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UNIVERSITY OF LOUISVILLE
GRAPHICAL ANALYSIS
"
OF CERTAIN DIFFUSION PROBLEMS

A Thesis

Submitted to the Faculty
of the Graduate School
of the University of Louisville
in Partial Fulfillment
of the Requirements
for the Degree of

MASTER OF CHEMICAL ENGINEERING

Department of Chemical Engineering

Wylie C. Kirkpatrick
"

September, 1947

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**GRAPHICAL ANALYSIS
OF CERTAIN DIFFUSION PROBLEMS**

Wyllie C. Kirkpatrick

Approved by the Examining Committee

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September, 1947

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ABSTRACT

An investigation is here made of the possibility of predicting the temperature, humidity, and enthalpy conditions of the exit gas from a non-adiabatic wetted wall tower.

It is found that single runs give results too uncertain for use. The averages of sets of several countercurrent flow runs were found statistically to be reasonably dependable, but the averages for parallel flow sets were unreliable.

Good heat transfer-mass transfer coefficient ratios were obtained from carefully selected runs.

A relationship is indicated between the humidity and velocity of gas and the evaporation rate of a liquid with which it is in contact.

It is concluded that the graphical method of analysis for non-adiabatic liquid-gas diffusion processes is too uncertain to be of general use.

INTRODUCTION

Whenever a volatile liquid is exposed to a moving gas, a physical change ensues. At any given temperature, liquids exert a definite vapor pressure; this may be visualized as a vapor blanket of a certain density in contact with the liquid. Unsaturated air blowing over moist earth, for example, blows away the vapor-rich blanket near the moist soil. The earth continues to lose moisture in this manner until the water is gone. When the air containing vapor meets cooler air, the temperature drops, saturation follows, and some vapor condenses. In addition to these changes, the water absorbs or gives off heat, depending on its temperature relations with the air. Many other liquids and gases follow the same procedure when exposed to a similar natural or artificial environment.

It may be shown theoretically, and it has been confirmed experimentally, that where a vapor has different concentrations at two adjacent points, the rate at which these concentrations tend to equalize is proportional to the difference between them. Similarly it may be shown that where different temperatures exist at two such points, the rate of heat flow is proportional to the temperature difference between them. That is, through a unit area between points a and b

$$dp = K_1 (P_a - P_b) \quad \text{and} \quad dt = K_2 (t_a - t_b)$$

where p represents vapor pressure, t the temperature and the K's are constants. It has been found that the logarithm of the vapor pressure of a liquid varies approximately as its latent heat of vaporization and inversely as its absolute temperature. Thus for the liquid-gas system at disequilibrium, the temperature changes depend on relative gas and liquid temperatures, on amount of liquid evaporated, and on heat flow from the surroundings; the liquid vapor pressure varies exponentially with its temperature; and the rate

of evaporation depends on two varying vapor pressures. These interrelated conditions indicate that liquid-gas relations may be somewhat complicated and this fact has been demonstrated, so that at present, no satisfactory simple way has been evolved for forecasting these interactions.

The primary purpose of this thesis is to determine whether a relatively simple graphical analysis method may be used to calculate the paths of liquids and gases as they interact. It is desired to find whether the condition of the gas leaving a given system may be predicted if the gas and liquid inlet and exit temperatures are known. Only nonadiabatic systems are considered in this study. If the final gas conditions can be accurately predicted, it should be possible to calculate readily other valuable information such as heat and mass transfer coefficients.

For the investigation a wetted wall tower was used. This apparatus uses a tubular tower section as the contact area for liquid and gas. The particular suitability of this equipment is that by proper choice of tower, dimensions, and flow conditions, almost any desired arrangement of liquid and gas in contact may be effected. Gas and liquid rates, temperatures, direction of gas flow relative to liquid flow, gas velocities, and area of contact may be selected. Two liquids were used in the investigations: water and methanol. Water was selected as the control liquid because of the detailed information on the properties of water, and water-air mixtures readily available. Methanol was selected as the other test liquid since a fair amount of data on methanol and methanol-air mixtures is available, it is not an expensive substance, and its moderate volatility and high latent heat of vaporization at room temperatures should give operating conditions fairly close to those of the water runs.

Some of the industrial equipment utilizing liquid-gas interaction are dryers, humidifiers, air conditioning equipment, cooling towers, and evaporation equipment. These installations generally are non-adiabatic in operation, and there is no dependable way to forecast what their performances will be. If this simple graphical method is satisfactory, a few test runs would enable prediction of heat and mass transfer coefficients and other characteristics of the apparatus for any range of operations. Furthermore, for a complicated non-adiabatic diffusion type apparatus, if water or some other liquid whose properties are well known is used to calibrate the machine, information may be provided to enable successful operation with other industrial liquids whose properties are little known.

HISTORICAL

Although industrial operations involving diffusion, drying, and humidification have been carried on for centuries, nearly all the progress in investigating the mechanism of these phenomena has been made in the past hundred years. The scientific information available usually may be considered as relating to diffusion in general, to relationships between heat transfer and mass transfer, and to the properties of liquid-gas mixtures.

In the late 1700's, Black (1) made quantitative measurements of the heat of vaporization of water and developed the concept of specific heat. In 1860, Maxwell (2) published the first theories on the diffusion of gases through each other, based upon calculations of the molecular properties of gases; these theories were extended to liquid-gas diffusion by Stefan (3) in the 1870's and by Sutherland (4) in the 1890's. Work proceeded rather slowly in developing diffusion and evaporation theory, and only a moderate amount of literature on the subject was published prior to 1915. A new impetus was given diffusion studies in 1923 when Whitman (5) proposed that a two film theory applied generally to cases of liquid and gas contact. This theory held that gas and liquid films at the interface controlled the heat and mass exchanges between the two phases. This film theory has been the basis for considerable work, although lately, appreciable discrepancies in recent investigations suggest that other factors may apply. Othmer (6) has developed equations expressing vapor pressure and latent heat as straight line functions of temperature.

Following Reynold's publication in the 1890's of an analogy between heat transfer and fluid friction for turbulent flow conditions, Prandtl (7) in 1910 advanced a theory that resistance to heat transfer is partly due to a laminar film resistance near the phase interface and partly due to fluid friction effects in the fluid core. He worked out a relationship between

the surface coefficient of heat transfer and the ratio of fluid velocities in the laminar and turbulent flow areas. Colburn (8) in 1930 derived for mass transfer an analogy to Prandtl's idea for heat transfer. Applying Reynold's ideas to the eddy diffusion zone, he reasoned that mass transfer was a pure diffusion process in the laminar flowing film. Arnold (9) then suggested that an interface film might be assigned a certain "effective thickness" which would be suitable for both heat and mass transfer. Colburn next developed a relationship for heat transfer and fluid friction, and Chilton and Colburn (10) in 1934 found an analogous relationship for gas-film diffusion rates and heat transfer in their equations relating heat and mass transfer coefficients. Gilliland and Sherwood (11) obtained very good results in deriving an equation for the effective thickness of a film of assigned properties, and in predicting the rate of transfer of liquids into moving vapor streams. In 1941 Barnett and Kobe (12) ran an extensive series of wetted wall tower calculations to check heat transfer, with and without simultaneous mass transfer, against the currently accepted equations. Their results agreed in general with the equations, differing somewhat on the constants involved.

The most logical substances for investigation in evaporation or liquid-gas diffusion problems were water and air. The earliest extensive work published on water-air properties was that of Regnault about 1846. Despite the tremendous growth in requirements for steam power in the nineteenth century no great amount of data on the properties of steam seems to have been gathered until the close of the century. From about 1890 considerable water-air data were presented by various workers, particularly German and American; these were collected and published in a set of steam tables by

Marks and Davis in 1908. In 1908 also, an important part of the groundwork for humidity calculations was published by Grosvenor (13) who proposed the plan of using a pound of dry air as a basis for humidity calculations and a chart relating temperature and humidity. In 1919, the *Wärmestabellen* were published in Germany providing enlarged and very reliable water-air data, particularly in moderate pressure and temperature ranges. Following an intensive research program on steam after the first World War and an international conference on the subject in 1935, the Keenan and Keyes steam tables (14) were published in 1936 and they remain at present the most authoritative work in the field. Recently printed have been Zimmerman and Lavine's psychrometric tables (15) which are particularly useful from an engineering point of view.

The fore-runner of present day adiabatic saturation and wet-bulb theory was first propounded by Ivory (16) in 1822 when he stated that air in contact with a wet-bulb cools to wet-bulb temperature and that its humidity increases until it becomes saturated at that temperature. A number of scientists enlarged on Ivory's premises during the early part of the nineteenth century and the investigation was apparently dropped until 1877 when Maxwell (17) presented a conclusion derived from diffusion theory: he claimed that the amount of heat flowing from the surrounding gas to the wet bulb just equaled that required to evaporate the liquid leaving the bulb. In 1911 Carrier (18) reintroduced the subject suggesting that the adiabatic cooling lines on the chart proposed by Grosvenor in 1908 (13) could be used in calculating humidities. In 1922 Lewis (19) made an analytical derivation of the psychrometric relation, stating that the rate of heat flow to the bulb was proportional to the difference in gas and wet bulb temperatures and that the rate of vaporization was proportional to the difference between saturation humidity at

wet-bulb temperature and actual humidity of the surrounding gas. Some derivations for wet-bulb temperatures were presented by Sherwood and Comings and by Colburn (20) in 1932 and again later by Sherwood (21), but very little supplementary information has appeared since. In 1941 Rodekofer (22) in a theoretical study presented a graphical method for predicting the final humidity and enthalpy conditions of the vapor in a water-air reaction tower. In 1943 Boelter (23) presented equations for determining the vapor diffusion in wetted wall towers as a function of certain determinable properties of the system. For the present thesis, there has been made a series of water-air runs in a wetted wall tower in an effort to check empirically the results of these types of runs against the theoretical values. The investigation has been extended to the methyl alcohol-air system to test the general applicability of the method to a liquid having different evaporation characteristics from water.

THEORETICAL

I. The Theoretical Basis

When a gas and a volatile liquid are brought in contact, the resulting action depends on many factors, each of which may vary widely. By considering changes over small intervals, however, the actions may be separated into a number of fundamental processes. There may be formulated for these processes enthalpy balances, material balances, heat and mass transfer rate equations, and saturated vapor-temperature relations.

Sherwood and Reed (24) have derived a group of mathematical relations governing the interaction of a gas and a volatile liquid. Their derivation applies to countercurrent flow, although the procedure may be applied equally well to parallel flow. The gas enters the tower at the bottom and flows past the liquid which runs down the walls. The tower is operating in a steady state with end concentrations, rates of flow, temperatures, and the like constant at any given point in the tower. Since conditions vary from point to point along the length of the tower, a differential analysis offers a means of making a survey at any chosen spot. In most operating towers, both heat and mass transfer may be expected to take place. The dimensions of all quantities used are given in English units. The following terms are used:

- H - humidity, lb. liquid vapor/lb. dry gas
- L' - liquid rate, lb. liquid/(ft.² area) (hour)
- G' - gas rate, lb dry gas/(ft.² area) (hour)

Assume a unit cross sectional area of height dy feet, as shown in Figure 1.

For a liquid material balance, the input and output in section dy is

Input of liquid vapor with air at a:	$G'H$
Input of liquid as liquid at b:	$L' + dL'$
Output of liquid as liquid at a:	L'
Output of liquid as vapor at b:	$G' (H + dH)$

Since there is no net gain in the section, input equals output

FIGURE 1: FLOW SCHEME OUTLINE

$$G'H + L' + dL' = L + G'H + G'dH \quad (1)$$

$$\text{so that} \quad dL' = G'dh \quad (2)$$

For a differential enthalpy balance, a base temperature, t_o °F., is selected at which the enthalpy of dry gas and of liquid in liquid form are each taken as zero, the enthalpy of vapor equals the latent heat needed to vaporize it at the base temperature. Specific heats may be considered constant over small temperature ranges. Using the symbols,

$$\begin{aligned} c_g &= \text{gas heat capacity, Btu/lb. dry gas} \\ c_v &= \text{vapor heat capacity, Btu/lb. vapor} \\ c_l &= \text{liquid heat capacity, Btu/lb. liquid} \\ h_{fg} &= \text{liquid heat of vaporization at } t, \text{ Btu/lb. liquid;} \end{aligned}$$

then at any temperature t (°F.) and constant total pressure, enthalpies may be closely approximated by

$$\begin{array}{lll} \text{gas:} & c_g (t_g - t_o) & \text{Btu/lb.} \\ \text{liquid:} & c_l (t_l - t_o) & \text{Btu/lb.} \\ \text{vapor:} & c_v (t_g - t_o) + h_{fg} & \text{Btu/lb.} \end{array}$$

An enthalpy balance across a differential cross section is

Enthalpy input by gas and its associated vapor at a

$$G'h_{fg} + G' (c_g + Hc_v) (t_g - t_o)$$

Enthalpy input by liquid at b

$$(L' + dL') (t_l + dt_l - t_o)$$

Enthalpy output by gas and its associated vapor at b

$$G' (H + dH)h_{fg} + G' (c_g + (H + dH)c_v) (t_g + dt_g - t_o)$$

Enthalpy output by liquid at a

$$L'(t_l - t_o)$$

Equating input to output

$$\begin{aligned} L'dt_l = & h_{fg}G'dH + G' (c_g + Hc_v)dt_g + G'c_vdH(t_g + dt_g - t_o) \\ & - dL' (t_l + dt_l - t_o) \end{aligned} \quad (3)$$

Replacing the term $(c_g + Hc_v)$ by the term c_s representing humid heat, the heat capacity of one pound of dry gas and its associated vapor, and

neglecting the second order differentials, which are relatively small compared with the other terms

$$L'dt_1 = h_g G' dH + G' c_g dt_g + G' c_v dH(t_g - t_o) - dL' (t_1 - t_o) \quad (4)$$

to simplify this term further, $G' c_v dH(t_g - t_o)$ may be taken as negligible compared with $h_g G' dH$, and by assuming the liquid rate to be nearly constant, the term $dL'(t_1 - t_o)$ is also negligible compared with the remaining terms, so that

$$L'dt_1 = h_g G' dH + G' c_g dt_g \quad (5)$$

Whitman's two-film theory that liquid and heat transfer between phases are controlled by the surface films at the phase boundary, as well as Prandtl's earlier assumption that the heat transfer between phases is effected by the conduction through the interface area and by eddy diffusion in the turbulent area, and Colburn's subsequent theory that mass transfer between phases is also effected by pure diffusion in the turbulent area provide a basis for believing that heat and material film transfer coefficients control rates of flow.

Since surface area in packed towers is not easily measured, a quantity a is designated as the surface area per unit volume of the tower. For the coefficient of heat transfer through the gas film, h_g (Btu/hr.ft.² of.) is used. Then in height dy of the tower, the heat transferred across the gas film must equal the sensible heat change of the gas stream in the same section; that is for unit cross section where $dV = dy$, and where t_1 equals liquid temperature at the interface,

$$h_g a (t_1 - t_g) dy = G' c_g dt_g \quad (6)$$

Where h_l is the coefficient of heat transfer through the liquid film, similarly

$$h_l a (t_1 - t_g) dy = -L dt_1 \quad (7)$$

For mass transfer the rate through the gas film is controlling since in a pure

liquid no concentration differences can be built up at points in the liquid. The rate of mass transfer is proportional to the difference between the partial pressure of the liquid vapor at the interface and that in the main gas stream. Over small ranges, assuming humidities proportional to partial pressures, with k (lb./hr.ft². (unit H)) as the mass transfer coefficient of the gas film, there is given per unit of volume

$$ka(H_1 - H)dy = G'dH \quad (8)$$

where H_1 is the humidity of the saturated gas at interface liquid temperature.

The humidity and p_1 , the vapor pressure at the interface, are easily related where P_A is the total atmospheric pressure and M_g and M_l are molecular weights of gas and liquid, p_1 may be found from the vapor pressure tables at various values of t_1 , then by definition

$$H = \frac{M_l p_1}{M_g (P_A - p_1)} \quad (\text{lb. liq./lb. dry gas}) \quad (9)$$

Six relationships have now been established governing the interaction:

$$dL' = G'dH \quad (2)$$

$$L'dt_1 = G'h_{fg}dH + G'e_s dt_g \quad (5)$$

$$h_g a(t_1 - t_g)dy = G'e_s dt_g \quad (6)$$

$$h_l a(t_1 - t_l)dy = -L'dt_1 \quad (7)$$

$$k'a(H_1 - H)dy = G'dH \quad (8)$$

$$H = \frac{M_l p_1}{M_g (P_A - p_1)} \quad (9)$$

Barnet and Kobe (12) have found evidence that the liquid film coefficient is so high that temperature gradient across the film is negligible. Most investigations have shown the gas film to be controlling for both heat and mass transfer in an apparatus of this type. Hence, equation (7) may be deleted as t_1 substantially equals t_l . Taking the saturation humidity at the average mean stream temperature of the water, there is found on dividing equation (8) by

equation (6)

$$\frac{dH}{dt_g} = \frac{ke_g(H_s - H)}{h(t_1 - t_g)} \quad (10)$$

In the case of the air-water system, (ke_g/h) has been found to be very close to unity and may be taken as such for the operating conditions of the thesis. Then for the air-water system, equation (10) becomes

$$\frac{dH}{dt_g} = \frac{H_s - H}{t_1 - t_g} \quad (11)$$

In the literature (25) the value of 1.22 is given for (h/ke_g) for the air-methanol system and for the air-methanol system equation (10) becomes

$$\frac{dH}{dt_g} = .82 \frac{H_s - H}{t_1 - t_g} \quad (12)$$

Having established the path of the humidity-temperature line, it is necessary to ascertain its end point to know the final vapor-air mixture condition. The enthalpy level of the starting vapor-air mixture may be calculated. In an adiabatic system, the enthalpy change of the liquid system must be compensated for by an equal change in the gas system. As the enthalpy change of the liquid is readily approximated, the end enthalpy of the gas system may be calculated, and its intersection with the humidity-temperature line of equation (12) will indicate final gas conditions. Figure 1 shows a material and enthalpy balance outline for both countercurrent and parallel flow. The derivation following is for countercurrent flow but may be applied equally well to parallel flow.

Using Figure 1, the material balance is

$$L_1 + GH_2 = L_2 + GH_1 \quad (13)$$

and the enthalpy balance is

$$L_1 (t_{11} - t_{10})c_1 + G_1 i_2 = L_2 (t_{12} - t_{10})c_1 + G_1 i_1 \quad (14)$$

where c_1 is liquid specific heat, i represents enthalpy, and t_{10} is the liquid

base temperature. Then placing equation (13) in equation (14)

$$(L_2 + G(H_1 - H_2)) (t_{11} - t_{10}) c_1 + G i_2$$

$$L_2(t_{12} - t_{10}) c_1 + G i_1 \quad (15)$$

Factoring terms

$$G(i_1 - i_2) = L_2(t_{11} - t_{12}) c_1 + G c_1 (H_1 - H_2) (t_{11} - t_{10})$$

or

$$i_1 = i_2 + \frac{L_2 c_1}{G} (t_{11} - t_{12}) + c_1 (H_1 - H_2) (t_{11} - t_{10}) \quad (16)$$

By substituting equation (13) again for the last group and simplifying

$$i_1 = i_2 + \frac{L_2 c_1}{G} (t_{10} - t_{12}) + \frac{L_1 c_1}{G} (t_{11} - t_{12}) \quad (17)$$

Except for extremely low liquid-gas ratios, the amount of liquid evaporated is only a small percentage of that passing through the apparatus so that L_1 and L_2 are substantially equal and

$$i_1 = i_2 + \frac{L_0 c_1}{G} (t_{11} - t_{12}) \quad (18)$$

Using the same symbols, it may be shown for parallel flow

$$i_2 = i_1 + \frac{L_0 c_1}{G} (t_{11} - t_{12}) \quad (19)$$

Equations (18) and (19) state that the enthalpy change of the vapor mixture in the system will just compensate for the enthalpy change of the liquid in the system. In other words, an increase of $\underline{x}$ Btu's in the liquid system will be accompanied by a decrease of $\underline{x}$ Btu's in the vapor system. This assumes, obviously, that because of the relatively large amount of materials flowing, the quantity of heat transferred from the surroundings will be negligible and that the system is considered approximately adiabatic. It also assumes that the liquid-gas ratio will be constant; however, since it has already been specified for this thesis that the liquid-gas ratio is not very small, the amount of liquid evaporated will be only a minor part of the liquid present and the L/G ratio

will be substantially constant, thereby giving very nearly linear relationship. Thus the assumption is made that the heat of vaporization of the liquid and the temperature change of the liquid are the controlling enthalpy changes of the system. This heat of vaporization varies widely from the comparatively high value for water to values about a sixth as large for many common organic liquids. No comprehensive quantitative statement may be made since different liquids have different degrees of volatility as well as different heats of vaporization.

Recapitulating, the two equations

$$\frac{dH}{dt_g} = \frac{ko_s(H_s - H)}{h(t_1 - t_g)} \quad (10)$$

and

$$i_1 = i_2 + (L_o/g) (t_{11} - t_{12}) \quad (18)$$

may be used together to obtain a graphical solution of the interaction of a gas and a volatile liquid if the above assumptions are proper.

II. The Psychometric Chart

The most convenient form of chart devised to date for showing liquid vapor-gas relationships is the temperature-humidity-enthalpy (THE) chart. Against the temperature as the abscissa, the humidity and enthalpy may be plotted as ordinates. With either of the latter as the vertical coordinate, values for the other will fall as slanting lines. When humidity is plotted as the vertical coordinate, changes of humidity with temperature may be readily graphed.

These charts are termed psychometric charts and as the name implies, the condition of a gas mixture is deduced from the associated physical phenomena. A variety of physical measurements are often recorded on psychometric charts: the line of saturation humidity, specific gas volume, humid gas volume, humid heat, latent heat of vaporization, and adiabatic cooling lines. The saturation

line is that showing the humidity of air holding all the water it can at various temperatures. While empirical equations have been derived for this line, they are involved and clumsy to use, and the best source of saturation data is the steam tables (14, 15) for air-water, and various handbooks or tables (29) for other liquid-gas combinations. When air and water are left in contact in an adiabatic system, the air will follow a definite temperature path as it becomes humidified. This predictable path, dependant only on initial air and water conditions, is called an adiabatic humidification line. Goodman (26) employs the following relation to locate end-points for these lines.

$$c_s(t_g - t_{gs}) = h_{fgs}(H_s - H) \quad (20)$$

where the s subscript terms refer to saturation conditions. A base temperature, t_o , must be chosen to compute enthalpy values; whenever possible, it should be selected so that phenomena, such as freezing, do not enter to complicate the calculations. The enthalpy of the liquid is zero at t_o and that of the liquid vapor is h_{fgo} , where h_{fgo} is the latent heat of vaporization (Btu/lb.) at t_o ; the enthalpy of the dry gas is also zero if the same base temperature is used for the gas. Values of h_{fgo} and vapor pressure, p , may be found in the literature. By substituting p for p_1 in equation (9), the humidities for any vapor pressure conditions may be found. The saturation line is located by plotting either the saturation humidity or saturation enthalpy points versus temperature on the graph. Since the enthalpy of a gas-liquid vapor mixture is the sum of the enthalpies of each plus a negligible heat of mixing, the value for any mixture can be calculated:

$$\begin{aligned} i &= i_g + i_v = c_g(t_g - t_o) + Hh_{fgo} + Hc_v(t_g - t_o) \\ i &= Hh_{fgo} + (c_g + Hc_v)(t_g - t_o) \end{aligned} \quad (21)$$

For the construction of the air-water psychometric chart, both air and water vapor are assumed to behave like perfect gases and the specific heats are taken as constants. At high temperatures or high pressures, these assumptions might lead to an appreciable amount of error, for both air and water vapor vary widely from the perfect gas laws. The specific heats of air and of water vapor also vary slightly with temperature. For this thesis, however, the chart is to be constructed for atmospheric pressure conditions and in a relatively low temperature range, so that calculations based on these assumptions will be well within the limits of accuracy of the humidity measurements made. Table I lists data for the air-water system, taken directly from the psychometric tables. A base temperature of 32 °F. is used for air and water, corresponding to the practice in the Keenan and Keyes tables (14) and to that in the Zimmerman and Lavine tables (15). This base temperature makes calculations a little more complicated, for observed temperatures must have thirty two degrees deducted before they can be used in equations. This base, however, avoids the even greater complications that would follow if the water-freezing point heat changes had to be considered in the computations.

Taking latent heat and specific heat values from tables in standard reference books (29, 30):

Enthalpy of air at t_0	0	Btu/lb.
Enthalpy of water at t_0	0	Btu/lb.
Enthalpy of water vapor at t_0	1075.1	Btu/lb.
c_a , specific heat of air	.24	Btu/lb. °F
c_v , specific heat of water vapor	.45	Btu/lb. °F

Enthalpy of dry air at t °F.

$$i_g = .24(t - 32) \quad \text{Btu/lb.}$$

FIGURE 2: T-H-E CHART FOR AIR - WATER SYSTEM

TABLE I: PROPERTIES OF WATER (14, 15, 27)

Temperature °F.	Latent Heat of Vaporization Btu/lb. water	Saturation Humidity lb. water lb. dry air	Enthalpy of Saturated Mixture Btu/lb. dry air
40	1071	.0052	7.54
50	1065	.0077	12.68
60	1059	.0111	18.77
70	1054	.0158	26.39
80	1048	.0223	35.99
90	1043	.0312	48.21
100	1037	.0431	63.98
110	1031	.0593	84.54
120	1025	.0813	111.65
130	1020	.1113	147.99
140	1014	.1530	197.59

TABLE I: PROPERTIES OF WATER (cont.)

I. Location of constant humidity lines

For a given humidity, by locating two temperature enthalpy points, the constant humidity line may be drawn.

Humidity	Humid Heat	Enthalpy at 60°	Enthalpy at 100°
<u>lb. water</u>	<u>Btu</u>	<u>Btu</u>	<u>Btu</u>
<u>lb. dry air</u>	<u>lb. dry air °F</u>	<u>lb. dry air</u>	<u>lb. dry air</u>
0	.2400	6.73	16.32
.01	.2445	17.60	27.38
.02	.2490	28.48	38.42
.03	.2535	39.36	49.47
.04	.2580	50.23	60.53
.05	.2625	61.11	71.59
.06	.2670	71.99	82.65
.07	.2715	82.87	93.69
.08	.2760	93.74	104.74
.09	.2805	104.62	115.79
.10	.2850	115.50	126.84

II. Location of adiabatic saturation lines

By establishing two temperature-humidity points, one on the saturation line and the other found at some selected humidity level, adiabatic saturation lines can be drawn.

Saturation Temperature	Saturation Humidity	Arbitrary Humidity	Corresponding Temperature
<u>°F</u>	<u>lb. water</u>	<u>lb. water</u>	<u>°F</u>
	<u>lb. dry air</u>	<u>lb. dry air</u>	
40	.0052	0	63.2
50	.0077	0	84.2
60	.0111	.005	87.0
70	.0158	.010	95.0
80	.0223	.020	89.7
90	.0312	.025	115.5
100	.0431	.040	112.4
110	.0593	.055	126.8
120	.0813	.080	124.8

Enthalpy of water vapor at $t^{\circ} \text{F}$

$$i_v = 1075.1 + .45(t - 32) \quad \text{Btu/lb.}$$

And the enthalpy of an air-vapor mixture containing H pounds of water per pound of dry air, at some temperature t above 32°F . is found by adding together the expressions above for the dry air enthalpy and for the enthalpy of the associated vapor, so that, in Btu's per pound of dry air in the mixture,

$$i = 1075.1H + (.24 + .45H)(t - 32) \quad (22)$$

Where atmospheric humidity is to be determined from wet and dry bulb readings, a plot of temperature versus humidity is the most convenient form. For making the calculations of this thesis, it is necessary to include temperature, humidity, enthalpy, saturation, and adiabatic cooling line information on the graph. It is generally most convenient to plot temperatures as abscissas and enthalpy as ordinates. Lines of constant humidity may be plotted between desired temperatures by finding the enthalpy from substitution of the given temperature and humidity in equation (22). The saturation line is plotted directly from steam table data. In plotting adiabatic cooling lines, any temperature, t , is selected; one end of the cooling line is located where the vertical line representing this temperature t , intersects the saturation curve at a point (t, H_s) . Substituting this value and some other chosen value of either t or H in equation (20) gives the point (t, H) which is the other end of the cooling line.

When a wet bulb is inserted in a gas stream, an equilibrium condition is soon reached wherein the amount of water which evaporates in a unit time uses heat just equal to that transferred to the wet bulb from the gas stream. In this state, the wet bulb is at a temperature known as the wet bulb temperature, t_w . The thermometer will record this temperature as long as the wick

covering the bulb is moist. In addition to the heat reaching the thermometer by conduction and convection, allowance must be made for a small amount of heat transferred by radiation. For the range of gas velocities covered in this thesis, an increase of only two or three percent is caused by radiation, so that it may be neglected in this case (25).

Thus where

- q - heat flow per unit time
- t_w - wet bulb temperature
- p_w - water vapor pressure at t_w
- h_{fgw} - heat of vaporization at t_w
- p - water vapor pressure in the gas stream
- A - wet bulb surface area

Letting heat transferred to the wet bulb equal heat lost by evaporation of liquid on the wet bulb

$$k'A h_{fgw} (p_w - p) = hA (t_g - t_w)$$

or

$$\frac{p_w - p}{t_g - t_w} = \frac{h}{k' h_{fgw}} \quad (23)$$

This form is not commonly used in humidity calculations. Since the vapor pressure is usually lower by far than the total pressure, humidity may be shown to equal approximately $(M_1 p / M_g)$ and the mass transfer coefficient k may be used, where k equals $(M_g k' / M_1)$. If the identity (c_s / c_g) is introduced

$$\frac{H_w - H}{t_g - t_w} = \frac{h c_s}{k h_{fgw} c_g} \quad (24)$$

It has been found that $(h / k c_s)$ for air-water nearly equals unity for the range involved (25). Hence the equation becomes

$$\frac{H_w - H}{t_g - t_w} = \frac{c_s}{h_{fgw}} \quad (25)$$

which is identical with equation (20) for the adiabatic cooling line. In other words, the wet bulb lines and the adiabatic cooling lines nearly coincide. In the case of liquids other than water and gases other than air,

(h/k_{e_g}) generally does not equal unity and the wet bulb and adiabatic cooling lines do not coincide. A psychometric chart for air-water at normal barometric pressure and ordinary temperatures is plotted as Figure 2. Table I lists values of properties plotted on this chart.

IV. Construction of the Air-Methanol TBE Chart

Values of the vapor pressure of methanol are available in various chemical handbooks and tables (29). From these, humidity values for air saturated with methanol vapor may be calculated by use of equation (9)

$$H = 1.105 \frac{p}{(760-p)} \quad (9a)$$

where

H - humidity, lb CH_3OH / lb. air
 p - methanol vapor pressure, mm. Hg.
 1.105 - molecular weight ratio, $\text{CH}_3\text{OH}/\text{air}$

For a base temperature for enthalpy calculations, 0 °F. is very convenient. As an average value for specific heat of methanol vapor, 0.38 (Btu/lb.) may be used for the temperature range considered and 519 (Btu/lb.) may be used for h_{fgo} (29). Enthalpy values for air-methanol mixtures of humidity, H , at temperature, t , are computed in the same manner as for air-water vapor; for the methanol mixture:

$$i = 519H + (.24 + .38H)(t - 0) \quad (26)$$

As before, constant humidity lines may be plotted on the chart by locating enthalpy values for the given humidity at some two temperatures. By using the vapor pressure values of methanol and equations (9a) and (27), the saturation line may be found. Wet bulb lines must be plotted, however, for in this system they do not coincide with the adiabatic saturation lines. The adiabatic saturation lines are found, as before, by substituting in equation (20). For the wet bulb lines, equation (25) holds as in the case for the air-

water system, but for the air-methanol system under the conditions encountered here, (h/kc_g) has been found to equal 1.22 (25). The slope of the wet bulb lines, therefore, equals 1.22 times that of the adiabatic saturation lines. Adiabatic saturation lines may be plotted, as usual, from points along the saturation lines; increasing the amount of slope 1.22 time gives the wet bulb lines from the same saturation line starting points. By using these wet bulb lines, humidity and other readings may be made directly from the TNE chart, just as was done for the air-water case. Figure 3 is a TNE chart for the air-methanol system; values used in plotting are listed in Table II.

V. Graphical Evaluation of Transfer Coefficients

In finding the coefficients of heat transfer and of mass transfer, the respective gas film coefficients h and k are assumed to be the limiting factors, and hence are used as the overall transfer coefficients. The literature generally supports this position and Barnett and Kobe (11) checked it experimentally. The following equations are the basis for the graphical derivation of transfer coefficients:

$$G_o dt = h(t_g - t_1) dA \quad (27)$$

and

$$G dH = k(H_s - H) dA \quad (28)$$

Then over the range of gas temperature and gas humidity, respectively, by integrating the terms

$$\frac{dt}{(t_g - t_1)} = s_1 \quad \text{and} \quad \frac{dH}{(H_s - H)} = s_2$$

the coefficients may be found by letting

$$h = \frac{G_o s_1}{A} \quad \text{and} \quad k = \frac{G s_2}{A}$$

The integrals are calculated by plotting $1/(t_g - t_1)$ versus t , and $1/(H_g - H)$ versus H , respectively, and by measuring the areas under the curves and equating them to s_1 and to s_2 respectively. Substituting the s 's in the equations above gives the coefficients sought. The value of (h/kc_g) may be compared with values available in the literature.

FIGURE 3: T-H-E CHART FOR AIR-METHANOL SYSTEM

TABLE II: PROPERTIES OF METHANOL (29,30)

Temp. OF	Vapor Pressure mm.Hg.	Latent Heat Vaporization Btu/lb.water	Saturation Humidity lb.water lb.dry air	Enthalpy of Saturated Mixture Btu/lb.dry air
20	17.5	514.8	.0250	17.93
30	26.3	512.5	.0397	28.20
40	40.8	510.1	.0628	43.05
50	53.8	508.0	.0845	57.40
60	76.0	505.8	.123	81.00
70	100.8	503.2	.169	107.9
80	134.5	500.0	.238	150.2
90	180	496.3	.343	210.6
100	234.5	493.1	.493	297.0
110	301	489.4	.724	429.4
120	382.5	485.8	1.12	656.

TABLE II: PROPERTIES OF METHANOL (cont.)

I. Location of constant humidity lines

For a given humidity, by locating two temperature-enthalpy points, the constant humidity line may be plotted.

Humidity	Humid Heat	Enthalpy at 50°	Enthalpy at 100°
lb. CH ₃ OH	Btu	Btu	Btu
lb. dry air	lb. dry air °F	lb. dry air	lb. dry air
0	.240	12.0	24.0
.05	.259	38.8	51.9
.10	.278	65.8	79.7
.20	.316	119.7	135.5
.30	.354	173.5	191.2
.40	.392	227.3	246.9
.50	.430	280.7	302.2
.60	.468	335.0	358.4
.70	.506	388.8	414.1
.80	.544	442.6	469.8
.90	.582	496.5	525.6
1.00	.620	550.3	581.3

II. Location of wet bulb lines

By locating two temperature-humidity points, one located on the saturation line and the other found at some arbitrary humidity level, the wet bulb lines may be drawn.

Saturation Temperature	Saturation Humidity	Arbitrary Humidity	Corresponding Temperature
°F.	lb. CH ₃ OH lb. dry air	lb. CH ₃ OH lb. dry air	°F.
20	.0250	0	62.9
30	.0397	.025	54.1
40	.0628	.050	60.2
50	.0860	.070	74.3
60	.123	.100	93.0
70	.169	.150	96.7
80	.238	.220	100.
90	.343	.320	115.
100	.493	.480	112.

EXPERIMENTAL

I. General Considerations

Following the example of other experiments in wetted-wall tower work, the apparatus was designed to use a wetted-wall section five and one half feet long in either parallel or countercurrent flow. Gravity flow provided adequate means for moving the liquid, and a pump was arranged to move the gas. The apparatus was ten feet long and twelve feet high overall. Air was used as the gas in all cases, and it underwent no preliminary treatment other than heating in some instances. For the runs with water, ordinary city water was used, since the possible accuracy of humidity measurements was far less than any error which might have been caused by impurities in the water. A commercial grade of methanol was used for part of the experiments. Ordinary iron piping was used in the apparatus, and no attempt was made to treat the inside surfaces other than to clean them before assembling the apparatus. The arrangement of the apparatus for countercurrent flow is shown in Figure 4, and for parallel flow, in Figure 5. Figures 6, 7, and 8 show details of three important sections of the apparatus.

II. The Apparatus

The most involved section of the wetted-wall apparatus was the air supply device. To obtain a non-pulsating flow of air, a small blower actuated by a constant speed electric motor was used. Air output was regulated by means of an outlet valve and a bypass around the blower. These arrangements gave a range of flow values in the lower turbulent range. An intake extension permitted drawing air either from the laboratory or from outdoors as desired. The air passed, for the most part, through one inch uninsulated iron pipes.

The liquid supply originated in a five gallon reservoir at the top of the apparatus. This reservoir was fitted with an overflow pipe to maintain

PRINCIPAL DIMENSIONS IN APPENDIX

FIGURE 4: SCHEMATIC DIAGRAM
OF COUNTERCURRENT FLOW APPARATUS

constant liquid level. Flow of liquid was regulated by a valve in the pipe connecting the reservoir and the wetted-wall section of the apparatus. For the case when water was used, the water was piped directly into the reservoir from the city water system. Water input was set to keep a constant overflow, and hence, constant water level. In the instances where methanol was the liquid used, it was not practical to utilize the overflow without setting up an elaborate recirculating device, so liquid flow measurements were made over the whole period of a run to obtain an accurate indication of the flow. For the moderate flow region with which this thesis is concerned, the decrease in head for a single run was very small and the variation in flow rate negligible.

One of the major problems of the thesis was to bring together and to separate the liquid and gas at the desired points. Assemblies housed in two-and-one half inch pipe couplings at each end of the wetted-wall section performed this service. (See Figures 6 and 7) In the upper coupling, a small piece of pipe extending up from the bottom had eighth inch holes drilled at small intervals completely around the circumference. These holes impeded liquid flow until a small head was built up, and hence they insured liquid flow on all sides of the pipe. Inside the lower coupling a one-half inch pipe nozzle extended from the bottom almost to the lower end of the wetted-wall. Liquid flowing down the walls of the tower passed into the coupling through the gap between the pipe wall and the nozzle, while the gas could pass either into or out of the nozzle, depending on the flow direction. A liquid outlet pipe extending from the side of the coupling had a two inch deep U-bend to prevent gas flow from this exit.

For the most essential part of the apparatus, the wetted wall, one-inch iron pipe was first tried. The heat exchange between this and the surround-

FIGURE 5: SCHEMATIC DIAGRAM
OF PARALLEL FLOW APPARATUS

ings and the impossibility of determining whether the pipe was completely covered with liquid made this arrangement unsatisfactory and a section of one-inch Pyrex tubing was substituted. While at first, liquid did not cover the whole tubing surface, a thin scale was deposited on the inside of the tube from the hard water and without interfering with the visibility, aided in securing a complete liquid film, even at very low rates of liquid flow. Regular one inch pipe couplings were attached to the separatory couplings just described and the glass tube ends fitted into these. The ends of the tube were taped to prevent cracking and to prevent heat conduction to the rest of the apparatus. Putty and other fillers failed to prevent leaks in these tube-coupling joints, particularly under gas flow conditions; rubber gaskets, however proved satisfactory, completely stopping leaks under all conditions.

The orifice was fitted into the air pipe between the blower and the wetted-wall section. A stainless steel plate with a center hole beveled on the downstream side was fitted into a pipe union to make the orifice, as shown in Figure 8. Taps were located at suitable distances up and down stream as recommended in the literature. (25) A straight section of pipe about one meter long preceded the orifice to insure uniform flow in the orifice section. Leads from the orifice were attached to a manometer consisting of a glass U-tube filled with water. The entire union and taps area was coated with paraffin to prevent leaks. A steam jacket about a meter long on the air flow pipe permitted air heating; also used at times was an electric heating plate placed directly under the air pipe. This latter unit allowed closer air temperature control than the steam jacket. To heat the liquid, a steam coil was used in the liquid reservoir together with a small stirrer to promote uniform temperature.

III. Measurements

For the measurement of liquid temperature, -10° to 110° C. thermometers graduated in degrees were used. The liquid temperature fluctuated but little over short periods and temperatures could be estimated to about two-tenths degree. A check with more accurate thermometers showed that the results obtained with the test thermometers were completely satisfactory.

Entering liquid temperature was measured in the reservoir just above the top of the pipe leading to the wetted-wall section. This pipe, together with the coupling at the top of the wetted-wall section, as encased in one-inch felt insulation to insure little temperature change from that measured. From the coupling at the lower end of the wetted-wall section extended a liquid outlet pipe with a U-shaped liquid trap into which a thermometer was inserted to measure liquid outlet temperatures. This coupling and the adjacent pipe also were encased in felt insulation.

For the measurement of air temperature, thermometers of the range -10° to 110° C. graduated in one-tenth degrees were used. Entering air temperatures were measured through holes in the air pipe just outside the wetted-wall section, as were wet bulb temperatures. A thermocouple encased in cotton was tried for wet bulb measurements: it gave identical results to those of the thermometers but was sluggish in reaching equilibrium temperature, more erratic, and remained at wet-bulb temperature for considerably less time than did the thermometers. The bulbs of the thermometers used in measuring air were about one-half centimeter in diameter by two centimeters long so that they extended across nearly the whole diameter of the air flow pipe. This undoubtedly aided in getting a representative reading since the air temperature probably varied across the diameter of the pipe. For the wet bulb readings, a small jacket was made from a double thickness of thin cotton of

2-1/2 INCH PIPE COUPLING

ONE INCH PIPE

LIQUID INLET

1/8 INCH HOLES

TO WETTED WALL SECTION

FIGURE 6: SCHEMATIC DIAGRAM
OF UPPER COUPLING

FIGURE 7: SCHEMATIC. DIAGRAM OF LOWER COUPLING

a moderately loose elastic weave. This jacket fitted snugly on the thermometer bulb and held a good quantity of liquid. A steady wet bulb temperature could often be maintained for up to three minutes with these jackets. For the wet bulb readings, the temperature was found by two or three successive readings which indicated the approximate equilibrium temperature. The final equilibrium point was determined by inserting the thermometer at about one half degree above and at one half degree below the equilibrium temperature thus approaching the true value from both sides. On some runs where the temperature oscillated somewhat, check readings were taken.

Air flow was measured by means of the orifice and its attached manometer. Runs were discontinued in the ranges where an increase in either the liquid or the air flow caused pulsations in the manometer reading. Liquid flow was found by volumetric measurement of the liquid leaving the apparatus. Liquid was collected over as much of the run as possible to minimize fluctuation effects in the flow rate.

IV. Experimental Procedure.

The specific gravity of the alcohol was checked with a Westphal balance. Daily measurements of the atmospheric pressure showed an average of 753 millimeters plus or minus two millimeters so that all calculations are based on normal atmospheric pressure, the slight deviation being outside the limits of accuracy of the measurements made. For all liquid and gas temperature readings, at least one-half hour was allowed for the apparatus to come to equilibrium, and at least fifteen minutes for each small change in either the liquid or the gas rate. All thermometers were calibrated against one of the large one-tenth degree thermometers. Since the purpose of the thesis was to test the reliability of measurements and subsequent graphical analysis for a

non-adiabatic system, insulation was used only for the conduits at the ends of the wetted-wall section where it was not practical to measure the temperature of the liquid directly. One-inch felt covering was used for these intermediate sections.

V. Graphical Calculation of Results

For the graphical calculation of results, the following data are needed for both the top and the bottom of the wetted wall tower: wet and dry bulb temperatures, liquid temperatures, liquid and gas flow rates, and calculated humidities for the entering and exit gas.

Letting the subscript 1 refer to the column top and 2 to the column bottom and using the nomenclature of Figure 1, a skeleton TBE chart is drawn to include t_{11} and t_{12} , t_{g1} and t_{g2} and the saturation curve between t_{11} and t_{12} . Using a countercurrent example, this is shown in Figure 9-a.

The exit and entrance gas enthalpy levels are next calculated by an equation such as equation (22) or (26),

$$i = 519H + (.24 + .38H) t \quad (26)$$

The exit enthalpy level is found by assuming that the change in the gas mixture enthalpy will just equal the measured change in the liquid enthalpy; that is

$$i_1 = i_2 + (L_1/G) (t_{11} - t_{12}) \quad (18)$$

The addition of these lines is shown in Figure 9-b.

The preliminary liquid and gas operating lines, $(t_{11}i_1, t_{12}i_2)$ and $(t_{g1}i_1, t_{g2}i_2)$, are next drawn. Also added is a set of parallel horizontal lines between the operating lines, as in Figure 9-c.

Sections of the liquid operating line, AL, and the gas operating lines, A"G, are shown in Figure 9-d. A vertical line is drawn from A to intersect

FIGURE 8: SCHEMATIC DIAGRAM OF ORIFICE ASSEMBLY

the saturation line at A' . Where the line $A'-A''$ intersects the horizontal line through B determines the location of B'' . A vertical line from B establishes B' on the saturation line. Line $B'-B''$ locates point C'' on the horizontal line through C , and this process is repeated until the actual air operating line $A''B''C''$... reaches the enthalpy level i_2 . This point represents the final predicted humidity and temperature of the gas mixture.

One modification of this last step must be made for combinations other than water-air. Where the (k_o/h) value does not equal unity, the vertical lengths AA' , BB' , ... are multiplied by the (k_o/h) value to give lengths AA''' , B''' , ... as in Figure 9-d. This multiplying factor is 0.82 for the methanol-air system so that AA''' equals $.82AA'$, BB''' equals $.82BB'$, and so on. The gas operating line then falls along the lines $A''A'''$, $B''B'''$, and the same procedure is used as before.

Two typical sets of operating lines are shown in Figure 9-e for parallel flow and Figure 9-f for countercurrent flow.

FIGURE 9: STEPS IN GRAPHICAL ANALYSIS

FIGURE 9: (CONTINUED)

VI a. The Effect of Liquid-Gas Ratio on Vapor Enthalpy Change

Since the setting of the final vapor enthalpy level is based on the change in enthalpy of the liquid in the system, it is of considerable interest to determine how close the measured vapor enthalpy change approached the measured liquid enthalpy change. In this investigation, the inlet and outlet liquid temperatures are measured and from this the theoretical change in liquid enthalpy is calculated for a definite time. The actual vapor enthalpy change is calculated from measurements of the inlet and outlet gas temperatures and humidities. In an adiabatic example, there would be no net enthalpy changes; in these examples, the change should be relatively small if the graphical analysis is to succeed.

The countercurrent runs proved markedly better than the parallel flow runs for both water and for methanol. For both liquids, the averages of the countercurrent run theoretical vapor enthalpy changes averaged within five percent of those actually measured, although individual runs varied over twenty five percent from the measured values. The theoretical values for parallel flow varied over thirty percent from the measured values, on the average, for methanol and for water. As might be expected, the runs having the very low liquid-to-gas ratios gave the most unsatisfactory results, the measured changes being several times larger than the calculated changes for parallel flow. Results are listed in Tables III and IV and graphed in Figures 10 and 11.

FIGURE 10: WATER-AIR RATIO INFLUENCE ON GAS ENTHALPY CHANGE

FIGURE 11: METHANOL-AIR RATIO INFLUENCE ON GAS ENTHALPY CHANGE

b. Statistical Analysis of the Preceding Section

In order to determine the probability of obtaining a measured vapor enthalpy change approximately equal to the theoretical enthalpy change, a statistical analysis of the data was made by the usual methods, (see Appendix m). This analysis presumed that the procedure and equipment used in this thesis are typical for this type of investigation, and that under ideal conditions the measured enthalpy change would just equal the theoretical enthalpy change. If an infinite number of sets of runs similar to those in this thesis were made, because of random errors their average measured enthalpy values would deviate from the theoretical values in a pattern shown by a probability that some one set of runs will vary by a given amount from the ideal value. It is indicated in Table V that substantially no likelihood exists that the measured enthalpy change will fall within 10% of the theoretical value for either water or methanol in parallel flow. For countercurrent flow, however, it is evident that nearly all the water runs and four fifths of the alcohol runs will have enthalpy changes falling within 5% of the theoretical value. This result is based on the assumption that the various data are employed without corrections for radiation and velocity changes.

c. Graphical Evaluation of Tower Performance

From the runs listed in Tables III and IV, twelve of the best were selected to find whether the end temperature and humidity conditions may be predicted by graphical means if data from very good runs is available. While it was desired to select the runs with the best theoretical to measured enthalpy change ratios, it was necessary to modify this somewhat so as to get a wider variety of Reynolds numbers and liquid to gas ratios. With one or two exceptions, the runs taken may be considered better than average. Assuming a vapor enthalpy change equal to that undergone by the liquid

TABLE III: A COMPARISON OF THERORETICAL AND MEASURED
VAPOR ENTHALPY CHANGES

Theoretical Values Computed From Liquid Temperature Changes

a. Parallel Air-Water Flow

Run No.	Reynolds Number	<u>Liquid</u> <u>Gas</u>	<u>Theoretical</u> <u>Measured</u>
1	9720	.316	.92
2	8480	.454	1.61
3	9260	.643	1.12
4	8100	.740	1.63
5	6760	.883	1.76
6	9620	.960	1.38
7	5220	1.14	1.00
8	9720	1.42	1.39
9	9020	1.63	1.10
10	8000	1.93	1.13
11	6950	2.21	1.29
12	5800	2.65	1.54

TABLE III (cont.)

b. Countercurrent Air-Water Flow

Run No.	Reynolds Number	<u>Liquid</u> <u>Gas</u>	<u>Theoretical</u> <u>Measured</u>
13	9960	.151	1.02
14	8580	.218	.93
15	9520	.328	1.05
16	6040	.403	1.01
17	8670	.516	.94
18	6630	.587	.98
19	6280	.700	.86
20	7290	.766	.95
21	7450	.808	1.00
22	7070	.910	.94
23	7920	.990	.78
24	8630	1.40	.87
25	7450	2.48	.92

TABLE IV: A COMPARISON OF THEORETICAL AND MEASURED

VAPOR ENTHALPY CHANGES

Theoretical Values Computed From Liquid Temperature Changes

a. Parallel Air-Methanol Flow

Run No.	Reynolds Number	<u>Liquid</u> <u>Gas</u>	<u>Theoretical</u> <u>Measured</u>
26	4810	.104	.279
27	10400	.115	.356
28	8220	.156	.478
29	8790	.159	.484
30	10500	.217	.485
31	10450	.364	.448
32	7230	.380	.460
33	7070	.410	.673
34	10650	.420	.645
35	8620	.435	.693
36	7100	.442	.572
37	4680	.640	.698
38	10700	.650	.742
39	6950	.872	.753
40	8020	.980	.771
41	4900	.990	.875
42	10950	1.12	1.02
43	7920	1.13	1.21
44	5050	1.29	.860
45	4490	1.74	1.09

TABLE IV (cont.)

b. Countercurrent Air-Methanol Flow

Run No.	Reynolds Number	<u>Liquid</u> <u>Gas</u>	<u>Theoretical</u> <u>Measured</u>
46	6520	.177	.710
47	9150	.185	1.03
48	6520	.221	.618
49	8050	.282	1.17
50	7310	.422	1.13
51	6710	.468	.695
52	7170	.538	1.39
53	6520	.618	.745
54	6560	.718	.788
55	7110	.785	1.00
56	7900	.885	1.10
57	5080	1.07	1.09

TABLE V: PROBABILITY FOR APPROXIMATELY ADIABATIC OPERATION

It is assumed that the apparatus and procedure employed are typical for the method investigated. If an infinite number of sets of runs similar to those of this thesis are made, the following table shows the probability that a run selected at random would have a (theoretical/measured) enthalpy ratio (T/M) within the limits shown. The probability is based on the data of the sets of runs listed in Tables III and IV. The letter P denotes parallel flow runs and CC denotes countercurrent flow runs.

Liquid	Flow	Number of Runs	T/M Ratio	Probability
When (t/M) equals $1.0 \pm 10\%$				
Water	P	12	1.35	0.05
"	CC	13	.95	99.0
Methanol	P	20	.68	0.00
"	CC	12	.96	79.0
When (T/M) equals $1.0 \pm 5\%$				
Water	P	12	1.35	0.00
"	CC	13	.95	50.0
Methanol	P	20	.68	0.00
"	CC	12	.96	46.0

involved at the same time, is the basis for the theoretical end enthalpy used in graphing. A graphical analysis of each of the runs was then made by the procedure outlined in the theoretical section. In Table VI, the predicted temperature and humidity changes indicated by the graphical analysis are listed, along with the temperatures and humidities actually measured. The ratio of predicted changes to measured changes are listed also in the table. A similar arrangement for enthalpy is included for comparison; these values are the same as those listed in Tables III and IV. Figures 12, 13, 14, and 15 show the graphical analysis of the four problems.

d. Derivation of h/k Ratios by the Graphical Method

Values of (h/k) and (h/ko_g) were calculated for one run of each of the four sets of data. The runs chosen were those from Table VI in which the graphically predicted temperature, humidity, and enthalpy results most closely approached the actual measured values. An anomaly from the previous findings is noted in this case, in that the parallel flow runs for both water and alcohol are considerably closer to the theoretical values of the literature than are the countercurrent runs. The results are indicated in Table VII. Graphs used in calculating the h and k values are shown in Figures 16 and 17.

TABLE VI: ACCURACY OF PREDICTED ENTHALPY VALUES

Enthalpy values in BTU Per Pound of Dry Air

Run	Start	Enthalpy Values		Predicted
		Measured End	Predicted End	<u>Enthalpy Change</u>
				<u>Measured</u> Enthalpy Change
Parallel Air-Water Flow				
1	9.45	10.6	10.5	.92
7	15.8	17.0	17.0	1.00
10	17.6	14.5	14.1	1.13
Countercurrent Air-Water Flow				
15	26.7	28.6	28.7	1.05
21	21.4	27.6	27.6	1.00
24	21.4	28.4	27.4	.87
Parallel Air-Methanol Flow				
39	35.7	57.5	52.1	.75
42	35.8	60.8	61.4	1.02
44	24.7	33.0	31.8	.86
Countercurrent Air-Methanol Flow				
50	33.5	41.0	42.0	1.13
55	43.0	54.5	54.5	1.00
57	39.2	54.3	55.7	1.09

TABLE VI: ACCURACY OF PREDICTED TEMPERATURE VALUES

Temperature In Degrees Fahrenheit

Temperature Values				Predicted
Run	Start	Measured End	Predicted End	Temperature Change
				Measured Temperature Change
Parallel Air-Water Flow				
1	60.8	50.4	47.6	1.27
7	73.0	62.2	62.2	1.00
10	66.9	54.7	55.2	.96
Countercurrent Air-Water Flow				
15	81.7	75.2	75.5	.94
21	81.1	75.0	75.3	.95
24	87.1	75.2	78.6	.71
Parallel Air-Methanol Flow				
39	87.8	64.0	73.8	.59
42	78.8	60.1	65.5	.71
44	67.1	45.0	47.5	.89
Countercurrent Air-Methanol Flow				
50	64.9	45.1	52.8	.61
55	77.7	51.8	53.4	.86
57	77.4	56.3	62.0	.73

TABLE VI: ACCURACY OF PREDICTED HUMIDITY VALUES

Humidities In Pounds Liquid Per Pound Dry Air

Run	Start	Humidity Values		Predicted
		Measured End	Predicted End	Humidity Change
				Measured Humidity Change
Parallel Air-Water Flow				
1	.0023	.0057	.0062	1.15
7	.0055	.0092	.0092	.97
10	.0078	.0086	.0079	.13
Countercurrent Air-Water Flow				
15	.0135	.0166	.0168	1.07
21	.0089	.0158	.0158	1.00
24	.0075	.0165	.0149	.82
Parallel Air-Methanol Flow				
39	.027	.080	.063	.68
42	.032	.088	.084	.93
44	.015	.042	.039	.89
Countercurrent Air-Methanol Flow				
50	.034	.058	.054	.83
55	.046	.080	.078	.94
57	.039	.078	.075	.87

FIGURE 12: GRAPHICAL ANALYSIS OF RUN 7

FIGURE 13: GRAPHICAL ANALYSIS OF RUN 21

FIGURE 14: GRAPHICAL ANALYSIS OF RUN 44

FIGURE 15: GRAPHICAL ANALYSIS OF RUN 55

TABLE VII: HEAT AND MASS TRANSFER COEFFICIENT RATIOS

a. The (h/ko_g) Ratio

Run	Liquid	Flow	Observed Ratio	Literature Value (25)
7	Water	P	.92	.99
21	Water	CC	.82	.99
44	Methanol	P	1.30	1.22
55	Methanol	CC	.83	1.22

b. The (h/k) Ratio

Run	Liquid	Flow	Observed Ratio	Literature Value (21)
7	Water	P	.22	.21-.27
21	Water	CC	.20	.21-.27
44	Methanol	P	.33	.29-.35
55	Methanol	CC	.22	.29-.35

FIGURE 16: GRAPHICAL INTEGRATION OF RUN 21

FIGURE 17: GRAPHICAL INTEGRATION OF RUN 44

VI. c. Evaporation Rates in Gas Streams

Recent British workers have done considerable work on the evaporation of various liquids into gas streams. Generally this subject has been approached from previous empirical work rather than on basic theoretical grounds. Wade (31) developed the equation,

$$\frac{Y}{P_s} = Ku^{.85} \quad (29)$$

where Y is the evaporation rate per unit area, P_s , the saturation pressure of liquid at its temperature, and u is the gas velocity passing over the liquid. A study was made of the evaporation rate of water in the wetted wall tower to determine whether a definite trend was shown. In the calculations H_s was used in lieu of P_s , since it would only affect the equation slightly and it employed figures used in other psychometric work. The results are listed in Table VIII; they show a definite trend, although both the constant and the exponent differ from Wade's values.

TABLE VIII: EVAPORATION RATES WITH VARYING GAS FLOW

Column	A	B	C	D	E
	Run	Gas Velocity	Measured Evap. Rate	Saturation Humidity	$\frac{\log C/D}{\log B}$
		ft./hr.	lb./hr.ft. ²	$\frac{\text{lb. H}_2\text{O}}{\text{lb. dry air}}$	
	58	36200	.38	.0076	.372
	59	36500	.30	.0076	.351
	60	38400	.29	.0070	.352
	61	39600	.35	.0070	.370
	62	43300	.78	.0088	.420
	63	49500	.46	.0076	.380
	64	52600	.47	.0070	.387
	65	52600	.49	.0070	.391
	66	57700	.39	.0076	.360
	67	60300	.45	.0076	.370
	68	63700	1.38	.0088	.457
	69	69000	.47	.0076	.370
	70	70400	.90	.0070	.436
	71	71300	.60	.0070	.399
	72	73000	.69	.0079	.398
	73	73000	1.55	.0088	.463

VII. Discussion

The chief limitation of the apparatus used was undoubtedly the small wetted wall tube diameter. This seriously restricted gas and liquid flow rates, for it was found that at Reynolds numbers greater than 10000, there was spray or flooding, and that the gas flow tended to pulsate.

The results do not indicate any generalities as to the minimum feasible liquid-gas ratio usable in this type experiment. Water, with a high latent heat of vaporization, gave results varying little with the liquid-gas ratio. The parallel flow air-methanol runs vary somewhat, but no clear cut implications are evident. In an apparatus of limited flow capacity such as that used, it seems likely that non-adiabatic runs will encounter too much heat flow from the surroundings to give satisfactory results. Starting with room-temperature methanol, differences as high as fifty degrees were encountered between the room and exit liquid temperatures. These temperature differences indicate that heat flow occurs and plays an important part in runs of highly volatile liquids.

The statistical analysis indicates that while good predictions may be made of the enthalpy changes of sets of countercurrent flow runs, sets of parallel runs will not give good averages. Furthermore, no dependence may be placed on individual runs, in either parallel or countercurrent flow.

The temperature and humidity predictions outlined in Table VI show that reasonably good forecasts may be made for both parallel and countercurrent flow of various liquids in some cases. However, the restrictions necessary to insure good runs must first be known before this method can prove of value; at present these criteria have not been ascertained.

The heat transfer and mass transfer coefficient ratios seem accurate enough to warrant use of this graphical system for their calculation. It

must be noted, though, that the runs used as examples were selected and that the method is not of general applicability.

The possibility of figuring evaporation rates and gas exit humidities by the very simple equation (29) suggests that further work in this direction may be promising. The differences from the British results are not considered significant since their operating conditions were entirely different from those of this thesis.

To further complicate analysis, the thesis confirmed certain literature notes (20) that water vapor in the air would give humidity readings with wet bulbs using other liquids, particularly liquids soluble in water. Both water alone and water mixed with other liquid vapor will give wet bulb readings indicating humidity, though the basis is not known.

SUMMARY AND CONCLUSIONS

The results of this thesis indicate that in a small non-adiabatic wetted wall tower apparatus, the heat transfer from the surroundings generally cannot be ignored. Sufficient variance in predicting temperature, humidity, and enthalpy results occurs so that a single run is of little value in forecasting operating conditions. It appears that a set of countercurrent runs gives findings near the theoretically expected values, but that sets of parallel flow runs vary widely from the theoretical values. Graphical calculation of accurate heat and mass transfer coefficients is possible for those cases which approximate theoretically expected conditions; parallel flow cases give somewhat better results than countercurrent flow cases. It is found that evaporation rates can be related to gas humidities and gas velocities.

In conclusion, it is believed that further work on this non-adiabatic graphical analysis method is promising at this time, in view of the uncertainty of the expected results, and the limited amount of background information in the literature.

LITERATURE CITED

LITERATURE CITED

1. Dampier, Sir William, "A History of Science", 3rd edition, p. 221, New York, MacMillan Company, 1940
2. Maxwell, J.C., Phil. Mag., 20, 21 (1860)
3. Stefan, Sitz.Acad.Wiss., Wien, 63, 2 (1871)
4. Sutherland, W., Phil. Mag., 38, 1 (1894)
5. Whitman, W.G., Chem. and Met., 29, No. 4, 146 (1925)
6. Othmer, D.F., Ind. Eng. Chem., 34, 1072, (1942)
7. Prandtl, L., Z. Physik, 11, 1072 (1910)
8. Colburn, A.P., Ind. Eng. Chem., 22, 967 (1930)
9. Arnold, J.H., Physics, 4, 255 (1933)
10. Chilton, T.H. and Colburn, A.P., Ind. Eng. Chem., 26, 1183 (1934)
11. Barnett, W.I. and Kobe, K.A., Ind. Eng. Chem., 33, 436 (1941)
12. Gilliland, E.R. and Sherwood, T.K., Ind. Eng. Chem., 26, 516 (1934)
13. Grosvenor, W.M., Trans. A.I. Ch. E., 1, 184 (1908)
14. Keenan, J.H. and Keyes, F.G., "Thermodynamic Properties of Steam", 1st edition, New York, John Wiley & Sons, 1936
15. Zimmerman, O.T. and Lavine, I., "Psychometric Tables and Charts", 1st edition, Durham, N.H., Industrial Research Service, 1945
16. Ivory, Phil. Mag., 60, 81 (1822)
17. Maxwell, J.C., Encyclopedia Britannica, 9th edition, 1877
18. Carrier, W.H., Trans. A.S.M.E., 33, 1005 (1911)
19. Lewis, W.K., Trans. A.S.M.E., 44, 325 (1922)
20. Sherwood, T.K. and Comings, E.W., Trans. A.I. Ch. E., 28, 88 (1932)
21. Sherwood, T.K., "Absorption and Extraction", 1st edition, pp. 52-57, New York, McGraw-Hill Company, 1937
22. Rodekohr, H.M., M.S. Thesis, Iowa State College, 1941

23. Boelter, L.M.K., Trans.A.I.Ch.E., 39, 557 (1943)
24. Sherwood, T.K. and Reed, C.E., "Applied Mathematics in Chemical Engineering", 1st edition, pp. 68-76, New York, McGraw-Hill Company, 1939
25. Walker, W.H., Lewis, W.K., McAdams, W.H. and Gilliland, E.R., "Principles of Chemical Engineering", 3rd edition, pp. 582-593, New York, McGraw-Hill Company, 1937
26. *ibid.*, pp. 56-68
27. Goodman, Wm., "Air Conditioning Analysis", 1st edition, chap. 17, New York, Macmillan Company, 1943
28. *ibid.*, pp. 285-323
29. International Critical Tables, 1st edition, Vol. V, pp. 113-138, New York, McGraw-Hill Company, 1928
30. Perry, J.H., "Chemical Engineer's Handbook", 2nd edition, p. 381, New York, McGraw-Hill Company, 1941
31. Croxton, F.E. and Cowden, D.J., "Applied General Statistics", 1st edition, p. 240, New York, Prentice-Hall Inc., 1946
32. *ibid.*, p. 307
33. *ibid.*, p. 873

APPENDIX

SYMBOLS USED

A	-	area, sq.ft.
c	-	specific heat, Btu/lb.°F.
c _g	-	humid heat, Btu/lb.dry air °F.
d	-	density, lb./cu.ft.
E	-	standard error of the mean
G	-	gas flow, lb./hr.
G'	-	mass gas rate, lb./hr.ft ² .
h	-	coefficient of heat transfer, Btu/hr.ft ² .°F.
h _{fg}	-	latent heat of vaporization, Btu/lb.
H	-	humidity, lb.liquid/lb.dry air
i	-	enthalpy, Btu/lb.
k	-	mass transfer coefficient, lb./hr.ft ² .unit H
k'	-	$M_L k / M_G$
K	-	a constant
L	-	liquid rate, lb./hr.
L'	-	mass liquid rate, lb./hr.ft ² .
M	-	molecular weight
N	-	number of items
p	-	pressure, atm. or lb./sq.in.
P _A	-	atmospheric pressure, atm.
q	-	rate of heat flow, Btu/hr.
R	-	Reynolds number
S	-	standard deviation
t	-	temperature, °F.

T	-	absolute temperature, °R.
u	-	velocity, ft./sec.
v	-	volume, cu.ft.
x	-	any value of a function
X	-	arithmetic mean
y	-	height, feet
Y	-	evaporation rate, lb./hr.ft ² .
z	-	symbol used in graphical integration

General Subscripts

g	-	refers to gas
i	-	refers to interface conditions
l	-	refers to liquid
o	-	refers to base temperature
s	-	refers to saturation conditions
v	-	refers to vapor
w	-	refers to wet bulb conditions

APPARATUS DIMENSIONS

Air Pipe:

	Parallel flow	C.C. flow
Standard one inch iron pipe		
Straight approach to orifice, cm.	90	180
Straight approach to wetted wall, cm.	100	90
Straight length from wetted wall, cm.	100	150
Air nozzle - one half inch pipe, 12 cm. long		
Steam jacket on air pipe, 100 cm.		

Wetted Wall Section:

Pyrex glass tubing,	
Overall length, parallel flow,	140 cm.
Overall length, countercurrent flow,	148 cm.
Wall thickness,	0.14 cm.
Inside diameter,	2.45 cm.

Orifice:

Stainless steel plate beveled on downstream side	
Upstream orifice diameter	1.00 cm.
Downstream orifice diameter	1.65 cm.
Plate thickness	0.28 cm.
Upstream tap center from plate edge	2.36 cm.
Downstream tap center from plate edge	1.96 cm.
Tap diameters	0.60 cm.

Reservoir capacity: 4 gallons

Wet and dry bulb thermometers:

Bulb diameter	0.50 cm.
Bulb length	2.0 cm.

SAMPLE CALCULATIONS

a. Orifice Calculations:

Reference (25) gives the orifice equation:

$$(u_o^2 - u_p^2)^{\frac{1}{2}} = .63(2gH')^{\frac{1}{2}}$$

where: u_o , u_p - orifice, pipe velocities, ft.
 g - force of gravity, ft./sec.²
 H' - head, ft. of liquid flowing

given: pipe diameter is 2.56 cm.
 orifice diameter is 1.00 cm.

$$\text{then: } u_p = (1/(2.56)^2)u_o = .153u_o$$

$$\text{and: } (u_o^2 - u_p^2)^{\frac{1}{2}} = u_o(1 - .0234)^{\frac{1}{2}} = .988u_o$$

Taking air at 75° F., the density is 1/13.475; then for each centimeter of water in the manometer reading, the air head at this temperature is:

$$\frac{(1)(62.4)(13.475)}{(2.54)(12)} = 27.6 \text{ ft. air/cm. water}$$

Where Z is the manometer reading in cm. of water the gas velocity is given by the orifice equation:

$$u_o = \frac{(.63)(62.4)^{\frac{1}{2}}(27.6 Z)^{\frac{1}{2}}}{(.988)} = 26.9 Z^{\frac{1}{2}} \text{ ft./sec.}$$

b. Gas Flow for Air at 75° F.:

$$G = \left(\frac{3.1416}{4}\right) \left(\frac{1}{(12)(2.54)}\right)^{\frac{1}{2}} \left(\frac{(3600)(26.9 Z^{\frac{1}{2}})}{13.475}\right)$$

$$= 6.12 Z^{\frac{1}{2}} \text{ lb. air/hr.}$$

c. Air Velocity in Wetted Wall Tower:

Assuming a water film of .025 cm., the i.d. is 2.40 cm.

$$u_t = \frac{(2.56)^2}{(2.40)^2} (.153)(26.9 Z^{\frac{1}{2}}) = 4.69 Z^{\frac{1}{2}} \text{ ft./sec.}$$

d. Reynolds Number in Wetted Wall Section:

Taking: tower diameter - .0787 ft.
 air density - $1/13.475 \text{ lb./ft.}^3$
 viscosity - .018 cp. (from (26) p. 689)

$$R = \frac{(.0787)(4.69 \text{ ZS})}{(.018)(.000672)(13.475)} = 2260 \text{ ZS}$$

e. Water Flow Rate:

$$100 \text{ cc./min.} = \frac{6000}{453.6} = 13.23 \text{ lb./hr.}$$

f. Enthalpy of Entrance Gas for Run 18:

$$t_g = 79.9^\circ\text{F.}$$

$$H = .0146$$

from equation (22):

$$i = (1075)(.0146) + (.24 + .45(.0146))(79.9 - 32) \\ = 27.5 \text{ Btu/lb.}$$

g. Calculation of Theoretical Gas Exit Enthalpy for Run 18:

Liquid entrance temp. - 77.5°F.
 Liquid exit temp. - 73.0
 Liquid flow rate - 10.2 lb./hr.
 Gas flow rate - 17.4

from equation (18):

$$i = 27.5 + \frac{10.2}{17.4} (77.5 - 73.0) = 30.1 \text{ Btu/lb.}$$

h. Location of THE Chart Constant Humidity Lines:

For a given humidity, two temperature-enthalpy points are located using equation (22) or equation (26). For the water vapor humidity line, $H = .020$, the method of (f) above is used to find:

$$\text{when } t = 60^\circ\text{F.} \\ t = 100$$

$$i = 28.48 \text{ Btu/lb.} \\ i = 38.42$$

i. Location of Adiabatic Saturation Line Endpoints:

A modified form of equation (25) is used, taking a saturation temperature for one end and either a constant humidity or a constant temperature line for the other.

$$t_g = t_{gs} + (h_{fgs}/c_s)(H_s - H_g)$$

For the air-water system, to find the adiabatic cooling line from a point of the saturation line where t is 60° :

$$t_s = 60^\circ \text{F.}$$

$$H_s = .0111 \text{ lb. water/lb. dry air}$$

$$h_{fgs} = 1053 \text{ Btu/lb.}$$

$$c_s = .2423 \text{ by computation}$$

Arbitrarily selecting the $H = .005$ line for the second end, solution of the equation shows that the adiabatic cooling line which starts on the saturation line at 60°F. will intersect the $H = .005$ line at a temperature of 57.0°F.

j. Graphical Integration for Heat Transfer Coefficients:

Taking Run 44 as an example, Figure 14 is used in this integration. By drawing horizontal lines at various intervals along the liquid operating line, their intersections with the graphically plotted gas operating line will indicate the temperature of gas in contact with the liquid at various points. The reciprocal of the difference of corresponding values of liquid and gas temperatures is plotted in Figure 17; values for the plot are:

t_g	t_l	$1/(t_g - t_l)$
67.1	45.7	.043
65	42	.044
63	40	.044
60	38.2	.046
57	36.7	.049
55	36	.053
52	35	.059
47.5	34.5	.077

Humid heat is found:

$$c_g = .24 + .38(.015 + .038)/2 = .2525$$

The area under the temperature curve in Figure 17 is found to equal 10.5 squares, each of which has a dimension of (.02) by (5), so that:

$$z_1 = (10.5)(.02)(5) = 1.05$$

where z_1 represents the integration area; the wetted wall tower area is 1.15 sq.ft. and the gas rate 13.3 lb./hr., so that from equation (27):

$$h = \frac{(13.3)(1.05)(.2525)}{(1.15)} = 3.06$$

k. Graphical Integration for Mass Transfer Coefficients:

Exactly the same procedure as in (j) is followed, except that corresponding humidities instead of corresponding temperatures are found. Values are:

t_g	H_g	H_s	$1/(H_s - H_g)$
67.1	.015	.072	17.6
65	.019	.068	20.4
63	.023	.063	25.0
60	.027	.059	31.4
57	.030	.056	38.4
55	.033	.054	47.6
52	.036	.052	58.8
50	.036	.051	66.7
47.5	.038	.050	83.3

The area under the humidity curve in Figure 17 is 4.05 squares with dimensions (20) and (.005):

$$z_2 = (4.05)(20)(.005) = .81$$

The wetted wall tower section area is 1.15 sq.ft. and the gas rate is 13.3 lb./hr. so that from equation (28):

$$k = \frac{(.81)(13.3)}{(1.15)} = 9.37$$

l. Evaporation Rate Check - Run 60:

Gas velocity - 38,400 ft./hr.

Measured evaporation rate - .29 lb./hr.ft.²

Saturation humidity at liquid temp. - .0070 lb.H₂O/lb.d.air

$$\log \text{ Gas Velocity} = 4.584$$

$$\log \frac{\text{Evaporation Rate}}{\text{Humidity}} = 1.617$$

$$\frac{\log (\text{Evap. Rate/Humidity})}{\log \text{ Gas Velocity}} = \frac{1.617}{4.584} = .352$$

m. Statistics Calculations:

Take: $\bar{x}$ - value of (T/M) ratio for one run
 N - number of runs
 $\bar{X}$ - arithmetic mean for N runs
 S - standard deviation for set of runs
 E - standard error of the mean

Using the parallel flow air-water set (runs 1-12):

the arithmetic mean may be calculated:

$$\bar{X} = \frac{(x_1 + x_2 + \dots + x_N)}{N} = \frac{16.17}{12} = 1.35$$

calculation of standard deviation (31):

$$S = \frac{((x_1 - \bar{X})^2 + \dots + (x_N - \bar{X})^2)^{\frac{1}{2}}}{N^{\frac{1}{2}}} = \frac{(.757)^{\frac{1}{2}}}{(12)^{\frac{1}{2}}} = .251$$

calculation of standard error of the mean (32):

$$E = S/(N-1)^{\frac{1}{2}} = .251/(11)^{\frac{1}{2}} = .0757$$

calculation of probability for (T/M) between .90 and 1.10:

$$\frac{X-.90}{E} = \frac{.45}{.0757} = 5.9$$

$$\frac{X-1.10}{E} = \frac{.25}{.0757} = 3.3$$

From the chart (33), the probability of the (T/M) ratio falling in this range is found to be .0005, or five in ten thousand.

WETTED WALL TOWER OBSERVATIONS

Temperatures - °F.
 Humidities - lb. liquid/lb. dry air
 Flow Rates - lb./hour

Run	Air In	Temperature Out	Liquid In	Temperature Out	Room Temperature
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Parallel Air-Water Flow

1	60.8	50.4	45.2	46.6	53.2
2	61.2	50.4	46.4	45.2	54.3
3	73.9	64.2	60.4	59.0	69.1
4	74.5	64.2	60.3	58.1	68.9
5	74.7	64.2	60.3	57.9	69.3
6	56.7	48.2	45.0	43.5	50.0
7	73.0	62.2	58.5	57.4	66.6
8	68.7	56.1	51.3	46.4	60.4
9	68.0	55.2	50.7	46.6	60.4
10	66.9	54.7	50.7	47.3	60.1
11	66.9	54.3	50.0	48.9	60.4
12	67.1	54.5	49.8	46.8	60.4

Countercurrent Air-Water Flow

13	75.9	66.6	71.2	61.5	71.6
14	81.3	71.2	75.2	64.2	75.9
15	81.7	75.2	77.4	71.8	76.1
16	72.7	67.1	71.2	61.5	72.0
17	82.0	73.8	75.6	66.2	76.5
18	79.9	76.1	77.5	73.0	76.3

Run	Humidity		Gas	Flow	
	In	Out		Gas	Liquid
Parallel Air-Water Flow					
1	.0023	.0057	25.0		7.9
2	.0025	.0057	21.8		9.9
3	.0072	.0102	24.7		15.9
4	.0070	.0102	21.5		15.9
5	.0068	.0102	18.0		15.9
6	.0025	.0053	24.8		23.8
7	.0050	.0092	13.9		15.9
8	.0084	.0085	25.8		38.4
9	.0084	.0085	24.0		39.1
10	.0078	.0086	21.2		40.9
11	.0085	.0085	18.5		40.9
Countercurrent Air-Water Flow					
12	.0082	.0085	15.4		40.9
13	.0072	.0107	26.5		4.0
14	.0084	.0130	22.8		5.4
15	.0135	.0166	25.0		8.2
16	.0074	.0122	16.1		6.5
17	.0078	.0144	23.1		12.9
18	.0146	.0179	17.4		10.2

Run	Air Temperature In	Out	Liquid Temperature In	Out	Room Temperature
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Countercurrent Air-Water Flow (cont.)

19	80.4	72.2	76.7	65.8	76.3
20	76.1	77.2	76.4	75.0	75.4
21	81.1	75.0	76.8	69.1	76.4
22	80.8	74.8	75.9	69.8	76.5
23	86.9	76.8	76.1	71.8	81.9
24	87.1	75.2	76.3	72.0	82.2
25	76.6	75.7	78.4	75.0	78.6

Parallel Air-Methanol Flow

26	71.4	47.3	67.6	40.8	72.0
27	72.3	47.3	68.0	34.5	72.1
28	71.4	48.4	67.6	39.2	71.1
29	74.1	49.6	91.4	37.2	66.2
30	71.4	47.5	47.1	34.0	72.0
31	90.3	57.9	61.7	40.6	73.4
32	71.6	48.6	53.6	37.4	72.3
33	74.8	52.3	89.6	39.9	67.1
34	71.2	50.9	91.9	37.6	64.4
35	86.9	55.4	60.8	40.1	71.2
36	84.0	55.2	67.6	42.8	73.6
37	74.5	36.7	68.9	46.2	73.9
38	64.9	44.1	45.5	31.6	64.9
39	87.8	64.0	88.2	50.9	74.7

Run	In	Humidity Out	Gas Flow	Liquid
Countercurrent Air-Water Flow (cont.)				
19	.0082	.0148	16.8	8.3
20	.0167	.0195	19.2	14.7
21	.0089	.0158	19.7	15.9
22	.0039	.0078	13.6	21.0
23	.0093	.0166	21.0	20.8
24	.0074	.0165	23.0	32.4
25	.0069	.0180	19.7	48.9
Parallel Air-Methanol Flow				
26	.024	.047	12.6	1.30
27	.018	.042	27.4	3.15
28	.019	.044	21.6	3.56
29	.008	.040	23.1	3.68
30	.023	.041	27.7	6.00
31	.021	.056	28.3	10.3
32	.023	.049	19.1	7.25
33	.009	.054	18.6	7.61
34	.008	.053	27.8	11.7
35	.023	.052	23.2	10.1
36	.025	.060	19.0	8.40
37	.031	.065	12.3	7.88
38	.011	.055	27.9	18.1
39	.027	.080	18.7	16.3

Run	Air Temperature		Liquid Temperature		Room Temperature
	In	Out	In	Out	

Parallel Air-Methanol Flow (cont.)

40	66.9	44.2	46.8	34.3	63.0
41	75.7	57.2	84.6	46.9	65.8
42	78.8	60.1	91.4	52.7	75.2
43	74.7	55.4	69.6	45.5	74.1
44	67.6	45.0	43.7	34.5	63.0
45	84.4	64.4	86.0	54.5	75.2

Countercurrent Air-Methanol Flow

46	77.4	53.1	71.6	47.8	75.9
47	61.9	42.1	65.5	35.6	70.3
48	77.4	53.8	71.6	50.9	75.7
49	67.6	45.9	65.1	35.4	69.3
50	64.9	45.1	67.5	34.0	69.3
51	75.4	57.9	71.4	47.3	73.9
52	66.9	46.9	71.2	35.6	69.3
53	77.7	55.8	71.4	41.4	75.7
54	77.4	54.1	71.6	44.2	75.7
55	77.7	51.8	65.7	41.4	73.6
56	75.6	51.6	66.0	41.5	74.1
57	77.4	56.3	70.5	44.8	75.6

Run	Humidity		Gas	Flow	
	In	Out			Liquid
Parallel Air-Methanol Flow (cont.)					
40	.010	.039	21.0		20.5
41	.009	.067	12.9		12.7
42	.032	.088	29.0		31.5
43	.031	.066	20.9		23.6
44	.015	.042	13.3		17.1
45	.024	.089	12.1		21.0
Countercurrent Air-Methanol Flow					
46	.045	.064	17.3		3.0
47	.032	.048	23.6		4.3
48	.040	.061	17.3		3.8
49	.035	.053	20.9		5.9
50	.034	.058	18.6		7.9
51	.045	.072	17.7		8.3
52	.006	.057	18.6		10.0
53	.040	.078	17.3		10.7
54	.036	.075	17.4		12.5
55	.046	.080	18.7		14.7
56	.050	.084	20.8		18.4
57	.039	.078	13.6		14.5

CALCULATED HEAT TRANSFER AND MASS TRANSFER COEFFICIENTS

Run	h	k
7	3.64	16.3
21	4.35	21.6
44	3.06	9.37
55	5.04	23.0

VITA

Wylie Charles Kirkpatrick was born in Medford, Massachusetts, on 2 March 1918, the son of Charles G. and Louise (Wylie) Kirkpatrick. He received his early education in the Medford public schools.

Kirkpatrick entered the Massachusetts Institute of Technology in 1936 and received the degree of Bachelor of Science in Chemical Engineering in 1940. He attended the University of Louisville in 1941-42 and 1946-47 and received the degrees of Bachelor of Arts in Mathematics and Master of Chemical Engineering in 1947. He attended Universite Laval (Canada) in 1946.

Kirkpatrick was employed by the explosives division of the duPont company from 1940 to 1942. He was ordered to active Army duty in 1942 and spent two years on engineer troop duty in the United States and two years on the staff of the Chief Engineer, European Theater, reverting to inactive duty in 1946.

He has been a member of the American Chemical Society since 1940 and is at present assistant division engineer of the reserve 100th Airborne Division.