HEAT TRANSFER IN A SUBMERGED CONDENSER

By

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PREFACE

This thesis describes an effort to characterize the heat transfer effects when a saturated vapor condenses in a pool of liquid, causing agitation in the pool.

The extreme complexities of such a system preclude any attempt to characterize the system by a purely mathematical analysis. Rather, an effort to predict heat transfer coefficients must rely upon experiment and the application of dimensional analysis. Such has been the approach for this study.

This has been, to the knowledge of the author, the first serious attempt to characterize this type of system. As with most first attempts, many blind alleys have been explored. Also, only the steam-water system at atmospheric pressure was used in the experiment. With this limited background, it is doubtful that any completely general correlations were developed. However, the theory behind the correlations seems physically correct and the correlations developed do indeed describe the experimental data quite well.

I hope that this study will open the door to more research in this same area, both studying the same system with another fluid and extending the studies of the present system.

I wish to thank my adviser, Dr. Kenneth J. Bell, for his confidence and patience, and for his rescuing me from blind alleys.

I would also like to recognize Mr. E. E. McCroskey for his contribution to the design and construction of the equipment.

My wife, Doris, also deserves special credit for her encouragment and her endurance of many boring evenings.

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CHAPTER I

INTRODUCTION

Condensation is an important commercial process in which a saturated vapor is brought into contact with a cooler surface, heat is transferred from the vapor to the surface, and condensate is formed.

Agitation is an important unit operation which greatly enhances heat transfer. Agitation may be internally introduced in a liquid system by means of a mechanical stirrer or nonmechanically by allowing gas bubbles to rise through the system. Research on the agitation by rising gas bubbles has primarily been conducted using non-condensable gases. The vapor-induced agitation by a condensable gas has thus received little if any serious study in the past.

This study represents an attempt to study the effects on heat transfer of vapor-induced agitation via condensation in a condenser. The ultimate goal is to develop correlations which will allow the prediction of the heat transfer coefficients of a condenser with vapor-induced agitation.

CHAPTER II

BACKGROUND AND LITERATURE REVIEW

First of all, the background as to why this study was initiated seems appropriate. A condenser was required for a certain process. It was designed and ordered. The condenser was a horizontal shell and tube type with shell side condensation; in such a case standard practice calls for the baffle cuts to be vertical. When mounted in this position, however, the unit failed to work satisfactorily. The unit was then rotated 90° and operated over design. One attempt to explain this phenomenon held that the rotated baffles caused pools of liquid to be formed and that vapor leakage through the space between the baffles and tubes caused an agitation effect which enhanced the normal condensation. To test this theory, a small, single tube, horizontal condenser was built so that steam could be introduced below the surface level of a liquid pool. The study of the variables affecting the heat transfer was the objective of this thesis research.

A search of the literature was made to find out how others had described and correlated data concerning condensation and agitation.

Discussion of condensation ranges from the theoretical and mathematical approach of Nusselt (1), to the experimental correlations of Carpenter and Colburn (2), to strictly empirical correlations for specific systems. Unfortunately the correlations are confined almost strictly to condensation on solid surfaces. What research of direct contact condensation that was found in the literature either describes an unrelated system (such as presented by McCabe and Smith (3)), or describes a system without presenting a correlation (exemplified by Friedlander's work (4)), or attempts to describe condensation in terms of transfer units (such as in Rai's presentation (5)). In any case, no correlation applicable to this study could be found. A closely related field which offers possibly applicable theory is that of boiling heat transfer. Boiling is in effect something like condensation, only with the heat flow in the opposite direction. Most of the treatments of boiling by such people as Rohsenow (6), Jakob (7), and Forster and Zuber (8) again pertain to solid surfaces and involve the number of nucleation sites, frequency of bubble formation, contact angle of the bubbles to the surface, and other such phenomena not related to this study. Of the correlations studied, only the dimensionless ratio analysis by McNelly (9) contains most of the physical properties which might seem to affect the transfer of heat and at the same time not be dependent upon surface characteristics. The McNelly correlation and its relation to the author's derived correlation will be discussed in the section on dimensionless correlations.

The study of the effects of agitation by rising gas bubbles on heat transfer was recently presented by Hart (10). Although the source of agitation was different for his research, the agitation effects and the dimensionless ratios correlating these effects should correspond to those of this researh. The study and modification of his correlation will also be presented in the section on dimensionless correlations.

CHAPTER III

THEORETICAL DISCUSSION

The development of any purely mathematical or theoretical analysis of this system is beyond the scope of a Master's level thesis. Rather, this chapter will be devoted to the discussion of fundamental concepts and observations. Also, a study of existing correlations and their relationship to this system will be presented.

Description of Condensing and Agitation Process

A look at Figures 1 and 2 along with a brief description will acquaint the reader with some of the physical features and phenomena of this study. Figure 1 shows some of the main features of the condenser while Figure 2 illustrates some of the flow patterns in the water surrounding the entering steam.

Saturated steam enters at the bottom of the condenser and condenses as it comes in contact with the pool of water. The steam enters with a certain kinetic energy due to its velocity and a potential energy due to its position relative to the pool. The steam imparts its kinetic and potential energy to the pool of water causing an agitation of the pool. The latent heat of condensation, which the pool is absorbing as the steam condenses, is then transmitted to the cooler tube wall and then to the water flowing through the tube.

The transfer of heat from the incoming steam to the cooling water

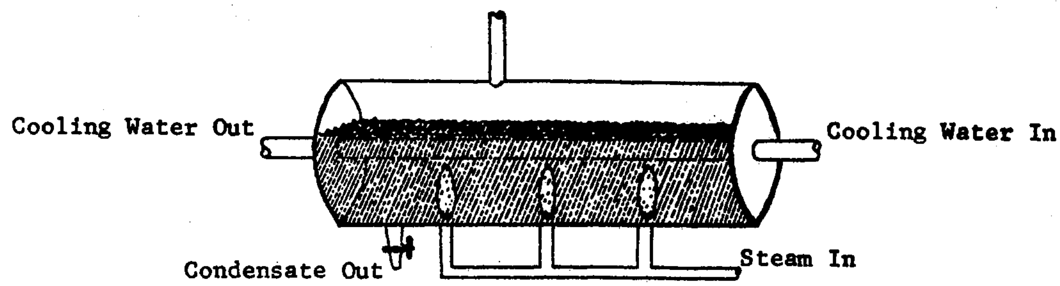
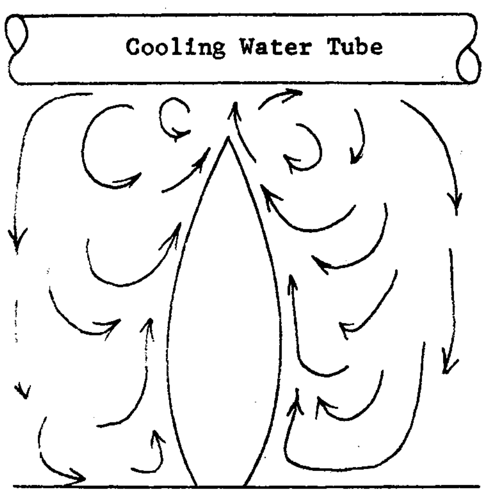
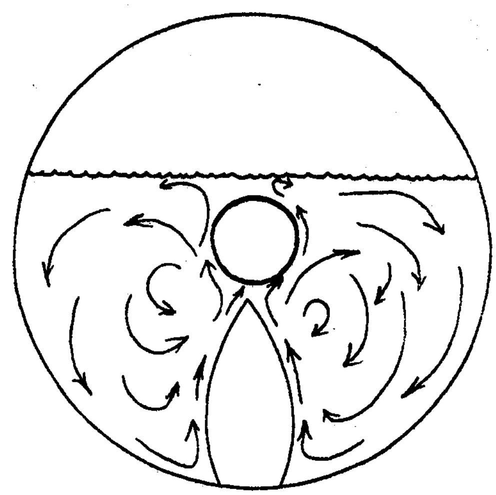


Figure 1. Main Features of Condenser Used in Research



(a)

Side View of Condenser



(b)

End View of Condenser

Figure 2. Liquid Flow Patterns in Condenser

in the tube can thus be broken into four steps: transfer of heat from the saturated steam to the agitated pool, transfer of heat from the agitated pool to the outside tube surface, transfer of heat through the tube wall, and transfer of heat from the inside tube surface to the cooling water flowing in the tube. The latter two heat transfer mechanisms have been well studied and correlated; thus, this study will be concerned only with describing the transfer of heat from the steam to the agitated pool, and from the agitated pool to the outside tube wall.

Liquid Flow Patterns

Figures 2-a and 2-b illustrate some of the fluid flow patterns existing in the pool of condensate. These flow patterns were observed by placing a small colored bead in the condenser and observing its motion. There were essentially three main channels of upward flow, these channels corresponding to the immediate perimeters of the entering vapor. The upward flow of liquid was interrupted by the cooling water tube situated immediately above the steam jet (see Figure 1). When the liquid reached the tube, it divided, part going up and around both sides of the tube, the other part traveling out and down (see Figure 2-b). The primary down circulation occurred along the shell wall. An example of the circulation just under the cooling water tube is shown in Figure 2-a. The circulation was not limited to strictly up or down motion; indeed there was much short circuiting. The system may thus be summarized by noting that there is an upward flow in the immediate vicinity of the steam jet, and a downward flow elsewhere, with a complicated radial flow pattern superimposed upon these two main streams.

Heat Transfer and Dimensionless Correlations

The complex hydrodynamics of this system obviously preclude a strictly mathematical or theoretical approach to the solution of the heat transfer problem. Rather, an attempt to correlate experimental data in terms of dimensionless ratios seems more fruitful. Dimensionless ratios may be constructed by grouping those properties which affect the process, or by writing in dimensionless form, the differential equations which describe the transport of heat and momentum. Both approaches are used in developing the correlations for this research.

The properties which most seem to affect condensation are related to the jet formed by the steam as it enters the liquid pool (See Figures 1 and 2). Affecting the jet are the steam velocity, V_s, (ft/hr), the viscosity of the pool liquid, μ, (lb/ft-hr), the thermal conductivity of the pool liquid, k, (Btu/hr-ft-$^{\circ}$F), the specific heat of the pool liquid, C_p, (Btu/lb-$^{\circ}$F), the enthalpy change of condensation, ΔH, (Btu/lb), the temperature driving force, ΔT, ($^{\circ}$F), the pressure of the system, P (lb_f/ft^2), the surface tension, σ, (lb_f/ft), the liquid density, ρ_l, (lb/ft^3), the vapor density, ρ_v, (lb/ft^3), and a characteristic length, d, (ft). From these properties are readily grouped the Reynolds and Prandtl numbers. Since the heat transfer coefficient, h, (Btu/hr-ft^2-$^{\circ}$F) is to be characterized, it can be incorporated in the Nusselt number. A fourth dimensionless ratio is constructed involving the pressure, surface tension, and the characteristic length. Of all the properties previously presented, those not used in one of the dimensionless ratios are ΔH and ΔT. Looking at basic Nusselt theory, the equation for laminar film condensation on a vertical surface is:

$$h = .943\left(\frac{\rho^2 g\lambda k^3}{\mu x\Delta T}\right)^{\frac{1}{4}} \tag{1}$$

where: g = acceleration due to gravity, ft/hr^2

λ = latent heat of condensation, Btu/lb

x = length of the condensing surface, ft

(other terms as defined above)

Although few of the requirements for Nusselt condensation are met, the equation does demonstrate that the heat transfer coefficient is proportional to $\lambda/\Delta T$. All of the nonconstant terms of equation 1 are used in the dimensionless ratios already chosen except for λ and ΔT. Since λ is approximately equal to ΔH, the proportionality becomes

$$h \propto \left(\frac{\Delta H}{\Delta T}\right)^{\frac{1}{4}} \tag{2}$$

With the addition of the specific heat term, a dimensionless ratio is formed. All of the properties previously mentioned may thus be combined into the following dimensionless ratio relationship:

$$\frac{hd}{k} = f\left[\left(\frac{Cp\mu}{k}\right), \left(\frac{dVs(\rho_1-\rho_v)}{\mu}\right), \left(\frac{Pd}{\sigma}\right), \left(\frac{\Delta H}{Cp\Delta T}\right)\right] \tag{3}$$

At this stage, one must assume some kind of functional relationship among the dimensionless groups. The most common assumption, one that is quite successful in many cases, is that the groups are related in a logarithmically linear manner, such as:

$$\frac{hd}{k} = a\left(\frac{Cp\mu}{k}\right)^b \left(\frac{dVs(\rho_1-\rho_v)}{\mu}\right)^c \left(\frac{Pd}{\sigma}\right)^e \left(\frac{\Delta H}{Cp\Delta T}\right)^f \tag{4}$$

The question arises as to what is the proper characteristic length. The diameter of the steam jet at its base, its height, its maximum diameter, or the depth of the pool are all possible choices for a characteristic length. The jet height and maximum diameter are transient values and difficult to measure. Since they are also functions of the jet diameter at its base, they would not seem to be appropriate choices for the characteristic length. Preliminary investigations also showed that the heat transfer was very insensitive to pool depth. Thus the diameter of the jet at its base, a value which directly affects the steam velocity, will be defined to be the characteristic length. The diameter of the holes through which the steam enters the condenser is considered to be the diameter of the steam jet at its base.

While studying the references concerning boiling heat transfer, it was found that McNelly had described boiling heat transfer in terms similar to those in equation 4. His correlation for boiling heat transfer is:

$$\frac{hd}{k} = a\left(\frac{Cp\mu}{k}\right)^{b}\left(\frac{Q/A\ d}{\lambda\mu}\right)^{b}\left(\frac{Pd}{\sigma}\right)^{(1-b)}\left(\frac{\rho_1}{\rho_v} - 1\right)^{c} \tag{5}$$

where: Q/A = heat flux, Btu/hr-ft^2

The first, second, and fourth ratios of the equation above are identical to the ratios used in equation 4. The ratio involving the heat flux is actually a modified Reynolds number with the heat flux being roughly equivalent to a velocity. The last term in McNelly's correlation was introduced to account for the change in volume when the liquid vaporized during boiling. The effect of this term has already been incorporated into the Reynolds number of equation 4. Thus it can be seen that the author's equation is equivalent to the McNelly correlation with a one ratio addition.

The heat transfer coefficient from agitated pool to tube wall may be characterized by using arguments similar to those of Hart (10).

Any heat transfer process which can be described by differential equations can be described in terms of the Nusselt, Reynolds, Grashof, Prandtl, and Froude numbers, or by special ratios of these numbers, such as the Peclet and Stanton numbers. All or some of the dimensionless ratios, depending upon the situation, may be needed to describe the transfer of heat. Usually, the Nusselt number is given as a function of the Prandtl number and one or more of the other ratios. I.e.,

$$Nu = f(Pr, Re, Gr, Fr) \tag{6}$$

Each of these ratios describe two forces at work in the system. For instance, the Reynolds number relates inertial and viscous forces, the Grashof number relates viscous and gravitational forces, and the Froude number accounts for the presence of inertial and gravitational forces. An agitated system obviously involves inertial and viscous forces, so the Reynolds number must enter the correlation. The gravitational forces are also important in a system with a free surface (see Bird (11)), The acceleration due to gravity is represented in the Froude and Grashof numbers. The Grashof number is important when only natural convection is occuring. When any forced convection is occuring, such as in the system being studied, the Grashof number is usually negligible and in this case, will be ommitted. Thus the acceleration due to gravity will be represented in the Froude number for this correlation. The correlation could be expected to take the form

$$Nu = f(Re, Fr, Pr) \tag{7}$$

Assuming a logarithmically linear relationship for these groups, the resulting equation is

$$\frac{hd}{k} = a\left(\frac{V_p D \rho}{\mu}\right)^b \left(\frac{V_p^2}{gD}\right)^c \left(\frac{C_p \mu}{k}\right)^d \tag{8}$$

where: V_p = superficial velocity of the steam, ft/hr

D = characteristic length, ft

The above is the form in which Hart presented his correlation. The superficial steam velocity is appropriate for this correlation since it relates agitation to the volume of the condenser, i.e., the larger the condenser, the lower the superficial velocity, which corresponds to a lesser agitation. Since the transfer of heat is occurring from the pool to the tube wall, the logical choice for the characteristic length is the tube diameter. Hart found in his research that the heat transfer characteristics were independent of any characteristic length, and thus modified equation 8 such that the exponents, b and c, were related by the equation, $b-c = 1$. Insufficient research has been done with this type of system to reach a similar conclusion for this study.

CHAPTER IV

DESCRIPTION OF APPARATUS

A flow diagram of the apparatus is shown in Figure 3. Additional detail of the condenser is provided by the scaled drawing in Appendix C.

The main piece of equipment for the study was the condenser. It was designed by the author and constructed from a 12 inch long, 4 inch diameter cylinder of pyrex glass such that agitation and condensation effects could be observed.

The copper tube running through the condenser had a 0.626 inch outside diameter and a 0.547 inch inside diameter.

The steam drier, a metal cylinder filled with aluminum turnings, was used to knock out any entrapped liquid in the incoming steam. This drier has been used in other heat transfer experiments and has been shown to be virtually 100% effective.

An orifice was placed in the pipe carrying the cooling water. A mercury filled, U-tube manometer was used to measure the pressure drop across the orifice, the pressure drop being proportional to the water velocity.

Five 16 gauge, copper-constantan thermocouples were used to measure the temperatures. The thermocouples were located as shown in Figure 3. The thermocouple providing the C reading was located approximately above the condensate removal valve, ½ inch from the pool surface and ½ inch from the tube. Thermocouple D was placed in the area between two steam

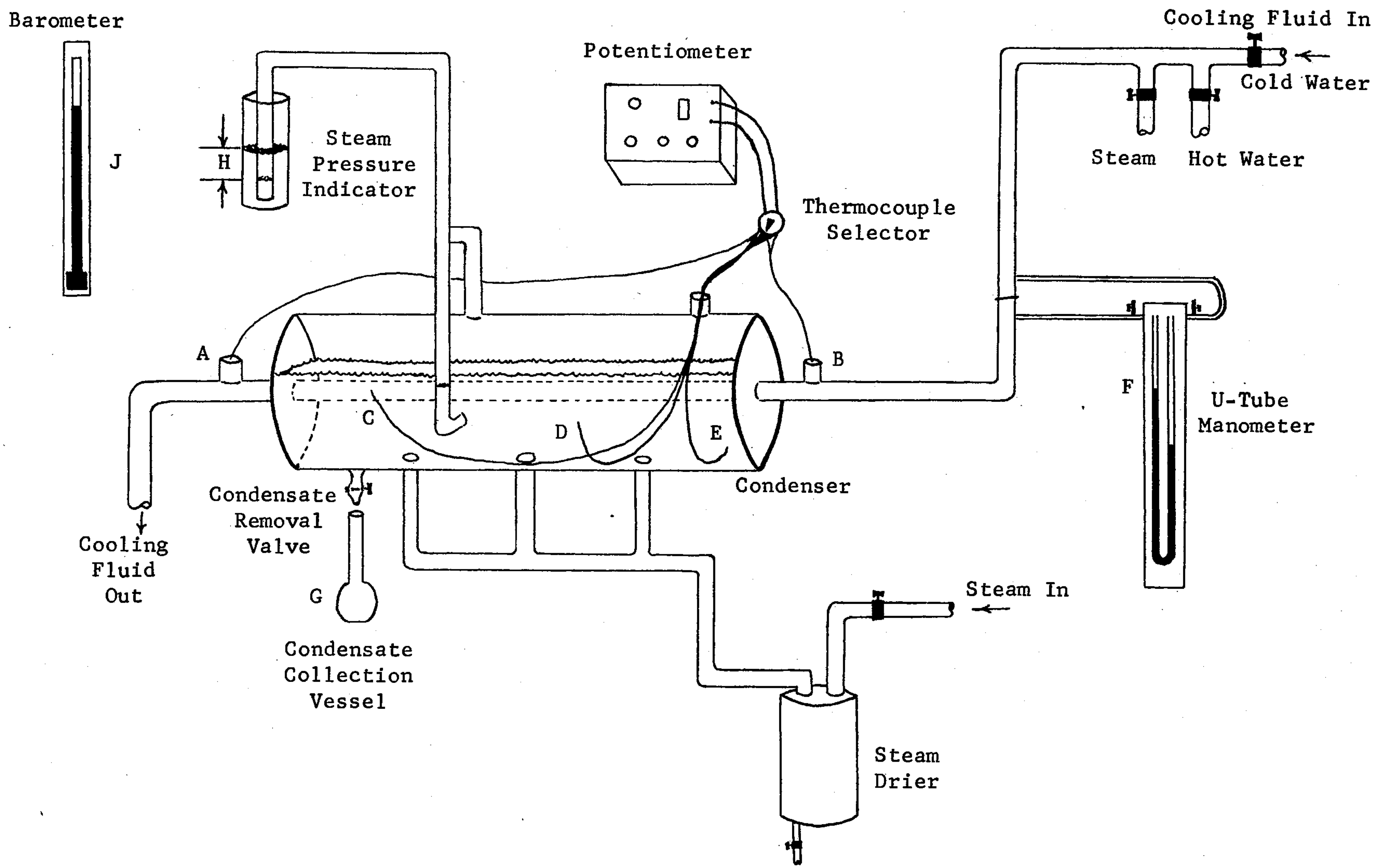


Figure 3. Apparatus For Submerged Condenser Research

jets, at approximately half the depth of the pool. Thermocouple E was located half way between the jet and the end of the condenser, approximately ¼ inch from the bottom of the pool.

The condensate collection vessel was a 500 milliliter volumetric flask.

The condensing pressure indicator was formed by running a piece of Tygon tubing from the condenser into a flask of water. Room pressure was measured by a mercury barometer.

Not indicated in the figure is the insulation. The steam drier and the line leading from the drier to the condenser was insulated using several layers of glass tape. A fiberglass wool was used to insulate the condenser. The steam line to the drier was also insulated.

CHAPTER V

EXPERIMENTAL PROCEDURE

This chapter is concerned with the calibration of the apparatus, the general operation of the system, the method of taking data, and the treatment of the data obtained. Figure 3 illustrates the system.

Calibrations

The components requiring calibration were the thermocouples and the orifice. The thermocouples were calibrated by placing the bare junctions in an ice bath and in boiling water. All thermocouples registered 32°F in the ice bath, but varied from 209.5 to 212°F in the boiling water. A linear variation was assumed over the range and correction factors added when converting the raw data to temperatures used in the calculations.

The orifice was calibrated by recording the position of the mercury in the U-tube manometer at various water velocities. The velocity of the water through the tube was calculated by weighing the quantity of water flowing through the system during a time period and dividing the volume of this quantity by the cross sectional area of the tube.

General Operation

The initial step was to establish a flow rate of water through the tube. The temperature of this water was controlled by the mixing of cold water, hot water, and steam at the water inlet (see Figure 3). The next step was to allow steam to enter the condenser. The condensate removal valve was closed until a liquid level could be established above the tube. The condensate removal valve was then opened just enough so that the liquid level remained constant, i.e., the condensate being formed just equalled the condensate being drained off. The system was then ready for the necessary measurements to be made.

Data Collection

For each set of runs, the temperature of the water flowing through the tube was held constant. All readings were taken at steady state conditions. Steady state was considered established if the temperature of the pool of condensate remained constant for a period of five minutes and the level of the condensate was constant. Referring to Figure 3, the measurements for one run were the cooling water temperatures, A and B, the pool temperatures, C, D, and E, the water flow rate, F, the condensate rate, G, the condensing pressure, H, and the room pressure, J. The five temperatures were recorded in the form of emf readings from the potentiometer. The water velocity was recorded in the form of a manometer reading. The level of the mercury tended to fluctuate occasionally during the run. Only the stabilized value was recorded. The condensate rate was measured by recording the time needed to collect 500 milliliters of the condensate. The condensing pressure was recorded by measuring the difference in water levels and the room pressure recorded by using a mercury barometer.

After the readings were recorded, the flow of steam to the system was changed. When the pool temperature and pool level once more stabilized, a new set of readings were taken. One set of runs consisted of several steam flow rates all taken at a fixed tube side temperature.

Treatment of Data

The thermocouple raw data were converted from the emf readings to degrees F, the appropriate corrections added, and the respective temperatures averaged to find the cooling water and pool temperatures used in the calculations. The condensate rate, recorded as pounds of condensate per minute, was converted to pounds of condensate per hour. The pressure measurements, H and J were combined to find the pressure at which the steam was condensing. If the water level in the tube and the beaker were equal, the system pressure was equal to atmospheric. If the height of the water in the tube were lower than the water height in the beaker, the system was that much above atmospheric. The temperature of saturated steam at this pressure was recorded and used in the calculations. The heat flux of the condenser was calculated by multiplying the condensate rate times the change in enthalpy of the steam and dividing this product by the outside tube area. The equation for heat flux is thus:

$$Q/A = W(H_{steam} - H_{pool})/A \tag{9}$$

where: Q/A = heat flux, Btu/hr-ft^2-$^\circ$F

W = condensate rate, lb/hr

H = enthalpy, Btu/lb

A = outside area of tube, ft^2

The heat transfer coefficient from steam to agitated pool is found by dividing the heat flux by the appropriate ΔT driving force:

$$h_{stap} = \frac{Q/A}{(T_s - T_p)} \tag{10}$$

where: h_{stap} = heat transfer coefficient from steam to agitated pool, Btu/hr-ft^2-°F

T_s = temperature of the steam, °F

T_p = temperature of the pool, °F

The heat transfer coefficient from agitated pool to tube wall is likewise found by dividing the heat flux by the appropriate ΔT driving force:

$$h_{aptw} = \frac{Q/A}{(T_p - T_w)} \tag{11}$$

where: h_{aptw} = heat transfer coefficient from agitated pool to tube wall, Btu/hr-ft^2-°F

T_w = temperature of the tube wall, °F

Since the temperature of the wall was not measured, it must be calculated before the previous equation can be solved. The temperature of the wall is found from the equation:

$$Q/A = h_{twtcw}(T_w - T_{cw}) \tag{12}$$

where: h_{twtcw} = heat transfer coefficient from the tube wall to the cooling water, Btu/hr-ft^2-°F

T_{cw} = temperature of the cooling water, °F

Rearranging equation 12,

$$T_w = \frac{Q/A}{h_{twtcw}} + T_{cw} \tag{13}$$

The heat transfer coefficient through the tube wall to the cooling water, referenced to the outside tube area, may be represented by:

$$\frac{1}{h_{twtcw}} = \frac{1}{\frac{(r_o - r_i)r_o}{k_{tw}\;\;r_m} + \frac{1}{h_{ts}}\frac{r_o}{r_i}} \tag{14}$$

where: k_{tw} = thermal conductivity of the tube, Btu/hr-ft^2-°F

r_o = outside radius of tube, ft

r_i = inside radius of tube, ft

r_m = mean radius of the tube, ft

h_{ts} = heat transfer coefficient from the inside of the tube wall to the cooling water, Btu/hr-ft^2- $^\circ$F

The tube side heat transfer coefficient for water being used as a cooling fluid is given by Coulson and Richardson (12). The equation is:

$$h_{ts} = 200(1 + .0157T_{cwc})\frac{V_w^{.8}}{d_i^{.2}} \tag{15}$$

where: T_{cwc} = temperature of the cooling water, $^\circ$C

V_w = cooling water velocity, ft/sec

d_i = inside diameter of tube, in.

Thus calculating the equations in the order, 15, 14, 13, the wall temperature can be found and the heat transfer coefficient from agitated pool to tube wall thus be calculated using equation 11.

Tabulated data for each of the runs may be found in Table B-1 in the appendix. A sample calculation may be found in Appendix D.

CHAPTER VI

RESULTS

Graphical and pictorial analyses of the results are to be found on the following pages. Specific data for each run may be found in Table B-I in Appendix B. Since the runs are grouped according to the cooling water temperatures, the wording, "a 67°F run", indicates the data corresponding to the cooling water temperature being from 67 to 73°F. The groupings of cooling water temperatures may be found on each figure.

ΔT, Pool Temperature - Wall Temperature

One of the most interesting features of this system can be demonstrated by plotting the driving force temperature difference (pool temperature minus the wall temperature) as a function of the heat flux. It may be noted that in general, the temperature difference is fairly constant throughout the heat flux range. See Figure 4.

The first data taken were in the 67°F group. The driving force varied only about 8°F over the entire heat flux range. The 98°F and 106°F runs provided similar plots. Although the 49°F run had a slightly negative slope, it was still fairly consistent. Data for the 109°F group were also very constant above a heat flux of about 58,000 or below a heat flux of about 43,000. Similar discontinuities were found for the runs with the higher cooling water temperatures. For instance, the plot of the 151°F run was almost exactly parallel to the 109°F run,

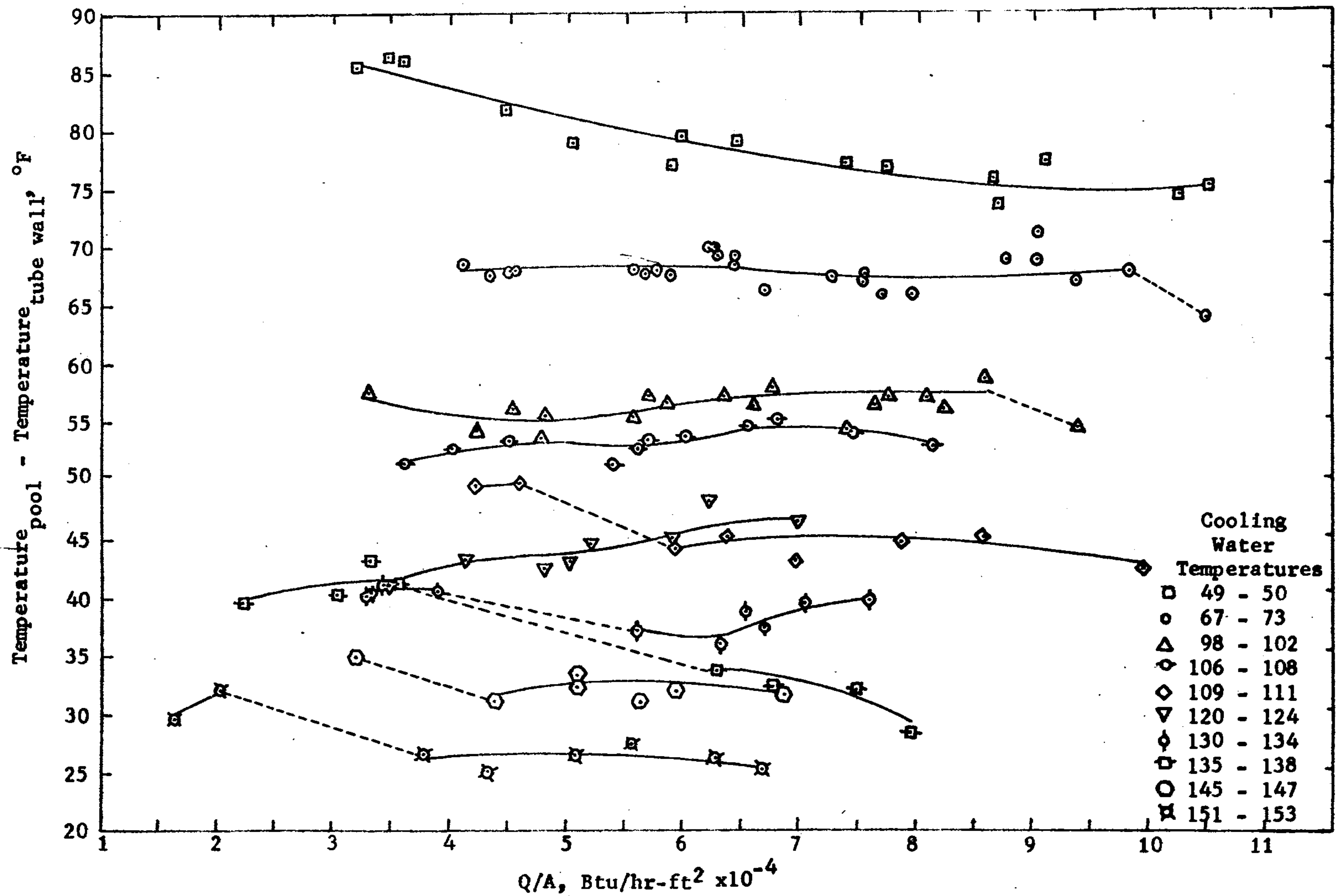
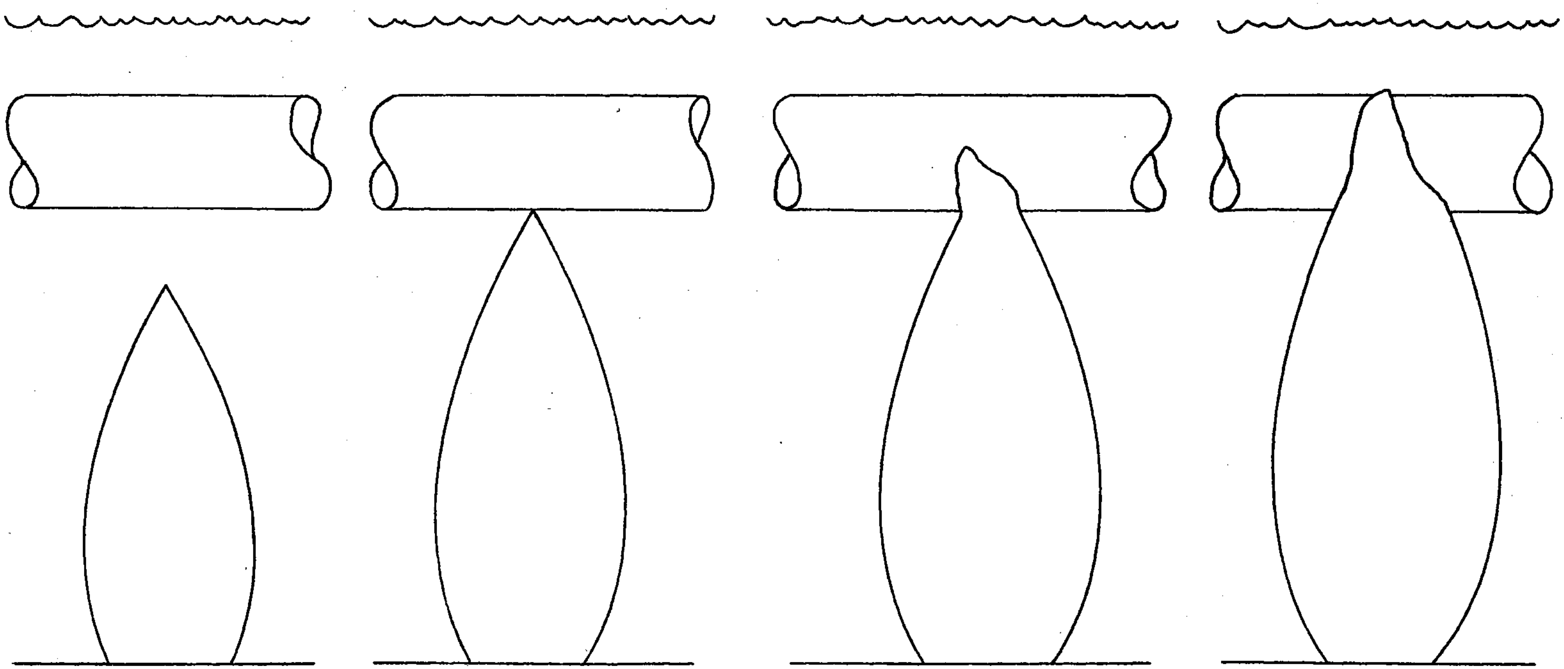


Figure 4. Temperature Difference Between Pool and Tube Wall as a Function of Heat Flux

except that the break points were at heat fluxes of 20,000 and 38,000.

A possible model which accounts for much of this behavior is based on the steam jet observations presented in Figure 5. Figure 5 illustrates different steam jet conditions which may occur in the condenser. Figure 5a shows the steam jet well down in the condensate pool, while Figures 5b, 5c, and 5d show the jet just starting to touch the tube, partially surrounding the tube, and completely surrounding the tube. When the steam jet is in a condition corresponding to Figures 5c or 5d, part of the jet is in contact with the pool, and the rest is in contact with the cooler tube wall. Thus, the jet is exposed to two different ΔT driving forces. With the increased temperature difference for the part of the jet in contact with the cooling water tube, the heat flux is suddenly increased without a marked rise in the pool temperature. Reference to equation 13 indicates that the wall temperature is calculated as a function of the heat flux. Hence, the calculated wall temperature would rise suddenly with no corresponding rise in the pool temperature. This accounts for the discontinuities in Figure 4. Low cooling water temperatures would cause the steam jet to condense faster, and thus to never reach the cooling water tube (which accounts for the lack of discontinuity in the 49°F run). The 67°F run has a marked discontinuity at the last point in the high flux region. This could correspond to the steam jet finally reaching the tube. Looking at the 98°F run, a similar discontinuity is found in the high flux region. There seems to be a definite pattern to the discontinuities (indicated by dashed lines) found in Figure 4. The discontinuity shifts to the left as the cooling water temperatures increase. If, indeed, the steam jet's contact with the cooling water tube is the correct explanation for the deviations,



(a) (b) (c) (d)

Figure 5. Various Steam Jet Conditions Which May Occur in Condenser

some criteria should exist that would determine at what conditions the steam jet would extend up far enough to contact the tube. Two variables which would logically seem to affect the steam jet height are the steam velocity and the pool temperature. However, there was no set temperature or velocity at which the discontinuity would occur. It would thus appear that either these are not the proper criteria, or that it is some complex relation of these two variables that affect the jet height.

Heat Transfer Coefficient, Agitated Pool to Tube Wall

With the discontinuities of Figure 4 in mind, the heat transfer coefficient from agitated pool to tube wall was plotted as a function of heat flux (see Figure 6). Again, the 49°F and 67°F runs showed very linear behaviors. However, as the cooling water temperatures increased, the discontinuities (again indicated by the dashed lines) became quite apparent. The data points both above and below the region of discontinuity are reasonably linear, but not with the same slopes. This suggests that the data be divided into two groups. It may be noted that in Figure 6, the discontinuities move to the left as the cooling water temperatures increase, but unlike Figure 4, the progression is in a horizontal manner. This suggests that there is possibly some value of the heat transfer coefficient which characterizes the point at which the steam jet reaches the cooling water tube. The value of 1400 Btu/hr-ft^2-°F seems to come closest to being a good indication of a jet-tube contact criterion. The data with a heat transfer coefficient from agitated pool to tube wall of less than 1400 Btu/hr-ft^2-°F can be correlated with an average deviation of 2.96% or less by the equation

$$Nu_{aptw} = .0385Re^{.975}Fr^{-.166}Pr^{.211} \tag{16}$$

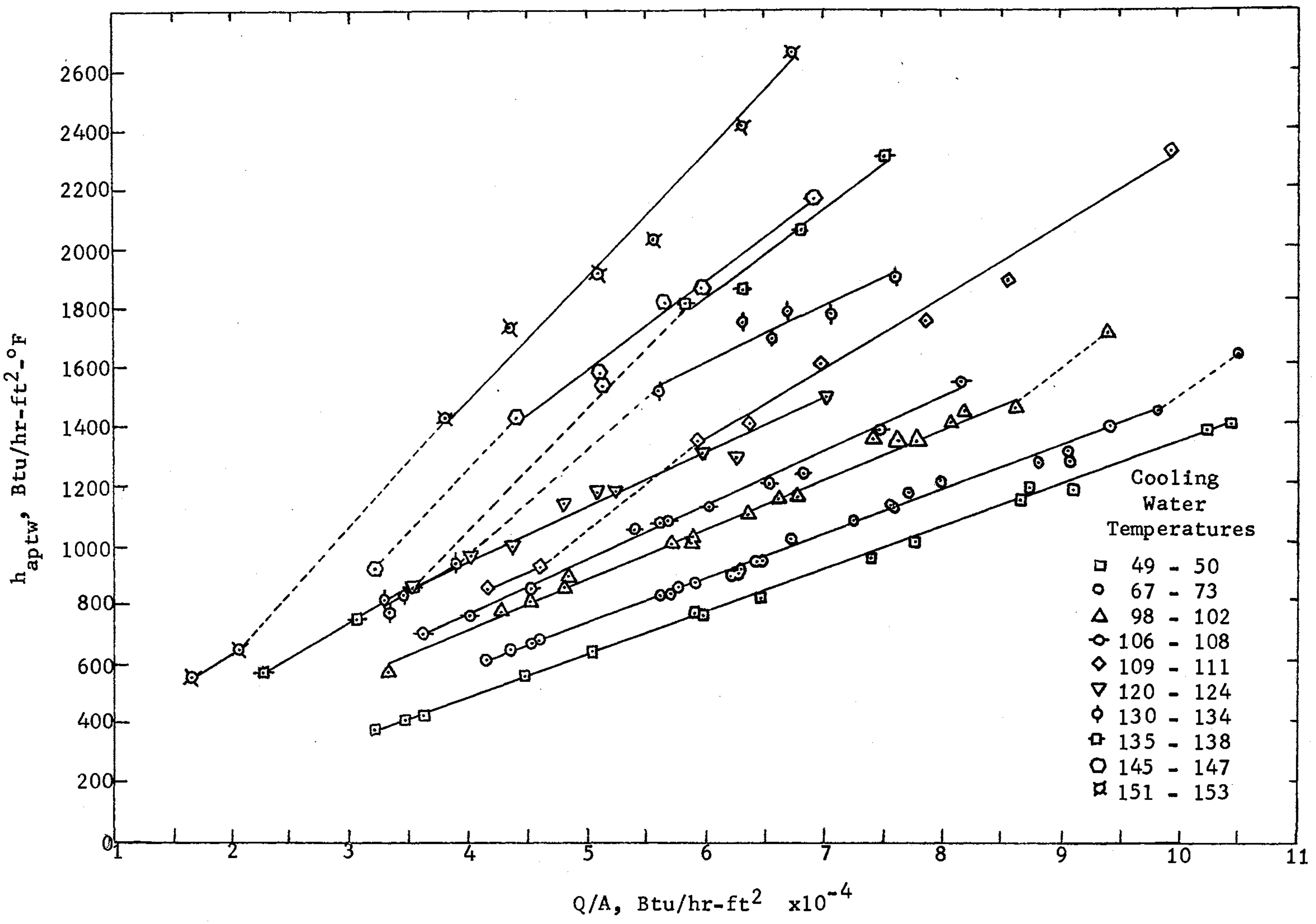


Figure 6. Heat Transfer Coefficient, Agitated Pool to Tube Wall, as a Function of Heat Flux

The data with a heat transfer coefficient of greater than 1400 Btu/hr-ft^2-°F can be correlated with an average deviation of 6.3% or less by the equation

$$Nu^*_{aptw} = .00369\ Re^{1.31} Fr^{-.356} Pr^{-.687} \tag{17}$$

However, according to the proposed model, this latter case represents the steam jet touching the cooling water tube, a condition with two different ΔT driving forces. Hence, this correlation could be expected to fit data only from this experimental apparatus. Application of equation 16 to the data above the 1400 Btu/hr-ft^2-°F division point, provides a conservative estimate of the heat transfer coefficient for this data.

It was found that the deviation of the correlation was minimum when the physical properties were evaluated at the wall temperature.

The quantities which varied most over the experimental range were the superficial velocity and the viscosity. Thus the exponents of the Reynolds and Froude numbers could be considered to well represent this system. However, since the specific heat and thermal conductivity varied so little over the range tested, no special significance may be attached to the value of the exponent of the Prandtl number.

One "rule of thumb" noted for this study is that for the data with h_{aptw} less than 1400 Btu/hr-ft^2-°F, the driving force ΔT, (pool temperature minus wall temperature) may be estimated by the equation

$$\Delta T \cong \tfrac{1}{2}(T_s - T_{cw}) \tag{18}$$

If this driving force remains constant throughout the heat flux range, then the limiting condensation rate would exist when the temperature of the wall was ΔT degrees less than that of the incoming saturated vapor (the maximum possible pool temperature).

This "rule of thumb" is, possibly, valid for only this experimental apparatus, but provides interesting possibilities if applicable to other systems.

Heat Transfer Coefficient, Steam to Agitated Pool

Figure 7 shows a log-log plot of the heat transfer coefficient from steam to agitated pool as a function of the heat flux. It should be noted that the cooling water temperatures have been grouped differently than the two previous graphs. The 49 - 73°F group shows a very consistent set of data in spite of the 24°F range of cooling water temperatures. The curves of the 98°F and 106°F groups are also consistent with the 49°F run. However, the 109°F run has a marked discontinuity indicated by the dashed line. This discontinuity may be explained by reference to Figure 5 and the previous discussion. Similar breaks are quite apparent for the 130°F, 135°F, and 151°F runs. The data for the 145°F run do not include a marked discontinuity, but there are insufficient data in the lower region to be conclusive. The 120°F run is interesting in that it is very consistent and obviously without the expected discontinuity. A look again at Figure 6 and the pattern of discontinuities indicates that this run has only one value above the arbitrary break point of 1400 Btu/hr-ft^2-°F. Whether this inconsistency indicates a data measurement error, a special set of conditions occuring only during this run, or a flaw in the proposed model is unresolved. It may be noted that of all the runs, the 124°F run had the lowest system pressures.

One other feature worth noting is the hyperbolic behavior of the steam to agitated pool heat transfer coefficient. While the condensing heat transfer coefficient limits the transfer of heat in most systems,

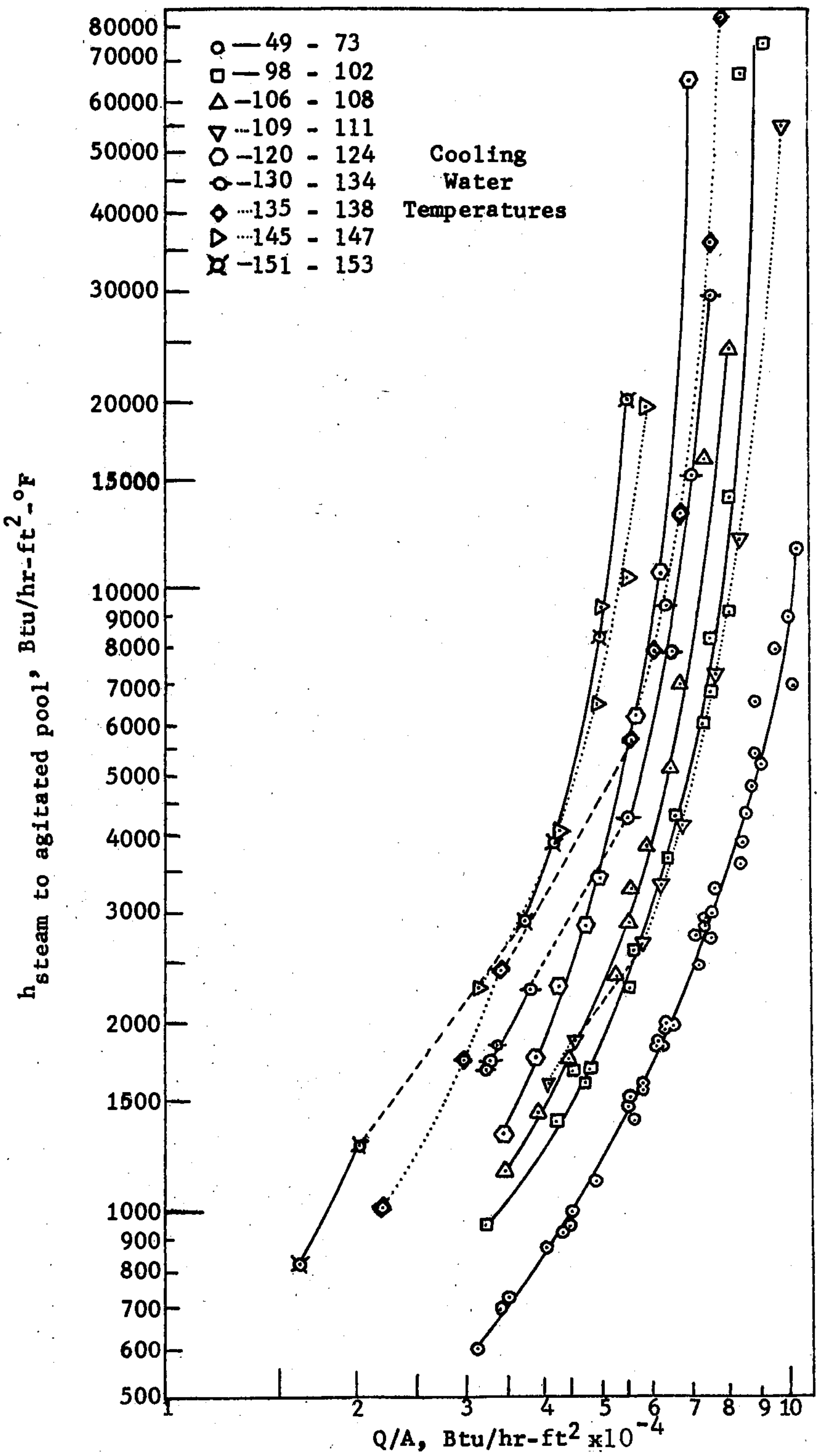


Figure 7. Heat Transfer Coefficient from Steam to Agitated Pool as a Function of Heat Flux

it tends to become a negligible resistance to heat transfer for this system at very low driving force temperature differences.

Since maximum heat flux is usually a design objective, a look at what conditions improve the heat flux is appropriate. Figure 7 indicates that, as expected, the maximum heat flux is obtained with the coolest cooling water. The 49°F curve is very consistent in spite of the 24°F variation in the temperature of the cooling water. This indicates that there is some temperature below which the heat flux is not significantly increased by decreasing the cooling water temperature. Assuming the proposed model to be valid, heat flux may also be increased by forcing the steam jet into contact with the cooling water tube. The effect of this is seen as the heat flux of the 109°F run becomes greater than the 98°F run after the discontinuity is passed.

The results of a non-linear regression curve fit indicates that the equation which best fits the data is

$$\frac{hd}{k} = 4.97\text{x}10^{-5}\left(\frac{dV_S(\rho_1-\rho_v)}{\mu}\right)^{1.025}\left(\frac{Cp\mu}{k}\right)^{.739}\left(\frac{Pd}{\sigma}\right)^{-.025}\left(\frac{\Delta H}{C_p\Delta T}\right)^{.99} \qquad (19)$$

This equation fits the data with an average deviation of 1.6%. For this equation, the physical properties were evaluated at the pool temperature. The reason for this close fit can be observed by selectively grouping the terms, V_S, ΔH, ΔT, those values which most significantly vary throughout the various runs. The grouping

$$\frac{V_S \Delta H}{\Delta T}$$

is roughly equivalent to

$$\frac{W \Delta H}{A \Delta T}$$

which is the form used in equations 9 and 10 to calculate the reported value for the heat transfer coefficient from steam to agitated pool. The problem with equation 19, is that it requires so many values to be known prior to the calculation, that the calculation in effect reveals very little new information.

The heat transfer coefficient from steam to agitated pool may best be summarized by its behavior, that it provides a negligible resistance to heat transfer as the ΔT driving force $(T_s - T_p)$ approaches zero.

Error Analysis

The objective of this discussion is to estimate the maximum possible error accruable in any single run.

For the heat transfer coefficient from agitated pool to wall, the measurements and possible deviations are:

Temperature measurements

Pool temperature, 3 thermocouples @ $\pm$ .3°F = .3°F

Wall temperature,

Cooling Water temperature, 2 thermocouples @ $\pm$ .3°F = .3°F

Water velocity $\pm$ 5% - a

Minimum $\Delta T, (T_p - T_w)$ measured - 26°F

Maximum Temperature Error $= \dfrac{.3^\circ F + .3^\circ F}{26^\circ F} = 2.3\%$ - b

Condensate rate $\pm$ 4% - c

TOTAL POSSIBLE ERROR = a + b + c = 11.3%

For the heat transfer coefficient from steam to agitated pool, the measurements and possible deviations are:

Temperature measurements

Pool temperature, same as above, .3°F

Steam temperature, $\pm$ .2°F

Minimum T $(T_s - T_p)$, measured - $.1^{o}F$

Maximum Temperature Error = $\frac{.3^{o}F + .2^{o}F}{.1^{o}F}$ = 500% - d

Condensate rate $\pm$ 4% - e

TOTAL POSSIBLE ERROR = d + e = 504%

The possible error of 504% should tend to raise a few eyebrows. However, considering that the heat transfer coefficients corresponding to a $.1^{o}F$ driving force are in the neighborhood of 150,000 Btu/hr-ft^2-^{o}F, this error seems more acceptable.

Considering only h_{stap} values of less than 10,000 Btu/hr-ft^2-^{o}F, the minimum value of $(T_s - T_p)$ is $7.2^{o}F$, which yields a maximum possible temperature error of 6.8% and hence a total possible error of 10.8%.

Again, considering only h_{stap} values of less than 5,000 Btu/hr-ft^2-^{o}F, the minimum $(T_s - T_p)$ is $10.8^{o}F$, the maximum possible temperature error is 4.63% and the total possible error is 8.63%.

The figures presented above represent estimates of the maximum possible errors involved. Perhaps a more realistic picture can be obtained by considering the reproducibility of the data obtained in the various runs. For instance, the data composing the 67 - $73^{o}F$ group of data was collected on 7 separate occasions and yet is very consistent.

CHAPTER VII

CONCLUSIONS AND RECOMMENDATIONS

The transfer of heat in a submerged condenser may be described by two heat transfer coefficients. The first is the heat transfer coefficient from steam to agitated pool, and the second is the heat transfer coefficient from agitated pool to tube wall. The proposed model that describes the heat transfer mechanisms divides the data into two groups. The first group of data corresponds to the steam jet being completely submerged in the condensate pool, while the second group corresponds to the steam jet being in contact with the cooling water tube. The correlation set forth for the transfer of heat from agitated pool to tube wall applies only to data in the first group. This data can be correlated with an average precision of 2.96% or less by the equation

$$Nu_{aptw} = .0385\ Re^{.975} Fr^{-.166} Pr^{.211}$$

Data in the second group are consistent but, according to the model, dependent upon two ΔT temperature differences. A conservative estimate of the heat transfer coefficients for this second group may be made by applying the above equation to the data.

The results of this study indicate that the heat transfer coefficient from steam to agitated pool can be correlated with an average precision of 1.6% by the following equation

$$\frac{hd}{k} = 4.97\times10^{-5}\left(\frac{dV_s(\rho_l-\rho_v)}{\mu}\right)^{1.025}\left(\frac{C_p\mu}{k}\right)^{.739}\left(\frac{Pd}{\sigma}\right)^{-.025}\left(\frac{\Delta H}{C_p\Delta T}\right)^{.99}$$

Two noteworthy features of this study are the ΔT driving force (pool temperature minus wall temperature) associated with the transfer of heat from the agitated pool to the tube wall, and the general behavior of the steam to agitated pool heat transfer coefficient. The ΔT driving force associated with a specific cooling water temperature tends to remain virtually constant throughout the range of heat fluxes. The heat transfer coefficient from steam to agitated pool has a hyperbolic behavior such that it provides a negligible resistance to heat transfer at very low ΔT driving force values.

It is recommended that the same system be reexplored with another fluid or possibly just using steam at a higher pressure. Since the heat transfer is superior when the steam jet is in contact with the cooling water tube, it is suggested that any data taken be concentrated in this region. Altering the position of the cooling water tube to a lower level, or moving it to the side of the steam jet could provide enlightening information. Altering the condenser to a vertical position and allowing the steam to enter near and rise up along the tube wall could provide some very interesting heat transfer research.

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APPENDIX A

NOMENCLATURE

A	-	outside area of tube, ft^2
aptw	-	subscript referencing heat transfer from agitated pool to tube wall
C_p	-	specific heat of pool liquid, Btu/lb-°F
d	-	characteristic length in h_{stap} correlation, ft
D	-	characteristic length in h_{aptw} correlation, ft
g	-	acceleration due to gravity, ft/hr^2
h	-	a heat transfer coefficient, Btu/hr-ft^2-°F
h_{aptw}	-	heat transfer coefficient from agitated pool to tube wall, Btu/hr-ft^2-°F
h_{stap}	-	heat transfer coefficient from steam to agitated pool, Btu/hr-ft^2-°F
h_{ts}	-	heat transfer coefficient from inside of tube wall to cooling water, Btu/hr-ft^2-°F
h_{twtcw}	-	heat transfer coefficient through tube wall to cooling water, Btu/hr-ft^2-°F
ΔH	-	enthalpy change as steam condenses, Btu/lb
k	-	thermal conductivity of pool liquid, Btu/hr-ft^2-°F/ft
k_{tw}	-	thermal conductivity of tube wall, Btu/hr-ft^2-°F
P	-	system pressure, lb_f/ft^2
Q/A	-	heat flux, Btu/hr-ft^2
r_i	-	inside radius of cooling water tube, ft
r_m	-	mean radius of cooling water tube, ft
r_o	-	outside radius of cooling water tube, ft

stap - subscript referencing heat transfer from steam to agitated pool

T_{cw} - temperature of the cooling water, ^{o}F

T_{cwc} - temperature of the cooling water, ^{o}C

T_p - temperature of the agitated pool, ^{o}F

T_s - temperature of the steam, ^{o}F

T_w - temperature of the tube wall, ^{o}F

ΔT - a temperature difference driving force, ^{o}F

V_p - superficial velocity of steam through the system, ft/hr

V_s - velocity of the steam as it enters the system, ft/hr

V_w - cooling water velocity, ft/sec

W - condensate rate, lb/hr

x - length of condensing surface, ft

λ - latent heat of condensation, Btu/lb

μ - viscosity of the pool, lb/ft-hr

σ - surface tension of liquid against its vapor, lb_f/ft

ρ_1 - density of the pool liquid, lb/ft^3

ρ_v - density of the steam vapor, lb/ft^3

Fr - Froude Number, $V_p{}^2/gD$

Gr - Grashof Number, $D^3\rho^2 gB\Delta T/\mu^2$

Nu - Nusselt Number, hd/k

Nu_{aptw} - Nusselt Number for heat transfer from agitated pool to tube wall, hd/k

Nu^*_{aptw} - Nusselt Number for heat transfer from agitated pool to tube wall with unknown ΔT

Pe - Peclet Number, Re•Pr

Pr - Prandtl Number, $C_p\mu/k$

Re - Reynolds Number, $\rho Vd/\mu$

St - Stanton Number, Nu/Pe

APPENDIX B

PHYSICAL PROPERTIES AND TABULATED DATA

Below are listed the sources of the physical properties of water used in this research. The data for this research are presented in Table B-I, which starts on the following page. The dashed lines in the data indicate groupings of cooling water temperatures.

PROPERTY	SOURCE
μ, viscosity	(13)
ρ_1, liquid density	(14)
ρ_v, vapor density	(14)
H, enthalpy	(14)
C_p, specific heat	(15)
σ, surface tension	(16)
k, thermal conductivity	(17)

TABLE B-I

TABULATED DATA

TEMPERATURES (°F) Cooling Water	Steam	Pool	Wall	COOLING WATER VELOCITY (ft/sec)	HEAT FLUX (Btu/hr-ft^2)	HEAT TRANSFER COEFFICIENTS (Btu/hr-ft^2-°F) steam to agitated pool	agitated pool to tube wall
49.5	212.0	158.6	73.4	9.6	31952.	599.	375.
49.6	212.0	161.9	75.6	9.6	34700.	693.	402.
49.6	212.0	162.4	76.5	9.6	35881.	724.	418.
49.1	211.7	166.0	87.0	9.6	50438.	1103.	639.
49.6	212.1	173.4	94.3	9.6	59630.	1542.	754.
49.4	212.0	182.0	104.7	9.6	73811.	2463.	956.
49.1	212.0	187.9	114.4	9.6	86807.	3603.	1181.
49.6	211.7	195.0	117.6	9.6	90749.	5439.	1172.
49.8	212.2	200.7	126.3	9.6	102321.	8891.	1376.
48.7	212.4	164.3	82.8	9.5	44729.	931.	549.
48.9	212.7	170.3	93.5	9.5	58842.	1387.	767.
48.9	212.4	176.6	97.6	9.6	64476.	1804.	816.
48.8	212.4	183.9	106.9	9.7	77340.	2713.	1005.
49.4	212.5	190.1	114.3	9.6	86552.	3870.	1142.
49.4	212.4	203.2	128.0	9.6	104824.	11433.	1394.
67.7	210.8	164.7	96.6	10.2	45721.	991.	672.
67.9	210.8	164.5	96.5	10.3	45378.	981.	667.
68.2	210.8	163.4	95.8	10.2	43526.	918.	644.
68.1	210.8	162.8	94.2	10.2	41353.	862.	603.
68.8	211.4	199.1	131.0	10.1	98063.	7967.	1440.
69.5	211.4	197.5	126.1	10.2	90419.	6509.	1265.
68.0	211.3	196.1	132.1	10.7	104861.	6891.	1638.
67.8	211.3	192.2	123.1	10.7	90366.	4742.	1306.
67.8	210.8	192.1	125.0	10.7	93712.	5016.	1396.
67.6	210.8	190.3	121.2	10.7	87802.	4292.	1270.
69.9	211.2	174.1	106.5	10.3	58806.	1585.	870.
69.8	211.2	174.0	105.9	10.2	57510.	1546.	846.
69.9	211.3	173.5	105.6	10.2	56846.	1505.	837.
69.7	211.0	172.9	104.8	10.2	55935.	1467.	821.
70.9	210.9	176.5	110.2	11.0	67053.	1948.	1012.
70.9	210.9	177.8	109.2	10.8	64341.	1941.	938.
70.6	210.9	177.4	108.1	10.9	63146.	1886.	910.
70.5	210.9	178.3	108.9	10.9	64652.	1983.	931.
70.4	211.0	177.0	107.0	11.0	62175.	1824.	889.
70.8	210.9	177.8	107.7	11.0	62776.	1894.	896.
72.2	211.2	185.1	179.9	10.4	75406.	2888.	1123.

TABLE B-I (CONTINUED)

TEMPERATURES (°F)				COOLING WATER VELOCITY (ft/sec)	HEAT FLUX (Btu/hr-ft^2)	HEAT TRANSFER COEFFICIENTS (Btu/hr-ft^2-°F)	
Cooling Water	Steam	Pool	Wall			steam to agitated pool	agitated pool to tube wall
72.2	211.0	185.3	117.5	10.6	75630.	2933.	1116.
73.2	211.0	186.9	120.9	10.5	79648.	3303.	1205.
73.2	211.2	185.3	119.4	10.5	77149.	2986.	1169.
73.2	210.9	184.4	116.8	10.5	72841.	2746.	1077.
98.8	211.4	189.5	131.9	9.1	57011.	2600.	991.
98.8	211.4	186.3	130.4	9.4	55870.	2224.	1000.
98.8	211.4	181.9	126.1	9.3	47732.	1617.	856.
99.7	211.6	176.2	118.3	9.3	32884.	929.	568.
99.3	211.6	193.6	136.5	9.4	65892.	3648.	1155.
98.8	211.4	195.4	137.0	9.4	67677.	4240.	1159.
98.4	211.5	199.9	142.2	9.3	77422.	6674.	1342.
98.8	211.5	202.4	145.8	9.3	82235.	9037.	1453.
102.3	211.3	180.6	126.0	9.3	42458.	1385.	777.
101.8	211.3	182.7	128.8	9.3	48075.	1679.	892.
101.8	211.0	183.7	127.3	9.3	45355.	1660.	805.
102.3	211.0	192.2	135.2	9.3	58686.	3121.	1029.
102.3	211.3	195.4	137.8	9.3	63327.	3989.	1099.
101.8	211.0	198.5	143.6	9.2	74007.	5880.	1350.
101.4	211.0	201.7	144.8	9.2	76419.	8222.	1343.
102.3	211.2	205.3	147.7	9.2	80704.	13738.	1402.
102.3	210.9	209.6	150.4	9.3	85771.	66008.	1448.
107.2	210.9	179.8	128.4	8.4	35910.	1156.	698.
107.6	211.2	183.7	131.1	8.4	40072.	1460.	762.
106.3	211.2	185.7	132.4	8.7	45232.	1778.	848.
105.9	211.2	188.8	137.6	8.5	53837.	2405.	1051.
106.3	211.2	192.0	139.4	8.4	56107.	2923.	1068.
107.2	211.0	194.0	140.7	8.4	56788.	3325.	1067.
106.3	211.0	195.6	142.0	8.4	60119.	3880.	1122.
105.9	211.2	199.1	144.6	8.4	65369.	5423.	1198.
105.9	211.0	201.5	146.4	8.4	68108.	7126.	1236.
106.3	211.0	204.6	150.4	8.4	74335.	11606.	1371.
106.8	211.0	207.7	154.9	8.4	81308.	23991.	1540.
110.6	210.6	184.1	134.3	8.7	42091.	1588.	846.
109.5	210.5	186.0	135.9	8.5	45927.	1873.	918.
110.2	210.6	188.1	143.9	8.7	59319.	2635.	1340.
110.0	210.8	191.4	146.2	8.7	63667.	3296.	1406.
110.8	210.5	193.6	150.1	8.7	69491.	4101.	1601.
110.0	210.6	199.8	154.8	8.6	78490.	7226.	1745.
109.3	210.8	203.6	158.3	8.6	85519.	11920.	1889.
109.3	210.6	208.8	166.2	8.6	99273.	55065.	2328.
122.0	210.3	191.3	147.8	7.6	43416.	2285.	999.
122.0	210.4	187.9	146.1	7.5	40182.	1779.	962.

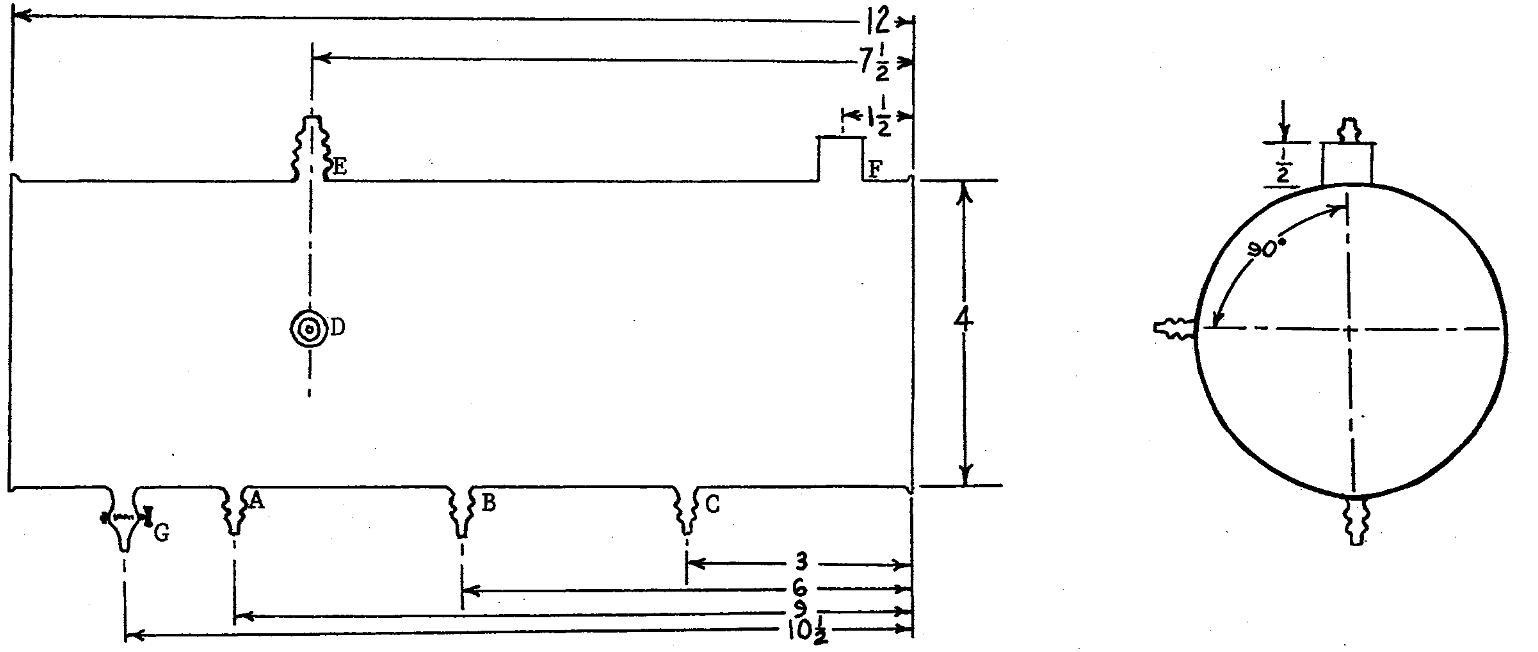
TABLE B-I (CONTINUED)

TEMPERATURES (°F)				COOLING WATER VELOCITY (ft/sec)	HEAT FLUX (Btu/hr-ft^2)	HEAT TRANSFER COEFFICIENTS (Btu/hr-ft^2-°F)	
Cooling Water	Steam	Pool	Wall			steam to agitated pool	agitated pool to tube wall
121.4	210.1	183.5	142.1	7.6	35015.	1317.	848.
122.4	210.6	193.8	150.9	7.6	48228.	2881.	1123.
122.4	210.4	195.7	152.3	7.6	50683.	3434.	1169.
123.7	210.6	199.4	154.4	7.6	52416.	4685.	1166.
121.6	210.6	201.1	155.6	7.9	59169.	6244.	1301.
119.7	210.6	204.6	155.9	7.9	62488.	10542.	1283.
122.2	210.7	209.6	162.9	7.8	70071.	65083.	1499.
133.7	211.7	192.1	151.6	7.9	32871.	1678.	811.
135.5	211.4	192.5	151.3	7.9	34336.	1813.	832.
132.5	211.7	194.4	153.7	7.9	38884.	2243.	957.
132.5	211.8	192.4	148.9	9.1	33293.	1713.	766.
132.9	211.4	198.1	161.0	8.9	56189.	4201.	1515.
132.5	211.8	199.9	163.8	9.0	63116.	5302.	1748.
131.8	211.4	202.8	165.2	9.0	66976.	7757.	1781.
131.8	211.3	204.1	165.3	8.8	65671.	9128.	1690.
131.6	211.6	207.0	167.3	8.9	70582.	15480.	1776.
130.2	211.6	209.0	169.1	8.8	75908.	29242.	1903.
136.8	211.2	188.9	149.2	7.6	22450.	1006.	566.
136.4	211.4	193.8	153.3	7.6	30500.	1739	752.
135.6	211.5	196.9	155.3	7.6	35465.	2425.	853.
137.0	211.4	201.1	169.1	7.6	58116.	5663.	1819.
137.6	211.2	210.3	181.7	7.6	79843.	81853.	2790.
134.7	210.9	202.7	168.7	8.0	63023.	7688.	1856.
136.2	211.4	206.2	173.4	7.7	67864.	13159.	2067.
135.8	210.9	208.8	176.4	7.9	74880.	36804.	2309.
146.3	211.2	197.0	162.1	8.4	32084.	2267.	918.
147.1	211.2	200.3	169.1	8.3	44232.	4077.	1418.
145.6	211.2	203.3	170.9	8.4	50949.	6512.	1572.
146.3	210.8	205.3	171.9	8.2	51030.	9320.	1528.
145.9	210.8	205.4	174.2	8.3	56576.	10580.	1811.
145.6	210.8	207.8	175.7	8.2	59710.	36804.	1863.
152.6	210.2	190.3	160.8	8.0	16445.	826.	558.
152.2	210.2	194.2	162.3	8.1	20505.	1288.	643.
151.4	210.0	197.0	170.3	8.1	38063.	2923.	1425.
152.4	210.3	199.4	174.3	7.9	43339.	3975.	1730.
151.6	210.2	204.0	177.4	7.9	50882.	8247.	1016.
151.6	210.2	207.4	179.9	7.9	55698.	20181.	2024.
151.0	210.0	209.4	183.1	7.9	63112.	93949.	2405.

APPENDIX C

DETAILS OF THE CONDENSER DESIGN

The following figures present the detailes of the condenser used in this research. Figure C-1 concentrates on the glass part of the condenser, while Figure C-2 presents information concerning the non-glass components of the condenser and the assembly of the total condenser.

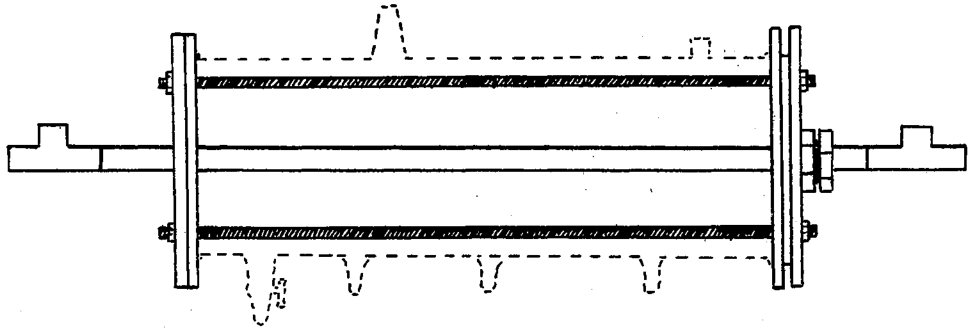


A, B, C, D - ¼" Nipple
E - ½" Nipple
F - ½" Pyrex Tubing, Reinforced End
G - Teflon Stopcock

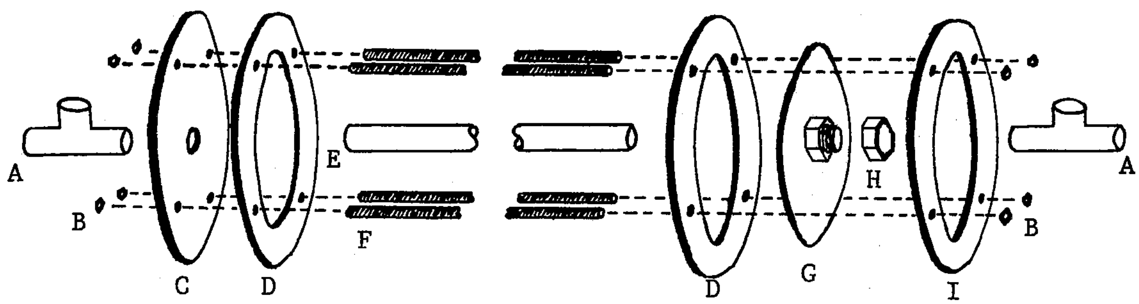
Note: Condenser - 4" Pyrex Tubing
Reinforced Ends

Scale: 1 cm = 1 in

Figure C-1. Scale Drawing of Glass Condenser



Parts in Assembled Condition



A - ½" copper tubing 'T'

B - 3/16" nut

C - 5½" diameter, 1/16" thick brass plate, 0.65" centered hole, 4 - ¼" diameter holes, equi-spaced, 3/8" from rim

D - 5½" outside diameter, 2½" inside diameter, 1/16" thick rubber gasket, hole specifications same as for C

E - 17" copper tube, 0.626" outside diameter, 0.547" inside diameter

F - 14" threaded steel rod, 3/16" diameter

G - 4" outside diameter brass plate, 0.65" centered hole, nipple end of ½" Swagelok fitting welded in place

H - ½" Swagelok fitting for nipple

I - steel plate, same dimensions as D

Figure C-2. Details of Non-glass Parts of Condenser

APPENDIX D

SAMPLE CALCULATION

The method of treatment of the data will be summarized by a sample calculation based on the first data point reported in Table B-I. The necessary data for the calculation is the following:

T_s, temperature of the steam	- 212.0°F
T_p, temperature of the pool	- 158.6°F
T_{cw}, cooling water temperature	- 49.5°F
T_{cwc}, cooling water temperature	- 9.71°F
A, area for heat transfer	- 0.166 ft^2
W, condensate rate	- 5.196 lb/hr
H_{steam}, steam enthalpy @ 212°F (14)	- 1150. Btu/lb
H_{pool}, pool enthalpy @ 158.6°F (14)	- 127. Btu/lb
k_{tw}, thermal conductivity of tube	- 225. Btu/lb
r_o, outside radius of tube	- 0.0261 ft
r_i, inside radius of the tube	- 0.0228 ft
r_r, mean radius of the tube	- 0.02445 ft
V_w, cooling water velocity	- 9.6 ft/sec
d_i, inside diameter of the tube	- 0.547 in

The heat flux is calculated using equation 9

$$Q/A = \frac{5.196(1150.-127.)}{.166} = 31952.\ \text{Btu/hr-ft}^2$$

The heat transfer coefficient from steam to agitated pool is calculated using equation 10

$$h_{stap} = \frac{31952.}{(212.-158.6)} = 599.\ \text{Btu/hr-ft}^2\text{-}^\circ\text{F}$$

The tube side heat transfer coefficient may be calculated using equation 15

$$h_{ts} = 200(1 + .0157(9.71))\frac{9.6^{.8}}{.547^{.2}} = 1593.\ \text{Btu/hr-ft}^2\text{-}^\circ\text{F}$$

Next, the heat transfer coefficient through the tube wall to the cooling water may be found using equation 14

$$\frac{1}{h_{twtcw}} = \frac{1}{\frac{(.0261-.0228)}{225.}\ \frac{.0261}{.02445} + \frac{1}{1593.}\ \frac{.0261}{.0228}} = .000748\ \text{hr-ft}^2\text{-}^\circ\text{F/Btu}$$

The wall temperature is then calculated using equation 13

$$T_w = (31952.)(.000748) + 49.5 = 73.4^\circ\text{F}$$

With the wall temperature calculated, the heat transfer coefficient from agitated pool to tube wall may be calculated from equation 11

$$h_{aptw} = \frac{31952.}{(158.6-73.4)} = 375.2\ \text{Btu/hr-ft}^2\text{-}^\circ\text{F}$$

VITA

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