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CONDENSATION OF A VAPOR  
IN THE PRESENCE OF A NON-CONDENSING GAS  
IN TURBULENT FLOW

by

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## TABLE OF CONTENTS

	Page
Abstract	1
Nomenclature	2
Introduction	5
Experimental Apparatus and Procedures	9
Results	15
Discussion	17
Conclusions	25
Bibliography	26
Appendix A - Experimental Data	28
Appendix B - Sample Calculations	33
Appendix C - Property Values	38
Appendix D - Details of Experimental Apparatus	40

## LIST OF FIGURES

		Page
Figure 1	Experimental Apparatus	10
Figure 2	Effect of Non-condensing Gas on Condensing Heat Transfer	16
Figure 3	Correlation of Data	20

## ABSTRACT

Rates of condensation of water vapor in the presence of air were determined in turbulent flow in a vertical tube. Wide ranges of vapor-gas composition and velocity were covered in the study.

A satisfactory correlation of mass transfer coefficients in terms of the Schmidt and Reynolds groups was made in the form predicted by analogy to turbulent forced convection heat transfer.

## NOMENCLATURE

A	area for heat transfer	(ft <sup>2</sup> )
c <sub>A</sub>	vapor concentration	(lb-mol/ft <sup>3</sup> )
c <sub>p</sub>	specific heat	(BTU/lb-°F)
D	tube diameter	(ft)
$\mathcal{D}$	diffusivity	(ft <sup>2</sup> /hr)
D/Dt	substantial time derivative	$(\frac{\partial}{\partial t} + v_x \frac{\partial}{\partial x} + v_y \frac{\partial}{\partial y} + v_z \frac{\partial}{\partial z})$
f	friction factor for fluid flow,	$\frac{\Delta P e g}{G^2} \frac{D}{2L}$
Fr	Froude number,	$\frac{v^2}{Dg_c}$
g	local acceleration due to gravity	(ft/sec <sup>2</sup> )
g <sub>c</sub>	gravitational conversion constant	32.174 (lb <sub>m</sub> /lb <sub>f</sub> )(ft/sec <sup>2</sup> )
G	mass velocity	(lb/hr-ft <sup>2</sup> )
Gr	Grashof number,	$\frac{L^3 \beta \Delta \theta g e^2}{\mu^2}$
Gr <sub>M</sub>	Grashof number for mass transfer,	$\frac{L^3 (\frac{\Delta \rho}{\rho}) e^2 g}{\mu^2}$
h	film heat transfer coefficient	(BTU/hr-ft <sup>2</sup> -°F)
J <sub>H</sub>	heat transfer group,	$\frac{Nu}{Re Pr^{1/3}}$
J <sub>M</sub>	mass transfer group,	$\frac{Nu_M}{Re Sc^{1/3}}$
k	thermal conductivity	(BTU/hr-ft-°F)

$K_g$	mass transfer coefficient	(lb-mol/hr-ft <sup>2</sup> -atm)
$L$	tube length	(ft)
$M_M$	mean molecular weight of vapor-gas mixture	
$Nu$	Nusselt number for heat transfer,	$\frac{hD}{k}$
$Nu_M$	Nusselt number for mass transfer,	$\frac{K_g DRT}{D} \frac{P_{2M}}{P}$
$Nu'$	Nusselt number based on length,	$\frac{hL}{K}$
$Nu'_M$	Nusselt number based on length,	$\frac{K_g LRT}{D} \frac{P_{2M}}{P}$
$P_i$	partial pressure,	(psia)
$P$	total pressure,	(psia)
$Pr$	Prandtl number,	$\frac{c_p M}{k}$
$q$	heat flux,	(BTU/hr-ft <sup>2</sup> )
$Q$	heat transfer	(BTU/hr)
$R$	universal gas constant,	0.73 (ft <sup>3</sup> -atm/lb-mol-°R)
$Re$	Reynolds number,	$\frac{DV\rho}{\mu}$
$Sc$	Schmidt number,	$\frac{\mu}{\rho D}$
$t$	time	
$T$	temperature	(°R)
$U$	overall heat transfer coefficient	(BTU/hr-ft <sup>2</sup> -°F)
$V$	velocity	(ft/hr)
$W$	general dimensionless transfer coefficient	
$x_A$	mole fraction vapor	

Z	general transport modulus	
$\alpha$	thermal diffusivity	(ft <sup>2</sup> /hr)
$\Delta$	driving force, change	
$\nabla$	vector differential operator,	$(\frac{\partial}{\partial x} + \frac{\partial}{\partial y} + \frac{\partial}{\partial z})$
$\epsilon$	eddy diffusivity	(ft <sup>2</sup> /hr)
$\lambda$	latent heat	(BTU/lb)
$\mu$	viscosity	(lb/ft-hr)
$\nu$	kinematic viscosity	$\mu/\rho$ (ft <sup>2</sup> /hr)
$\rho$	density	(lb/ft <sup>3</sup> )

Subscripts not otherwise defined:

1,A	vapor (condensing component)
2,B	gas (non-condensing component)
b	property in bulk flow region
c	property of cooling water
F	function for momentum transfer
g	property in gas phase
H	function for heat transfer
L	property in liquid phase
M	function for mass transfer or mean value of property
s	property at liquid surface
v	property in vapor phase
w	property at wall of condenser tube
<u>    </u>	wavy underline denotes vector quantity
x,y	directions of transport, vector or tensor components
$\bar{\quad}$	bar overline denotes average value of function for turbulent flow
*	asterisk superscript denotes dimensionless variable formed with constant reference values



## INTRODUCTION

For a long time, large reductions in condensing heat transfer have been observed when even a small percentage of non-condensing gas was present in the vapor. This effect is due to the formation of a film of a mixture of non-condensing gas and vapor next to the condensate layer on the condenser surface. The vapor concentration in the gas film is lower than in the main body of the mixture due to removal of the vapor at the condensate surface. Since the condensing vapor must be transported through the gas film, the determination of the overall condensing heat transfer coefficient gives a combination of the heat transfer resistance of the liquid film and the mass transfer resistance of the gas film.

Some previous experimental results have been expressed in empirical equations restricted to the particular systems studied.

Meisenburg's study<sup>(19)</sup> of the air-water system with a stagnant gas film produced a correlation resembling Nusselt's well-known theoretical equation, but using a concentration term raised to a power. Langen<sup>(16)</sup> condensed steam at rest on the outside of a horizontal tube with results applicable only to experimental conditions. Hampson<sup>(11)</sup> made a study which resulted in a linear decrease of the heat transfer coefficient with the weight ratio of nitrogen to steam.

A widely accepted procedure for calculating the overall transfer coefficient when condensing vapors from vapor-gas mixtures was originated by Colburn and Hougen<sup>(6)</sup>. They calculate the heat transfer and mass transfer separately as series phenomena, balancing the heat transferred from the condensate surface toward the cooling water against the heat transferred as latent and sensible heat through the gas film to the condensate surface. The result is:

$$U \Delta T = h_o (T_s - T_c) = h_g (T_b - T_s) + K_g M \lambda (P_b - P_s) \quad (1)$$

This equation is evaluated by trial and error guesses at  $T_s$  at several points in a desired condenser to permit integration of

$$dA = dQ / U \Delta T \quad (2)$$

for the total area required. The data of Cairns<sup>(4)</sup> and of Smith and Robson<sup>(26)</sup> are in good agreement with this method.

Considerable data is available for the determination of the heat transfer coefficients in equation 1, but the mass transfer data is less plentiful. For forced convection, Colburn and Chilton<sup>(5)</sup> extended the Reynolds analogy between fluid friction and heat transfer to include mass transfer. This suggested that

$$\frac{f}{2} = \frac{\Delta P e g}{G^2} \frac{D}{4L} \quad (3)$$

$$j_H = \frac{h}{C_p G} \left( \frac{\mu c_p}{k} \right)^{2/3} \quad (4)$$

$$j_M = \frac{K_g P_{2M}}{G/M_M} \left( \frac{\mu}{\rho D} \right)^{2/3} \quad (5)$$

could all be represented as a single function of the Reynolds number.

Some experimental results fit this analogy fairly well. The data of Little<sup>(17)</sup> for condensing vapors from vapor-gas mixtures in laminar flow gives

$$Nu_M = 2.52 (Re Sc D/L)^{1/3} \quad (6)$$

which resembles the laminar flow heat transfer equation of Sieder and Tate<sup>(25)</sup>,

$$Nu = 1.86 (Re Pr D/L)^{1/3} \left( \frac{\mu_b}{\mu_w} \right)^{0.14} \quad (7)$$

Gilliland and Sherwood<sup>(10)</sup> evaporated vapors into a turbulent air flow. Their data gives

$$Nu_M = 0.023 (Re)^{0.83} (Sc)^{0.44} \quad (8)$$

which compares well with the corresponding turbulent heat transfer equation recommended by

McAdams<sup>(20)</sup>,

$$\text{Nu} = 0.023 (\text{Re})^{0.8} (\text{Pr})^{0.33} \quad (9)$$

Crawford<sup>(7)</sup> applied the analogy to heat and mass transfer in free convection. His work resulted in

$$\text{Nu}_M' = 1.02 (\text{Gr}_m \text{Sc})^{0.373} \quad (10)$$

which resembles the equation recommended by McAdams<sup>(20)</sup>,

$$\text{Nu}' = 0.13 (\text{Gr} \text{Pr})^{0.33} \quad (11)$$

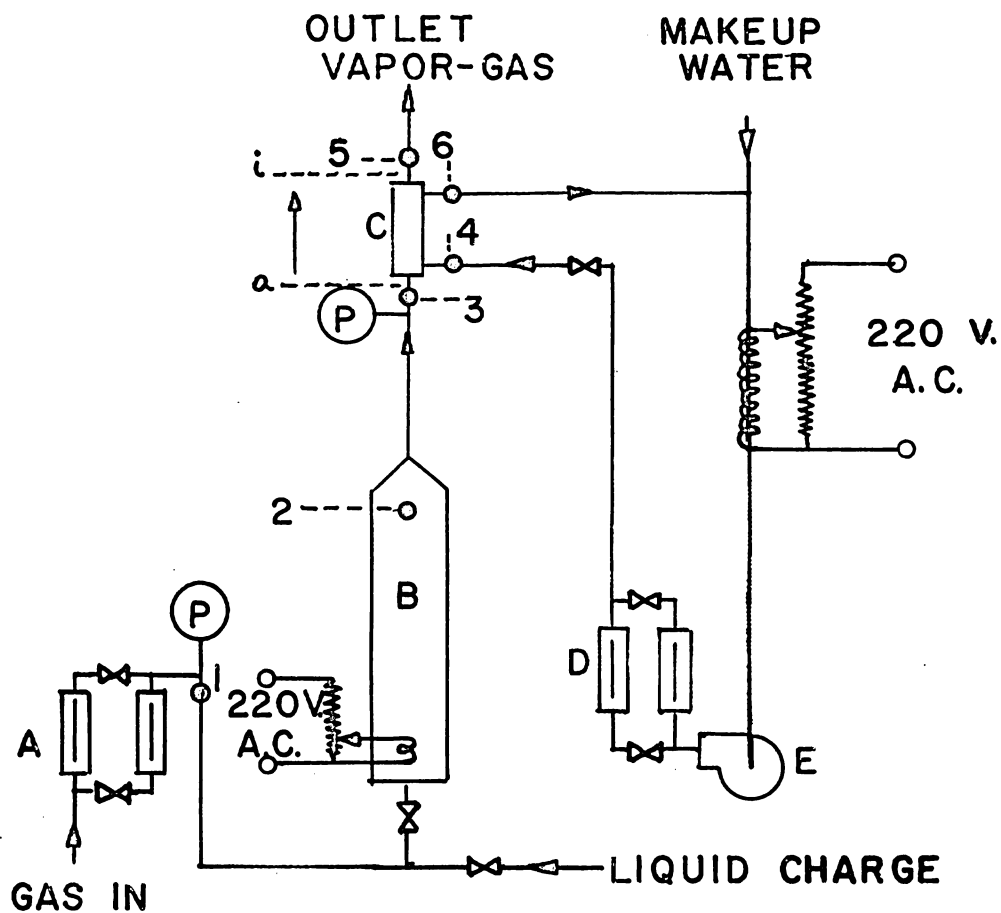
The purpose of this work was to study condensing vapor in a vertical tube in the presence of non-condensing gas in turbulent flow. Observations were made of the effects of vapor velocity, liquid loading, temperature driving force, and vapor-gas composition. A primary objective was to examine the forced convection analogy between heat and mass transfer in this type of condensation process.

## EXPERIMENTAL APPARATUS AND PROCEDURES

The apparatus used for this investigation was originally constructed by Crosser<sup>(8)</sup> and Rosson<sup>(24)</sup> and modified to a vertical condenser by Little<sup>(17)</sup>. The apparatus included a vapor generator, a vertical condenser, and auxiliary equipment such as electric heaters, cooling water pumps, potentiometers, thermocouples, and flow meters for gas and liquid. A schematic diagram of the equipment is shown in Figure 1.

The vapor generator consisted of a 6 kw. Chromalux immersion heater inserted through the flanged bottom of a four foot length of vertically mounted four inch schedule 80 pipe. Liquid capacity was three and one-half gallons. The vertical condenser was a one foot length of 5/8 inch Anaconda Type K copper water tubing inserted in a brass cylinder two inches in diameter. The cylinder was threaded to accommodate a close winding of 3/8 inch copper tubing for cooling water. The high conductivity of the brass bar provided a nearly uniform temperature over the entire condensing surface. Seven differential thermocouple junctions were placed down the length of condenser next to the 5/8 inch copper tube through holes drilled between threads of the brass cylinder. The entire system was well insulated; six inches of magnesia protected the condenser section.

The flow of non-condensing gas was measured



MAJOR COMPONENTS

- A. GAS ROTAMETERS
- B. VAPOR GENERATOR
- C. VERTICAL CONDENSER
- D. COOLING WATER ROTAMETERS
- E. COOLING WATER PUMP

T MEASURED

AT 1-5

$\Delta T$  MEASURED

AT 3-a, b, c, d, e, f, i

4-6

in a rotameter and fed into the bottom of the vapor generator. The gas passed upward through the liquid being vaporized by the immersion heater. The saturated vapor-gas mixture was carried from the vapor generator to the condenser through a vertical two foot length of copper tubing to provide fully developed flow in the condenser. Only a small fraction of the vapor was condensed to provide a nearly constant bulk composition. A bulk vapor-gas temperature drop of between 0.4 and about 4°F was allowed in order to obtain a significant heat flux. The outlet temperature was controlled by the condenser temperature.

A Brown multipoint recording potentiometer was used to measure these temperatures:

- a) vapor generator liquid temperature
- b) vapor-gas bulk temperature at condenser inlet
- c) vapor-gas bulk temperature at condenser outlet
- d) non-condensing gas temperature at rotameter inlet
- e) inlet cooling water temperature.

Pressure of the system was essentially atmospheric and was indicated by a gauge calibrated in ounces per square inch.

An electronic potentiometer with scale subdivisions of 0.002 millivolts was used for differential readings giving the condenser wall temperatures relative to the

inlet vapor-gas temperature and the cooling water temperature rise relative to the inlet cooling water temperature. A differential thermocouple was located one inch from each end of the condenser section to examine end effects.

Electric heaters were used to heat the cooling water to give small temperature driving forces in the condenser and nearly uniform surface temperatures.

The general approach was to obtain equilibrium data at wide ranges of non-condensing gas flow while holding the vapor-gas bulk composition (bulk temperature) nearly constant by decreasing the electrical energy input at each decrease of inlet non-condensing gas flow. This was done at many different vapor-gas bulk compositions in order to examine the effect of composition as well as velocity upon the condensing process. At two of the compositions, the temperature driving force was varied while holding velocity nearly constant to examine liquid loading and the importance of the liquid film resistance.

The bulk vapor-gas temperature was found by averaging inlet and outlet values. The average differential along the tube wall was subtracted from the inlet temperature to determine the wall temperature. This gave the temperature difference which was used as the overall driving force. The thermocouples were in contact with the outside of the condenser tube, but at the low heat fluxes measured,



the indicated temperatures were essentially the same as the inside wall temperatures.

Flow of cooling water as measured in a rotameter was used with the temperature rise of the water to calculate heat transferred,  $Q$ . The overall transfer coefficient was then

$$U = \frac{Q}{A(T_b - T_w)} = \frac{Q}{A \Delta T} \quad (12)$$

An equation from the correlations of Akers and Rosson<sup>(1)</sup> was used for the liquid film coefficient to calculate the condensate surface temperature,  $T_s$ .

$$Nu = 0.125 (Pr)^{1/3} \left( \frac{\lambda}{c_p \Delta T} \right)^{1/6} \left[ \frac{DG}{\mu} \left( \frac{\rho_L}{\rho_V} \right)^{1/2} \right]^{2/3} \quad (13)$$

Thus finding  $T_s$ , the composition at the liquid surface was calculated assuming a saturated mixture. This now gave the partial pressure driving force,  $\Delta P_2$ , for mass transfer across the vapor-gas film.

Sensible heat transfer through the film (according to the method recommended by McAdams<sup>(20)</sup>) was negligible. Cooling of the condensed liquid to the film temperature and heat transfer to tubing at the ends of the condenser were not important.

Velocities and Reynolds numbers were found from measured rates of non-condensing gas flow and bulk vapor-gas compositions. Values of  $\mathcal{D}$  and  $e$  of the mixture were taken from Reid and Sherwood<sup>(23)</sup>. The

Wilke equation was used for finding the bulk vapor-gas mixture viscosity. Knowing the heat flux,  $Q/A$ , and the partial pressures  $P_{2b}$  and  $P_{2s}$ ,  $K_g$  could be calculated directly in lb. moles/hr-ft<sup>2</sup>atm. and then the dimensionless group,  $\frac{K_g D R T}{\rho} \frac{P_{2M}}{P_T}$ , or  $Nu_M$ , was determined.

The precision in experimentally measured quantities was:

- a) temperatures  $\pm 0.4$  °F
- b) temperature differences  $\pm 0.04$  °F
- c) system pressure  $\pm 0.03$  psi
- d) cooling water flow  $\pm 2\%$
- e) non-condensing gas flow  $\pm 4\%$

All heat transfer coefficients calculated from these measurements were expected to have a maximum error of 8 per cent.

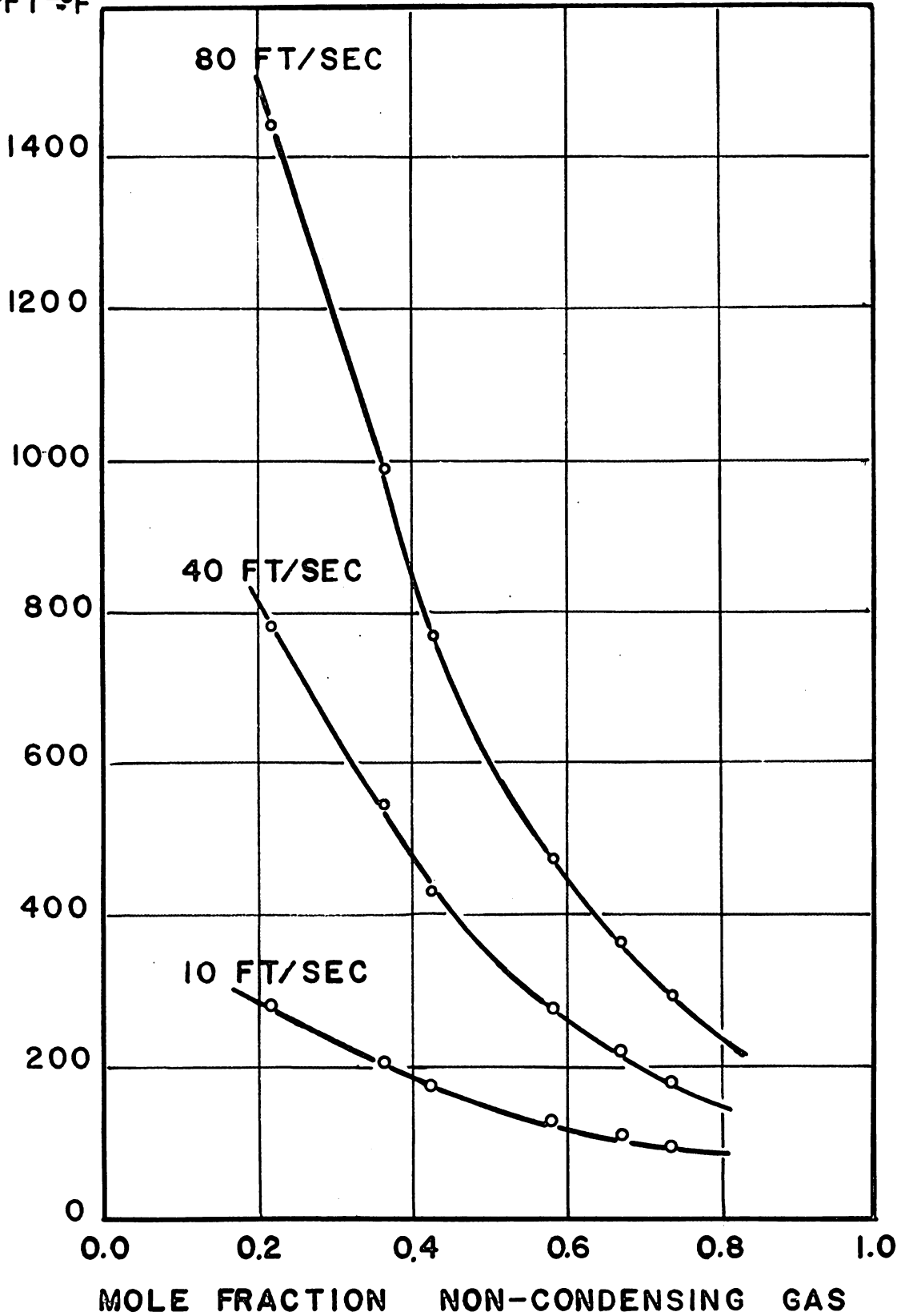
All calculations were performed on the experimental data using an LGP-30, a medium speed digital computer.

## RESULTS

Water was condensed in the presence of air in turbulent flow at six bulk compositions from about 21 to 75 per cent air. Velocities varied from 5 to 108 feet per second while Reynolds numbers ranged between 1200 and 28,000. For constant vapor velocity and composition, the temperature driving force was varied.

At constant composition, the overall condensing coefficients were essentially linear with vapor velocity; the slope decreased as per cent noncondensable was increased. Figure 2 shows the effect of the non-condensing gas on the transfer coefficient at constant velocity. With both velocity and composition constant, an increase in temperature driving force of about 320 per cent made no important decrease in the overall coefficient.

$U$   
 $\frac{\text{BTU}}{\text{HR-FT}^2\text{-}^\circ\text{F}}$



## DISCUSSION

Several approaches<sup>(3), (9), (12)</sup> have been used to arrive at analogies among the processes of transport of momentum, heat, and mass, but each of these approaches includes sufficient simplifying assumptions to reduce the equations of change and related boundary conditions to the same form and thus indicate a common solution.

For the case of forced convection with constant physical properties within a single-phase binary mixture of A and B, with no chemical reaction and neglecting viscous dissipation, radiation, absorption, and secondary mechanisms of diffusion and heat conduction, the following dimensionless equations of change are given by Bird<sup>(3)</sup>:

$$\text{Continuity: } (\nabla^* \cdot \underline{V}^*) = 0 \quad (14)$$

$$\text{Motion: } \frac{D\underline{V}^*}{Dt^*} = \frac{1}{Re} \nabla^{*2} \underline{V}^* - \nabla^* P^* + \frac{1}{Fr} \frac{\underline{g}_A}{g_c} \quad (15)$$

$$\text{Energy: } \frac{DT^*}{Dt^*} = \frac{1}{RePr} \nabla^{*2} T^* \quad (16)$$

$$\text{Continuity of A: } \frac{Dx_A^*}{Dt^*} = \frac{1}{ReSc} \nabla^{*2} x_A^* \quad (17)$$

Further restrictions are that the pressure gradient gives hydrostatic equilibrium,

$$-\nabla^* P + \underline{e} \underline{g}_A = 0 \quad (18)$$

that rates of mass transfer are very small, and that there is no slip at the phase boundary.

Now, equations 15, 16, and 17 have the same form. If  $v^*$ ,  $T^*$ , and  $x_A^*$  have the same boundary conditions, and a transport modulus,  $Z$ , is defined

$$Z_F = \frac{\mu/\rho}{\mu/\rho} = 1, \quad Z_H = \frac{\mu/\rho}{k/\rho c_p} = Pr, \quad Z_M = \frac{\mu/\rho}{D_{AB}} = Sc, \quad (19)$$

a common solution to the three equations is found to be a function only of the space coordinates, time, phase geometry,  $Re$ , and  $Z$ . For steady state flow and the same geometrical system, this predicts that the dimensionless transport coefficients,  $W$ , are all the same function of  $(Re, Z)$ . For

$$\begin{aligned} Z_F = 1, \quad W_F &= \frac{f}{2} = W_F(Re, 1) \\ Z_H = Pr, \quad W_H &= \frac{Nu}{RePr} = W_H(Re, Pr) \\ Z_M = Sc, \quad W_M &= \frac{Nu_M}{ReSc} = W_M(Re, Sc) \end{aligned} \quad (20)$$

There are adequate experimental data to support

$$\frac{f}{2} = 0.023 (Re)^{-0.2} \quad (21)$$

for turbulent flow in tubes ( $5000 < Re < 200,000$ ). Several writers<sup>(15), (20)</sup> report a similar heat transfer equation,

$$\frac{Nu}{RePr} = 0.023 (Re)^{-0.2} (Pr)^{-2/3} \quad (22)$$

and so it seems natural to predict a mass transfer equation,

$$\frac{\text{Nu}_M}{\text{ReSc}} = 0.023 (\text{Re})^{-0.2} (\text{Pr})^{-2/3} \quad (23)$$

which indicates the common solution for this system is

$$W = 0.023 (\text{Re})^{-0.2} (z)^{-2/3} \quad (24)$$

applicable to the transfer of momentum, heat, and mass.

The analogy between heat and mass transfer is valid in the presence of pressure gradients and irregular boundaries, but the friction factor analogy is limited to parallel boundaries, such as in a tube.

The evaporation data of Gilliland and Sherwood<sup>(10)</sup> is reported as

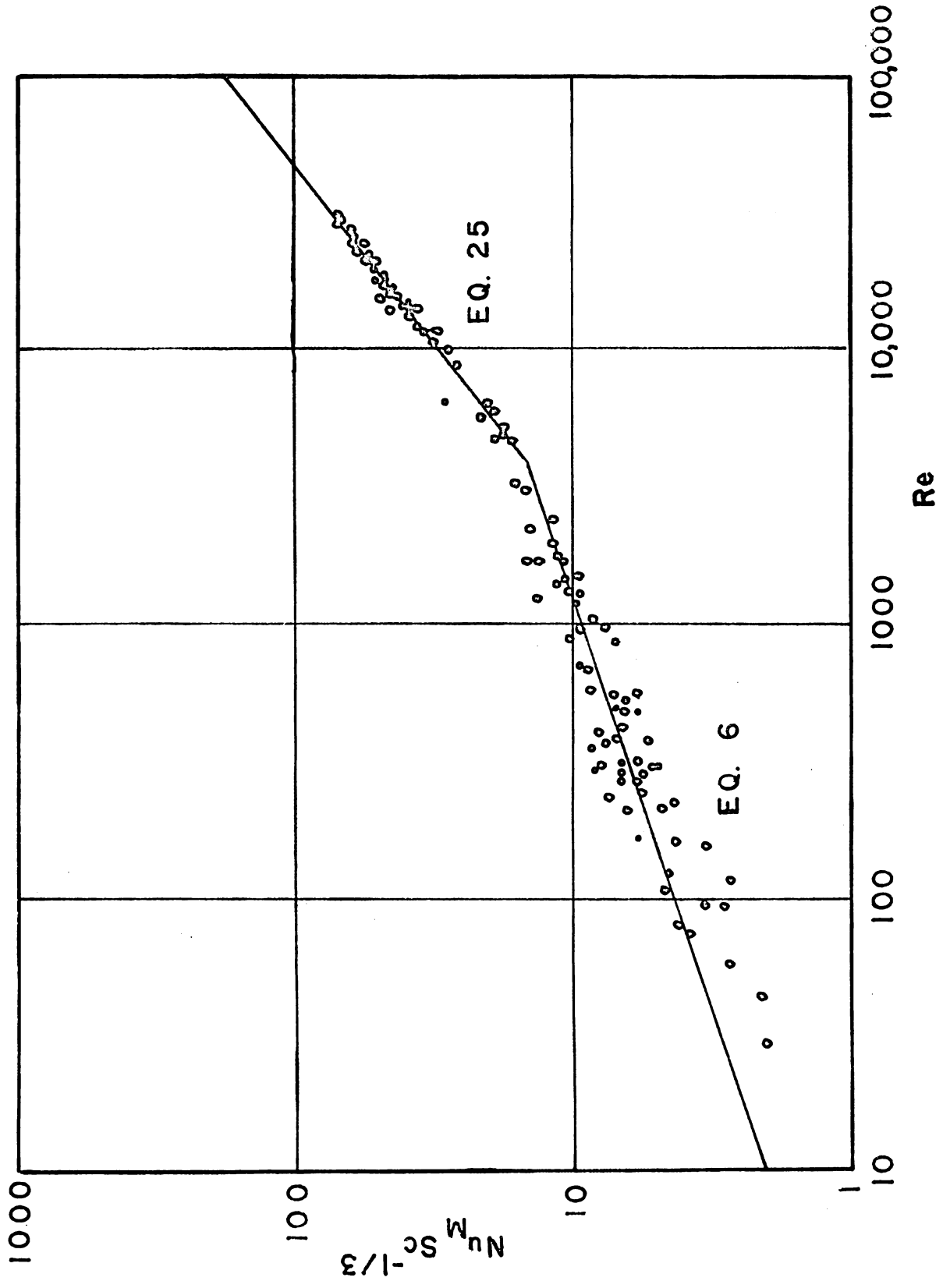
$$\text{Nu}_M = 0.023 (\text{Re})^{0.83} (\text{Sc})^{0.44} \quad (8)$$

The condensation data in this study is well represented by

$$\text{Nu}_M = 0.023 (\text{Re})^{0.78} (\text{Sc})^{0.33} \quad (25)$$

for Re greater than 4,000. Figure 3 shows these results for turbulent flow along with Little's laminar flow data<sup>(17)</sup>.

It seems logical that the closer the assumptions made are realized experimentally, the better the results will support the analogy. Thus, Crawford's free convection correlation<sup>(7)</sup> with a stagnant film and considerable net mass movement toward the condensing





surface is analogous to heat transfer in form but with a constant 8 times larger and exponent 12 per cent larger. Little's work<sup>(17)</sup> with a laminar flow mechanism in a tube still required a radial mass movement that was not negligible compared with bulk flow up the tube. His mass transfer correlation differs from the analogous heat transfer equation only by a constant which is 35 per cent higher. At the higher bulk velocities in this study, the radial mass movement was relatively nil, so that the assumptions were met well enough to give a mass transfer equation essentially the same as the related heat transfer expression.

It is well-known that frictional effects, mean velocity distributions, rate of spreading, and other features of turbulent flow bear little resemblance to those found in laminar flow. These differences result from a diffusiveness of turbulence which is much greater than molecular diffusion and is more intimately connected with the mean flow and thus more closely related to the Reynolds number.

Except near a wall or other phase boundary with zero relative velocity, the transport by turbulent motion surpasses that by molecular motion so much that the latter has little overall effect. If molecular diffusion cannot be neglected, it is assumed to be additive to turbulent diffusion; thus the "eddy diffusivities" for momentum, heat, and mass transport

are defined so that

$$\begin{aligned}\bar{\tau}_{yx} &= - \frac{\rho}{\epsilon_c} (\nu^{(m)} + \epsilon_s) \frac{d\bar{V}_x}{dy} \\ \bar{q}_y &= - \rho c_p (\alpha^{(m)} + \epsilon_H) \frac{d\bar{T}}{dy} \\ \bar{N}_{A,y} &= - \frac{1}{RT} (D^{(m)} + \epsilon_M) \frac{d\bar{C}_A}{dy}\end{aligned}\tag{26}$$

At the end of the laminar zone (as a function of Re), the "eddy" coefficients begin to increase in significance and when fully developed turbulent flow is reached, the eddy term is all that is significant.

This difference in laminar and turbulent transport mechanisms is demonstrated in the empirical correlations as a difference in Reynolds number dependence,  $(Re)^{1/3}$  and  $(Re)^{0.8}$ .

Use of the 1/3 exponent with the Schmidt number in both mechanisms is supported by much experimental data with the Prandtl number and by theoretical work by Eckert<sup>(9)</sup> assuming a cubic parabola form for velocity and temperature profiles.

The effect of mass removal and addition may be the source of  $(Re)^{0.78}$  in this condensation study and  $(Re)^{0.83}$  in Gilliland's evaporation work<sup>(10)</sup>. In the evaporation mechanism, transfer in a vapor-rich region near the surface would be enhanced more by turbulent diffusion than would be the case for the corresponding vapor-lean region for the condensation mechanism, although according to Stewart's<sup>(3)</sup> analysis,

the radial velocity component increases the mass transfer rate in the condensation process and decreases the rate for evaporation. As previously pointed out, this radial mass flow effect is negligible in this turbulent study.

Throughout each set of runs at constant composition, the overall  $\Delta T$  was controlled fairly closely at the smallest practical value, in order to promote uniform wall temperatures and minimize bulk composition changes. On two composition runs,  $\Delta T$  was increased by as much as 320 per cent while keeping velocities about the same. As no significant change resulted in the overall transfer coefficient, it appears that liquid loading was not important due to the relative unimportance of the liquid film resistance compared with the gas film resistance to diffusion.

In this study, close experimental limits on changes in condenser surface temperature, bulk composition, velocity, and  $\Delta T$  provide results which should be considered as local transfer coefficients which could vary widely throughout industrial equipment in which these controlled variables changed appreciably. In general, point to point calculations would be needed for design work. Another stumbling block preventing direct application is the amplifying effect of large mass fluxes as described by Stewart<sup>(3)</sup>.

The data of Gilliland and Sherwood<sup>(10)</sup> for evaporation in tubes and of Maisel and Sherwood<sup>(18)</sup>

for evaporation from solid spheres, cylinders, and plates, offer evidence for the Colburn and Chilton analogy<sup>(5)</sup> between heat and mass transfer, but all condensation studies previous to this one and Little's<sup>(17)</sup> were more concerned with providing a model typical of commercial usage in order to verify the design method credited to Colburn and Hougen<sup>(6)</sup>. Analogy - supporting data concerned with flat plates and other solid objects do not provide a widely useful conclusion, as functions such as  $Re_x$ ,  $Nu_x$ , etc., have rather limited definition and application.

For condensation in tubes, the results of Little<sup>(17)</sup> and of this study cover the laminar and turbulent flow regions, respectively. This is the first experimental support for the Colburn-Chilton analogy for heat and mass transfer that is based on condensation in tubes. The necessary restrictions limit rather severely the direct applications of the analogy, but a useful starting point is provided for solving problems in mass transfer using the analogous heat transfer solution.

## CONCLUSIONS

1. When vapors are condensed in the presence of a non-condensing gas, the overall rate may be represented as a series phenomena in which the resistance of the liquid film is expressed by an applicable pure vapor equation and the mass transfer coefficient may be expressed by

$$(a) \quad \frac{K_g DRT}{\rho} \frac{P_{2M}}{P} = 2.52 (Re Sc D/L)^{1/3} \quad (\text{Little})$$

for Re less than 4,000

and (b)  $\frac{K_g DRT}{\rho} \frac{P_{2M}}{P} = 0.023 (Re)^{0.78} (Sc)^{1/3}$

for 4,000 < Re < 100,000

for flow up a vertical tube, as suggested by analogy to forced convection heat transfer.

2. The overall condensing coefficient varies linearly with velocity for high Re.

3. Liquid film resistance and effect of  $\Delta T$  are not important for high Re.

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APPENDIX A



## EXPERIMENTAL DATA

Table A

Run	$T_b$ (°F)	$T_s$ (°F)	$T_w$ (°F)	V (ft/sec)	Q (BTU/hr)
1-1	150.50	144.51	143.75	85.0	351.2
2	151.40	144.65	143.74	73.4	373.8
3	152.70	145.01	143.94	66.7	400.5
4	153.00	144.88	143.75	59.8	389.1
5	152.10	143.33	142.33	55.9	358.9
6	152.90	144.13	142.95	48.9	350.0
2-1	160.20	153.65	152.58	101.0	549.0
2	161.00	154.78	153.74	92.0	508.4
3	162.40	155.19	153.95	84.4	554.2
4	162.50	154.88	153.56	74.8	538.6
5	160.90	153.28	152.01	69.3	493.4
6	161.10	153.14	151.84	61.9	463.9
7	162.40	153.06	151.60	54.7	465.0
8	162.50	153.72	152.12	49.7	481.5
3-1	171.30	165.25	163.88	108.3	749.0
2	171.90	165.28	163.78	96.6	750.6
3	171.60	164.67	163.05	86.0	739.7
4	172.10	164.73	162.96	76.2	734.3
5	172.00	163.85	161.97	66.9	704.5
6	172.00	162.85	160.67	55.3	700.1
7	171.50	161.64	159.25	48.1	688.1
8	171.20	163.78	161.89	40.2	510.0
9	171.80	164.26	162.46	18.4	288.3
10	173.10	160.97	157.91	66.9	1052.7
11	172.50	168.69	167.86	68.3	364.2
12	173.40	169.46	168.53	69.2	406.8
4-1	184.60	180.71	179.27	93.2	758.5
2	185.10	180.36	178.58	88.5	873.8
3	186.50	181.63	179.77	81.2	857.4
4	186.50	181.26	179.12	71.9	894.8
5	186.20	180.86	178.76	62.6	798.3
6	185.60	180.09	177.96	54.8	738.1
7	186.00	180.47	178.64	49.2	594.1
8	186.20	180.63	178.74	41.8	549.5
9	186.30	180.26	178.02	35.6	574.6
10	186.60	181.97	170.58	26.0	511.9

Run	$T_b$ ( $^{\circ}\text{F}$ )	$T_s$ ( $^{\circ}\text{F}$ )	$T_w$ ( $^{\circ}\text{F}$ )	V (ft/sec)	Q (BTU/hr)
5-1	191.40	186.46	183.55	72.4	1208.9
2	190.60	185.48	182.50	63.6	1128.8
3	190.10	184.93	181.98	57.0	1037.8
4	190.30	184.95	182.41	48.7	809.0
5	190.60	186.25	184.22	42.1	610.1
6	190.45	187.47	186.00	65.5	632.6
7	190.70	183.29	179.51	65.6	1378.4
8	191.20	184.06	180.41	66.8	1353.2
6-1	199.80	196.84	193.72	62.1	1216.2
2	200.20	195.84	192.36	27.7	753.1
3	200.00	196.37	191.56	25.8	776.3
4	199.90	195.63	191.63	23.8	776.3
5	199.90	194.92	190.03	22.3	729.9
6	200.20	195.34	191.09	21.1	746.8
7	200.20	194.93	190.56	19.60	723.2
8	200.20	194.61	189.94	17.75	714.1
9	200.50	195.39	189.78	13.89	722.4
10	200.20	193.44	187.21	10.45	639.6
11	199.90	192.54	184.53	7.62	644.7
12	200.50	193.28	184.17	5.24	564.0

Table B

Run	$y_2$ bulk	U (BTU/hr-ft <sup>2</sup> -°F)	Re	Nu <sub>M</sub> Sc <sup>-1/3</sup>
1-1	0.747	304.3	22,497	57.6
2	0.741	385.3	19,381	53.6
3	0.733	267.3	17,574	49.1
4	0.731	346.0	15,723	44.9
5	0.736	214.8	14,723	40.2
6	0.731	205.7	12,843	37.7
2-1	0.682	421.4	26,250	62.5
2	0.675	409.5	23,814	59.2
3	0.664	383.5	21,739	53.8
4	0.652	352.3	19,223	49.4
5	0.675	324.5	17,872	47.6
6	0.672	394.0	15,923	42.6
7	0.664	251.8	14,014	35.6
8	0.662	271.3	12,730	38.7
3-1	0.588	590.3	27,411	64.1
2	0.582	540.6	24,396	58.0
3	0.583	506.0	21,636	54.9
4	0.578	469.8	19,108	50.5
5	0.578	410.8	16,758	44.5
6	0.577	361.4	13,825	38.7
7	0.582	328.5	12,040	37.3
8	0.584	320.3	10,057	35.7
9	0.578	180.5	4,593	19.47
10	0.567	405.2	16,690	45.3
11	0.572	459.1	17,053	45.0
12	0.564	488.5	17,250	47.2
4-1	0.442	832.2	22,686	57.4
2	0.436	783.8	21,501	53.9
3	0.418	745.0	19,647	48.4
4	0.418	709.1	17,374	47.1
5	0.420	627.5	15,118	41.8
6	0.428	564.9	13,245	38.6
7	0.422	472.1	11,872	30.3
8	0.420	430.7	10,088	27.7
9	0.419	405.8	8,590	26.9
10	0.416	426.5	6,281	30.0

Run	$y_2$ bulk	U (BTU/hr-ft <sup>2</sup> -°F)	Re	Nu <sub>M</sub> Sc <sup>-1/3</sup>
5-1	0.353	900.6	17,311	52.8
2	0.374	814.9	15,227	49.5
3	0.369	747.4	13,627	46.3
4	0.638	599.7	11,660	34.8
5	0.364	559.2	10,073	31.1
6	0.365	831.3	15,685	45.8
7	0.362	717.8	15,660	43.5
8	0.357	733.4	15,980	43.4
6-1	0.228	1169.8	14,536	48.4
2	0.217	561.8	6,444	20.25
3	0.221	537.9	6,005	20.20
4	0.222	549.0	5,540	21.80
5	0.222	475.9	5,191	18.00
6	0.217	479.4	4,905	18.27
7	0.217	438.7	4,560	16.60
8	0.217	407.0	4,126	15.55
9	0.212	394.1	3,227	16.53
10	0.217	287.9	2,429	11.96
11	0.222	245.3	1,773	11.57
12	0.212	202.0	1,217	9.77

APPENDIX B

## SAMPLE CALCULATIONS

Run 3-2

1.  $T_{b \text{ in}} = 172.70^{\circ} \text{ F}$  by recording potentiometer  
 $T_{b \text{ out}} = 171.10^{\circ} \text{ F}$ 

$$\underline{T_b = 171.9^{\circ} \text{ F}}$$
2.  $T_{b \text{ in}} - T_w = 8.92^{\circ} \text{ F}$  by differential thermocouple
3.  $\Delta T = 171.90 - 163.78 = \underline{8.12^{\circ} \text{ F}}$
4. Cooling water rotameter = 67.7 lb./hr.  
 $\Delta t_{\text{water}} = 11.09^{\circ} \text{ F}$  by differential thermocouple
5. Heat removed,  $Q = (67.7)(11.09)(1) = \underline{750.6 \text{ BTU/hr.}}$
6. Overall heat transfer coefficient  

$$U = \frac{750.6 \text{ BTU/hr}}{(8.12^{\circ} \text{ F})(0.171 \text{ ft}^2)} = \underline{540.5 \text{ BTU/hr-ft}^2\text{-}^{\circ} \text{ F}}$$

Calculations involving vapor pressures, densities and viscosities were done on the computer with the physical properties expressed as functions of temperature and composition.

7. Vapor pressure of water,  $P_1$  (psia) as  $f(T^{\circ} \text{ R})$ 

$$\ln P_1 = 16.2727 - \frac{9122.3}{T}$$
8. Mole fraction water vapor,  $y_1 = \frac{P_1}{P_T}$   
Mole fraction air,  $y_2 = 1.00 - y_1$
9. Viscosity of pure water vapor,  $\mu_1^{\circ}$  (lb<sub>m</sub>/ft-hr) =  $f(T^{\circ} \text{ R})$ 

$$\mu_1 = 0.02885 + 0.0000499 (T - 609.7)$$

10. Viscosity of pure air,  $\mu_2^0$  (lb<sub>m</sub>/ft - hr) as  
 $f(T^{\circ}R)$ ,  $\mu_2 = 0.0491 + 0.000058 (T-609.7)$

11. Average viscosity of bulk vapor-gas mixture  
 (Wilke equation)

$$\mu_{vb} = \frac{\mu_{1b}}{1+y^2/y_1(1)} + \frac{\mu_{2b}}{1+y^1/y_2(1)} \quad (\text{lb}_m/\text{ft-hr})$$

12. Average density of bulk vapor-gas mixture

$$\rho_{vb} = \frac{M_M}{359.05} \times \frac{491.69}{T_b} \times \frac{P_T}{14.696}$$

$$= (0.093182)(P_T/T_b)(18.02y_{1b} + 28.97 y_{2b})$$

13. Density of liquid water  $\rho_L$  (lb/ft<sup>3</sup>) =  $f(T^{\circ}R)$

$$\rho_L = 73.98 - 0.0210 T_w$$

14. Viscosity of liquid water  $\mu_L$  (lb<sub>m</sub>/ft-hr) =  $f(T^{\circ}R)$

$$\ln \mu_L = \frac{2820.73}{T_w} - 4.5861$$

15. Air flow rate from rotameter, 3.34 ft<sup>3</sup>/min at  
 30 psia and 75°F (535 °R)

16. Bulk velocity of vapor-gas mixture,  $V_b$  (ft/sec)

$$V_b(\text{air flow, } 3.34 \text{ ft}^3/\text{min}) \left( \frac{T_b}{535} \right) \left( \frac{30.0}{P_T} \right) \left( \frac{1}{y_2} \right) \left( \frac{1}{60 \times 0.02331} \right) =$$

$$\frac{96.6 \text{ ft/sec}}{14.99 \times 0.582}$$

17. Reynolds number for vapor,  $\frac{DV_b \rho_{vb}}{\mu_{vb}}$

$$Re = (0.0545 \text{ ft})(3600 \text{ sec/hr})(96.6 \text{ ft/sec}) \left( \frac{\rho_{vb}}{\mu_{vb}} \right)$$

$$= \underline{24,396}$$

18. Modified Reynolds number for  $h_L$  equation of

Akers and Rosson, 
$$Re_V = \frac{D v_b \rho_{v_b}^{1/2} \rho_L^{1/2}}{\mu_L}$$

19. 
$$F(T \text{ } ^\circ R) = k_L (Pr_L)^{1/3} \left( \frac{\lambda}{(C_P)_L} \right)^{1/6}$$

$$F(T) = 3.698 - 0.003298 T_W$$

20. Liquid film coefficient,  $h_L$  (BTU/hr-ft<sup>2</sup>-°F)

$$h_L = 2.29 (F(T)) (Re_V)^{2/3} \left( \frac{1}{(\Delta T)} \right)^{1/6}$$

$$= \underline{2920 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}}$$

21. Liquid film surface temperature,  $T_S$

$$T_S = T_W + U/h_L (\Delta T)$$

$$= 163.78 + \frac{540.5(8.12)}{2920} = \underline{165.28^\circ\text{F}}$$

22. Diffusion coefficient,  $D_{P_T}$  (ft<sup>2</sup>-psia/hr) =

$$f(T_b \text{ } ^\circ R), \quad D_{P_T} = 0.001192 (T_b)^{3/2}$$

23. Schmidt number,  $Sc$

$$Sc = \frac{\mu_{v_b} P_T}{\rho_{v_b} (D_{P_T})} = \underline{0.6157}$$

24.  $\Delta P_2$  (psia) driving force across the gas film. Calculate  $P_{1_b}$  and  $P_{1_s}$  from  $T_b$  and  $T_s$  using (7);  $\Delta P_1 = \Delta P_2$  since  $P_T$  is constant.

25. 
$$\frac{K_g DRT}{D} = \frac{P_T QDRT}{(D_{P_T}) \lambda M A \Delta P_2} = \underline{80.5}$$



$$26. \quad P_{2M}/P_T = \Delta P_{2/P_T} \ln \frac{P_T - P_{1s}}{P_T - P_{1b}} = \underline{0.613}$$

$$27. \quad Nu_M = (80.5)(0.613) = \underline{49.3}$$

$$28. \quad Nu_M Sc^{-1/3} = \frac{49.3}{(0.6157)^{1/3}} = \underline{58.0}$$

APPENDIX C

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## PROPERTY VALUES

All values taken from references 2, 13, and 23

VAPOR PHASE

T( $^{\circ}$ F)	<u>150</u>	<u>200</u>	<u>250</u>	
Air Viscosity (cp)	0.0203	0.0215	0.0227	(2)
Water Vapor Viscosity (cp)	0.01192	0.01296	0.01398	(2)
Diffusivity of Water-Air	at $66^{\circ}$ F is $0.260 \text{ cm}^2/\text{sec}$			(23)

LIQUID PHASE

(13)

T( $^{\circ}$ F)	<u>140</u>	<u>160</u>	<u>180</u>	<u>200</u>
Thermal Conductivity $k_L$ (BTU/hr-ft- $^{\circ}$ F)	0.378	0.384	0.388	0.392
Prandtl Number	2.98	2.51	2.16	1.88
Specific heat $c_p$ (BTU/lb- $^{\circ}$ F)	1.00007	1.00147	1.00336	1.00583
Viscosity $\mu_L$ (cp.)	0.467	0.398	0.345	0.303
Density $\rho$ (lb/ft $^3$ )	61.39	61.01	60.58	60.13
Heat of Vaporization $\lambda$ (BTU/lb)	1014.1	1002.3	990.2	977.9

APPENDIX D

## DETAILS OF EXPERIMENTAL APPARATUS

### Vapor Generator

The vapor generator was constructed from a four foot length of Schedule 80 four inch pipe mounted vertically. A 6 kw. Chromalux immersion heater was inserted through the flanged bottom. The generator was equipped with a gauge glass for detection of the liquid level. The generator was insulated with two inches of magnesia. Liquid capacity was three and one-half gallons.

### Condenser

The condenser was mounted vertically approximately three feet above the vapor generator. The one foot length condenser consisted of 5/8" Anaconda Type K copper water tubing inserted in a cylindrical brass bar two inches in diameter. The brass bar was threaded with 3/8" copper tubing wound into threads. Seven thermocouple junctions were mounted flush against the central 5/8" copper tube through holes drilled in the outer sleeves. Two additional thermocouples were soldered to the copper tube one inch from each end of the condenser. Construction and other details are discussed by Rosson<sup>(24)</sup> and Crosser<sup>(8)</sup>.

### 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 Northup 30 gauge iron-constantan duplex.

#### Temperature Measuring Instruments

All emfs generated were read by use of a Brown Electronix Potentiometer, Model No. Y 156 x 15 (VH)-X-(V), which read millivolts in 0.002 millivolt subdivisions. Direct temperature measurements were also recorded by a Brown Multipoint Recorder, Model No. 153x62P12-X-16, which read directly in  $^{\circ}\text{F}$  from  $100^{\circ}\text{F}$  to  $300^{\circ}\text{F}$  in  $1^{\circ}\text{F}$  subdivisions.

#### Cooling Water

Cooling water was supplied to the condenser at a constant temperature maintained by electric heaters. The flow rate was given by two calibrated Fischer and Porter rotameters. Cooling water circulation was provided by a centrifugal pump. Copper thermocouple wells were located in the inlet and outlet cooling water streams to measure the temperature rise.

#### Air Supply

Dry, clean air was supplied at nearly constant pressure and temperature by a DeVilbiss transformer and pressure regulator with a capacity of 50 SCFM. Two calibrated Fischer and Porter rotameters were used to meter the air flow.