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HEAT TRANSFER DURING CONDENSATION
INSIDE A HORIZONTAL TUBE

by

Harold F. Rosson

A THESIS
SUBMITTED TO THE FACULTY
IN PARTIAL FULFILLMENT OF THE
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ABSTRACT

Heat transfer data were taken for methanol and Freon-12 condensing inside a horizontal tube over considerable ranges of pressure, temperature driving force, liquid loading, and vapor velocity.

A single mechanism for heat transfer could not explain the behavior of this and other data over the entire range of the variables. For condensation in a horizontal tube, three primary regions of flow are postulated: (1) semi-stratified flow (annular condensation and run-down superimposed on stratified flow), (2) laminar annular flow, and (3) turbulent annular flow.

A semi-theoretical equation,

\[ \frac{N\nu}{Pr^{1/3}} = 0.388 \frac{M^{1/2}}{\left( \frac{\rho_{v}}{\rho_{f}} \right)^{1/4}} \left[ \frac{DG_{v}}{\mu} \right]^{1/4} \]

is developed and shown to be applicable to both semi-stratified and laminar annular flow. Other equations must be used for turbulent annular flow.
NOMENCLATURE

A  area for heat flow, \( \text{ft}^2 \)
B  defined following equation (17)
C  undetermined constant
\( C_p \)  specific heat, \( \text{BTU/}^\circ\text{F} \)
D  inside tube diameter, ft
f  friction factor, dimensionless
G  mass flow, \( \theta/\text{hr ft}^2 \)
\( g_c \)  gravitational conversion factor, \( \theta \text{ mass ft/}^\theta \text{ force hr}^2 \)
\( g_L \)  acceleration of gravity, \( \text{ft/hr}^2 \)
h  heat transfer coefficient, \( \text{BTU/hr ft}^2 \^\circ\text{F} \)
\( k \)  thermal conductivity, \( \text{BTU/hr ft}^\circ\text{F} \)
L  tube length, ft
M  defined by equations (31) and (31a), dimensionless
\( \text{Nu} \) Nusselt number, \( hD/k \), dimensionless
\( \text{Pr} \) Prandtl number, \( C_p\mu/k \), dimensionless
q  heat flow, \( \text{BTU/hr} \)
r  radius, ft
\( r_0 \) inside tube radius, ft
\( r_s \) radius to surface of laminar film, ft
\( \text{Re} \) Reynolds number, \( DG/\mu \), dimensionless
\( \text{Re}^* \) Reynolds number defined by equation (10)
\( t_s \) saturated vapor temperature, \(^\circ\text{F}\)
\( t_w \)  
wall temperature, \( ^\circ F \)

\( \Delta t \)  
temperature difference across condensate film, \( ^\circ F \)

\( \Delta T \)  
temperature difference between saturated vapor temperature and cooling water inlet temperature, \( ^\circ F \)

\( T_w \)  
shear stress at the wall, \( \# \) force/ft\(^2\)

\( u \)  
velocity, ft/hr

\( u_0 \)  
velocity at surface of laminar film, ft/hr

\( v \)  
volume, ft\(^3\)

\( x \)  
defined by equation (23), dimensionless

\( x_0 \)  
thickness of laminar film, ft

\( z \)  
tube length, ft

\( \Gamma \)  
mass flow, \( \# \)/hr

\( \Theta \)  
angle, radians

\( \lambda \)  
latent heat, BTU/\( \# \)

\( \mu \)  
viscosity, \( \# \) mass/ft hr

\( \rho \)  
density, \( \# \)/ft\(^3\)

Subscripts not otherwise defined:

none \quad on fluid properties refers to liquid

\( v \) \quad on fluid properties or \( G \) refers to vapor

\( \Theta \) \quad refers to \( \Theta \) direction

\( m \) \quad indicates mean value
INTRODUCTION

Nusselt in 1916 was the first to develop mathematical expressions for the process of heat transfer during condensation. Nusselt assumed that only latent heat was transferred through a laminar liquid film flowing under the influence of gravity and derived equations for condensation on vertical tubes, vertical plates, and the outside of horizontal tubes. He also developed an expression taking sub-cooling into account which was later shown to be in error by Bromley (1). Nusselt's equations are applicable in areas where conditions do not vary widely from his assumptions. They may not be applied when the effects of liquid sub-cooling or vapor shear become important or when the liquid film becomes turbulent.

Several modifications to Nusselt's original work have been made. Bromley, Brody, and Fishman (2) included terms to account for the temperature change of the liquid as it flows around a horizontal tube. Peck and Reddie (15) computed the effect of the acceleration of the liquid film on the condensing coefficient.

Bromley (1) and Rohsenow (17) introduced corrections to Nusselt's equations to account for the sub-cooling of the liquid. Seban (19) has discussed the effects of turbulent flow on condensation and Rohsenow, Webber, and Ling (16) have treated the effect of vapor velocity on laminar and turbulent film condensation.

Haseldon and Prosad (8) condensing nitrogen and oxygen on a vertical cylinder at low temperature differences, obtained data that agreed quite well with the Nusselt equations. Condensing coefficients for more viscous liquids measured at higher temperature differences may differ from the Nusselt values as much as a factor of two. Katz (11, 12) and Short and Brown (20) reported coefficients for vertical rows of horizontal tubes higher than the Nusselt values. Othmer and Berman
condensing alcohols, esters, and ketones at a constant vapor velocity, found that the effect of the temperature difference upon the heat transfer was not as predicted by the Nusselt relations.

Jakob, Erk, and Eck found the condensing coefficient of steam inside a vertical tube to vary directly with the velocity of the vapor. Tepe and Mueller (21), condensing benzene and methanol alcohol inside a horizontal and an inclined tube, reported coefficients 50% higher than the Nusselt values. Schmidt (18) condensed carbon dioxide inside a vertical tube at pressures up to the critical. The experimental values were about twice the Nusselt values and showed a marked dependence on vapor velocity. Carpenter and Colburn (3) studied the effect of vapor velocity on condensation inside a vertical tube. The coefficients varied directly with the average mass velocity of the vapor. Crosser (4), condensing propane and Deans (5) condensing Freon inside a horizontal tube found coefficients dependent on the average vapor velocity raised to a power ranging from 0.2 to 0.8.

The work of Crosser (4) and Deans (4) has adequately correlated condensing heat transfer data in those areas where a turbulent boundary layer is thought to exist. The purpose of this work was primarily to study condensing heat transfer inside a horizontal tube when this transfer takes place through a laminar boundary layer. Equipment was constructed to allow the effects of vapor velocity, liquid loading, and temperature driving force to be studied independently.
EXPERIMENTAL APPARATUS

The apparatus used for this investigation was originally constructed by Crosser (4) and was modified in order to better investigate the effects of liquid loading and to obtain more accurate temperature profiles. The major components were

(a) a vapor generator
(b) a pre-condenser
(c) a short differential condenser
(d) a vapor-liquid separator
(e) an exit vapor condenser
(f) condensate and condensed exit vapor rotameters.

The major components were supplemented by the necessary cooling water flow metering devices, temperature measuring instruments, thermocouple elements, and electrical power measuring instruments to determine heat loads.

All sections of the equipment containing vapor were operated adiabatically by use of adiabatic heaters. A schematic diagram of the equipment is shown in Figure 1 and details of all components and construction are given in Appendix F.

The vapor from the vapor generator rose vertically approximately ten feet where it entered a horizontal condensing section. In this condensing section it passed first through a five foot double pipe pre-condenser where the extent of liquid loading was controlled by partial condensation. The flow then passed through a one foot differential condenser where all condensing coefficients were measured.

The exit flow from the differential condenser was separated into vapor and condensate in the vapor-liquid separator. The liquid from the separator passed through a rotameter and then was returned to the vapor generator. The vapor from the separator
was condensed in the exit vapor condenser, passed through a rotameter, and then re- turned to the vapor generator. The liquid level in the separator was measured directly by a high pressure differential gauge and, during any given run, this level was held constant.

All power to the vapor generator was electrical and was measured using a wattmeter, voltmeter, and ammeters.

The cooling water to the pre-condenser could be operated as either a closed or an open system while the differential cooling water was always operated as a closed system. Cooling water rates to both of these condensers were measured by rotameters. Cooling water temperature rises were measured by differential thermocouple elements inserted in the entrance and exit water streams.

The differential condenser had a series of differential thermocouple elements located down its length for determination of temperature profiles. There were also two differential thermocouple elements located one inch from either end of the condenser to measure end effects. The hot junction for all these elements was located in the vapor leaving the vapor generator. Voltages generated by all differential thermocouples were measured by an electronic potentiometer.

Temperatures throughout the system were measured by an electronic multipoint recording potentiometer and included:

(a) vapor generator exit vapor temperature
(b) pre-condenser inlet cooling water temperature
(c) differential condenser inlet cooling water temperature
(d) condensate temperature at the condensate rotameter
(e) condensed vapor temperature at the condensed vapor rotameter
(f) temperature of the combined condensate and condensed vapor as it entered the vapor generator.

The pressure of the system was measured by a calibrated test gauge.
OPERATION OF EQUIPMENT

Charging Procedure:

Prior to charging the system with either methanol or Freon, it was first evacuated with a water aspirator. Liquid methanol could then be sucked into the system directly. Liquid Freon was introduced by heating an inverted Freon storage cylinder connected to the charge valve on the vapor generator. A heat lamp was used to supply the necessary heat. The system required a total charge of about three gallons of liquid.

Pressure Control:

For a given power input to the vapor generator, the system pressure was controlled by the operation of the exit vapor condenser. A coarse pressure control was accomplished by varying the height of condensed liquid maintained in the condenser and a fine control was accomplished by varying the flow of cooling water to the condenser. This cooling water was taken from a constant head tank whenever possible as steadier operation resulted. However, at the higher heat loads the constant head tank did not furnish sufficient water and cooling water had to be drawn directly from the line supply. For easier control, pressure was determined from the system temperature rather than from the pressure gauge.

Temperature Difference:

The temperature difference between the differential condenser tube wall and condensing vapor was controlled by controlling the inlet temperature of the differential condenser cooling water. This was done by use of the condenser water cooler and/or the condenser water heater incorporated in the system. The temperature rise of the cooling water passing through the differential condenser was usually held
somewhere between 1 °F and 5 °F by controlling the water flow rate through the condenser. It was desirable to have this water rate as high as possible without making the temperature rise too small to measure accurately.

**Flow Rate:**

The total flow rate through the condensing sections was controlled by the power input to the vapor generator. Constant power input at a given pressure resulted in fairly constant total flow over wide ranges of exit vapor flow.

**Liquid Loading:**

The extent of liquid loading in the differential condenser was controlled by the amount of condensation in the pre-condenser. This was controlled primarily by the temperature of the cooling water in the pre-condenser which in turn was controlled by the amount of waste water removed from the cooling system (See Figure F-2). For no liquid loading, no cooling water was allowed to pass through the pre-condenser.

**Vapor-Liquid Separator:**

The liquid level in the vapor separator was held constant during any given run by adjustment of the condensate return valve.

**Adiabatic Sections:**

The vapor riser, horizontal tubing, condensing sections, and vapor-liquid separator were maintained adiabatic by adjustment of the current through the adiabatic heaters to give a zero deflection of the galvanometer in the differential thermocouple circuits. Control was facilitated by use of calibration charts. (See Appendix D)
EXPERIMENTAL MEASUREMENTS

All runs were coded to facilitate data processing and with each run there is associated a series of four numbers separated by dashes. The first of the numbers represents the system pressure in psig for methanol and psia for Freon. The second represents in round numbers the temperature difference between the condensing vapor and the inlet differential condenser cooling water. The third number represents the per cent of maximum power supplied to the vapor generator (maximum power 8.7 kw). The last number represents the average per cent liquid (average liquid flow/total flow) present in the differential condenser.

For each run the following were recorded:

(a) power input to the vapor generator
(b) condensate flow rate
(c) condensed vapor flow rate
(d) pre-condenser cooling water rate
(e) differential cooling water rate
(f) temperatures of
   (1) vapor generator exit vapor
   (2) pre-condenser inlet cooling water
   (3) differential condenser inlet cooling water
   (4) condensate at its rotameter
   (5) condensed vapor at its rotameter
   (6) combined condensate and condensed vapor at the entrance to the vapor generator

(g) Pre-condenser cooling water temperature rise
(h) differential condenser cooling water temperature rise

(i) the temperature difference profile down the length of the differential condenser.

The general procedure of operation was to run at a fixed pressure, fixed temperature difference between condensing vapor and inlet differential condenser cooling water, and fixed power input, varying the amount of condensation in the pre-condenser. Then the power input was changed and the procedure repeated. Eventually $\Delta T$, and finally the system pressure were also changed following the same procedure.

From the data taken, heat balances could be made around the vapor generator, pre-condenser, and the differential condenser. Balances around the generator and differential condenser checked well, but due to the use of an improper size rotameter in the pre-condenser cooling water system, the balances around the pre-condenser did not always check well. Heat balances were not made for all runs.

For several of the Freon runs the condensate rate exceeded the limits of the condensate rotameter and in these cases the condensate rate was taken as the difference between the total flow corresponding to the given power input and the measured condensed vapor rate.

Experimental condensing coefficients were determined from the relation

$$h = \frac{q}{A \Delta t}.$$ 

$q$ was taken to be the product of the cooling water flow rate multiplied by the cooling water temperature rise. The temperature differences measured at the thermocouple
positions were plotted versus their corresponding position in feet down the condenser.
The temperature difference profile was then sketched in, extrapolating to zero at each
end, and the area under the curve determined with a planimeter. The product of this
area multiplied by the inside tube circumference was taken as the term $A \Delta t$.

Average flow rates through the condenser were taken as arithmetic averages
of the flows entering and leaving the condenser. $q/\lambda$ was taken to be the condensa-
sion rate in the differential condenser.

The accuracy of all experimentally measured quantities was as follows:

(a) power. $\pm 0.02$ kw

(b) condensate and condensed vapor flow rates. $\pm 4\%$

(c) differential condenser cooling water flow rate. $\pm 2\%$

(d) temperatures. $\pm 0.5^\circ F$

(e) temperature differences. $\pm 0.04^\circ F$.

All heat transfer coefficients as calculated from experimental measurements were
expected to be accurate to at least $\pm 7\%$. Large coefficients were more accurate
than small ones.

Heat transfer coefficients were calculated on the basis of the inside area of
the tube although temperature differences were measured on the outside of the tube.
At the highest heat flux encountered, this introduced an error of only 1%; hence
temperature differences were not corrected to the inside tube wall temperature.
EXPERIMENTAL RESULTS

Methanol was condensed at pressures of 27, 55, and 110 psia at temperature differences across the film of from 4°F to 80°F. Superficial vapor mass velocities ranged from 700 to 23,000 lbs/hr ft² and superficial liquid mass velocities ranged from 200 to 21,000 lbs/hr ft².

Commercial grade Freon-12 was condensed at pressures of 320 and 400 psia at temperature differences across the film of from 70°F to 35°F. Superficial vapor mass velocities ranged from 60,000 to 290,000 lbs/hr ft².

In all runs except those of methanol at 27 psia the coefficients were essentially linear with vapor velocity and independent of liquid velocity. The effect of a decrease in either pressure or Δt for all runs was to increase the condensing coefficient, other things being equal.

Although Deans (5) found "breaks" in the curves of h vs. $G_v$ for Freon, no such breaks were apparent from these data.

The behavior of methanol at 27 psia did not show the coefficient to be independent of liquid loading as was true for all other pressures. For constant total flow, an increase in the liquid flow at first caused a sharp drop in h when plotted vs. $G_v$ and then leveled off and became parallel to the "dry" curve as the liquid flow was further increased. The "drops" were less severe at higher temperature differences.
**Figure 5 - Plot of Experimental Data**

- **h**: BTU/hr ft$^2$°F
- **q**: $10^{-3}$ lbs/hr ft$^2$
- **Methanol 58 PSIA**
- **10° AT**
- **20° AT**
- **40° AT**
- **80° AT**
DISCUSSION

The first mathematical expression for the condensation process was derived by Nusselt assuming

(1) a film of condensate covered the cooling surface,
(2) only latent heat was transferred through the film,
(3) negligible surface curvature, and
(4) negligible vapor shear on the surface.

Nusselt's derived equation is

\[ h = 0.948 \left[ \frac{k^3 \rho (\rho - \rho_v) g_v \lambda}{\mu L \Delta t} \right]^{1/4}. \]  \hspace{2cm} (1)

Equation (1) was later rearranged by Kirkbride (13) yielding

\[ h_m \left[ \frac{g_v \mu^2}{\rho (\rho - \rho_v) k^3} \right]^{1/3} = 1.47 \left( \frac{4W}{\mu} \right)^{-1/3}. \hspace{2cm} (2) \]

Nusselt also presented an analysis for condensation on a horizontal tube:

\[ h_m = 0.728 \left[ \frac{k^3 \rho (\rho - \rho_v) g_v \lambda}{\mu \varnothing \Delta t} \right]^{1/4}. \hspace{2cm} (3) \]
Except for limited applications, these equations do not adequately fit experimental data. Deviations have been attributed to turbulence in the liquid film and/or the effect of vapor shear on the condensate surface.

Carpenter and Colburn (3) have studied the effect of vapor velocity on condensation inside a vertical tube. They assumed that all resistance to heat transfer was in a liquid laminar sublayer which they characterized by use of the universal velocity profile. For turbulent vapor flow, they derived the equation

\[ h = 0.065 \left( \frac{c_p \rho L f}{2 \mu \rho_v} \right)^{1/2} \left( \frac{\Delta T_{m}}{T} \right)^{1/2} \]  

which gave only a fair correlation for their data. The equation cannot be applied without modification to horizontal tube condensation.

Rohsenow, Webber, and Ling (16) have also presented a discussion of the effect of vapor velocity on film condensation. Their resulting equation is quite complex requiring a very laborious numerical solution and is directly applicable only to vertical surfaces.

Crosser (4) modified the results of Carpenter and Colburn to apply to condensation inside a horizontal tube and obtained the relation

\[ \frac{hD}{k} \rho_v^{-1/5} = B \left[ \frac{D \Delta T_{m}}{\mu} \left( \frac{\rho}{\rho_v} \right)^{1/2} \right]^n \]  

(5)
with \( n \) varying from 0.2 at low Reynolds numbers to a constant value of 0.8 at \( \text{Re} = 10^5 \).

Two phase flow in a horizontal tube has been studied by Gazley (7). Co-current flow of air and water was observed over a wide range of flow rates and four flow regions were observed: (1) stratified flow where the liquid runs along the bottom of the tube under low vapor velocity; (2) wave flow where the surface is disturbed by a periodic vertical motion without the formation of eddies; (3) slug flow where the surface of the liquid is picked up by the vapor and thrown down the tube; and (4) annular flow where the liquid forms a ring around the periphery of the tube with the vapor flowing through the center.

For horizontal two phase flow with condensation one would expect to find some type of annular flow superimposed on the flow pattern which would occur if condensation did not take place. For low vapor flow rates a relation similar to Nusselt's analysis for no vapor shear should control with the condensing coefficient being a function of \( \Delta t \). At vapor rates sufficient to cause turbulent annular flow, a relation similar to that of Carpenter and Colburn should control with the coefficient being a function of vapor flow rate and not of \( \Delta t \). For a purely laminar annulus of condensate, the extent of liquid loading would also be expected to have some effect.

Following Crosser's argument we will assume an annular ring of condensate surrounding a core of vapor with all resistance to heat flow contained in a laminar condensate sub-layer. Assuming a linear velocity profile through this layer, we may write

\[ \tau_w g_c = \mu \frac{\mu_o}{\mu_o} \] (6)
where \( U_o \) is the velocity at the outer boundary of the sub-layer and \( \kappa_o \) is the thickness of the sub-layer.

Further we assume that the vapor shear on the condensate surface is transmitted undiminished to the wall and may be expressed

\[
\tau_w q_c = \frac{f \frac{a}{2}}{2 \rho_v} \tag{7}
\]

where \( f \) is the two phase fluid flow friction factor.

Equating (6) and (7) and rearranging

\[
U_o = f \frac{\kappa_o \frac{a}{2}}{2 \rho_v \mu} \tag{8}
\]

If we now define \( U_o^* \) to be the hypothetical laminar liquid velocity at the center of the tube if the tube were running full of liquid with a velocity of \( U_o \) at \( \kappa_o \), then

\[
U_o^* = \frac{f D \frac{a}{2}}{4 \rho_v \mu} \tag{9}
\]

Define

\[
Re^* = \frac{D U_o^* \rho}{\mu} = f \left[ \frac{D G_v}{\mu} \left( \frac{\rho}{\rho_v} \right)^{1/2} \right]^2 \tag{10}
\]

23.
Assuming \( f \) to be independent of vapor velocity and expressing

\[
N_u = C \left( \frac{Re^*}{\mu} \right)^{m'}
\]

which would be analogous to single phase flow, then

\[
N_u = C \left[ \frac{DG_v}{\mu} \left( \frac{\rho}{\rho_v} \right)^{1/2} \right]^n. \quad (11)
\]

Equation (11) does not adequately correlate the data obtained during this investigation. Analysis of the data indicates that \( C \) is not constant but is a function of \( \Delta t \).

If we assume a laminar condensate film on the inside of the tube wall we may determine the functional relation between the point film thickness and the point Nusselt number.

\[
dg = -k \frac{d \phi}{dz} r \frac{dt}{dr} \quad (12)
\]

Rearranging and solving the differential equation,

\[
\frac{r_n}{r_o} = - \frac{k}{\frac{dg}{dz}} \frac{d \phi}{dz} \left( t_s - t_w \right). \quad (13)
\]

\[
dg = h r \phi dz \left( t_s - t_w \right). \quad (14)
\]
Substituting (14) into (13) and rearranging gives

$$\ln \frac{r}{r_0} = -\frac{2}{N\nu}$$

(15)

or

$$r_r = r_0 e^{-\frac{2}{N\nu}}.$$  \hspace{1cm} (16)

If we look at a differential volume element of the condensate film:

Assume

1. no radial liquid velocity
2. no vapor shear
3. no acceleration terms.

$$dA_2 = r \, d\phi \, dz$$

$$dV = \frac{1}{2} \left( r^2 - r_r^2 \right) d\phi \, dz.$$  

Writing a force balance in the $\Theta$ direction,

$$\frac{1}{2} \frac{q}{\delta c} (\rho - \rho_v) (r^2 - r_r^2) \sin \Theta \, d\phi \, dz$$

$$= \mu \left( \frac{\partial u_\Theta}{\partial r} \right) r \, d\phi \, dz.$$  \hspace{1cm} (17)
Letting \( B = \frac{g_e}{2g_c} (\rho - \rho_v) \) and solving (17)

\[
U_0 = \frac{B}{2} \sin \theta \left( r^2 - r_e^2 \right) + B \sin \theta \frac{r_e^2}{m} r + C.
\]

\( C \) may be evaluated since \( U_0 = 0 \) at \( r = r_o \).

\[
C = B \sin \theta \frac{r_e^2}{m} r_o - B \frac{r_o^2}{2} \sin \theta \frac{r_o^2}{r}.
\]

and

\[
U_0 = \frac{B}{2} \sin \theta \left( r^2 - r_o^2 \right) + B \sin \theta \frac{r_e^2}{m} \frac{r_e}{r}.
\] \( \tag{18} \)

To determine the mean velocity in the \( \Theta \) direction,

\[
(U_0)_m = \frac{1}{r_o - r_e} \int_{r_e}^{r_o} U_0 \, dr.
\] \( \tag{19} \)

Solving (19)

\[
(U_0)_m = \frac{B \sin \theta}{r_o - r_e} \left[ \frac{g r_o^2 r_e^2}{6} - \frac{2 r_o^3 - r_e^3}{6} + r_o r_e^2 - r_e^3 \left( 1 + \ln \frac{r_o}{r_e} \right) \right].
\] \( \tag{20} \)

We can now make a heat balance on the differential element.
where \( \Gamma \) is the mass flow into the volume element, \( d \Gamma \) is the mass of vapor condensed, and \( dq \) is the heat removed from the volume element. For unit length of tube,

\[
\Gamma' = \rho' (r_o - r_e) \left( \omega \right) \frac{d\theta}{r_e}.
\]  

(21)

Substituting (20) into (21) and differentiating,

\[
d\Gamma = \frac{B \rho}{6} \left\{ \left[ 3 r_o^2 r_e - 2 r_o^3 + 6 r_o r_e^2 - r_e^2 \left( 7 + 6 \ln \frac{r_o}{r_e} \right) \right] \cos \theta \; d\theta \\
+ \left[ 3 r_o^2 + 12 r_o r_e - 3 r_e^2 \left( 5 + 6 \ln \frac{r_o}{r_e} \right) \right] \sin \theta \; dr_e \right\}
\]

(22)

Substituting (15) and (16) into (22) and letting

\[
\chi \equiv \frac{2}{\nu_w},
\]

(23)

then

\[
d\Gamma = \frac{B \rho \epsilon r_o^3}{6} \left\{ \left[ 3 \; e^{-\chi} - 2 + 6 \; e^{-2\chi} - e^{-3\chi} \left( 7 + 6\chi \right) \right] \cos \theta \; d\theta \\
- \left[ 3 + 12 e^{-\chi} - 3 \; e^{-2\chi} \left( 5 + 6\chi \right) \right] \; \sin \theta \; d\chi \right\}
\]

(24)

Assuming that only latent heat is transferred,

\[
dq = \lambda \frac{d\Gamma}{dr} = h \; r_o \; d\theta \; \Delta t.
\]

(25)
Substituting (24) into (25) and letting

\[ M = \frac{\sqrt{\frac{B}{\tau_0^3}} P \lambda}{6 \lambda \Delta t} \]  

(26)

and rearranging, then

\[ \frac{d\chi}{d\Theta} = \frac{3 e^{-\chi} - 2 + 6 e^{-2\chi} - e^{-3\chi}(7 + 6\chi)}{3 e^{-\chi} + 12 e^{-2\chi} - 3 e^{-3\chi}(5 + 6\chi)} \cot \Theta \]

\[ - \frac{1}{M \left[ 3 e^{-\chi} + 12 e^{-2\chi} - 3 e^{-3\chi}(5 + 6\chi) \right]} \chi \sin \Theta . \]  

(27)

To approximate the solution of (27) we may neglect \( \Delta \chi \) with respect to 5 and 7 (for \( Nu = 100, x = 0.02 \)) and expand \( e^{\chi} \approx 1 + \chi \). After collecting terms, equation (27) becomes

\[ \frac{d\chi}{d\Theta} + \frac{I}{18 M \sin \Theta} \left( \frac{1}{\chi} \right)^2 = \frac{\cos \Theta}{3 \sin \Theta} . \]  

(27a)

This is a non-linear differential equation which must be solved numerically. However if we express the solution of (27a) in the form

\[ \chi = 2 M^m \frac{f(\theta)}{f(\theta)} \]  

(28)

then

\[ Nu = \frac{M^m}{f(\theta)} . \]  

(28a)
Integrating around the tube and assuming no condensate collection in the bottom of the tube,

\[
(Nu)_{m} = \frac{M}{2\pi} \int_{0}^{2\pi} \frac{d\theta}{f(\theta)}.
\]  \hspace{1cm} (29)

Hence

\[
(Nu)_{m} = C \cdot M^{m}.
\]  \hspace{1cm} (30)

Expanding \( M \),

\[
M = \frac{g_{L} (\rho - \rho_{v}) \rho \lambda r_{o}^{3}}{12 \lambda \mu \Delta t}.
\]  \hspace{1cm} (31)

Jacob (10) shows that for condensation on the outside of a horizontal tube, the exponent for \( M \) is 1/4.

Equation (30) is applicable only in the case of no vapor shear. When shear is present we may expect \( C \) and possibly \( m \) to be functions of shear. Combining equations (11) and (30) and including the Prandtl number to the one-third power,

\[
Nu \cdot P_{r}^{-\frac{1}{3}} = C' \cdot M^{m} \left[ \frac{\Delta S_{v}}{\mu} \left( \frac{\rho}{\rho_{v}} \right)^{\frac{1}{2}} \right]^{\frac{m}{3}}.
\]  \hspace{1cm} (32)

The one-third exponent of the Prandtl number is suggested by theoretical (6) and
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experimental attacks in order to relate the temperature and velocity profiles.

Using values of \( m = 1/6 \) and \( n = 1/4 \), equation (32) correlated the data within \( \pm 20\% \). The final equation was taken to be

\[
N_u \; Pr^{-1/3} = 0.388 \; M^{1/6} \left[ \frac{Dk}{\mu} \left( \frac{\rho}{\rho_v} \right)^{1/2} \right]^{1/4}
\]  

where \( M \) is now defined

\[
M \equiv \frac{g \lambda \left( \rho - \rho_v \right) \rho \lambda r_o^3}{k \mu \Delta t}.
\]  

(31a)

All physical properties of the liquid were arbitrarily evaluated at the tube wall temperature while vapor properties were evaluated at the saturation temperature. The Nusselt and Prandtl groups were built with liquid properties.
DISCUSSION OF CORRELATION

With the exception of five points, equation (33) correlates all data taken during this investigation to within ± 20%. The correlation is shown plotted in Figure 7. It may be noted that for values of \( \sqrt{\frac{\rho \nu}{\mu}} \left( \frac{\rho}{\rho_v} \right)^{1/2} \) greater than 6 x 10^4, there appears to be a tendency for an increasing slope. This tendency is shown more clearly by the data of Crosser (4) and Deans (5) and suggests some kind of flow transition in this region.

The data of Crosser and Deans at moderate pressures, shown plotted in Figure 8, fit the correlation quite well. These data are the average values of condensing coefficients over a five foot length of 3/4" stainless steel condenser tube. At pressures nearing the critical, their data do not fit equation (33), but can be correlated by omission of the term \( M^{1/6} \), indicating that for these pressures temperature driving force is not important.

The majority of the data taken during this investigation show no effect of liquid loading, but the data of methanol at 27 psia exhibited a marked lowering of the heat transfer coefficient by the presence of liquid. This tendency was also shown to a smaller extent by Freon at 320 psia. For these data, \( \Delta t \) is still a primary variable, and the effect of liquid loading is also some function of \( \Delta t \), being less important at high values of \( \Delta t \).

If we postulate "semi-stratified" flow, i.e., annular condensation and rundown superimposed on stratified flow, at low vapor velocities, followed by laminar annular flow and turbulent annular flow at increasing vapor velocities, then the behavior of the data can be explained.
In all cases, the heat transfer coefficient will be a function of the thickness of the laminar liquid boundary layer. In regions of semi-stratified flow, this thickness will be a function of $M \, (\Delta t)$, and $G_v$, and only slightly of liquid loading as excess liquid will tend to be in the stratified layer. In regions of laminar annular flow, $M$, $G_v$, and liquid loading will all be important factors. Liquid loading is important here as now excess liquid will be in the annulus rather than in the bottom of the tube.

For regions of turbulent annular flow, the thickness of the boundary layer will be independent of $\Delta t$ as all condensation will take place at the turbulent liquid surface. Here liquid loading can aid rather than deter the transfer of heat by contributing to the turbulence. This has been shown by Akers, Deans, and Crosser (22) and the data in this region can be correlated by using an equivalent liquid mass velocity in the relation

$$Nu \, Pr^{-\frac{1}{3}} = A \left( \frac{DG_e}{\mu} \right)^{\alpha}$$

where

$$G_e = G_{l} + \left( \frac{D}{P_e} \right)^{\frac{1}{2}} G_v.$$
CONCLUSIONS

1. Previous analyses of condensing heat transfer inside horizontal tubes do not completely express the effect of all variables, particularly in the regions of moderate vapor velocities and pressures away from the critical. In the regions of semi-stratified and laminar annular flow, the effects of vapor velocity, temperature driving force, and, to some extent, liquid loading are important.

2. The semi-theoretical equation

\[ N_u \rho_s^{3/4} = 0.35 \frac{\rho}{\mu} \left[ \frac{\eta}{\gamma} \left( \frac{\rho}{\rho_s} \right)^{3/4} \right]^{1/4} \]

adequately correlates the data for regions of semi-stratified flow. Although it does not contain a liquid loading factor, it also correlates the data for laminar annular flow to within ±20%. For turbulent annular flow, a relation similar to that of Crosser (4) must be used.

3. No criterion has been established to determine the flow pattern from the system variables. However, at pressures well away from the critical, and at moderate vapor velocities, the flow can be expected to be semi-stratified. At values of \((\rho \gamma/\mu) (\rho/\rho_s)^{3/4} > 6 \times 10^4\), the flow seems to be turbulent annular.
APPENDIX A

BIBLIOGRAPHY
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1. Bromley, Inds. & Eng'g. Chem., 44, 2966, (1952)

2. Bromley, Brodkey, and Fishman, Inds. & Eng'g. Chem., 44, 2962, (1952)


APPENDIX B

SUMMARY OF DATA
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<td>Gv x 10^-4</td>
<td>Gl x 10^-4</td>
<td>q</td>
<td>Lat °F</td>
<td>h</td>
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<td>196</td>
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<td>1273</td>
<td>31.8</td>
<td>234</td>
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<td>137</td>
<td>508</td>
<td>22.73</td>
<td>4.93</td>
<td>1425</td>
<td>31.8</td>
<td>262</td>
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<tr>
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<td>590</td>
<td>26.29</td>
<td>0.94</td>
<td>1507</td>
<td>29.8</td>
<td>296</td>
</tr>
</tbody>
</table>
APPENDIX C

SAMPLE CALCULATIONS
SAMPLE CALCULATIONS

Run Number 40-40-63-34

1. Condensate rotameter reading: 10.8 (gallons)

2. Temperature in rotameter: 115 °F

3. Condensate flow: 13.6 gph

4. Exit Vapor rotameter reading: 14.3 (gallons)

5. Temperature in rotameter: 80 °F

6. Exit vapor flow: 18.5 gph

7. Pre-condenser water rotameter reading: 17.0

8. Pre-condenser water flow: 380 gph

9. Differential condenser water rotameter reading: 62.0

10. Differential condenser water flow: 547 gph

11. Total power input to generator: 5.5 kw or 18,800 BTU/hr

12. Total alcohol flow: 13.6 + 18.5 = 32.1 gph

13. Temperature of alcohol return: 95 °F

14. Heat content of return: 22 BTU/g

15. Temperature of exit vapor: 218.2 °F

16. Heat content of exit vapor: 539.2 BTU/g

17. Heat balance around generator:

<table>
<thead>
<tr>
<th></th>
<th>IN (BTU/hr)</th>
<th>OUT (BTU/hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power</td>
<td>18,800</td>
<td>17,308</td>
</tr>
<tr>
<td>Fluid</td>
<td>706</td>
<td>2,198</td>
</tr>
<tr>
<td>Losses (by diff):</td>
<td></td>
<td>19,506</td>
</tr>
<tr>
<td>Total:</td>
<td>19,506</td>
<td>19,506</td>
</tr>
</tbody>
</table>

18. Inlet temperature of pre-condenser water: 198 °F

19. Temperature rise of pre-condenser water: 0.195 mv or 7.5 °F

C-2.
20. Heat removed in pre-condenser: \(7.5 \times 380 = 2850\) BTU/hr

21. Inlet temperature of differential condenser water: \(178.5^\circ\)F

22. Heat removed in differential condenser: \(4.3 \times 547 = 2350\) BTU/hr

23. Temperature rise of differential water: \(0.109\) mv or \(4.3^\circ\)F

24. Latent heat: \(430\) BTU/\#

25. Condensers heat balance:

\[
\begin{array}{cc}
\text{Fluid:} & \text{IN (BTU/hr)} & \text{OUT (BTU/hr)} \\
\text{Pre-condenser Water:} & 13,800 & 2,850 \\
\text{Diff'l cond. water:} & 13,800 & 2,350 \\
\text{Total:} & 13,800 & 13,155 \\
\end{array}
\]

26. Alcohol condensed in diff’l cond. \(\frac{q}{\lambda}\): \(2350/430 = 5.5 \frac{\#}{hr}\)

27. Average vapor flow: \(18.5 + 5.5/2 = 21.2 \frac{\#}{hr}\)

28. Average condensate flow: \(13.6 - 5.5/2 = 10.9 \frac{\#}{hr}\)

29. Tube cross-sectional area: \(0.002331\) ft\(^2\)

30. Average vapor mass flow: \(21.2/0.002331 = 9.09 \times 10^3 \frac{\#}{hr} ft^2\)

31. Average condensate mass flow: \(10.9/0.002331 = 4.68 \times 10^3 \frac{\#}{hr} ft^2\)

32. Temperature difference profile:

<table>
<thead>
<tr>
<th>Switch Position</th>
<th>L (ft)</th>
<th>(\Delta t) (mv)</th>
<th>(\Delta t) (^\circ)F)</th>
</tr>
</thead>
<tbody>
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<td>10</td>
<td>-.0833</td>
<td>.013</td>
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<td>.440</td>
<td>17.0</td>
</tr>
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<td>4</td>
<td>.146</td>
<td>.640</td>
<td>24.7</td>
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<td>5</td>
<td>.271</td>
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<td>6</td>
<td>.469</td>
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</tr>
<tr>
<td>11</td>
<td>1.0833</td>
<td>.088</td>
<td>3.2</td>
</tr>
</tbody>
</table>

33. Area of \(\Delta t\) vs. L plot: \(26.5\) ft \(^\circ\)F

34. Inside circumference of tube: \(0.1711\) ft

35. Heat transfer coefficient: \(2350/0.1711 \times 26.5 = 518\) BTU/hr ft\(^2\) \(^\circ\)F
APPENDIX D

EQUIPMENT CALIBRATION

D-1.
ROTAMETER CALIBRATION CURVE

Tube: 3F 1/4 20-4
Float: 316 Stn. Stl.

Flow Rate #/hr Methanol

FIGURE D-3
ROTAMETER CALIBRATION CURVE

Tube: B3-24-127/21
Float: BSX 35A

Flow Rate #/hr Water

FIGURE D-5
ROTAMETER CALIBRATION CURVE

Tube: B4-27-5
Float: DSVT-44

Flow Rate #/hr Water

FIGURE D-6
ROTAMETER DENSITY CORRECTION FACTORS FOR FREON-12

FIGURE D-7
ROTAMETER CALIBRATION CURVE

Pre-Condenser Cooling Water

Flow Rate #/hr Water

FIGURE D-8
ROTAMETER CALIBRATION CURVE

Diff'l. Condenser Cooling Water

Flow Rate#/hr Water

FIGURE D-9
Figure D-10

Curves from left to right:
Separator
Entrance Tubing
Condensing Sections
Vapor Riser

Current Setting Amps
APPENDIX E

PROPERTY VALUES
FREON-12
LATENT HEAT OF VAPORIZATION

From "Thermodynamic Properties of F-12," ASRE Circular No. 12, 1931, and Ref. JLR. 69-13 No. 7, S4 19243, Kinetic Chemicals, Inc.

Latent Heat BTU/#

Temperature °F

FIGURE E-2
HEAT CAPACITY OF LIQUID FREON-12

From "Thermodynamic Properties of F-12," ASHE Circular No. 12, 1951, and Ref. JLR 69-13 No. 7 SN 19243, Kinetic Chemicals, Inc.

FIGURE E-4
FIGURE E-6

VISCOSITY OF FREON-12
THERMAL CONDUCTIVITY OF LIQUID METHANOL

Data from Riedel (1951)

FIGURE E-7
THERMAL CONDUCTIVITY OF LIQUID FREON-12

From "Thermal Conductances and Heat Transmission Coefficients of 'Freon' Refrigerants," 1942, Kinetic Chemicals, Inc.

Temperature °F

FIGURE E-8
DENSITY OF SATURATED ETHANOL VAPOR

Calculated using compressibility factors

FIGURE 5-10
DENSITY OF SATURATED LIQUID
AND VAPOR FREON-12

From "Thermodynamic Properties of
F-12," ASRE Circular No. 12, 1931,
and Ref. JLR 69-13 No. 7 SN 19243,
Kinetic Chemicals, Inc.

FIGURE E-11
APPENDIX F

DETAILS OF EXPERIMENTAL APPARATUS
APPENDIX F - DETAILS OF EXPERIMENTAL APPARATUS

**Vapor Generator:** The vapor generator was constructed from a 4 foot length of Schedule 80 4 inch pipe mounted vertically. A 6 kw Chromalux immersion heater was inserted through the flanged bottom. Copper plates were silver soldered to the heater elements in order to reduce the watt density and thereby lower the metal surface temperature. Two auxiliary heaters, made of 18 gauge nichrome wire, insulated with ceramic beads, were wound around the generator and supplied an additional 2 kw of heat. The generator was equipped with a gauge glass for detection of the liquid level. The generator was insulated with 2 inches of magnesia. As a safety precaution relay switches operating from a temperature controller were installed in all heater circuits. These switches opened if the system temperature exceeded a preset value. The wiring diagram for the generator heaters is shown in Figure F-1.

**Pre-condenser:** A common section of 5/8" Anaconda Type K copper water tube approximately 9 1/2 feet long joined the top of the vapor riser and the vapor-liquid separator and all condensation took place inside this tube. The pre-condenser was constructed by passing the copper tube through a 4.7 foot length of 1" galvanized pipe with packing glands at each end. The pre-condenser cooling water flowed counter-currently through the annulus formed between the tube and pipe; thermowells were installed in water inlet and outlet to measure temperature rise. Cooling water was circulated through the system by a small centrifugal pump during operation as a closed system and by line water pressure when operating as an open system. Waste water could be removed from the system at a controlled rate by discharge through a rotameter. The piping diagram for the pre-condenser is shown in Figure F-2.
FIG. F-1  VAPOR GENERATOR HEATERS WIRING DIAGRAM
Differential Condenser: The differential condenser was located approximately 1 1/2 feet downstream from the pre-condenser and was constructed as follows. A 15 inch length of 2" cylindrical brass bar stock was drilled laterally to allow the 5/8" copper tube to pass through with 1/64" clearance. The bar stock was then counter-bored 3 inches with a 1" drill and this 3 inch section was later cut off making the condenser 12 inches long. (The counter-bored section was necessary for chucking in the lathe.) A 3/16" radius round-bottomed thread 3/16" deep on a pitch of 2 threads per inch was cut in the outside of the brass sleeve. Holes were drilled and tapped for 8-32 brass screws in the shoulders between threads, perpendicular to the axis of the sleeve, at seven points down the length of the sleeve for insertion of thermocouples. The sleeve was then sweated to the copper tube.

A 3/8" x 1/2" roller was made with a 3/16" radius groove 3/16" deep cut around the roller. The sleeve was chucked in the lathe and the roller mounted in the tool post with the proper helix angle to line up with the sleeve thread. Then with the lathe split nut engaged and utilizing the roller as one half of a tube bender, 3/8" copper tubing was wound in the thread by operating the lathe very slowly. Finally the 3.8" copper tubing was sweated firmly in the thread. Thermocouples were constructed by drilling 8-32 brass screws laterally to allow insertion of the thermocouple lead wires. Copper-constantan thermocouple wires were passed through the screw, then through a small teflon washer, and the ends soldered to a small copper washer. These screws were then screwed into the holes tapped in the brass sleeve which firmly seated the thermocouple junctions against the outside of the central copper tube. Two additional thermocouples were soldered to the copper tube 1 inch from
FIG. F-2  PIPING DIAGRAM FOR PRE-CONDENSER COOLING WATER

FIG. F-3  DIFFERENTIAL CONDENSER ASSEMBLY
each end of the sleeve to measure end temperature effects.

A photograph of the completed differential condenser assembly is shown in Figure F-3. Thermocouple construction is shown in Figure F-4, and the thermocouple switching diagram is shown in Figure F-6. Physical locations of temperature profile thermocouples are shown in Figure F-5.

Cooling water for the differential condenser was circulated through the coil of 3/8" copper tubing. The large mass of brass smoothed the temperature fluctuations between loops of the cooling coil and effected a very nearly constant-temperature condenser wall. The cooling water was circulated in a closed circuit and flow rate measured with a rotameter. Temperature rise was measured from thermowells made of 1/8" copper tubing inserted about 3 inches into the 3/8" tubing. Physical contact between the thermowells and the 3/8" copper tubing was prevented by use of small teflon "spiders" around the thermowells. The piping diagram for the differential condenser cooling water is shown in Figure F-7.

**Vapor-Liquid Separator**: The condensate and exit vapor passed directly into a vapor separator constructed from a 2 foot length of 4" pipe mounted vertically. The flow entered at the center of the length and the vapor passed upward through 6 inches of copper shavings. A Barton high pressure differential gauge mounted on the floor below the separator was used to determine liquid level.

**Exit Vapor Condenser**: The exit vapor condenser was a 12 foot double pipe exchanger mounted vertically. The cooling water passed through the annulus and could be taken either from a constant head water tank or directly from the line water supply. The piping diagram is shown in Figure F-8.
FIG. F-4 EXPLODED VIEW OF TEMPERATURE PROFILE THERMOCOUPLE

FIG. F-5 LOCATION OF DIFFERENTIAL CONDENSER TEMPERATURE PROFILE THERMOCOUPLES
FIG. F-6 DIFFERENTIAL THERMOCOUPLE SELECTOR SWITCH

FIG. F-7 PIPING DIAGRAM FOR DIFFERENTIAL CONDENSER COOLING WATER
Condensate and Condensed Exit Vapor Rotameters: Fischer and Porter Tri-flat Tube Flowrators 2F 1/4-20-5 with interchangeable glass and 316 stainless steel floats were used for all alcohol runs. Use of both glass and stainless steel floats resulted in accurate metering of all flow ranges encountered. These meters were not calibrated experimentally but calibration curves were calculated following the instructions in the Fischer and Porter Catalog 10-A-91 (August 1954).

The Fischer and Porter high pressure rotameters used by Crosser (4) and Deans (5) were used for all Freon-12 runs. These rotameters had been previously calibrated and were not re-checked for this investigation.

Calibration curves for all rotameters are contained in Appendix D.

Thermocouples: All differential thermocouple elements were made from Leeds and Northrup 30 gauge copper-constantan duplex. All temperature measuring couples used in conjunction with the Brown multipoint recorder were Leeds and Northrup 30 gauge iron-constantan duplex.

Temperature Measuring Instruments: Emf's generated by all differential thermocouple elements were read by use of a Brown Electronik Potentiometer, Model No. Y156x15(VH) → X- (V), which read millivolts in 0.002 millivolt subdivisions. Direct temperature measurements were recorded by a Brown Multipoint Recorder, Model No. 153x62P12-X-16, which read directly in degrees Fahrenheit from 100°F to 300°F in 1°F subdivisions.

Electrical Power Measuring Instruments: Watt meters and ammeters necessary for power measurements are shown in Figure F-1. Voltages were obtained by use of a Simpson Meter.
**FIG. F-8** PIPING DIAGRAM FOR EXIT VAPOR CONDENSER COOLING WATER

**FIG. F-9** SCHEMATIC REPRESENTATION OF ADIABATIC SECTION

1 - SECTION TO BE MAINTAINED ADIABATIC
2 - DIFFERENTIAL THERMOCOUPLE ELEMENT
3 - MAGNESIA INSULATION
4 - NICHROME HEATING WIRE
5 - AMMETER
6 - GALVANOMETER
Adiabatic Sections: All sections of the equipment containing vapor were maintained adiabatic. This was accomplished by placing a differential thermocouple element in contact with the section to be controlled. This thermocouple was then covered by a layer of magnesia, another differential thermocouple element, and another layer of magnesia. Nichrome wire was then wrapped around the unit with coils about 1 inch apart, and a final layer of insulation was added. Most layers of insulation were 1 inch thick, although in some cases it was necessary to use 1 1/2 inches to obtain proper physical fit.

A galvanometer was placed between the two thermocouple elements and current passed through the nichrome wire to obtain O emf between couples. A schematic drawing of a typical adiabatic section is shown in Figure F-9.

Four separate sections were so maintained: the vapor riser, horizontal tubing, condensing sections, and the vapor-liquid separator. The resistance of the nichrome wire was found to be relatively constant over the temperature range so it was possible to calibrate the sections for O emf and plot \( (T_{\text{unit}} - T_{\text{room}}) \) vs. heater current. Calibration curves are in Appendix D.

Piping: With the exception of the horizontal copper tube, all piping was Schedule 80 grade B carbon steel seamless pipe, and the fittings were all standard screw 3000 psi forged steel. For the methanol runs, piping to the condensate and condensed exit vapor rotameters was standard galvanized pipe.