calculated from the rate of condensate flow and the initial and final enthalpies of ammonia. The MTD was determined by plotting actual temperature of ammonia and water as ordinates, and position along the pipe as abscissa. By finding the inclosed area for a given run the mean coordinate in degrees, and the mean temperature difference $\overline{\Delta T}$ as determined above was checked by the Eq.4.13.

The mathematical relation of mean temperature difference and U was expressed by the equation

$$U = \frac{C}{H.T.D.^{0.7}} \qquad 4.13$$

where C is a constant

When velocity is taken into account C varies with the velocity of water ($V f.p.m.$)as:

$$C = 230 + V/1.22 \qquad 4.14$$

U was then expressed as

$$= \frac{(230 + V/1.22)}{(H.T.D.)^{0.7}} \quad .... \quad 4.15$$

The Eq. 4.15 was in good agreement with the results obtained except for low velocities. From the results obtained, the following conclusions were made :(1) The condenser tonnage is a function of the water rate and the H.T.D. (2) The ammonia film surface coefficient of conductance varies as U varies, and inversely as some function of the total quantity of heat transfer. The film thickness varies as some function of the weight of ammonia condensed per square foot (3) The coefficient of heat transfer varies as the velocity to the 1.22. (4) The coefficient of heat transfer varies inversely as the mean temperature difference to the 0.7 power.

The mean temperature difference, condenser tonnage, U, and velocity were plotted as a common set of coordinates, the condenser tonnage as abscissa, which resulted the performance curves which can be used for the design of a similar type of exchangers.

4.5. Heat Transfer in a Multitube-Multipass Condenser.

Kratz-et-al[23] investigated the heat transfer in a multitube-multipass type of condenser, which consisted of two shells placed horizontally containing seven tubes of 12.82 ft. long. The tubes were made of charcoal iron, and each tube had an outside diameter of 2.00 in. and an inside diameter of 1.31 in. The tubes were arranged in the shell so that the water passed through the seven tubes in series. The two shells were connected in parallel and equal amounts of water passed through the tubes in each shell. A series of tests were conducted with varying pressure of ammonia.

The ammonia-film heat transfer coefficient as determined from the Wilson plot, was 1635 BTU/hr/sq.ft./°F. The water film coefficient was found, from the same plot, to be, $h_w = 211\ v^{0.8}$. The authors found out an emperical relation, based on the dimensional analysis and taking into consideration the effect of entrance of water into the tubes, which was expressed as

$$h_w = \frac{0.02965\, K_w}{D_i}\left(1+\frac{50}{r}\right)\left(\frac{\rho D_i v}{\mu}\right)^{0.8} \quad \ldots 4.16$$

where r = ratio of actual tube length to actual inside diameter and k_w = Conductivity of water, BTU/hr/sq.ft./oF per ft. of thickness.

From the above equation, the value of h_w for the condenser tested, was found to be 194 $v^{0.8}$. The calculated values of U, based on the above result, were within limits of about 10%, which is probably an acceptable degree of accuracy for the practical purpose of condenser design. The overall coefficients of heat transfer varied from 130 to 450 as the pressure of ammonia increased from 125 to 175 psi.

Zumbro[40] conducted tests on a vertical shell and tube type condenser using ammonia as the working fluid. The condenser was 36 in. outside diameter and contained 108-2 in. seamless steel tubes of No - 10 gauge and 14 ft. long. The number of tubes in the condenser was varied from time to time by plugging the tops with wooden plugs. The tests were conducted with initial condensing water temperatures of approximately 85°, 75°, 65° and 55°F and with varying vapour pressures and water velocities.

From the actual test data, it was found that the overall heat transfer coefficient increases as a straight line function of the quantity of condensing water per tube and plot of the tons of refrigeration and condenser pressure also showed a straight line relationship. This type of condenser is capable of handling large over loads by increasing the supply of water, and this is obtainable due to gravity fall without any additional expenditure of power to force the water through condensor, although additional horse-power is required to pump the water against the static head. The test data can be used conveniently for the design purposes and would also enable one to prophesy, with reasonable accuracy, the performance under various conditions.

4.6. Design Aspects of Horizontal Type Refrigerant condensors.

In this section the design of horizontal type refrigerant condensers is discussed both qualitatively and quantitatively.

The process of condensing vapours on cold surfaces is one which has been very susceptible to theoretical attack, and great progress has been made with such analytical approaches, beginning as far back as 1916 when Nusselt first formulated the mathematics involved in condensing pure vapours. As greater understanding has been developed of processes of flow, diffusion, and heat transfer in general, more precise methods of calculation of heat transfer in condensors of both pure and mixed vapours have been possible. At the same time, these nicetics have made the problems more complex so that, from the design aspect, the precise procedures have been undesirable in consumption of time, as a result there has been a need for approximat methods which could be solved readily and which would yield results sufficiently accurate for most purposes.

<u>Qualitative Analysis.</u>

In the design of heat exchangers, a number of factors have to be specified. Usually the apparatus is designed to meet the requirements of the safety codes of the ASME. The following aspects are to be considered in the design of a condenser.

A shell-and-tube exchanger with fixed tube sheets is suitable only for those cases in which the fluid in the shell does not foul the tubes. The large differences in temperature between the shell and tubes may produce severe temperature stresses, and when this is encountered, a type of construction that will allow independent expansion of either the shell (an expansion joint) or tubes (floating heads) should be provided. Alternatively, temperature stresses may be avoided by the use of U -shaped tubes, which, however, are more difficult to clean inside by mechanical means. If frequent cleaning is required, the construction employed should facilitate this. For this reason many exchangers are provided with removable tube bundles.

If one of the fluids fouls the surface much more rapidly than the other, it should be routed through the tubes, since the inside surface may be cleaned without removing the tube bundle from the shell. If both fluids are equally non-fouling and only one is under high pressure, it should flow inside the tubes to avoid the expense of high pressure shell. Where only one of the fluids is corrosive, it should flow inside the tubes to avoid the expense of special metal for both shell and tubes. If one of the fluids is much more viscous than the other, it may be routed through the shell to increase the overall coefficient. At times, because of a limitation in available pressure drop, it is necessary to route a given stream through the shell.

Although tube lengths of 4 to 22 ft. or more are obtainable , one often selects a standard length of 8, 12, or 16 ft, using more than one pass where necessary, depending on the available space and when large shell diameters are required. The tube diameter is to be selected depending on the fouling factors and velocity of the fluid to be used. In some cases fouling may be reduced by use of high velocity, thus permitting the use of tubes of moderate diameter. The thickness of tube wall should be selected not only to withstand working pressures and extreme temperatures and to provide allowance for corrosion but also to facilitate expanding the tubes into the tube sheets.

The baffles, used to increase velocity on the shell side, should have less clearance with the shell to reduce the leakage. Sometimes the minimum clearance between the tubes and edges of holes in the baffles is 1 percent of the tube diameter, but since tubes may have a diameter tolerance of 2 percent, the maximum clearance may be 3 percent; consequently, some fluid will flow through these clearances. The baffle is a fixture, whereas the tubes can be replaced individually, and the baffles may be subjected to considerable abuse when the tube bundle is being withdrawn; consequently the baffle should be at least twice as thick as the wall of the tube. Sufficient space between baffles should be provided to facilitate cleaning.

The initial cost of a tubular heat exchanger, expressed per square foot of heat-transfer area, depends upon a large number of factors; the total heat-transfer surface, the tube size and length, the material of construction of the tubes and the shell, the working pressure, the degree of baffling, and the cost of special features, if any.

Quantitative Analysis for the Design.

In the previous chapters, quantitative relations, either derived analytically or correlated from experimental data, have been given which can be used in the design of tubular heat transfer apparatus such as heaters, coolers, and condensers. In many cases the designer is free to employ the optimum velocity at which the total costs are at a minimum; however, this may lead to impractical proportions of apparatus. At times, because of process requirements or for other reasons a fixed pressure drop is available, and the exchanger is designed to meet this situation. Alternatively, one may fix the tube length and diameter and use whatever velocity and pressure drop are necessary. Regardless of which factors are fixed in advance and

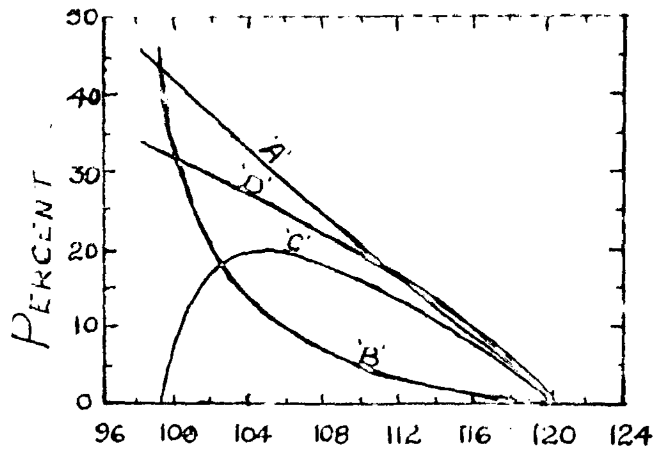


FIG. 4.4 OPTIMUM CONDENSING TEMPERATURE ON A SPECIFIC FREON-12 CONDENSING UNIT.

SUCTION PRESSURE CORRESPONDING TO 40°F WITH WATER OFF AT 85°. PERCENTAGE INCREASE IN TONS WAS COMPUTED ON CONSTANT MOTOR LOADING, AT ALL CONDENSING TEMPERATURES, WITH VARIABLE SPEED.

CURVE A INCREASE IN TONS — PERCENT
CURVE B. INCREASE IN UNIT COST — PERCENT
CURVE C. DECREASE IN UNIT COST PER TON — PERCENT
CURVE D. DECREASE IN BRAKE H-P. PER TON — PERCENT

which are computed, the same basic relations are always involved.

The following section presents the application of heat transfer data to practical design of horizontal shell type refrigerant condenser, as presented by Pownall[28], using Freon-12 as refrigerant. Considerable research was done on heat transfer coefficients in condensers, but their application was largely empirical. The practice was to establish a fixed number of square feet per ton of refrigeration, regardless of other influencing factors. Purchase of a condensing unit is on the basis of the tonnage which the unit can deliver under a given set of conditions For a given motor loading, at a fixed water temperature leaving the condenser, the unit operating with the lowest condensing temperature is the one capable of producing the most refrigerating capacity.

Fig. 4.6 shows that condenser performance means to a specific F-12 condensing unit at a fixed evaporator temperature and a fixed leaving condenser water temperature. Curve A shows the percentage increase in tons which can be expected as the condensing temperature is decreased from the selected top limit of 120°F. Curve B shows the percentage increase in unit cost which would be expected using this particular type surface in order to obtain the various condensing temperatures.

By subtracting curve B graphically from curve A, curve C is obtained, showing the percentage decrease in the cost per ton , as the condensing temperature balances below 120°F.

The best cost per ton in this case is 103°F to 107°F where it is practically constant. Curve D shows the percentage decrease in brake h.p. per ton as the condensing temperature is reduced below 120°F. If this were operated at 107°F, condensing temperature, the cost per ton would be decreased approximately by 10% and the b.h.p. per ton would be decreased by about 23% of their respective values at 120°F. If the unit were operated at 103°F, the decrease in these values will be respectively 10% and 28%. Thus, in reducing the condensing temperature from 107°F to 103°F, there would be no gain in cost per unit ton, but initial cost would increase about 12%. The power decrease resulting from the 4°F reduction will be approximately 5%. The cost of power would be influential in determining whether the higher initial unit cost could be justified.

In order to make such comparisons, one must have some means for analyzing condenser performance. Pownall[28] presented one method. All analyses were made on the basis of a single tube in order to combine the variables of condenser length and pass arrangements for a multi-pass shell and tube condenser. Also, sq.ft. per ton was used to take care of loading, so that for any given duty the number of tubes per pass may be determined readily for any size condenser.

Terminal difference, or difference between temperature corresponding to condensing pressure and outgoing cooling water, was used as the basis of comparison for condenser performance, since by using this value the effect of g.p.m. per ton, or water range, may be segregated. In all calculations, a heat removal in the condenser of 15000 BTU per hr. per ton was used; this value represents an average condition for common refrigerants.

Effect of Water Pressure Drop.

Overall heat transfer will improve as tube diameter is decreased, but to arrive at an economical surface for practical use, it is necessary to carry analysis considerably further than merely obtaining heat transfer values. Condenser performance can be measured by the terminal difference obtainable for a given square feet of condensing surface per ton of refrigeration. On all heat transfer surfaces, water pressure drop may limit the maximum permissible water velocity, which should be utilised in selecting the arrangement of passes.

Effect of Tube Diameter.

With the maximum allowable pressure drop and terminal temperature difference being fixed over a particular length of travel, the rate of flow of water can be determined from which the tube diameter can be selected. Comparison of performance of different diameters shows that the smaller the tube diameter, the shorter the total length of travel necessary in order to maintain the same terminal temperature difference and loading.

Effect of Scaling on Condenser Tubes.

Normal waters will have little or no effect upon non-ferrous tubes after a thin oxide film has been formed. However, waters sometimes termed "temporary hard water", or waters which will deposit salts upon heating can occasionally cause trouble. Maintaining higher velocities will help, as the water will have a tendency to sweep the salts out, as they are precipitated.

Scaling has a greater effect upon fin tube than on bare tube surface, so it should always be considered in the former case. In the case of non-ferrous tubes, the scale factor varies from 1000 to 4000. Ferrous tubes, unlike non-ferrous tubes, seem to have an increasing scale resistance, because of the continuous oxidation of inside tube surfaces, even with normal waters. Therefore, it is impossible to establish a fixed value for scale conductivity.

As the condenser tubes become more and more scaled or corroded, the resistance to heat transfer continually increases, so that for the same load the temperature difference in the condenser must be increased, causing a corresponding increase in head pressure and in compressor horse power. The effect of temperature difference may be reduced by cleaning the condenser, causing a corresponding reduction in power requirements, and an economical balance must be struck between the power saving and the cost of cleaning. Sherwood[31] derived a formula for the optimum period between cleanings, which may be stated as

$$X = \sqrt{\frac{2C}{PC'}}$$

where x = Optimum period between cleanings, days,

c = Total cost of each cleaning of the condenser,

c' = Power cost per h.p-day (same units as above)

p = Average daily necessary increase in power, due to the rusting of the condenser tubes.

4.7. Design of Condensing Heat-exchanger.

In this section the design of condensing heat exchanger, capable of taking up a load of 3/4 ton of refrigeration at evaporator, is illustrated. Freon-12 is taken as the working fluid. It is assumed that the evaporator pressure is 20 psia ; condenser pressure 150 p.sia with a saturation temperature (t_s) of 109.448°F. The vapours leaving the compressor are taken to be at 200°F (t_{sup}) and the liquid is sub-cooled by 6°F in the condenser.

With the above assumptions, the refrigerant can take up a load of 57.75 BTU per pound, which requires the rate of flow of refrigerant to be 155.9 lb./hr.

The inlet temperature of the cooling water (t_1) is assumed to be 70°F. It happens that the temperature of the tube wall is above the saturation temperature of the pure vapour near the vapour inlet and below the saturation temperature at the outlet. Under these conditions, the vapour will be cooled without condensation in the first part of the apparatus, will condense in the second part, and will be sub-cooled in the latter part, the effect of which may be combined with second part since the sub-cooling is assumed to be very small. Since the coefficient of heat transfer in the desuperheating region is very small compared with that in the condensing region, the design of the heat exchanger is divided into two stages: desuperheating and condensing.

The heat transferred in the desuperheating region is

$$q_1 = W\,(H_{sup} - H_{sat})$$
$$= 155.9\ (104.25 - 87.8) = 2300\ \text{BTU/hr.}$$

while that in the condensing region is

$$q_2 = W\,(H_{sat} - H_{Liq})$$
$$= 155.9\ (87.8 - 32) \qquad = 8698\ \text{B/hr.}$$

The following data have been further assumed.

Type of exchanger: Double-pipe heat exchanger.

Inner tube: 3/8 in. O.D. copper tube.

(D_1) Diameter (inside) - 0.307 in. - 0.0253 ft.

(D_2) Diameter (outside)- 0.375 in. - 0.03125 ft.

Wall thickness- 0.034 in. - 0.002833 ft.

Fluid- Freon-12.

Jacket Tube: 3/4 in. O.D. Copper Tube.

(D_S) Diameter (inside) = 0.672 in. - 0.056 ft.

(D_o) Diameter (outside)= 0.75 in.- 0.0625 ft.

Annulus:

(D_e) Equivalent diameter = 0.297 in. = 0.02475 ft.

Fluid - cooling water.

Velocity of cooling water: 4 ft. per sec.

$\therefore$ Rate of cooling water $(w) = \frac{\pi}{4 \times 144} (0.672^2 - 0.375^2)$

$\times 4 \times 3600 \times 62.4$

$= 1521.6$ lb/hr.

After assuming the above data, the design is simplified to the determination of the length of the heat-exchanger. As stated earlier, the design is split up into two parts: desuperheating and condensing. So the lengths of heat exchanger required for the two regions are calculated separately and the sum of these two gives the total lengths of exchanger.

Desuperheating Region.

The coefficient of heat transfer in this section may be evaluated from the equation 2.1. Since the Prandtl numbers of common gases do not vary widely, the following equation is recommended by McAdams[26]

$$h_L = 0.0144\ C_p\ G^{0.8} / D_i^{0.2} \qquad \ldots.\ 4.17$$

provided the effects of natural convection are not important.

The heat transferred (q1) can be equated to

$$q_1 = h_L (\pi\ D_i\ L_1)\ \Delta t_L \qquad \ldots..4.18$$

At the condenser pressure, C_p = 0.1816 B/°F

and $G = \dfrac{165.9}{\pi/4 \times 0.0256^2}$

= 302,000 lb/hr.sft.

Substituting all the values in Eq. 4.17 the coefficient of heat transfer is found to be

h_L = 131.6 BTU/hr.sq.ft./°F

The heat transferred (q_1) can also be equated to that absorbed by cooling water, i.e.

$$q_1 = WC_p\,(t_2 - t_1) \qquad \ldots.\ 4.19$$

On substitution of the values in Eq. 4.19 the temperature of the cooling water at outlet (t_2) of the desuperheating section comes out to be :

t_2 = 71.71 °F

The long-mean-temperature difference (Δt_L) is calculated from the equation

$$\Delta t_L = \frac{(t_{sup} - t_1) - (t_{sat} - t_2)}{\log_e\left(\dfrac{t_{sup} - t_1}{t_{sat} - t_2}\right)} \qquad \ldots.\ 4.20$$

which yields a value of Δt_L = 74.6 °F.

Substitution of the above values in Eq. 4.10 gives the length (L_1) of heat-exchanger in the desuperheating region equal to 3.8 ft.

Condensing Region.

The length of heat exchanger in the condensing section is calculated using the equation (4-21) for which the overall coefficient of heat transfer is to be determined first, which, in turn, requires the determination of film coefficient to heat transfer of condensing Freon-12, and water. The first one is predicted from the Nusselt's equation, as given by 4.21 since the tube is assumed to be horizontal, while the second one is predicted from the equation 4.22, the value of which is increased by 15 percent, as recommended by Katz et al.[17] compared to that of Eq. 2.1.

Condensing Film Coefficient.

The film coefficient for a temperature drop of 34.805°F is first calculated and then modified by a trial and error method to take into account the temperature drop through the water film and copper wall.

This should be written after calculating ΔT.

The properties of F-12 are taken from the standard tables corresponding to a temperature $t_f = t_{sv} - \frac{3}{4}\Delta t$ where $\Delta t = t_{sv} - t_w$ while t_w is the average temperature of water and the value of λ corresponds to the saturation pressure i.e. 160 p.s.i.a.

The temperature rise of cooling water in the condensing region is $\frac{q_c}{UC_p} = \frac{8608}{1282.6 \times 1} = 6.7°F$. The inlet water temperature in the condensing region being t_2 71.71°F, the outlet water temperature (t_3) is 71.71 + 5.7 = 77.41°F

The average water temperature $t_w = \frac{t_2 + t_3}{2}$ = 74.53°F

$\therefore \quad \Delta t$ = 109.445 - 74.53 = 34.885°F

and t_f = 109.445 - $\frac{3}{4}$ x 34.885

= 82.295°F

The following values are taken at a temperature of t_f =82.295°F

$\therefore K_f$ = 0.0508 BTU/hr/ft./°F;

μ_f = 0.628 lb/hr-ft.

ρ_f = 81.158 lb/cft.

λ = 56.4 BTU/lb.

D_i = 0.0253 ft.

Substitution of the above values in the equation 4.21

$$h_{ave} = 0.725\left[\frac{K_f^3 \rho_f^2 g \lambda}{\mu_f D_i (t_f - t_s)}\right]^{1/4} \quad \text{.... 4.21}$$

(For $t_f - t_s$ = 34.885F)

gives h_{ave} = 312 BTU/hr. °F.sq.ft. of inside surface area.

Water Film Coefficient.

The water film coefficient of heat transfer is predicted from the equation

$$h_w = 1.15 \times (0.023)\left[\frac{D_e G}{\mu}\right]^{0.8}\left[\frac{C_p \mu}{K}\right]^{0.4} \quad \text{... 4.22}$$

The properties of water are taken at an average temperature $t_W = 74.56\,°F$

$G = 4 \times 3600 \times 62.4$

$= 900{,}000\ lb/ft^2/hr.$

$\mu = 2.2326\ lb/hr.ft.$

$K = 0.3477\ BTU/hr.ft.°F$

$C_p = 1\ BTU/lb.°F.$

Substitution of the values in Eq. 4.22 gives h_W based on the inside surface of the annulus = 1178 BTU/hr.ft.^{2}°F

Overall Coefficient of Heat Transfer.

The overall coefficient of heat transfer based on the ~~inside~~ outside surface area of annulus is calculated from the equation.

$$\frac{1}{U} = \frac{D_1}{D_i\, h_{ave}} + \frac{D_1}{2}\,\frac{\log_e D_1/D_i}{K_{COPPER}} + \frac{1}{h_w} \qquad 4.23$$

Substitution of the values already determined and $K_{copper} = 222$, in eq. 4.23 we get

$$\frac{1}{U} = 0.003915 + 0.00014 + 0.000849$$

$$= 0.004778$$

The temperature drop through each of the film is proportional to the resistance of each film, or the reciprocal of the heat transfer coefficient.

The temperature drop through the condensing film

$$= \frac{0.003916}{0.004778} \times 34.885 = 28.6 \text{ F.}$$

The final results of the trial and error solution for the overall coefficient and the condensing film coefficient are:

h_{avo} (at $(t_f - t_s) = 28.45$) = 322.5 BTU per hr.sq.ft.°F

$$\frac{1}{U} = \frac{0.375}{0.307 \times 322.5} + 0.000014 + 0.000349$$

$$= 0.004643$$

Temperature drop across condensing film =

$$\frac{0.00378}{0.004643} \times 34.885$$

$$= 28.45 \text{ F.}$$

∴ Overall coefficient of heat transfer for a mean temperature difference of 34.885F, based on the outside surface area of the tube is

$$U = \frac{1}{0.0046443} = 215.5 \text{ BTU per hr-sq.ft.-}^{\circ}\text{F}$$

The heat transferred, q_2, can be written as

$$q_2 = U(\pi D_1 L_2)\Delta t \qquad \ldots . 4.23$$

where L_2 is the length of the heat exchanger in the condensing region.

Substituting the values in Eq. 4.24 the length, L_2 is calculated to be

$$L_2 = 11.78 \text{ ft.}$$

The total length (L) of the condensing heat exchanger is equal to the sum of the lengths determined individually for desuperheating and condensing regions.

The total length of condensing heat exchanger,

$$L = 11.78 + 3.3 = 15.08 \text{ ft.}$$

The condensing heat exchanger has been fabricated for a length of 16.33 ft. and with the remaining particulars as assumed earlier. The sketch of the exchanger is given in the next chapter.

CHAPTER -5

TESTING OF CONDENSING HEAT EXCHANGER.

ABSTRACT.

This chapter deals with the description of the apparatus used for the experimental purpose along with the instruments used for the measurements. The procedure adopted for the experiment on condensation of Freon-12 vapours is also outlined. A summary of the data taken is presented. The method of calculation of heat transfer coefficients is indicated.

6.1. Scope of Investigation.

In order to obtain data of a more general nature, overall heat-transfer measurements were made for the condensation of Freon-12 vapours on the inside surface of a $\frac{3}{8}$ in. plain copper tube. The data were obtained to find out the effect of the following on the condensing-film coefficients and overall heat-transfer coefficients.

a) The temperature of the cooling fluid at inlet.

b) The velocity of the cooling fluid.

c) The direction of flow of the cooling fluid.

d) The average temperature difference between the condensing vapour and the cooling fluid.

In all the above cases, the cooling fluid, viz. water, was circulated in the annular space formed between 3/4 in. tube and 3/8" in. tube.

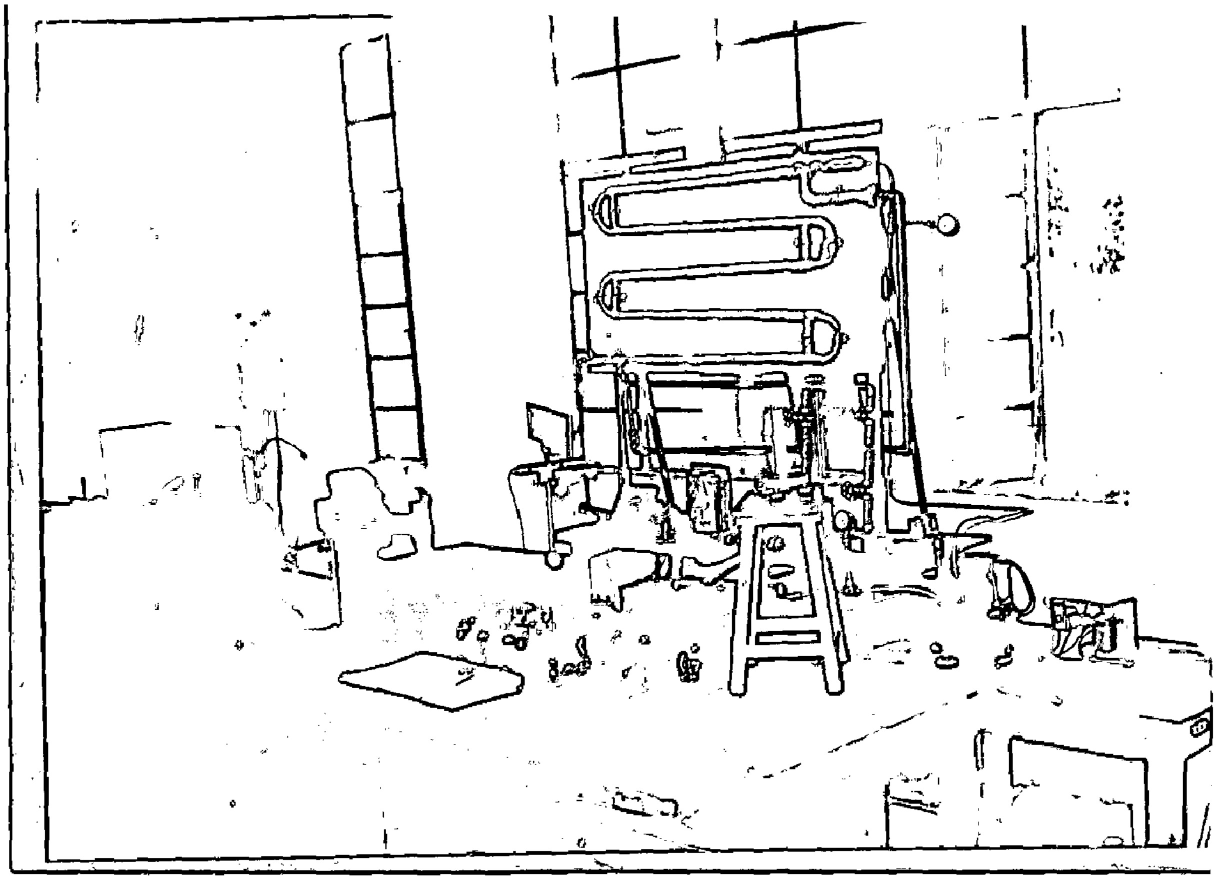
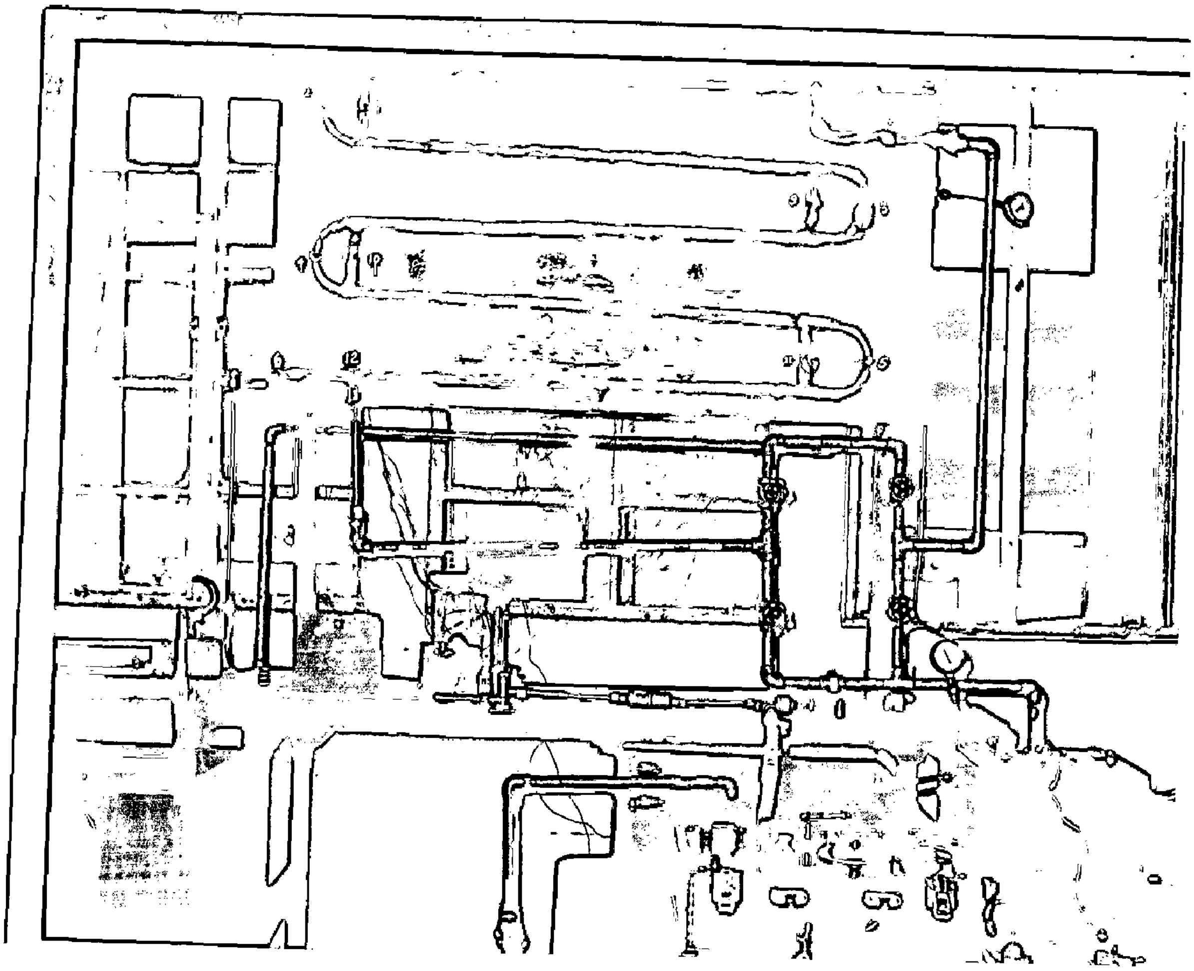


Fig. C-1 (top). Experimental Set up

Fig. C-2 (Bottom). Condensing Heat Exchanger.



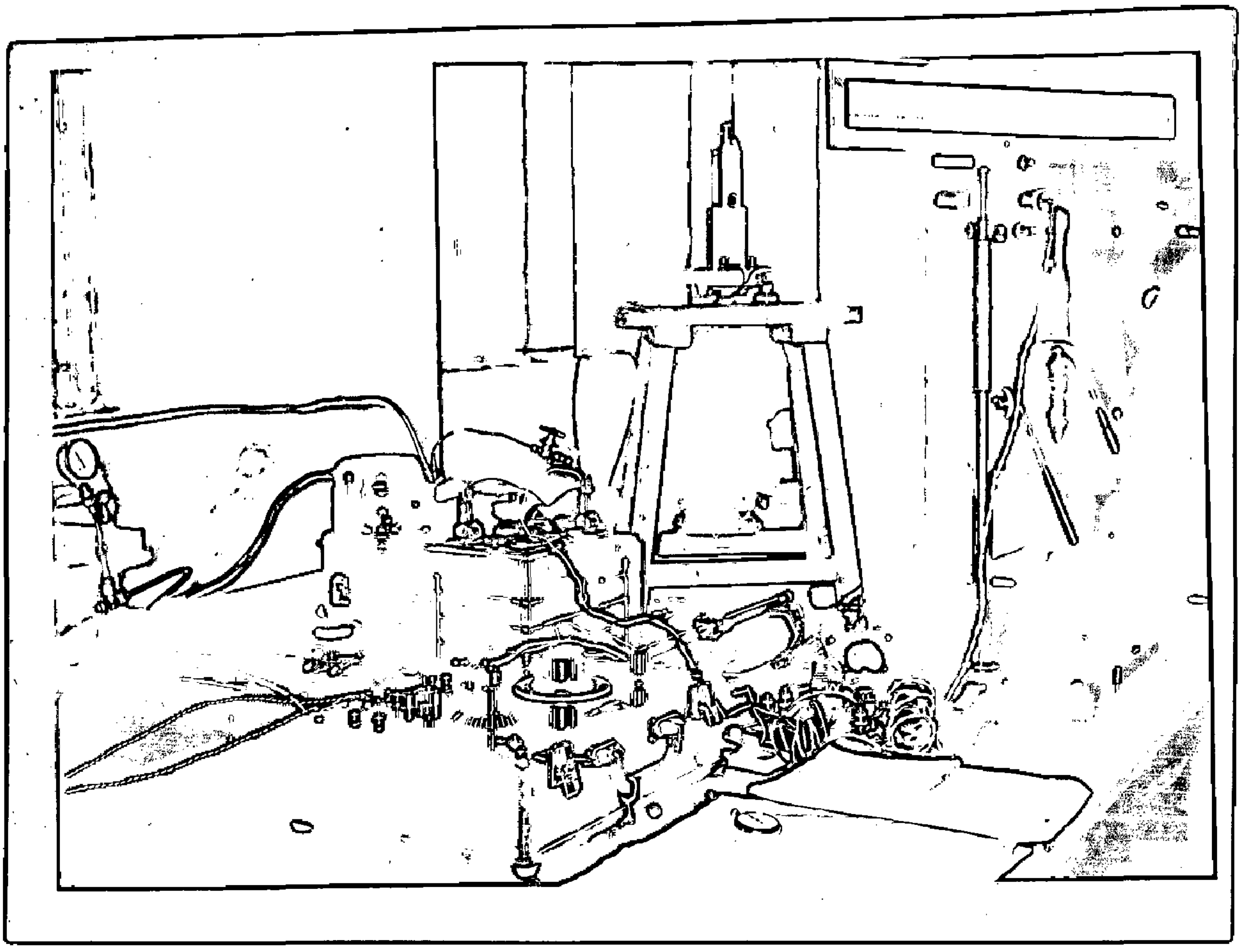


Fig. 3-8 Potentiometer and other Accessories

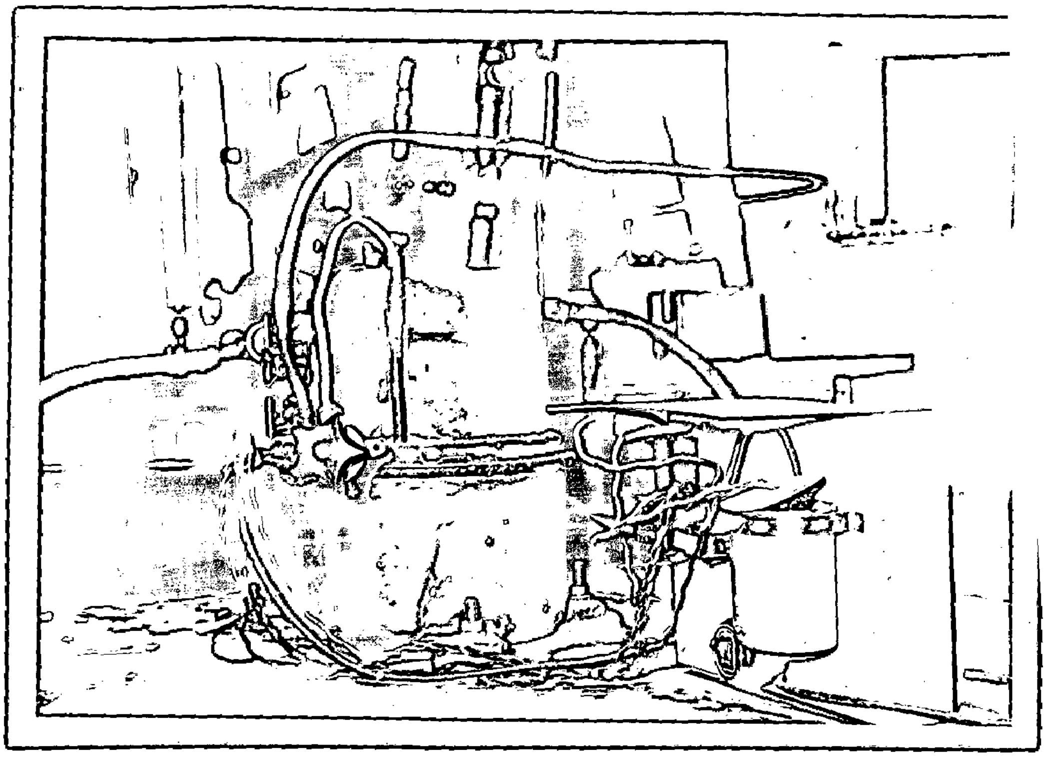
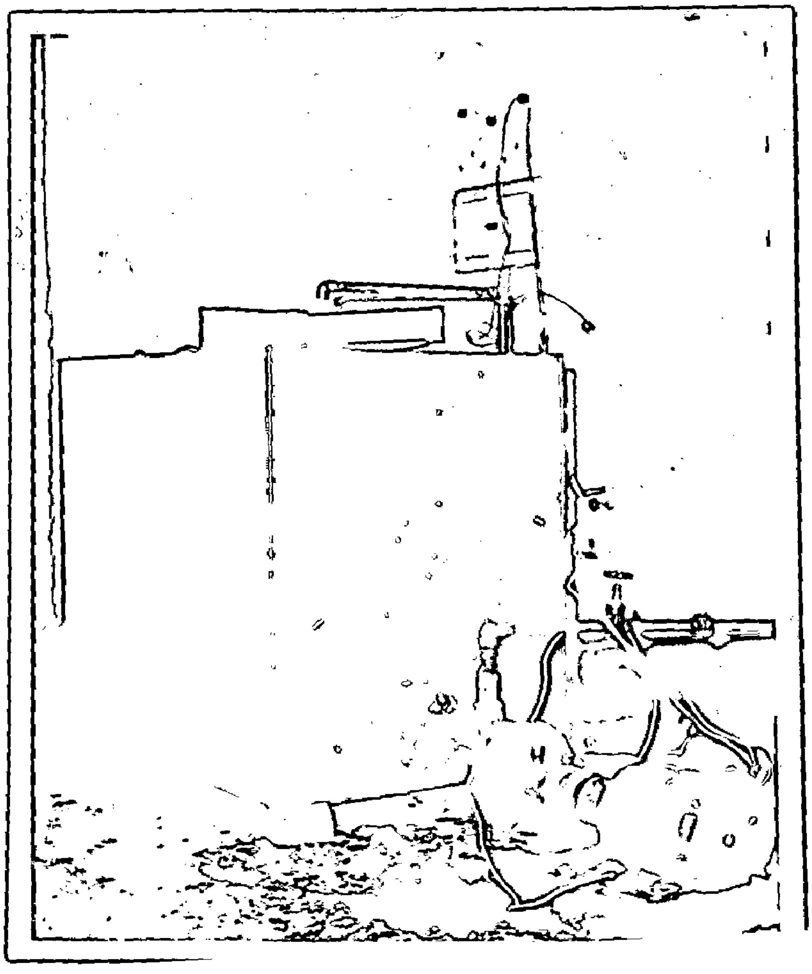


Fig. 8-4 (Top) Hermetically Sealed Compressor and its Controls.

Fig. 8-6 (Bottom) Water Tank and Circulating Pump.



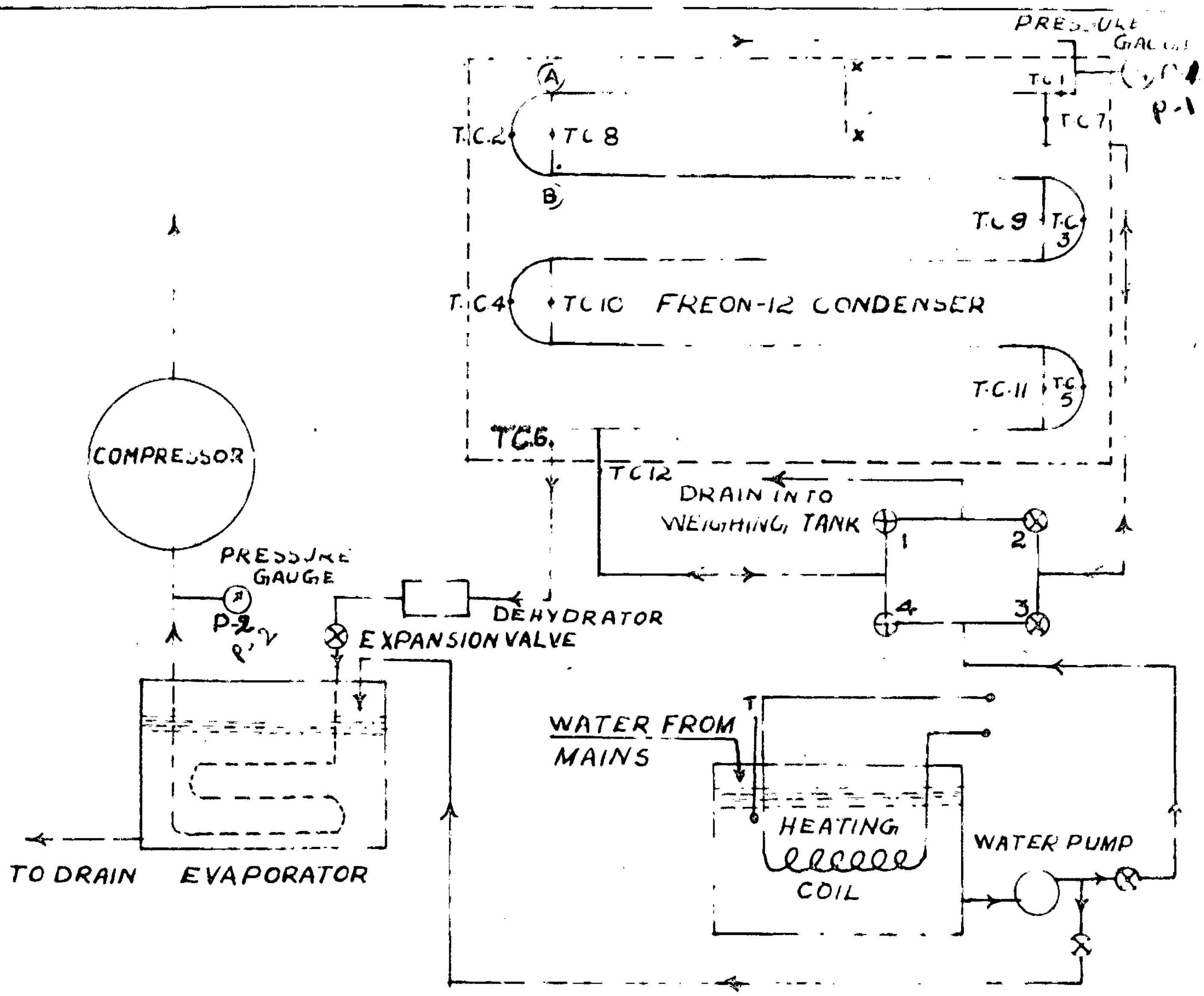
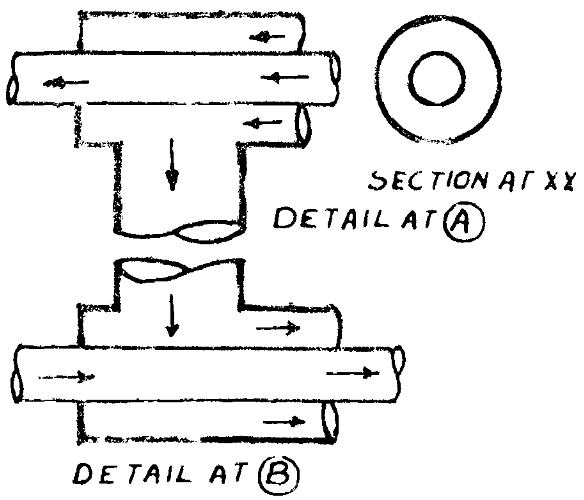


FIG. 5.6 FLOW DIAGRAM OF CONDENSING DOUBLE-PIPE HEAT EXCHANGER



Condensing-film coefficients were determined from the overall coefficients by the use of Wilson-type plot[33]. These condensing coefficients have been compared with the calculated values from the Nusselt's analysis[27].

The water-film coefficients were also determined from the slope of the Wilson-type plot and compared with the correlations already made by some investigators.

5.2. Experimental Apparatus.

The apparatus is shown in Fig. 5.1 through 5.5 and a diagrammatic flow sheet is given in Fig. 5.6. The equipment consisted of a condenser, expansion valve, evaporator, and compressor in a closed circuit connected by a high pressure vapour line, a condensate line, and a low pressure vapour line.

The condenser under investigation consisted of a double-pipe parallel or counter-flow coil type with 3/8 in. plain copper tube inside a 3/4 in. plain copper tube having an effective total length of 16 ft. 4 in. The complete details of the condenser are given in Table 5.1.

Table 5.1. Dimensions of Condenser.

Inner tube: 3/8 in. plain copper tube.

Inner diameter (D_i) = 0.307 in.
Outer diameter (D_1) = 0.376 in.

Outer tube: 3/4 in. plain copper tube.

Inner diameter (D_2) = 0.672 in.
Outer diameter (D_o) = 0.75 in.

Annulus.

Equivalent diameter = D_0 = $D_2 - D_1$

= 0.297 in. or 0.02475 ft.

Area based on the inside surface. = 0.00169.2 sq.ft.

Length.

Effective total length = 16 ft. 4 in.

The high-pressure, superheated Freon-12 vapours from the compressor were admitted into one end of the top 3/8 in. inner tube of the condenser and the condensate was removed from the other end of the bottom tube after the removal of super-heat and latent heat by the cooling water. This high pressure liquid was passed through a dehydrator and then through an automatic expansion valve where it was throttled to a lower pressure. The mixture of low-pressure liquid and vapour was circulated in the evaporator in which it absorbs its latent heat from water. The low-pressure vapours from the evaporator were then supplied to a hermetically sealed type of compressor of 1 H.P. capacity in which the vapours were compressed and supplied to the condenser.

Two pressure gauges, one (p-1) on the discharge and the other (p-2) on the suction lines of the compressor, were provided to record the pressures of the refrigerant.

The first one gives the pressure of the refrigerant in the condenser. In order to measure the temperature of the refrigerant, six thermocouples were provided at inlet, outlet, and at all return bends of the condenser.

Water for the condenser was drawn from the mains. In order to get different inlet water temperatures, the water was preheated in a tank by immersion coils and circulated by a pump. Water velocity in the condenser was regulated by the manipulation of the regulating valve. Since the pump was having more capacity than that required by the condenser, a part of the water from the pump was fed back to the tank.

The heat transfer process in the condenser could be made either parallel or counter-flow with the help of the valves 1,2,3, and 4, provided in the water line for the condenser. For parallel flow, the valves 1 and 3 were opened and valves 2 and 4 were closed, the water entered at the same point where the refrigerant entered and leaves at the same point as that of refrigerant. For counter-flow, the valves 2 and 4 were opened and valves 1 and 3 were closed.

The cooling water for the condenser was passed through the annular space formed between the two tubes of 3/4 in. and 3/8 in. size. During its passage in the condenser, it absorbs heat from the refrigerant thus removing the superheat and latent heat of the refrigerant.

A collecting tank was also provided at the outlet of the cooling water to determine the rate of flow of cooling water with the help of a platform scale which had an accuracy of measuring weight upto 1/8 lb. In order to measure the temperature of the cooling fluid , six thermocouples were provided at inlet, outlet and at all the return bends of the water line, so that these values would correspond to those of the refrigerant along the length of the condenser.

The evaporator was fed with water either from the mains or from the pump. As the water was cooled, it was drained off and fresh water was supplied.

The heat-exchanger was well insulated to prevent heat transfer to the surrounding atmosphere.

5.3. Flow-measurements.

In order to determine the rate of heat-transfer in the condenser, from the vapour to the cooling fluid, the rate of flows of cooling fluid was measured. After the steady-state conditions had been reached, the condenser water was turned into the weigh-barrel and a run started. At all rates of flow, the water was collected for not less than 10 minutes and weighed on a platform-scale from which the exact rate of water flow was determined.

5.3. Temperature-measurements.

Measurement of the temperatures was by the use of thermo-couples. Temperatures of condensing fluid and cooling fluid were recorded at six points each along the length of the condenser. The exact locations of these thermocouples are represented by the points 1 to 12 in Fig. 5.6. Chromel and Alumel wires of 1 mm standard size were used as thermocouple wires. The soldered-joints of these two wires, which formed the hot junction, were placed at the centre of the tube for each of the fluid so that they would give the average temperature of the fluid at that point. They were all located in the same manner and were believed to give fair values of the temperature difference from point to point. All the thermo-couple joints were sealed off and were tested for leakages upto a maximum pressure of 200 p.sig.

The Alumel wires from all the junctions were connected to a selector switch which was in turn connected to the potentiometer by Alumel wire. The chromel wires from all the junctions were soldered to one end of another Alumel wire, which formed the cold junction, and the other end of it was connected to the potentiometer. The cold junction was kept in an ice box and was maintained at a temperature of 32°F.

The voltage in millivolts, recorded by a semi-precision potentiometer, was converted into temperatures (°F) using the standard conversion tables.

5.4. Experiment Procedure.

Before charging the plant, the refrigerant line was evacuated by means of a vacuum pump and it was maintained for about 24 hours. A cylinder of Freon-12 was connected to the evacuated apparatus by means of copper tubing. Water was turned into the condenser tube and the fluid evaporated from the cylinder and condensed in the apparatus. When some erroneous data were gathered, purging was done as some air was known to be present in the apparatus.

When the apparatus had been charged and purged, the water rate through the condenser tube was regulated to the desired value. The inlet water temperature was adjusted, when possible, so that it was at 73.25°F. In some cases the available water supply was already above this and no adjustment was made. The compressor was started and the desired pressure of the condensing vapour was achieved by adjusting the throttle value and was obtained on gauge p-1. Equilibrium was considered to have been established when the pressure gauge, p-1, measuring the pressure of the condensing vapour was steady for at least 20 minutes.

When equilibrium had been reached, the condenser water was turned into the weigh -barrel and a run started. The water was collected for at least 10 minutes.

At the beginning and the end of a run, readings were made of the vapour and condensate temperatures at different points and pressure. Similarly, readings were made of the cooling water temperature at different points by means of a potentiometer as explained earlier.

For each rate of flow of the cooling water, runs were made for both parallel flow and counter flow. Runs were also made at a series of different water rates but with nearly constant overall temperature difference, $\Delta t = 10°F$, between the condensing vapour and the average water temperature.

This procedure was necessary in order to have satisfactory data for Wilson plot of reciprocal of overall coefficient of heat transfer versus reciprocal of water velocity to the 0.8 power. The water velocity range was from a minimum of about 2 ft./sec. to about 7 ft./sec.

Data were obtained for three more temperature differences, between water and Freon, 27°F, 31°F, and 36°F with inlet water temperatures of 77°F, 80°F and 84°F respectively, to find out the effect upon the coefficients and condenser tonnage.

5.5. Data and Calculation of Coefficients.

The experimental data and calculated overall heat transfer coefficients are recorded in tables 5.2 through 5.5. This data correspond to an average rate of flow of 150 lb./hr. of Freon-12. Typical observed results of temperature distributions are shown in Fig. 5.7.

The total heat load in the apparatus was calculated from the water rate and temperature rise. An example calculation of the heat transfer coefficient from the original data is given in Appendix 1. The following equation was used for calculating overall coefficients of heat transfer.

$$q = U_o A_o (\Delta t_o) \qquad \text{.... 5.1.}$$

in which

q = heat transferred, BTU per hour.

U_o = overall coefficient of heat transfer based on the outside surface area of the 3/8 in. tube, BTU per hr. °F. sq. ft.

A_o = Outside area of 3/8 in. tube, sq. ft.

Δt_o = difference of saturation temperature and average water temperature, °F.

The mean temperature difference could be readily determined by averaging equidistant point values of temperature difference over the condensing length. The water temperature rose almost linearly with length in most of these experiments so that the arithmetic average of the terminal temperature difference represented the true mean in such cases. So the Δt_o in Eq. 5.1 was taken as the difference between the average water temperature and the saturation temperature of the Freon-12 corresponding to the pressure in the condenser, which is recorded in item 15.

From the measurements ~~th~~ of water flow rate, the velocity of cooling water (ft./sec) was determined using the equation 5.2.

$$V = \frac{\text{Rate of flow of water (lb./min.)}}{60 \times 62.4 \times \text{Area of cross-section of Annulus (sq. ft.)}} \qquad \text{.... 5.2}$$

This value is recorded in item 13. Item 11 is the temperature difference between saturation temperature of Freon-12 corresponding to the condenser pressure (item 4) and the inlet water temperature (item 7).

5.6. Correlation of Data.

To determine condensing-film resistances from the values of overall thermal resistance, use was made of the method suggested by Wilson[53], as discussed in Chapter 4. In this procedure, the overal resistance (1/U) was plotted against the reciprocal of the 0.8 power of the water velocity in ft. per sec. Wilson pointed out that from equation 4.32 such a plot should yield a straight line if the water film coefficient, h_W, is the only coefficient varied. The straight lines thus obtained through the data were extrapolated to infinite water velocity. Table 5.6 lists, for several overall temperature differences, the Wilson plot intercepts which correspond to infinite water velocity less the resistance of the copper tube wall. Thermal resistance R_W of the copper tube wall was calculated by Eq. 5.3.

$$R_W = \frac{D_1}{2} \frac{\log_e (D_1/D_i)}{K_W} \qquad ...5.3.$$

Using a thermal conductivity of 222 for copper, the metal wall resistance was calculated to be 0.000014 hr, °F, sq.ft. per BTU . The reciprocal of the Wilson plot intercept minus the thermal resistance of the tube wall gives the film coefficient of heat transfer.

Film coefficient for condensing Freon-12 was computed using the Nusselt equation 1.41 for horizontal tube and an example of calculations is given in the appendix 2. The overall coefficient of heat transfer was determined by using Eq. 4.32 for an average mean temperature difference between saturation temperature of Freon-12 corresponding to the condenser pressure and the average temperature of water corresponding to a velocity of 4 ft./sec. Using this overall coefficient and the individual resistences the temperature drop across the condensing film was calculated. This temperature drop was used again for determining the coefficient of heat transfer for condensing Freon. By trial and error solution for the overall coefficient and the condensing film coefficient based on the inside surface area was made. The condensing film coefficient based on the outside area was again calculated.

The calculated film coefficients are compared with the experimental results in Table 5.6.

For turbulent flow of water in a given apparatus, the water film coefficient may be taken as $C_1 V^{0.8}$, Eq. 3.8, where C_1 is an empirical constant and may be considered as the apparent individual coefficient of heat transfer from tube to water. The reciprocal slope C_1 for each set of tests was found to be 380, indicating that the water side coefficient, at a water velocity of 1ft/sec, was 380 BTU/hr.sq.ft.°F, difference from tube to water. Using this value, the water film coefficients

h_w, were calculated at different velocities by using the formula $h_w = C_1 V^{0.8}$ and are shown in Table 5.7. Taking the physical properties from standard tables, the value of $(h_w De/K)/(Cp\mu/K)^{0.4}$ and DeG/μ were calculated for all the sets of readings and are shown in table 5.7.

The values of $(h_w De/K)/(Cp\mu/K)^{0.4}$ were plotted against Reynolds number, $N_{RE} = \frac{DeG}{\mu}$, and are shown in Fig. 5.9. These values were compared with the values obtained by the equation.

$$\frac{h_w De}{K} = 0.023\left(\frac{DeG}{\mu}\right)^{0.8}\left(\frac{Cp\mu}{K}\right)^{0.4} \quad \text{.... 5.4}$$

in which

h_w = water film coefficient, BTU/hr, °F, sq.ft. of inside surface of annulus.

De = Equivalent diameter of the annulus, $(D_2 - D_1)$, ft.

K = Thermal conductivity of water, BTU/hr.ft.°F.

G = Mass rate of flow, lb.per hr. sq.ft.

μ = Viscosity of water, lb.per ft. hr.

Cp = Specific heat of water, BTU per lb. °F.

In addition to the above plots, the following plots were also made, which are discussed in the next chapter.

1. A plot of water velocity versus overall coefficient of heat transfer, Fig.5.10, 2. A plot of M.T.D. versus overall coefficient of heat transfer, Fig. 5.11, 3. A plot of condenser heat transfer and M.T.D., Fig. 5.12., 4. A plot of M.T.D' versus Temperature difference (item 11), Fig. 5.13, 5. A plot of velocity versus water temperature rise (item 9), Fig. 5.14.

	1	2	3	4	[illegible]	[illegible]	7	8	[illegible]	10
[illegible] Operation (Condition)	P	C	C	P	C		P	C	C	
[illegible] Pressure, PSIa	107	107	100	100	112		100	100	[illegible]	[illegible]
[illegible] Temperature, °F	[illegible]	84.270	80.13	80.13	81.322	[illegible] 23	80.13	80.12	[illegible]	[illegible]
[illegible] Pressure, PSIG	15	15	12	13	15		15	15	15	[illegible]
[illegible] Temperature	[illegible]	212.5	185.81	182.33	188.[illegible]	[illegible]	[illegible]	.8[illegible]	[illegible]	21[illegible]
[illegible] Water [illegible] Temp.	[illegible]	77.25	72.93	73.22	[illegible]	75	74.0	.5	.38	[illegible]
Mean Temperature	77.25	77.50	73.00	76.382	73.75		76.5	.75	[illegible]	82.[illegible]
[illegible] Temperature [illegible]	[illegible]	.75	3.27	3.112	3.[illegible]	[illegible]	2.25	2.25		[illegible]
Average [illegible]	75.125	75.575	.550	74.803	75.[illegible]		75	625	[illegible]	[illegible]
Sat. [illegible] Heat [illegible] Rate	20.2[illegible]	21.123	17.4	13.3	17.68	.78	1	.65	[illegible]	8
[illegible], lb./min.	28 [illegible]	2[illegible].3	[illegible].05	21.8	36.35		[illegible]	.3[illegible]		12.[illegible]
[illegible] Flow (V), gpm	[illegible].43	[illegible]	5.50	5.05	5.73		5.[illegible]	[illegible]		2.[illegible]
[illegible]	7780	7280	6800	5950	[illegible]			[illegible]		[illegible]
[illegible] Btu/hr	18.3[illegible]	13.902	15.765	13.324	[illegible]	[illegible]		.5	[illegible]	[illegible]
[illegible]	[illegible]	1.608	1.608	1.308	1.[illegible]	[illegible]		.3[illegible]		[illegible]
[illegible]	33.[illegible]	[illegible]	[illegible]	242.3	28[illegible]					[illegible]
	.38[illegible]	[illegible]	.384	.412	.2[illegible]		[illegible]	.8		
	[illegible]	[illegible]	[illegible]	.2755	.2[illegible]		.21[illegible]		[illegible]	

3

[illegible]										
[illegible]	132	132	132	132	131	132	[illegible]	[illegible]	[illegible]	13
[illegible] Temperature, °F	1.8	[illegible]	107.8	107.5	107.283	107.8	[illegible]	6.775	[illegible]	[illegible]
[illegible] Pressure, [illegible]	25	[illegible]	25	25	25	25	37			47
[illegible]	[illegible]	[illegible]	201.82	157.72	[illegible]	186.82	[illegible]	1.23	[illegible]	[illegible]
[illegible] Temperature, °F	78.3	[illegible]	78.10	78.5	73.72	77.27	[illegible]	.33	[illegible]	[illegible]
[illegible] Te[illegible], °F	[illegible]	[illegible]	[illegible].77	[illegible]	[illegible]	34.84	[illegible]	11	[illegible].403	[illegible]
[illegible]	3.182	3.[illegible]	[illegible].68	6.78	5.[illegible]5	7.27	5.[illegible]	15	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]1.472	32.[illegible]	[illegible]	3[illegible].30	[illegible]	.008	[illegible]	[illegible]
[illegible]	[illegible]	[illegible].893	25.62	2[illegible]	[illegible]	20.5[illegible]	[illegible]	658	[illegible]	25
[illegible]	[illegible]	.58	[illegible].5	25.1	32.0	31.[illegible]0	[illegible]	575	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	5.28	5.[illegible]	5.85	5.[illegible]	5 [illegible]		[illegible]	[illegible]
[illegible]	[illegible]		8725	10220	[illegible]	12875	[illegible]	[illegible]	[illegible]	
[illegible]	27.38	[illegible]	15.325	20.776	2[illegible].133	25.855	[illegible]	575	[illegible]	[illegible]
[illegible]	3.[illegible]	[illegible]	[illegible].503	1.603	[illegible]	1.60[illegible]	[illegible]	03	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	250	247	23[illegible].5	322	[illegible]		[illegible]	[illegible]
	467	.173	[illegible]	.405	.12[illegible]	.3108	[illegible]			
	[illegible]	[illegible]	[illegible].035	[illegible]	[illegible]	.2725	[illegible]			

[illegible]

TABLE 5-4

[illegible]		[illegible]	22	23	24	25	26	[illegible]	[illegible]		
[illegible]	[illegible] Conn.	[illegible]	0	0	?	0	[illegible]		0	[illegible]	
[illegible]	[illegible] a PSIG	[illegible]	[illegible]	.132	.13	.145	[illegible]	[illegible]8	.145	[illegible]	[illegible]
[illegible] at	[illegible]	[illegible]	114.18	117.22	118.2	[illegible]	[illegible]	[illegible].57	[illegible]	114.18	[illegible]
[illegible]	[illegible] PSIG.	[illegible]	2.	[illegible]	13	40	[illegible]	[illegible]	[illegible]	13	
[illegible]	[illegible] °F	[illegible]	2[illegible].330	[illegible]	205.3	[illegible]	[illegible]	[illegible]	151.[illegible]	123.	[illegible]
[illegible]	[illegible] perature °F,	[illegible]	31.365	30.[illegible]	81.135	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	80.[illegible]	57.3	8[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
[illegible]	[illegible] e, °F	[illegible]543	[illegible]	.7[illegible]	[illegible].363	5.[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	34.[illegible]	[illegible].315	[illegible]	[illegible]	[illegible]	35.[illegible]	37.[illegible]	[illegible]
[illegible]	Temp. [illegible] Inlet [illegible]	[illegible].25	[illegible]	[illegible]	.034	[illegible]	[illegible]	[illegible]	24.332	[illegible]	
[illegible]	[illegible]	[illegible]22	[illegible]2		24.[illegible]	37.[illegible]	[illegible]	[illegible]	31.[illegible]	[illegible].2	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible].12	5.[illegible]	[illegible]	
[illegible]	[illegible]		[illegible]	[illegible]	55.[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	25.[illegible]	[illegible]	[illegible]	[illegible]	3[illegible].331	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible].308	[illegible].E05	1.6[illegible]	[illegible]	3.	[illegible].6[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	125	[illegible]	5.[illegible]	[illegible]	[illegible]	[illegible]	322.[illegible]	[illegible]	
[illegible]		[illegible]	[illegible]		.483	.[illegible]	[illegible]	32[illegible]	.31	[illegible]	
[illegible]		[illegible]	[illegible]	[illegible]	.224	.230	[illegible]	[illegible]	.[illegible]	[illegible]	

Run No.	[illegible]	32	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible] Alkon. C-Gramo. [illegible]	[illegible]	6	6	[illegible]	[illegible]	6	[illegible]	[illegible]	[illegible]	[illegible]
[illegible], PSIG	175.9	[illegible]	[illegible]	172	[illegible]	[illegible]	[illegible]	170	[illegible]	[illegible]
Saturation Temperature, °F	127.74	128.75	125.4	125.6	125.08	[illegible]	[illegible]	125.4	12[illegible]	[illegible]
[illegible] Pressure, PSIG	[illegible]	32	30	28	[illegible]	[illegible]	[illegible]	[illegible]	17	[illegible]
[illegible] Temperature, °F	[illegible]	222.34	203.37	228.18	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	56.25	56.25	[illegible]	84.5	[illegible]	82.	72	[illegible]	[illegible]
[illegible] Temperature, °F	[illegible]	23.334	24.54	[illegible]	[illegible]	[illegible]	85.	[illegible]	0.[illegible]	35
[illegible] rise, °F	[illegible]	9.334	6.78	[illegible]	[illegible]	[illegible]	5.[illegible]	[illegible].505	[illegible]	[illegible]
[illegible] Temperature, °F	[illegible]	[illegible]	[illegible] 325	87.[illegible]	3[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible] Ratio	[illegible].505	42.412	38.15	47.5[illegible]	4[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible] lb/min.	20[illegible]	20.35	25.2	20.5	[illegible]	[illegible]	[illegible]	20.[illegible]	[illegible]	[illegible]
[illegible] of Water (ml)/sec.	3.2[illegible]	3.21	3.075	3.[illegible]	[illegible]	[illegible]	3.2[illegible]	[illegible]	[illegible]	[illegible]
[illegible] Heat Transfer, [illegible]	[illegible]	11700	12520	1158[illegible]	1[illegible]	[illegible]	51[illegible]	[illegible]	[illegible]	[illegible]
[illegible] Difference, [illegible]	[illegible]	37.72	38.004	3[illegible].356	3[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	1.3[illegible]	1.302	1.373	1.303	1.	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible] (U)	[illegible]	153.5	220.5	130	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	.503	.435	.367	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	.334	.331	.333	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

TABLE 5-6

Comparision of Experimental Values for Film Coefficients with calculated values.

S.No.	Saturation Temperature, °F	Inlet Water Temperature, °F	Mean Temperature Difference °F.	Wilson plot intercept minus wall resistance.	Experimental film Coefficient BTU/hr. °F.sq.ft.	Calculated Film Coefficient BTU/hr. °F.sq.ft.	Deviation from experimental values%
1	91	73.25	16	0.003406	293.5	299.0	1.872
2	107	77.00	27	0.003546	282.0	281.5	-0.1772
3	114	80	31	0.003986	251.0	267.5	6.57
4	123	84	36	0.004136	242.0	247.5	2.27

TABLE 6.7

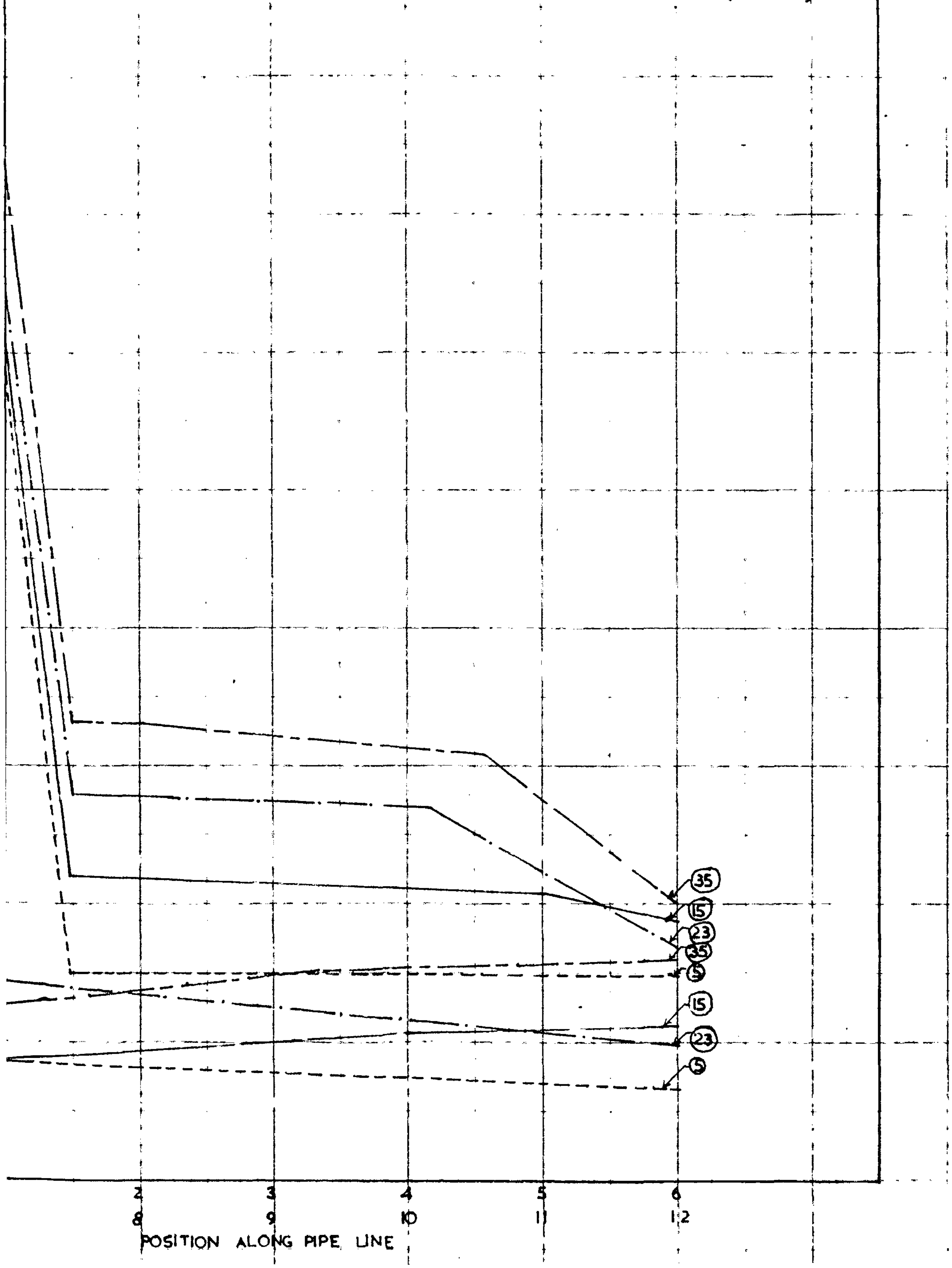
Water Side Coefficients.

D_e = 0.02475 ft., C_1 = 380

Run No.	Velocity v, fps	$G/10^3$ lb./hr. sq.ft.	$h_w = C_1 v^{0.8}$	Average Water Temperature, °F	$\frac{D_e G}{\mu}$	$\frac{h_w D_e / K}{(C_p \mu / K)^{0.4}}$
1	[illegible].43	[illegible]35	12[illegible]7	75.125	11480	42.7
2	[illegible].38	[illegible]	1238	75.875	11340	42.4
3	5.3	[illegible]	1438	7[illegible].335	13720	[illegible]
4	[illegible]	[illegible]	[illegible]73	74.805	13[illegible]20	[illegible]
5	[illegible].76	[illegible]	15[illegible]	75.195	14[illegible]	52.[illegible]
6	[illegible].90	1530	1595	75.375	[illegible]	5[illegible]
7	6.86	15[illegible]2	1[illegible]73	75.125	17770	60.[illegible]
8	[illegible]58	1503	1735	[illegible].625	17300	58.7
9	[illegible]	[illegible]18	624	[illegible].34	4930	21.55
10	[illegible]	[illegible]68	[illegible]	77.36	5170	25.8
11	[illegible]	[illegible]30	5[illegible]3	80.90[illegible]	8170	31.7
12	[illegible]5	63[illegible]	[illegible]25	83.[illegible]	8450	32.3[illegible]
13	[illegible].98	5[illegible]	1[illegible]0	8[illegible].475	10570	39.[illegible]
14	[illegible].93	390	1143	[illegible].024	10830	30.9
15	[illegible]05	112[illegible]	[illegible]753	78.545	1[illegible]440	47.5
16	[illegible]2	1123	[illegible]	80.905	1[illegible]540	47.5
17	[illegible]	1375	[illegible]	[illegible].225	164[illegible]	55.[illegible]
18	[illegible]	1373	[illegible]	[illegible].97[illegible]	1650[illegible]	55[illegible]
19	[illegible]	1535	7[illegible]	7[illegible].03[illegible]	180[illegible]	59.[illegible]
20	[illegible]	163[illegible]	[illegible]708	[illegible].998	18[illegible]	[illegible]

[illegible]	[illegible]	$G/10^3$ lb./hr. sq.ft.	E_{ND} [illegible] $^{0.8}$	[illegible] Temperature, °F	$D_e G / \mu$	$\dfrac{h_w D_e/K}{(C_p \mu/k)^{0.4}}$
[illegible]	[illegible]	347	32[illegible]	35,2[illegible]	81[illegible]	21.[illegible]
[illegible]	[illegible].35	650	502	[illegible],500	52[illegible]	31.35
	[illegible].71	830	1[illegible]	[illegible]	11[illegible]	37.3
	[illegible].05	[illegible]	1[illegible]	[illegible]	11[illegible]	40.0
26	[illegible].03	[illegible]	1[illegible]	82,[illegible]	[illegible]	55.[illegible]
[illegible]	[illegible]	1[illegible]	[illegible]	83,[illegible]	[illegible]	[illegible]
[illegible]	4.[illegible]	11[illegible]	[illegible]	83.08	13[illegible]	47.[illegible]
28	[illegible]3	[illegible]	1[illegible]83	[illegible],020	[illegible]	[illegible]
	[illegible]	[illegible]	1[illegible]	[illegible]	1881	33.[illegible]
[illegible]	[illegible]1	[illegible]	1805	[illegible]	18[illegible]	[illegible].2
[illegible]	.23	[illegible]	[illegible]	[illegible].26	88[illegible]	35.[illegible]
2	[illegible]	[illegible]	[illegible]	[illegible]	8[illegible]	34.[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible].85	[illegible]	41.[illegible]
34	[illegible]	[illegible]	1[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]10	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	1.23	3[illegible]	1[illegible]	[illegible]
[illegible]	[illegible]		[illegible]	8[illegible]2	[illegible]	58.[illegible]
[illegible]	[illegible]		[illegible]382	[illegible],[illegible]20	[illegible]	58.[illegible]
[illegible]	[illegible]4		1[illegible]	[illegible],708	[illegible]50	[illegible].5
40	[illegible]	[illegible]	[illegible]	[illegible].142	2[illegible]	[illegible]

FIG 5.7 ACTUAL TEMPERATURE DIFFERENCE CURVES (4RUNS)



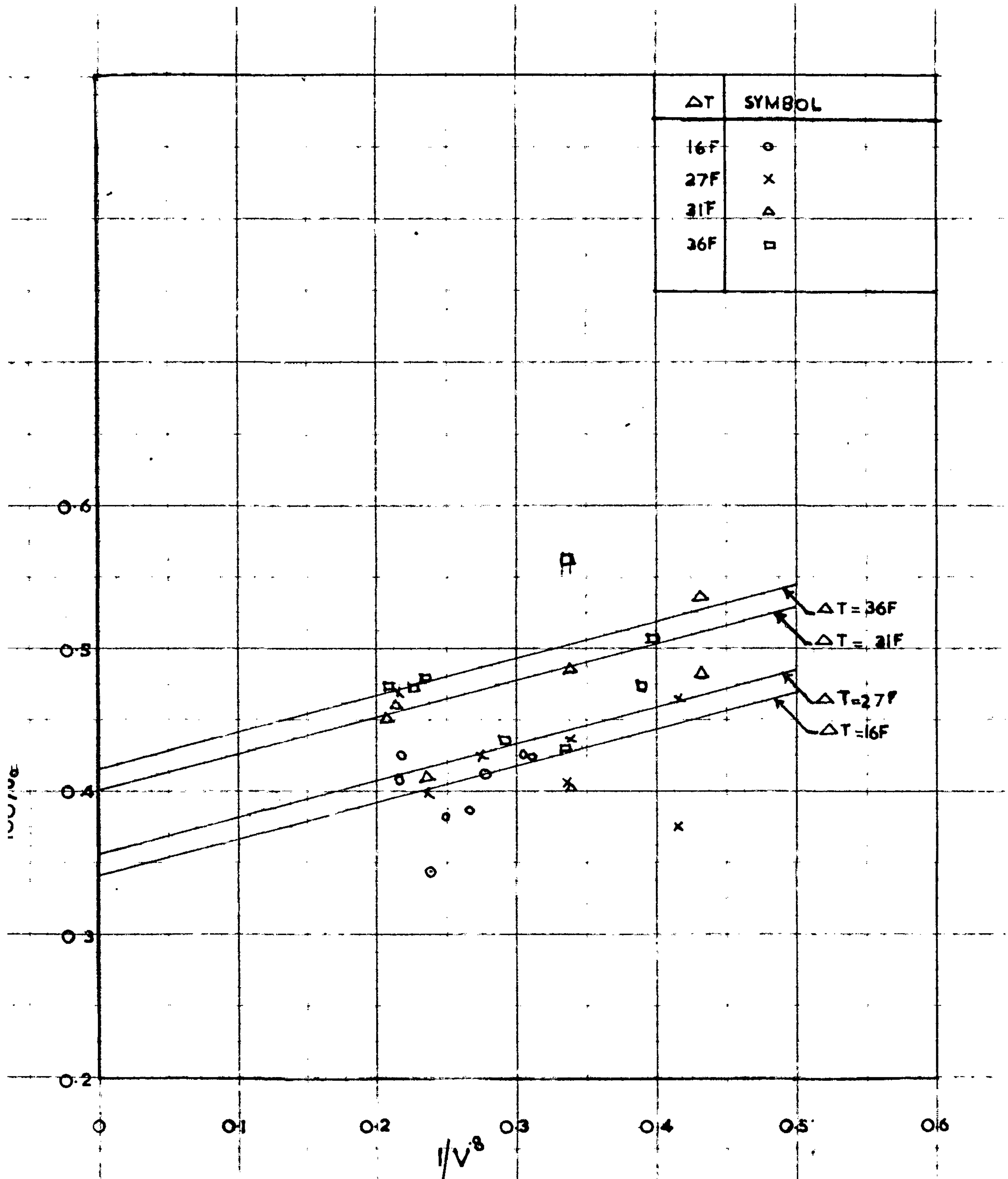
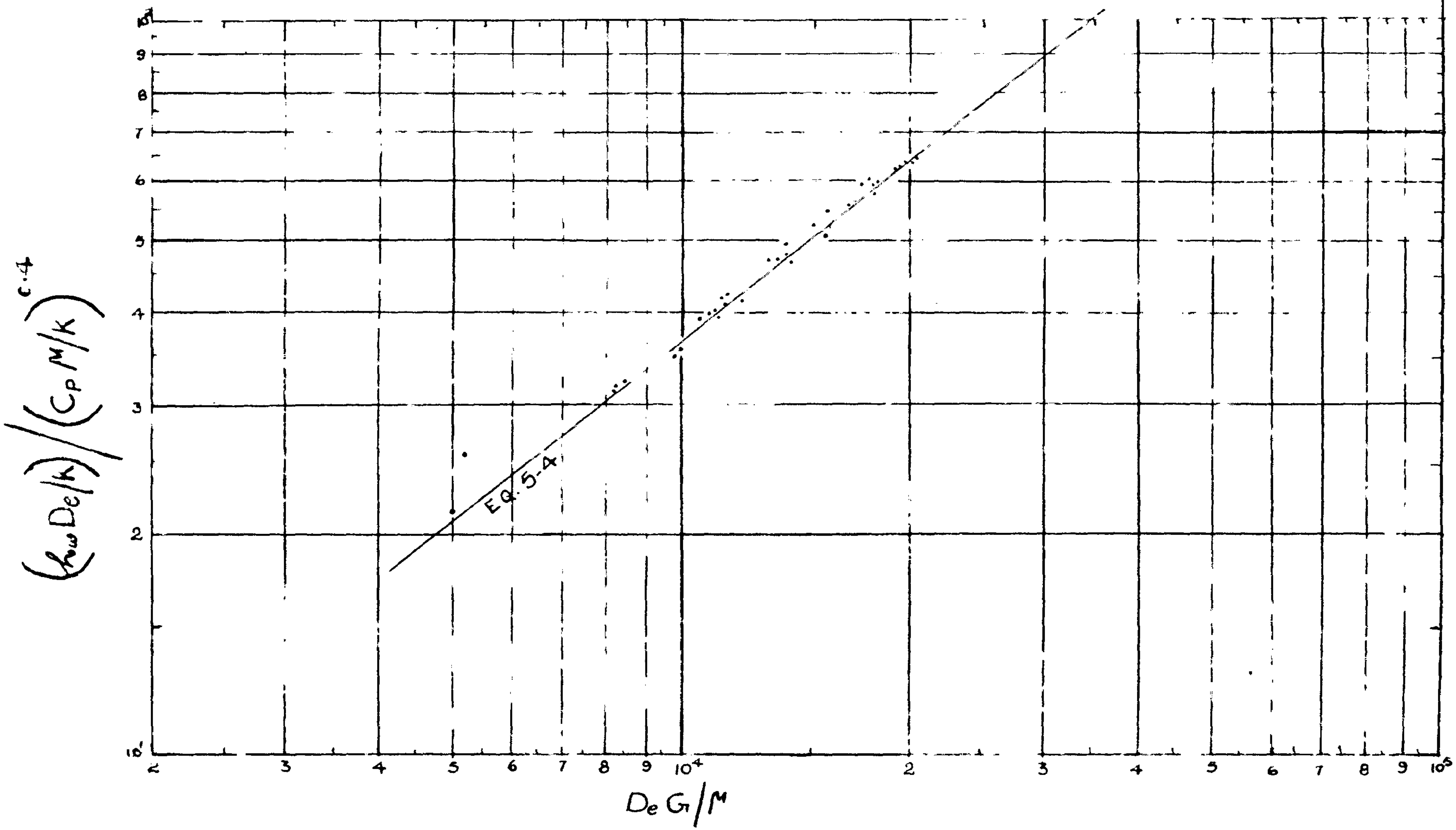
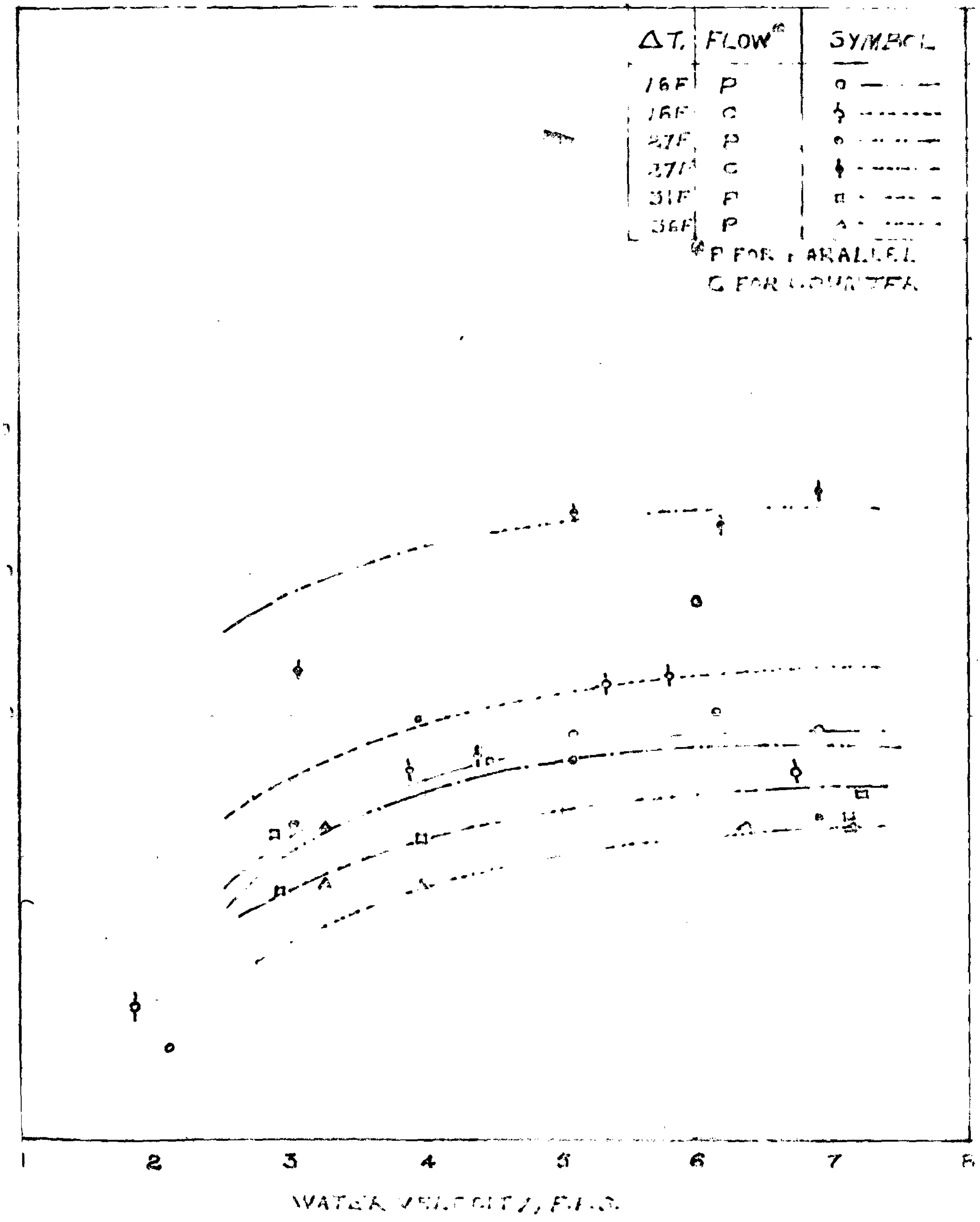


FIG 5·8. WILSON PLOT

FIG. 5-9 WATERFILM COEFFICIENTS IN THE ANNULUS VERSUS REYNOLD'S NUMBERS WITH A COMPARISON OF VALUES COMPUTED BY EQ. 5-4.

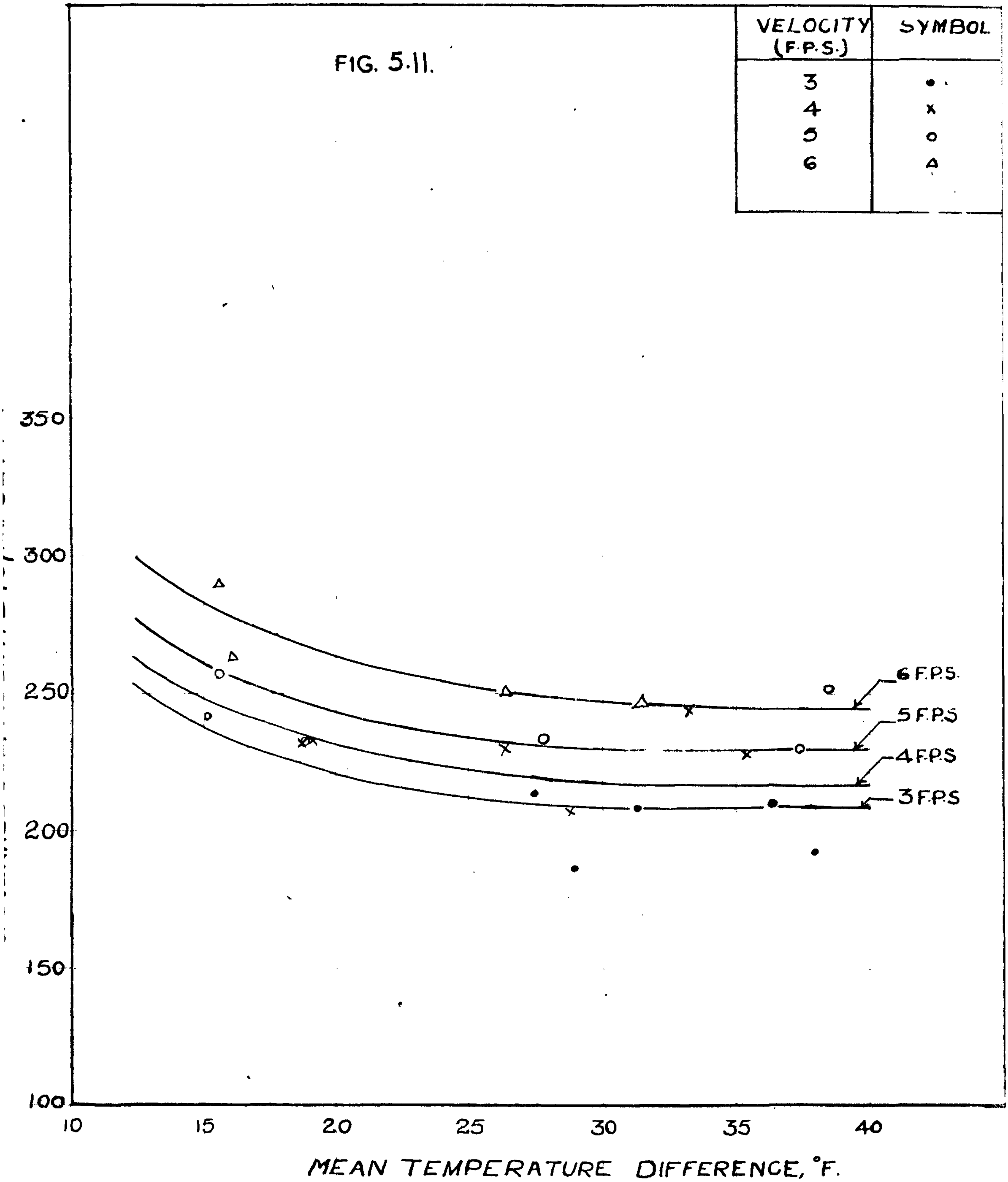




ΔT, FLOW*, SYMBOL
16F P
16F C
27F P
27F C
31F P
36F P
*P FOR PARALLEL
C FOR COUNTER
1 2 3 4 5 6 7 8
WATER VELOCITY, F.P.S.

FIG. 5-10

FIG. 5.11.



VELOCITY (F.P.S.)
SYMBOL
3
4
5
6
6 F.P.S.
5 F.P.S
4 F.P.S
3 F.P.S
350
300
250
200
150
100
10
15
20
25
30
35
40
MEAN TEMPERATURE DIFFERENCE, °F.

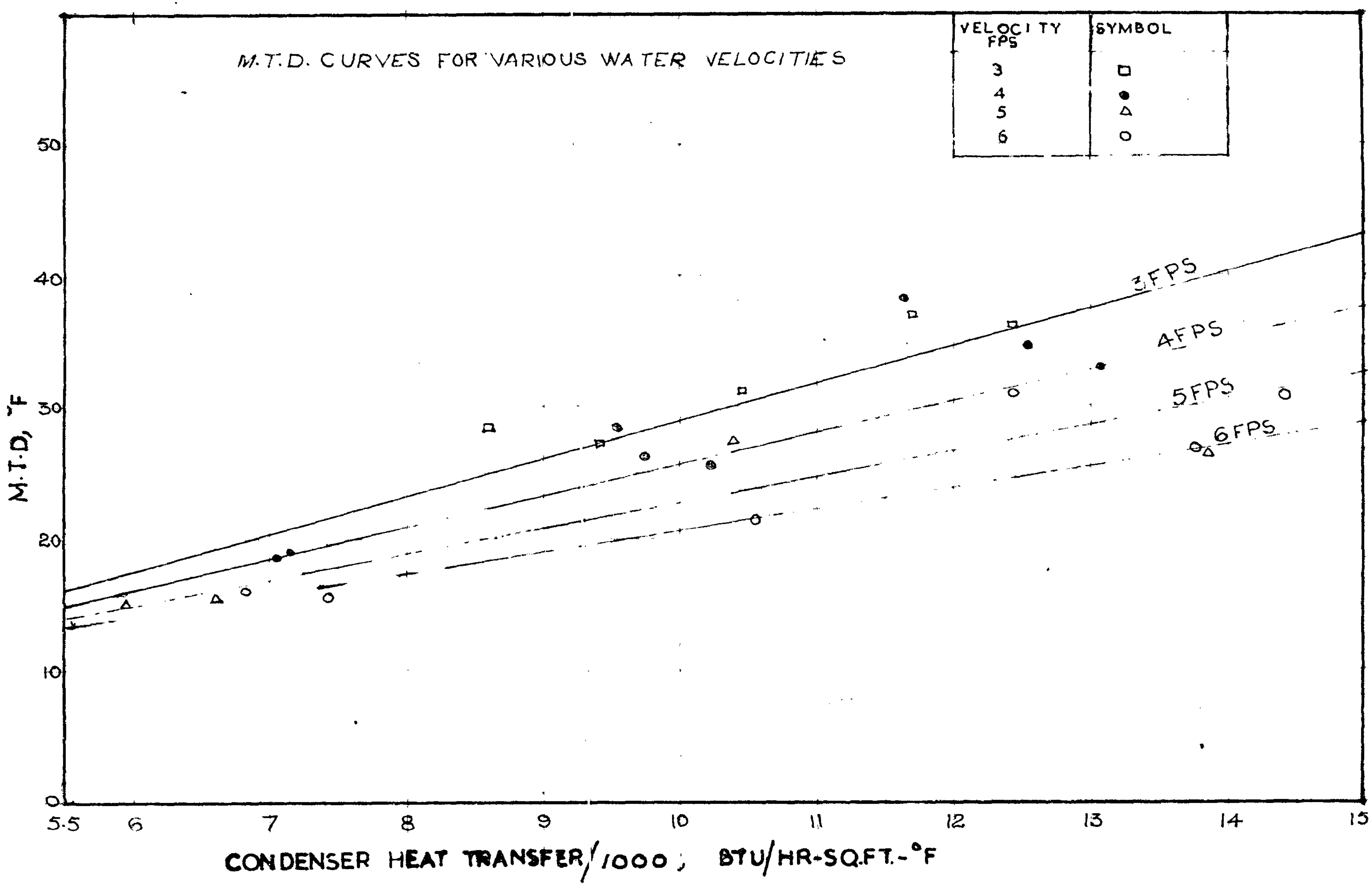
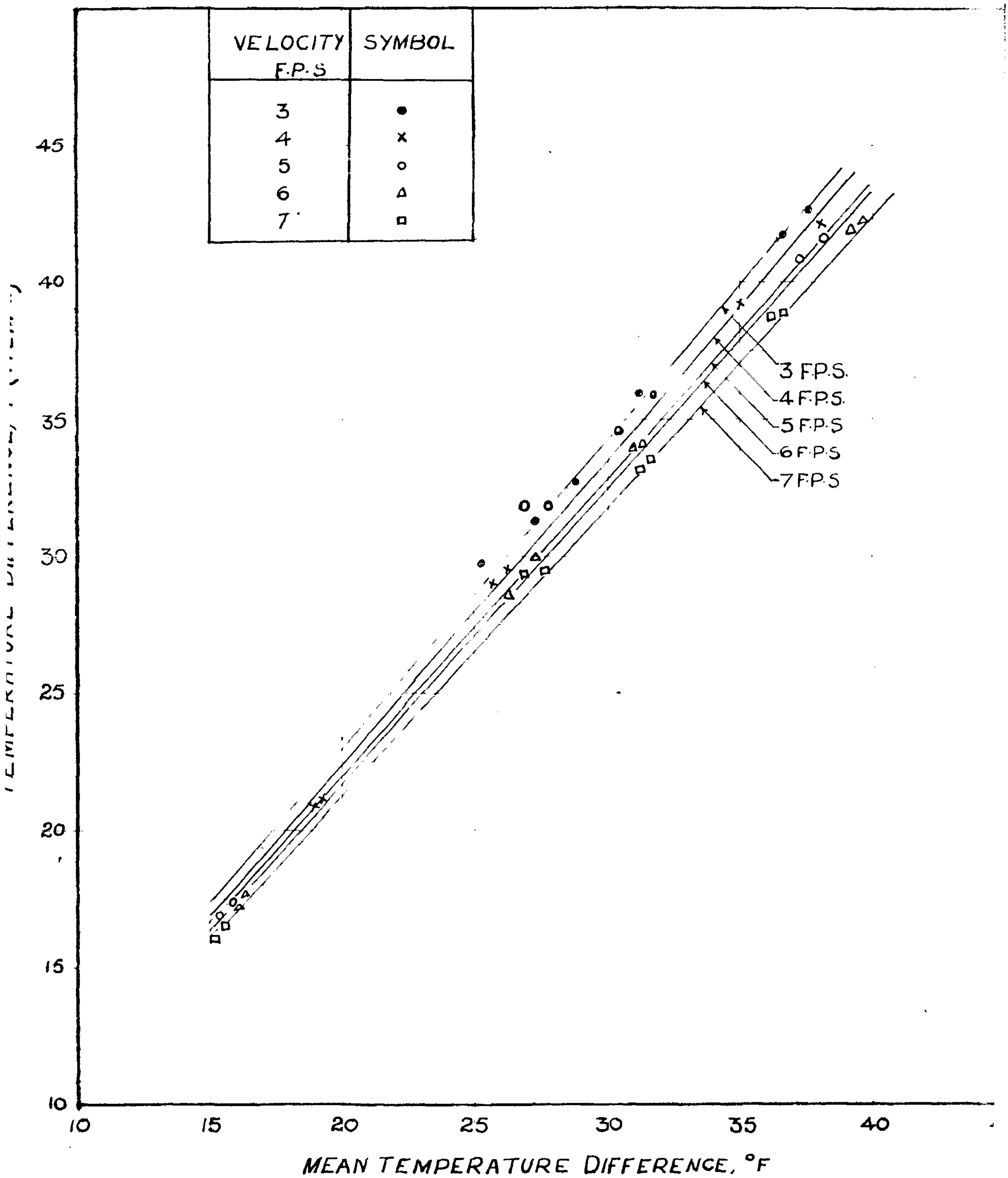
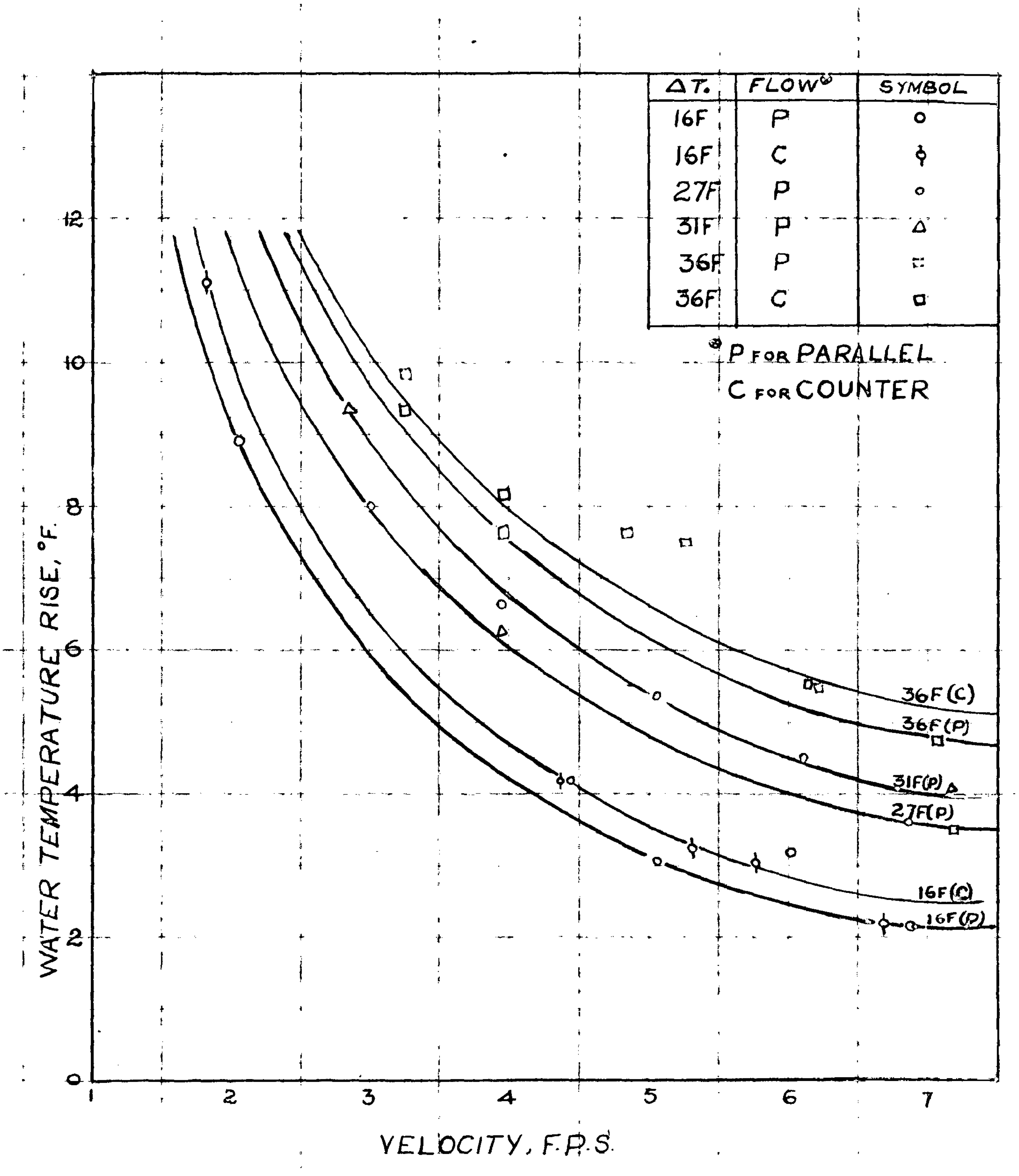

M.T.D. CURVES FOR VARIOUS WATER VELOCITIES
VELOCITY FPS
SYMBOL
3
4
5
6
3 FPS
4 FPS
5 FPS
6 FPS
M.T.D., °F
50
40
30
20
10
0
5.5
6
7
8
9
10
11
12
13
14
15
CONDENSER HEAT TRANSFER/1000, BTU/HR-SQ.FT.-°F

FIG 5-12



VELOCITY F.P.S
SYMBOL
3
4
5
6
7
3 F.P.S.
4 F.P.S.
5 F.P.S
6 F.P.S
7 F.P.S
10
15
20
25
30
35
40
45
MEAN TEMPERATURE DIFFERENCE, °F

FIG. 5-13



ΔT.
FLOW*
SYMBOL
16F
P
16F
C
27F
P
31F
P
36F
P
36F
C
* P FOR PARALLEL
C FOR COUNTER
36F (C)
36F (P)
31F(P)
27F(P)
16F(C)
16F(P)
WATER TEMPERATURE RISE, °F.
VELOCITY, F.P.S.
0
2
4
6
8
10
12
1
2
3
4
5
6
7

FIG. 5-14

CHAPTER 6.

DISCUSSION OF RESULTS.

ABSTRACT.

The results of the observations made during the experiments are discussed. The experimental condensing and water film coefficients are in good agreement with the theoretical predictions. Application of the results for the design purposes is given.

6.1. Wilson Plot.

Fig. 6.8 is a plot of the reciprocal of the overall coefficient versus the reciprocal of the 0.8 power of the water velocity in ft. per sec. The straight lines through the data are extrapolated to infinite water velocity. The data include observations made for temperature differences of 16°F, 27°F, 31°F and 33°F and with initial water temperatures of 73.25°F, 77°F, 80°F, and 84°F respectively. As the temperature difference increases the straight lines passing through the respective data are moving upwards showing that the wilson plot intercept increases as Δt increases. This is in agreement with the theory of condensation.

6.2. Condensing Film Coefficients

Table 6.6 lists the values of wilson plot intercepts less the copper tube wall resistance corresponding to different values of Δt, the experimental and calculated values of condensing film coefficients.

The calculated film coefficients are compared with the experimental results in the Table 5.6. The agreement is very good.

The experimental film coefficients are in good agreement with the calculated values, the deviation being from about -0.2% to 6.6% from experimental values. This shows that Nusselt's analysis can be used for predicting condensing film coefficients for this type of arrangement.

6.3. Water Film Coefficients.

The reciprocal slope (C) of the straight lines in Wilson plot was found to be 330. Using this value of C_1, the water film coefficients were calculated by using the equation 6.1.

$$h_w = C_1 V^{0.8} \quad \text{.... 6.1}$$

and these values are tabulated in Table 5.7 along with values of $(h_w De/k)/(C_p \mu/k)^{0.4}$ and $De G/\mu$.

Fig. 5.8 is a plot of water film coefficients expressed in terms of Nusselt's and Prandtl numbers versus Reynolds number, $De G/\mu$, and shows that the experimental values are in quite good agreement with those obtained by the equation 5.4. Actually, the experimental values slightly exceed the calculated values by about 4 percent, possibly due to the turbulence in the water as it entered the tube. But, in general, the agreement of the experimental results is very good with those of the calculated values within the experimental errors.

6.4. Overall Coefficient of Heat Transfer.

Fig. 5.10 is a plot of overall coefficient of heat transfer versus water velocity for different mean temperature differences both for parallel and counter flow. This shows that the overall coefficient is higher at low temperature differences for the same water velocity. The overall coefficient of heat transfer is more for counterflow than that for parallel flow for the same M.T.D. and velocity of cooling water.

The values of overall coefficient of heat transfer are plotted as ordinate and the mean temperature difference as abscissa for a given velocity of water. These curves appear in Fig. 5.11. The value of U_0 decreases as the mean temperature difference increases, for a given quantity of cooling water as is shown in the figure. For the same mean temperature difference the value of U_0 increases as the rate of cooling water increases, as pointed out earlier.

6.5. Condenser Heat Transfer.

Fig. 5.12 shows the set of curves obtained by plotting the mean temperature difference as ordinate and the condenser tonnage for a given water rate as abscissa. For a given quantity of water, there is ~~as~~ a straight line relationship between the condenser heat transfer and mean temperature difference. These straight lines tend to converge to a common point at zero mean temperature difference. If runs were made at very low mean temperature differences, these lines may bend and pass through the origin.

For the same mean temperature difference, the condenser heat transfer increases as the velocity of the cooling water increases.

6.6. Temperature Difference.

The curves in Fig. 5.13 show the relation between the mean temperature difference, and the difference in temperature of the saturation temperature of Freon-12 and the entering water temperature (Item 11) for a given velocity of cooling water. It may be observed that the relationship between these two is a straight line for a given velocity. These lines appear to converge at low M.T.D. and diverge as the M.T.D. increases.

6.7 Temperature Rise.

Fig. 5.14 is a plot of cooling water velocity versus temperature rise of cooling water for a given mean temperature difference, both for parallel and counter flow. The curves are taking the shape of hyperbola nearly. It may be observed from the set of curves that the temperature rise of cooling water, for a given velocity and a given mean temperature difference, is more in counterflow than in parallel flow. It has been observed in the case of counterflow the subcooling of the condensate is more than that in parallel flow and hence this increase in temperature rise.

6.8. <u>Application of Results.</u>

1. The data shown on Fig. 5.9 are based on a diameter ratio of D_2/D_1 = 1.793. A comparision with the experimental results obtained indicates that this data may also be used with reasonable accuracy for other narrow annular spaces of diameter ratios of D_2/D_1 from about 1.5 to 2.0 in the turbulent region for an average temperature of cooling water from about 75 to 87°F.

2. The curves of Fig. 5.10 are based on an average temperature difference from about 16 F to 36 Ft. This data may be used for predicting the overall coefficient of heat transfer corresponding to a particular velocity and average temperature difference or the same may be done with the use of Wilson plot i.e. Fig. 5.8.

3. The curves of Fig. 5.12 may be used to predict the mean temperature difference, if the total heat transferred by the condenser and the velocity of cooling water are fixed earlier.

4. If the saturation temperature of Freon-12, inlet water temperature and the velocity of the cooling water are known earlier, the curves of Fig. 5.13 may be used with reasonable accuracy to determine the mean temperature difference, from

which the overall coefficient of heat transfer and heat transferred by the condenser can be determined as stated above.

G.D. Conclusions.

The friction data obtained indicate that the proper Reynolds number to be used is D_eG/μ, where D_e is the equivalent diameter of the annulus, which is equal to the difference of the two diameters of annular space.

The friction data in the turbulent region are within reasonable accuracy of the calculated values, and it is believed that this correlation can be used for other sizes of annuli.

The results of the experimental film coefficients have been found to be in good agreement with predictions from standard heat transfer equations. Though this data can be used, with reasonable accuracy, for the design of smaller units, it is to be confirmed whether or not this data can be used for the design of higher capacity plants.

APPENDIX -1

Example Calculations of Heat Transfer Coefficients.

Run No.16.

Data Recorded.

Freon-12 pressure in the condenser	- 131 psig.
	-145.7 p.sia.
Temperature of Freon in vapour line.	- 185.91 F.
Saturation temperature of Freon (From table of thermodynamic properties).	- 107.283 F.
Inlet water temperature	- 76.82 F.
Outlet water temperature-	- 82.27 F.
Rate of flow of cooling water	- 32.00 lb./min.

Calculations of Overall Coefficient of Heat Transfer.

Temperature rise of cooling water	= 82.27 - 76.82
	= 5.45F.
Heat transferred in condenser	= 32.00 x 60 x 5.45
	= 10,470 BTU per hr.
Average temperature of cooling water	= $\frac{76.82 + 82.27}{2}$
	= 79.545 F.

Average temperature difference between saturated Freon vapours and cooling water (t) $= 107.283 - 79.545$

$= 27.738$ F.

Outside area of tube $= \pi \times \frac{0.375}{12} \times 16.33$

$= 1.603$ sq.ft.

Overall coefficient of heat transfer $= \frac{10,470}{1.603 \times 27.738}$

$= 235.5$ BTU/hr.sq.ft.°F

Values for Wilson Plot.

Reciprocal of coefficient $\frac{1}{234.5}$ $= 0.00427$

Area of cross section of the annular space $= 0.001692$ sq.ft.

Velocity of cooling water $= \frac{32.00 \text{ (lb.min)}}{60 \times 0.001692 \times 62.4}$

$= 5.05$ ft./sec.

Reciprocal of 0.8 power of velocity $= \frac{1}{5.05^{0.8}}$

$= 0.274$

Temperature Difference.

Temperature difference between saturated Freon vapours and inlet water $= 107.283 - 76.82 = 30.463$ F.

Appendix - 2.

Example Calculation of Film and Overall Coefficients of Heat Transfer for Single horizontal tube.

Data.

Freon Pressure = 130 psig.

Average temperature difference between saturated Freon vapours and cooling water (Δt) = 27 F.

Average water temperature = 80 F.

Velocity of cooling water = 5 ft/sec.

Tube data from Table 5-1.

Water-film Coefficient.

The water film coefficient, h_w, is calculated from the equation:

$$h_w = (1.15)(0.023)\left(\frac{De\,G}{\mu}\right)^{0.8}\left(\frac{C_p\mu}{K}\right)^{0.4}\left(\frac{K}{De}\right)$$

The physical properties of water are taken at an average temperature of 80°F.

$$D_e = \frac{0.672 - 0.375}{12} = 0.02476 \text{ ft.}$$

$$G = 5 \times 3600 \times 62.4 = 1{,}125{,}000 \text{ lb/sq.ft. hr.}$$

$$\mu = 2.08 \text{ lb/hr.ft.}$$

$$C_p = 1.0 \text{ BTU/lb.deg.F.}$$

$$K = 0.351 \text{ BTU/hr.ft.}^{\circ}\text{F}$$

$$h_V = (1.15)\ (0.023) \left[\frac{0.02370 \times 1,125,000}{2.00}\right]^{0.8} \left(\frac{1 \times 2.08}{.351}\right)^{0.4} \frac{0.351}{.02370}$$

$= 1940$ BTU/hr. °F – sq.ft. of inside surface of annulus.

Condensing Film Coefficient.

The condensing film coefficient of heat transfer is calculated from the equation:

$$h_V = 0.725 \left(\frac{K_f^3 \rho_f^2 g \lambda}{\mu_f D_i \Delta t}\right)^{1/4}$$

The values of K, ρ, & μ for Freon-12 are taken at the film temperature, which is the temperature of the condensing vapours minus $\frac{3}{4}\Delta t$. The value for λ is taken at the temperature of the condensing vapours.

The film coefficient for a 27°F Δt is first calculated and then modified by a trial and error method to take into account the temperature drop through the water film and copper tube wall.

$t_f = 107 - \frac{3}{4} \times 27 = 86.75$°F

$D_i = \frac{0.307}{12} = 0.0256$ ft.

$K_f = 0.01960$ BTU/hr. ft. °F

$\mu_f = 0.622$ lb/hr.ft.

$\rho_f = 80.572$ lb/cft.

λ = 54.806 BTU/lb.

g = 4.17×10^8 ft./hr.^{2}

$$h_v = 0.725 \quad \frac{0.04966^3 \times 89.572^2 \times 4.17 \times 10^8 \times 54.806}{0.622 \times 0.0256 \times 27}$$

= 329.5 BTU/hr.deg.F.sq.ft. of inside surface of inner tube.

Overall Coefficient of heat transfer is determined from the equation:

$$U_o = \frac{1}{h_w} + \frac{D_1}{2}\,\frac{\log_e(D_1/D_i)}{K_{copper}} + \frac{D_1}{D_i}\,\frac{1}{h_v}$$

U_o - overall coefficient based on the outside surface area of the inner tube.

D_1 = $\frac{0.375}{12}$ = 0.03125 ft;

D_i = 0.0256 ft.

K_{copper} = 222 BTU/hr.ft.deg.F.

$$\therefore U_o = \frac{1}{1548} + \frac{0.03125}{2}\,\frac{\log_e \frac{0.03125}{0.0256}}{222} + \frac{0.03125}{.0256 \times 329.5}$$

= 0.00065 + 0.000014 + 0.0037

= 0.004364

The temperature drop through each of the film is proportional to the resistance of each film.

The final results of the trial and error method for the condensing film and overall coefficients of heat transfer are:

h_v (at 22.7° t) = 344 BTU/hr (deg.F) (Sq.ft. of inside surface)

$$\frac{1}{U_o} = 0.00065 + 0.000014 + \frac{0.03125}{0.0256 \times 344}$$

$$= 0.000664 + 0.003545$$

$$= 0.004209$$

U_o = 237.5 BTU/hr. deg.F sq.ft. of outside surface area.

Temperature drop across condensing film $= \frac{0.003545}{0.004209} \times 27$

= 22.7F.

Condensing film coefficient based on outside surface area $= \frac{0.0256 \times 344}{0.03125}$

= 281.5 BTU/hr. deg.F sq.ft. of outside surface area.

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31-5-63.
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