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PSYCHROMETRIC STUDIES AT EXTREME HUMIDITIES

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Loren Jens Hov

(M.S. Thesis)

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Contents

Abstract 3
Introduction 4
Theory
Heat and Mass Transfer. 5
Psychrometry 11
Experimental
Plan of Investigation 16
Equipment 17
Procedure 25
Results 27
Discussion and Conclusions. 34
Acknowledgment. 38

Appendix
Properties of the Air-Water Vapor System. 39
Sample Calculations. 50
Experimental and Calculated Results 54
Calibration 54
Nomenclature 57

Bibliography 60

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ABSTRACT

To determine the effect of the film-pressure factor on the psychrometric ratio in the air – water-vapor system, wet- and dry-bulb determinations were made over a range of film-pressure factors from 0.941 to 0.0865. When the film-pressure factor did not deviate far from unity the data agreed with the expression of psychrometric ratio by Lynch and Wilke:

$$\frac{k_G M_m p_f}{h} C_p = 0.93 \left(\frac{C_p \mu}{k} \Big/ \frac{\mu}{\rho D} \right)^{1/2}$$

In this expression the heat-transfer coefficient h has been corrected for simultaneous mass transfer according to the Ackermann correction as extended by Mickley and associates. The data disagree sharply as the film-pressure factor is reduced. At values of p_f below 0.7 the disagreement exceeds 10%, and exceeds 90% at $p_f = 0.1$.

The above equation would more nearly agree with the data if the film-pressure factor entered to the zero power, but no simple power function can adequately explain the data. Further psychrometric work on other systems is needed to express the dependence on the film-pressure factor, and until this is accomplished a plot of psychrometric ratio versus film-pressure factor is offered for psychrometric calculations for the air–water-vapor system.

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INTRODUCTION

Since early in the nineteenth century with Apjohn's work on vaporization,[3] attempts have been made to quantitatively determine rates of mass and heat transfer and, primarily with the air – water-vapor system, the simultaneous effects of both and their effect on each other. Extensive studies have been made by industry, universities, and the U. S. Weather Bureau.[16] Although many accurate data have been compiled – enough to establish and maintain associated multimillion dollar industries such as air conditioning and product drying, and to assist in weather observation and prediction – reliable simultaneous heat- and mass-transfer data over more than 10% or 15% of the composition range are lacking.

Theoretical mass-transfer equations applicable to such studies postulate the film-pressure factor, a corrective term for nondiffusing fluids. Previous experimental work indicates a significant effect of simultaneous mass transfer on corresponding heat-transfer equations. The degree to which these enter into relationships describing simultaneous heat and mass transfer has been little investigated.

In view of the inadequacy of the existing data, this work was undertaken to (a) investigate simultaneous heat- and mass-transfer phenomena in the air – water-vapor system over as much of the composition range as was feasible, (b) determine what effect, if any, the film-pressure factor had on mass-transfer equations, and (c) determine the effect of simultaneous mass transfer on the heat-transfer equations.

HEAT AND MASS TRANSFER

The basic rate equation for gaseous diffusion was postulated by Maxwell as a driving force consisting of activity drop or of a partial-pressure difference of the diffusing gas overcoming a resistance to diffusion. This resistance is proportional to tbe number of molecules of diffusing gas and gas through which the molecules must pass, the velocity difference between the diffusing gas and other gases in the net diffusional direction, and the length of path in the diffusional direction. For a binary mixture this becomes

$$-dp_A = \alpha_{AB} \frac{\rho_A \rho_B}{M_A M_B} (u_A - u_B) \quad dx, \tag{1}$$

where

p_A = partial pressure of diffusing gas A,

α_{AB} = proportionality constant for diffusion of A and B,

ρ_A, ρ_B = partial densities of gases A and B,

M_A, M_B = molecular weights of gases A and B,

u_A, u_B = velocities of A and B in the net diffusional direction,

x = length of the diffusional path.

If we consider $N_A = \frac{u_A \rho_A}{M_A}$, $D_M = \frac{RT}{\alpha_{AB}}$, for an ideal gas $\rho_A = \frac{p_A M_A}{RT}$, and gas B stagnant so that $u_B = 0$, the above equation can be reduced to

$$N_A = -\frac{D_M dp_A}{p_B dx}, \tag{2}$$

where, now,

N_A = transport rate of gas A - $\frac{\text{lb moles}}{\text{hr ft}^2}$,

D_M = molal diffusivity - $\frac{\text{lb moles}}{\text{hr ft}}$,

p = atmospheres,

x = feet.

The above equation is theoretically applicable to mass transfer by molecular diffusion, but its practical application is limited because of uncertainty in the molal diffusivity caused by turbulence in fluid flow and by insufficient information available on the length of the diffusional path.

As we speak of molecular diffusion, so may we speak of eddy diffusion, wherein mass is transferred by fluid swirls or eddies in the core of fluid flowing under such conditions that turbulence exists. With fluids flowing under such conditions, there are three zones in the flow pattern: (a) the boundary or laminar layer where viscous forces are important and mass transfer is primarily controlled by molecular diffusion, (b) the turbulent or central region wherein eddy diffusion is the paramount factor, since material is transferred by mass motion, and (c) the buffer or transition region wherein laminar and turbulent flow shade into each other and both molecular and eddy diffusion are important. Thus in the turbulent core one may write a similar equation for eddy diffusional mass transfer, using E_m = eddy diffusivity (in lb moles/hr ft):

$$N_A = -\frac{E_m dp_A}{p_B dx} . \tag{3}$$

Since for mass transfer in a turbulently flowing fluid system, Eqs. (2) and (3) would be applicable in their respective zones and a weighted or averaged value for diffusivity would have to be used through the buffer region, and all these equations have unknown or uncertain factors, an empirical simplification of the problem has been applied.

We may now integrate Eq. (2) for a binary mixture with gas B stagnant, realizing for this case $\frac{dp_A}{dx} = -\frac{dp_B}{dx}$,

and obtain

$$N_A = \frac{D_m (p - p_i)}{B_g p_f} , \tag{4}$$

where B_g = the equivalent film thickness of a hypothetical model in which the resistance to mass transfer of the laminar layer is equal to the resistances of the laminar,

turbulent, and buffer layers in each actual model, in ft,

p = partial pressure of component A in the main stream, in atmos,

p_i = partial pressure of component A at the surface or interface, in atoms,

p_f = film-pressure factor, equal to the log mean partial pressure of the inert gas in this case.

The empirical mass-transfer coefficient is then defined as

$$k_g = \frac{D_m}{B_g p_f}, \tag{5}$$

and Eq. (4) now can be written as

$$N_A = k_g (p - p_i), \tag{6}$$

where k_g = gas film mass-transfer coefficient, in lb moles/hr ft^2 atmos.

Thus the gas film mass-transfer coefficient, (k_g), assumed equal to the over-all coefficient (K_G) upon neglecting the liquid film resistance, is found by the Maxwell-type equation to be inversely proportional to the film-pressure factor.

Mass-transfer coefficients for various models with varying conditions were experimentally measured, and ultimately workable prediction systems were devised. Chilton and Colburn[12] have used the data of others, as well as their own, to form a correlation based on physical properties of the system as well as flow properties in turbulence. Their correlation yields

$$k_g = \frac{0.023\ G}{M_m p_f\ Re^{0.2} Sc^{2/3}},$$

where

G = mass velocity, lb/hr ft^2,

M_m = mass molecular wt of gas,

$Re = \frac{dG}{\mu}$,

$Sc = \mu/\rho D$,

μ = viscosity in lb/hr ft,

ρ = density in lb (m)/ft^3,

D = molecular diffusivity in ft^2/hr,

d = diameter in ft.

As with the mass-transfer coefficient, the heat-transfer coefficient (h = Btu/hr ft^2 $^\circ$F), originally experimentally determined for a system, was postulated for the transfer of heat through a film and has been correlated by Colburn[13] as

$$h = \frac{0.023\ C_p\ G}{Re^{0.2} Pr^{2/3}},$$

where C_p = specific heat in Btu/lb $^\circ$F,

$Pr = C_p\ \mu/k$,

k = thermal conductivity in Btu/hr ft^2 $^\circ$F/ft.

In evaporative studies on a wetted wall column at pressures of 0.145 to 3.06 atmospheres and Re > 2000, Gilliland and Sherwood[17] found

$$k_g = 0.023\ Re^{0.83} Sc^{0.44} \left(\frac{D}{RTd}\right) \left(\frac{P}{p_f}\right).$$

They found little variation with pressure and concluded that k_g was inversely proportional to p_f. Their experimental ratio of P/p_f was varied only slightly, so that little value can be placed on this conclusion.

The work of Barnett and Kobe,[6] after suitable rearrangement of groupings, results in correlations of

$$k_g = \frac{0.025\ G}{Re^{0.2} Sc^{2/3} M_m p_f},$$

and

$$h = \frac{0.031\ C_p\ G}{Re^{0.2} Pr^{2/3}}.$$

A 24% difference is found between their heat transfer without simultaneous mass transfer and their mass-transfer data. Their work, also, was done without significant variation in p_f.

O'Brien and Stutzman found,[27] in evaporation studies from a plane surface with Re between 2600 and 22000,

$$k_g = \frac{0.11}{Re^{0.29} Sc^{2/3}} \left(\frac{u}{RT} \right) \left(\frac{P}{p_f} \right)$$

for five different liquids, but water was a great deviant. Also P/p_f remained nearly equal to unity for all their runs. It is interesting to note that they also conclude that $k_g p_f$ is not a simple correlation with diffusivity or Sc. They give a second equation as

$$k_g p_f = 0.041 \; D^{1.1} G^h ,$$

where

$$h = 0.54/D^{0.3}.$$

Beddingfield and Drew performed evaporation studies with "wet bulbs" of volatile solids and correlated their results,[8] along with other experimenters', as

$$k_g = \frac{0.281 \; \beta \; G}{Re^{0.4} Sc^{0.56} M_m}$$

and

$$h = \frac{0.281 \; C_p \; G}{Re^{0.4} p_f^{0.56}} ,$$

where β is a corrective term for a net mass flow across the film boundary, usually as found by the authors to vary from unity by less than 1% but with the possibility of being as high as 1.23%.

In a recent paper Westkaemper and White described the results of a study of concentration level on mass transfer rate in the air – carbon tetrachloride system.[36] The variation of film-pressure factor used in this study was from 0.810 to 0.287. They correlate their data

as the product

$$\frac{k_g RTd}{D_m} \quad Sc^{-0.44}$$

with and without film-pressure factor versus several functions of Reynolds number and also correlate with Cairns and Roper.[11] They conclude little from their data other than that there is adequate correlation on Reynolds numbers with the film-pressure factor entering to the 0, 0.83, and 1.00 powers. Indeed, plotting their data as

$$\frac{\frac{k_g RTd}{D_m} \quad \frac{p_f}{P}^{0,0.83,1.00}}{0.0014\ Re^{1.08}} \quad \text{versus} \quad \frac{p_f}{P}$$

yields, instead of a correlation on Sc^n ($Sc^{0.44}$), only a scatter of data. Their Schmidt number was varied fivefold.

Westkaemper and White in their theoretical treatment, suggest that eddy diffusivity (and thus mass-transfer rate) is a function of concentration level, and, in a system wherein kinematic viscosity is relatively constant with respect to concentration, eddy diffusivity is relatively constant with respect to concentration. Further, since the air — water-vapor system has a gas kinematic viscosity largely independent of concentration, it would be expected to show little effect of concentration level on mass-transfer rates. This is not borne out by the work of Cairns and Roper,[11] but each of their runs encompassed an exceptionally wide range of temperatures and thus a range of kinematic viscosity.

PSYCHROMETRY

Hygrometry, as a special case of psychrometry in which heat and mass transfer are occurring simultaneously in the air – water-vapor system, has probably been used more extensively for determining the humidity of air than any other means. In general, air is passed over the bulb of a thermometer which is covered with a water-saturated wick. Heat is transferred from the main gas stream to the wick while mass in the form of vaporizing water is transferred from the wick to the main gas stream until a dynamic balance is established in which, during any time interval, the quantity of heat transferred to the wick equals the quantity of heat absorbed in vaporizing the water plus the sensible heat needed to raise this water vapor to main-stream temperature conditions.

If the mechanisms of heat and mass transfer in this case are exactly analogous, any velocity dependency should enter the rate equations equally and therefore the ratio of the mass-transfer coefficient to the heat-transfer coefficient is functionally dependent only on the Schmidt and Prandtl numbers.

Colburn[13] and Chilton and Colburn[12] defined j numbers, analogous to the friction factor in momentum transfer, for heat and mass transfer as

$$j_h = \frac{h}{C_p G} \left(\frac{C_p \mu}{k} \right)^{2/3}$$

and

$$j_d = \frac{K_G M_m p_f}{G} \left(\frac{\mu}{\rho D} \right)^{2/3} ,$$

and, assuming them equal, obtained the psychrometric ratio

$$\frac{K_G M_m p_f}{h} C_p = \left(\frac{C_p \mu}{k} \Big/ \frac{\mu}{\rho D} \right)^{2/3} .$$

Arnold,[4] following Prandtl's equation[33] for heat transfer in pipes, modified the Sc, Pr correlation to include a corrective term for laminar flow at the surface of the thermometer so as to obtain

$$\frac{k_g M_m p_f}{h} \quad C_p = \frac{1 + \frac{u_B}{u_m}\left(\frac{C_p \mu}{k} - 1\right)}{1 + \frac{u_B}{u_m}\left(\frac{\mu}{\rho D} - 1\right)} ,$$

where

u_m = velocity of main stream,
u_B = velocity at edge of laminar layer.

His experimental work showed this velocity dependency to be a function of the Reynolds number to the 0.07 power, but, since he used a short wick which allows errors due to conduction by the thermometer stem, this dependency is questionable.

In a very carefully run series of experiments investigating the deviation of the wet-bulb temperature from the temperature of adiabatic saturation, Dropkin found no variation with either temperature or Reynolds number.[15]

While determining mass-transfer rates with water from various shapes, Powell attained excellent agreement with Dropkin up to Reynolds numbers of 10,000.[32]

Beddingfield and Drew conclude also that there is no dependency on Reynolds number, but gave a 0.56-power dependency to the Sc, Pr group.[8] Their work is quite accurate but they use an assumed fixed point of

$$Sc = 0.70 \quad \frac{k_g M_m p_f}{h} = 4.16$$

in calculating their slope, which disagrees with data by about 10%.

In a recent résumé of psychrometric results of other investigations as well as their own, Lynch and Wilke conclude[24] that the best expression for the psychrometric ratio is

$$1.10 \frac{k_g M_m p_f \gamma}{h'} \quad C_p = \left(\frac{C_p \mu}{k} \Big/ \frac{\mu}{\rho D} \right)^{1/2}$$

The corrective term γ, equal to $\frac{h'}{h}$ and originally defined as $1 - C_m k_g \Delta p/h$, has been shown by Olander[28] to be correctly expressed as $1 - C_m k_g \Delta p/2h$ for

small values of $C_m k_g \Delta p/h$. Lynch found in his determinations of three different gases with water vapor that his corrective term γ deviated less than 2% from unity; his film-pressure factor was 4% lower than unity. Lynch's and Wilke's value of 1/2 for the exponent has been accapted for this work, and values herein are reported on that basis. Recalculation of their data, using the Θ corrective term described in the next paragraph, yields a coefficient of 1.08, which is in excellent agreement with this work and is used herein.

Mickley et al., performing theoretical and experimental studies of boundary-layer phenomena in sucking or blowing air through a pourous flat plate with a main air stream flowing parallel to the plate, postulated a corrective factor for the effect of mass transfer on the heat-transfer coefficient.[26] They represent the temperature profile normal to the wall in a dimensionless form as

$$\beta_H = \frac{t_s - t}{t_s - t_a} ,$$

and show

$$\Theta = \frac{(d\beta_H/dy)s}{(d\beta_H/dy)s^*} = \frac{\phi}{e^{\phi} - 1} = \frac{h}{h^*} ,$$

where

$$\phi = \frac{v_o \rho C_p}{h^*} ,$$

y = normal distance from wall or plate,

h = heat-transfer coefficient with mass transfer,

h^* = heat-transfer coefficient in absence of mass transfer,

v_o = mass-transfer velocity (blowing or sucking).

During blowing their value of Θ ranged as low as 0.2 ($\phi_H = 2.7$).

Their theoretical development is based on dimensionless profiles of velocity, temperature, and mole fraction, as β_H above, which may all be reduced to the form

$$\frac{d^2\beta}{dm^2} - \Gamma \left(\frac{d\beta}{dm}\right) = 0,$$

where Γ = dimensionless measure of mass-transfer rate for each of the three cases (Γ is related to ϕ),

m = dimensionless distance from wall, y/Δ,

Δ = effective film thickness.

This form is solved, by using film theory with the simplification that $\Delta/\Delta^* = 1$, resulting in the reduced solution Θ above. The heat-transfer coefficient solution is as above; the friction-coefficient solution is

$$\Theta = \frac{C_f}{C_{f^*}},$$

where now

$$\phi = \frac{2\, v_o}{u_1 C_{f^*}},$$

C_f = friction coefficient with mass transfer,

C_{f^*} = friction coefficient in absence of mass transfer, and

u_1 = main-stream velocity;

and the mass-transfer coefficient solution is

$$\Theta = \frac{K}{K^*},$$

where now

$$\phi = \frac{v_o\, \rho}{K^*\, M_m},$$

K = mass-transfer coefficient with mass transfer,

K^* = mass-transfer coefficient in absence of mass transfer ($\lim_{K} \to 0$).

Additionally, solutions are obtained by using boundary-layer theory.

Excellent experimental evidence is presented which correlates nicely with the theoretically developed Θ corrective term, based on film

theory, for friction coefficient and heat-transfer coefficient in the presence of mass transfer. The boundary-layer-theory corrections are somewhat astray. Because of the firm theoretical and experimental establishment of the Mickley et al. Θ corrective term, it is applied in this work to correct the heat-transfer coefficient with the simultaneous mass transfer. The theoretical correction of the mass-transfer coefficient with mass transfer is not applied to this work. If used it would amount to about 0.1% change in the majority of the runs.

It is interesting to note that the runs of Mickley et al. at zero mass transfer correlate very nicely with the modified Chilton-Colburn[12] empirical relation

$$h = \frac{0.0288\ C_p\ G}{Re^{0.2} Pr^{2/3}}$$

for Reynolds number from 6,500 to 3,300,000. The Prandtl number, however, was not far from unity.

Mickley et al. state that equations somewhat similar to theirs were first presented by Ackermann.[1] Ackermann's work, a purely theoretical one, was developed through the use of dimensional parameters for the profiles of velocity and temperature in the boundary layer in the presence of mass transfer. Ackermann arrives at the same differential equation, in dimensional form, as does Mickley et al., but carries only his heat-transfer equations out to the same type of conclusion. Ackermann arrives at a solution identical with that of Mickley et al. for the use of suction only. His blowing solution, in the nomenclature of Mickley,

$$\Theta = \frac{\phi\ e^{\phi}}{e^{\phi} - 1} ,$$

differs by the factor e^{ϕ}, which was apparently introduced into his calculations in an attempt to not change the gradient $\left(\frac{dT}{dy}\right)_s$ in the case of the heat quantity carried through the film by diffusing mass during blowing. This factor change would have changed the theoretical curve of Mickley at the highest value of ϕ by a factor of fifteen in the value of Θ.

PLAN OF INVESTIGATION

The threefold purpose of this investigation as specified in the introduction was simplified, because of available equipment, to the use of wet- and dry-bulb measurements to determine the psychrometric ratio for the air — water-vapor system. In the large-scale equipment available, specific heat- and mass-transfer coefficients could not be determined, but the ratio of the two for a specific psychrometric ratio could be determined quite accurately. If this ratio followed in general the predictions of previous investigators into the high-water-vapor mole-fraction range, then any deviations could be quantitatively attributed to the effect of changing film-pressure factor, or to the various corrections for mass transfer on the heat-transfer coefficient, or both. It was felt that Lynch and Wilke had established the psychrometric ratios' dependence on Schmidt and Prandtl numbers, leaving just film-pressure factor and corrective terms undetermined experimentally.

EQUIPMENT

The equipment used to carry out this investigation was basically that used by Lynch, as shown in the accompanying diagrams (Figs. 1, 2, and 3) with several modifications. Gas circulation was provided by a Roots-Connersville positive-displacement blower with maximum output of 700 cfm. The gas temperature was maintained by steam-heated coils and the wet-bulb temperature by direct steam injection, the supply for both being controlled by two Brown Electronic temperature-recorder controllers. With this arrangement, it was possible to keep temperature fluctuations at the control point with ± 0.5°F, while the flow chamber and ducting provided damping to reduce this fluctuation at the wet-and-dry-bulb setup to within ± 0.2°F. As the investigation progressed, it was evident that even this small fluctuation would produce appreciable experimental errors in the extremely high-mole-fraction water-vapor region. For these latter runs the recorder controllers were bypassed and temperature was maintained by a fixed setting on the primary steam-pressure control valve and fixed settings on the bypass valves to the steam coil and direct steam injector, allowing up to 6 hours for steady-state conditions to be attained. With this arrangement temperature fluctuations at the wet and dry bulbs were held to within ± 0.04 °F.

The heated humidified gases entered a 20-inch diameter section (Fig. 2) containing straightening vanes and a flow nozzle of the Bean, Buckingham, and Murphy type,[7] calibration for which appears in Fig. 16 in the Appendix. The gases then passed the wet-and-dry-bulb setup. An orifice was placed in the ducting ahead of the thermometers to provide a uniform high-velocity flow across the thermometer bulbs. A thermocouple was placed in the ducting wall at the thermometer setup to determine wall temperature for radiation corrections. A pressure tap was also placed in the wall to determine total pressure at the wet-and-dry-bulb arrangement.

The wet-and-dry-bulb assembly is shown in Fig. 3. The thermometers used were precision thermometers with 0.1°C graduations. They were

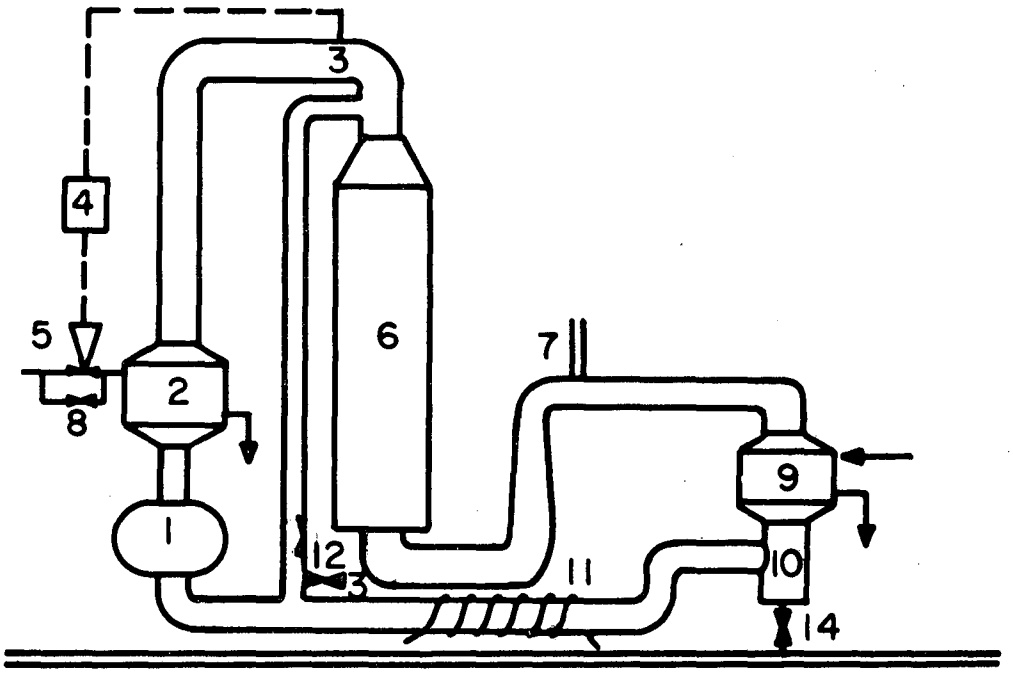

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Fig. 1.

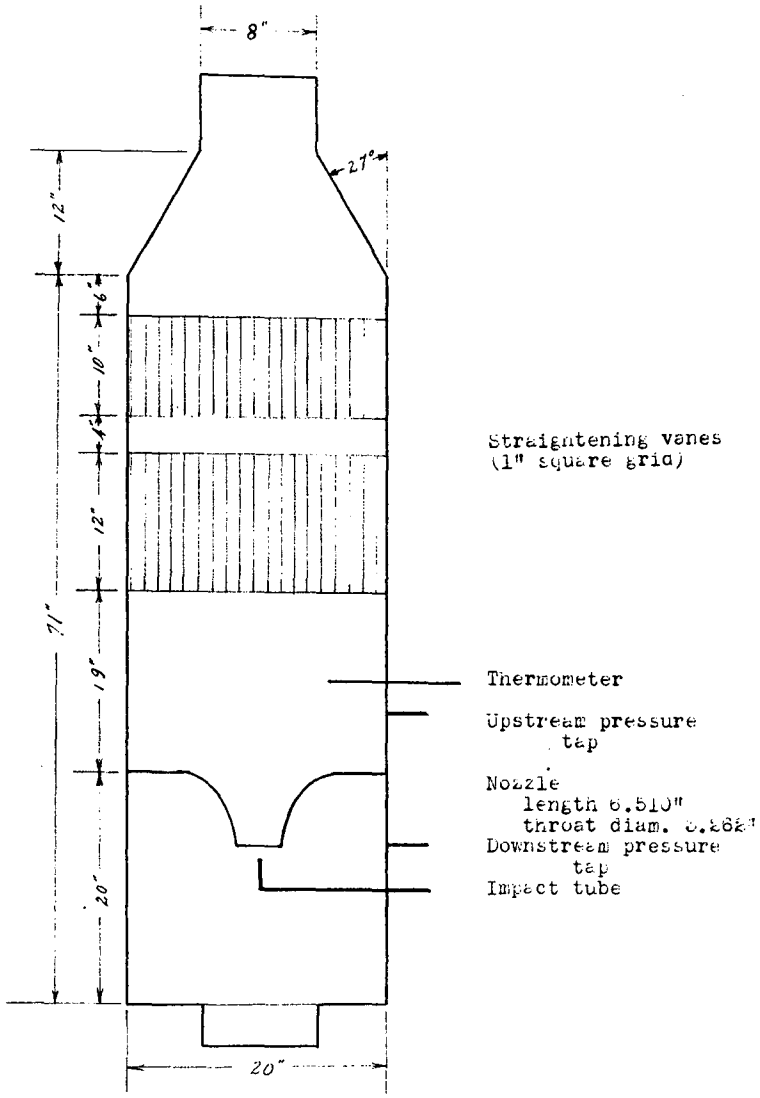

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Fig. 2. Flow-measuring chamber.

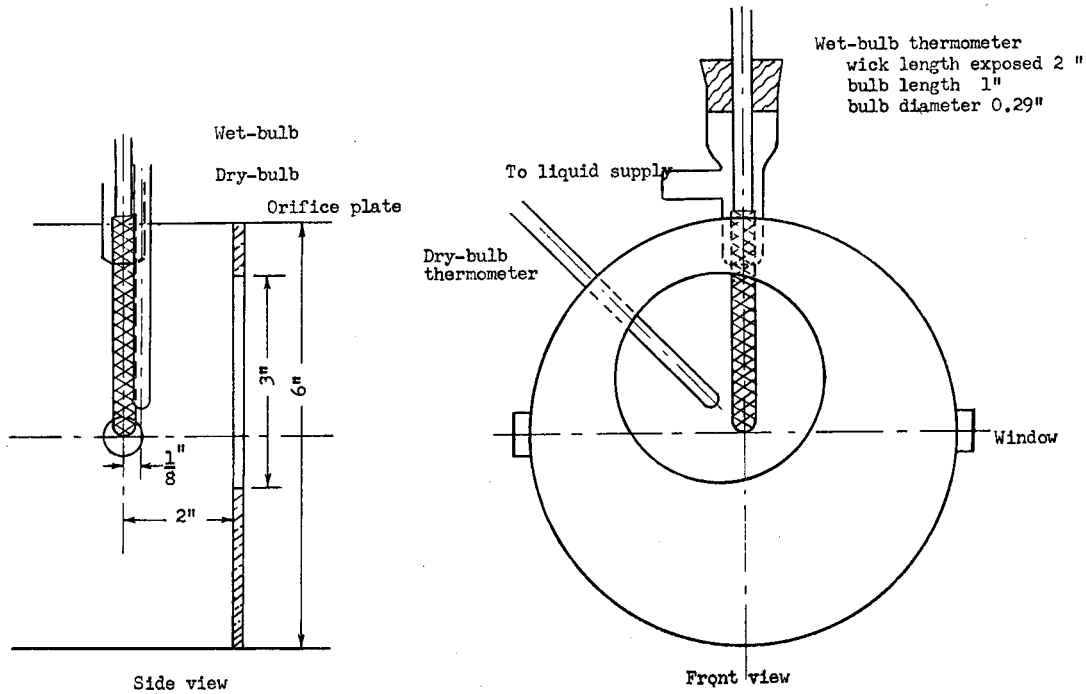

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Fig. 3. Wet- and dry-bulb thermometer arrangement.

calibrated to the nearest 0.01°C by the National Bureau of Standards and the calibrations checked against another NBS-calibrated thermometer as well as at the freezing and boiling points of water and the sodium sulfate triple point. The wet-bulb wick was made from a cotton sleeve which was carefully sewn closed at one end to fit snugly over the thermometer. The wick was changed periodically with no apparent effect on the temperature readings. Windows in the ducting were provided at the thermometer position so that positioning and wetting of the wick could be observed during operations. The windows were small enough so that they would not necessitate a radiation correction. The ducting at this point, as well as the previous six feet of run, was covered with a 1-inch-thick layer of magnesia insulation. The water supply for the wick was a 2-liter bottle mounted above the equipment with the water lead passing around the insulation to preheat the water.

Because the accurate determination for a stem correction of the air envelope surrounding the thermometer stems was difficult; because this stem correction for many of the runs would have exceeded 1°C; and because the temperatures, particularly of the wet bulb, must be accurate to within ± 0.04°F, it was felt that this uncertain stem correction must be eliminated. To assure total immersion of the thermometers, insulated glass tubes with entrance and exit ports at either end were placed over the length of the protruding thermometer stems and controlled-temperature heated air was blown through the annulus to maintain the temperature of the surrounding air envelope at the same level as that which the thermometers were reading. Thermocouples were inserted into the annulus and calibrated with position of the stem so as to insure the elimination of the need for stem correction.

A sampling tap was placed in the gas stream just ahead of the wet-and-dry-bulb assembly and led through copper tubing to the sampling oven (Fig. 4) and thence to a condenser and vent. The connecting tubing between the main gas stream and the sampling bulbs was controlled, electrically heated, and lagged with magnesia insulation. The sampling oven was electrically heated and well insulated. The oven incorporated

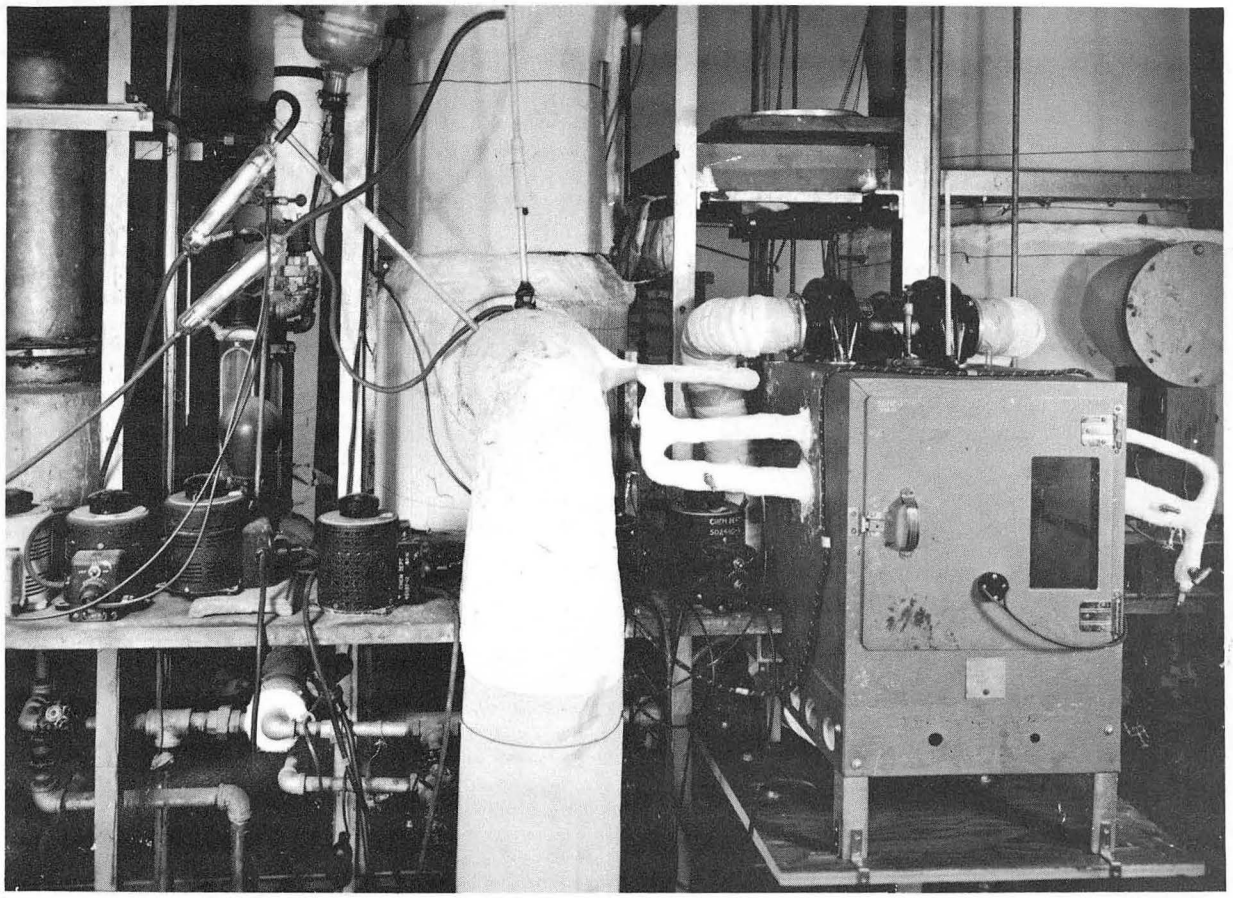

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Fig. 4. Thermometer and sampling assemblies.

a high-velocity re-circulation blower, baffled to insure constant temperature throughout.

In addition to the main sample line, the oven was provided with a sample-bypass line through it. To determine gas temperatures, thermocouples were placed in the sample line just before and just after the sample bulb. A pressure tap was placed in the sample line near the bulb to determine bulb pressures. The sample bulbs were of Pyrex of approximately 500 cc capacity, with stopcocked entrance and exit tubes, (Fig. 5).

The thermocouples used in determining all temperatures were calibrated against a Bureau of Standards thermometer. The emf's were read on a Leeds and Northrup K2 potentiometer.

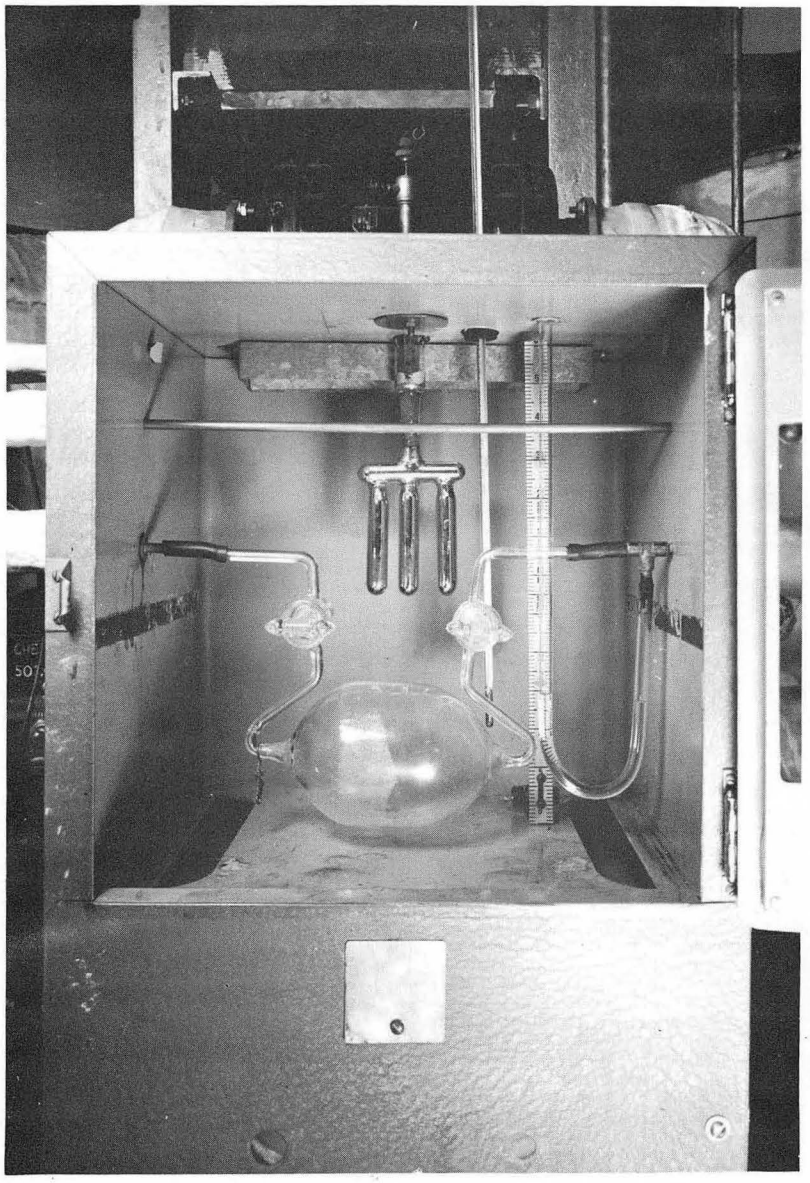

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Fig. 5. Sample bulb and oven.

PROCEDURE

For low-water-vapor mole-fraction determinations, at least 3 hours were required between equipment startup and first runs to obtain steady-state conditions. The high-mole-fraction runs took up to twice that period. Subsequent changes took from 1 to 3 hours to achieve steady state. This state was indicated when wet- and dry-bulb temperatures were constant, the temperature of gas surrounding the thermometers was at the wet- and dry-bulb temperatures, and the entrance and exit gas temperatures at sample bulb were equal to oven temperature. The wet-bulb wick was fed water at a rate to maintain a pendent drop at its tip; however, a slight overfeeding produced no observable temperature change. Following each run, the water supply was shut off and the bulb allowed to dry off. The temperature in all cases remained constant for several minutes before beginning to rise.

A sample bulb, which previously had been cleaned, degreased, and carefully calibrated as to volume, was evacuated to a pressure of less than 100 microns. It was then weighed on a sensitive balance to the nearest 0.1 milligram, with care being taken to allow no grease from the hands to contaminate it. Cleaned bulbs were kept in a desiccator except during the actual time of determinations. When steady-state had been achieved in the apparatus, the bulb was inserted into the closed sample line in the temperature-controlled oven and allowed to reach equilibrium temperature. The oven temperature was always set above 100°C. During this time sample gas was being drawn from the main gas stream through the bypass line in the oven. Gas flow was caused by a positive pressure in the apparatus and was accelerated by the sealed condenser ahead of the sample vent.

When sufficient time had passed to achieve equilibrium temperature, the bypass line was closed and the sample line opened, allowing the gas sample to continuously flow through the bulb. The sample-line temperatures had been in agreement with the oven temperature for sufficient time to completely replace the gas in the bulb. With all the temperatures of the equipment remaining at steady state during this

period, the sample-entrance stopcock was closed and the exit from the oven closed, the pressure in the bulb read by a butyl phthalate manometer, and the exit stopcock closed.

The sealed sample bulb was removed from the oven and allowed to attain room temperature in a desiccator in the balance section. The bulb was then reweighed and the mole fraction of water vapor in the sample could be calculated from the weight, pressure, and temperature readings and the known volume of the bulb. Because this method of determining mole-fraction water vapor in the sample was dependent upon small differences in large weights, it was usable in the low-mole-fraction water-vapor runs only. A more novel and refined method was used in the high-mole-fraction runs, in which a small error in mole fraction could cause a serious error in the calculated partial-pressure driving force for mass transfer.

A covered beaker of distilled water was kept in the balance compartment. When the bulb had reached equilibrium temperature with the balance compartment, one stem of the bulb was inserted into the beaker of water, and water was allowed to enter, replacing the vacuum created by condensing the water vapor in the sample and cooling the air in the bulb. To assure accurate filling, since temperatures would be slightly changed with pressureization and further condensation in the bulb, the bulb was allowed to again reach equilibrium temperature, reopened to the water, and then weighed. This was continued until no change in weight was observed upon subsequent fillings. Barometric pressure was taken and thus the volume of water condensed could be calculated, correcting for the air dissolved in the water (water previously saturated with air), the water vapor in the remaining air (assumed saturated), and the change in the bulb's volume with temperature. The difference between level of water in the bulb and in the filling beaker was also measured to correct for the difference in final pressure between the bulb and the atmosphere.

RESULTS

Three calibration runs were made in which the bulbs were filled with a sample of dried air (passed through magnesium perchlorate) and the procedure for a run determination was followed. The volume of air in the bulbs, determined in this manner, agreed to within 0.065% of the volume as determined by calibration filling with water and weighing. Since the method, as employed, actually determines volume of air in the bulb, these dry runs should show the maximum error compared with high-humidity runs in which the volume of air was smaller. Three calibrations of volume filling with water agreed to within 0.016 cc or 0.003%.

Four series of determinations had been made prior to the one reported in this paper. In these earlier series, various other methods were used to determine temperatures and absolute humidities. A single thermocouple — and later a four-unit thermopile — was used for wet-bulb temperature measurements, but temperatures were slightly erratic because of local hot spots where the thermocouple tips extended into the wick. Measured volumes of gas samples were drawn through condensers, silica gel, Drierite, and magnesium perchlorate by displacement of water and mercury; but it was soon seen that as p_f's decreased, the size of the water-removal devices increased drastically, as did the error possibilities. Also it was seen that, as p_f was reduced below 0.6, extreme accuracy was needed in determining wet-bulb temperatures and absolute humidities, because the partial-pressure driving force was apparently tending towards zero. The results from these four previous series in general agree with the data reported in this paper.

The determined values of the psychrometric ratio,

$$\frac{k_g M_m p_f}{h} C_p \quad \text{vs.} \quad p_f ,$$

as obtained in this work, are plotted in Fig. 6. The best curve has been drawn through the data and has an average deviation of 4.5%. It is compared with the theoretical curve of psychrometric ratio versus p_f.

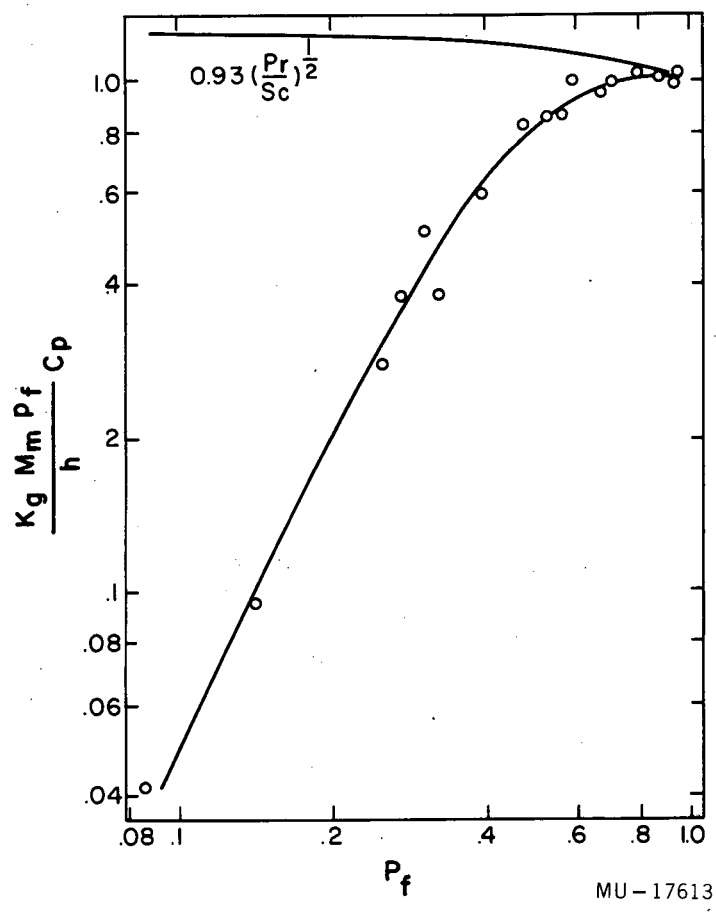
$0.93\left(\frac{Pr}{Sc}\right)^{\frac{1}{2}}$
$\frac{K_g M_m P_f}{h} C_p$
1.0
.8
.6
.4
2
.1
.08
.06
.04
.08
.1
.2
.4
.6
.8
1.0
P_f
MU-17613

Fig. 6.

The theoretical inclusion of the first power of the film-pressure factor in the psychrometric ratio becomes seriously in error when the film-pressure factor departs very far from unity. The theoretical ratio would be more nearly correct if the zero power of the film-pressure factor were used, but it is apparent that the psychrometric ratio does not depend upon a simple power function of the film-pressure factor. A plot of psychrometric ratio vs. p_f is shown in Fig. 7. Because of a complete absence of data on other than the air – water-vapor systems with a film-pressure factor other than approximately unity, it is difficult to ascribe the deviation from previous theory to physical properties of the system with changing film-pressure factor.

Two groups of researchers have performed experiments on the air – water-vapor system at film-pressure factors departing appreciably from zero. Cairns and Roper did work on heat and mass transfer at high humidities in a 0.90-inch i.d. by 37.25-inch-long wetted-wall column.[11] They concluded that the mass-transfer coefficient was a function of the -0.83 power of p_f and the heat transfer coefficient was a function of the -0.73 power of p_f. Although they did not combine their equations or data into the psychrometric ratio, if this is done their correlations yield a 0.10-power dependency. Their individual data have been recalculated into the psychrometric group

$$\frac{k_g M_m p_f}{h} C_p ,$$

and plotted versus p_f in Fig. 8. The dissimilarity between intercept at $p_f = 1$ and this work is caused by their changed Pr/Sc ratio at the much higher temperature at which they worked. From this plot of their data, no true correlation is apparent.

Their work, although carefully done, suffers from the wetted-wall column's lack of steady-state conditions. An over-all steady-state may be achieved, but driving forces vary drastically from one end of the column to the other. Their sample run calculations show, as an example, that their temperature driving force for convective heat transfer varies from 254 F^o (difference) at the bottom to 613 F^o (difference) at the top

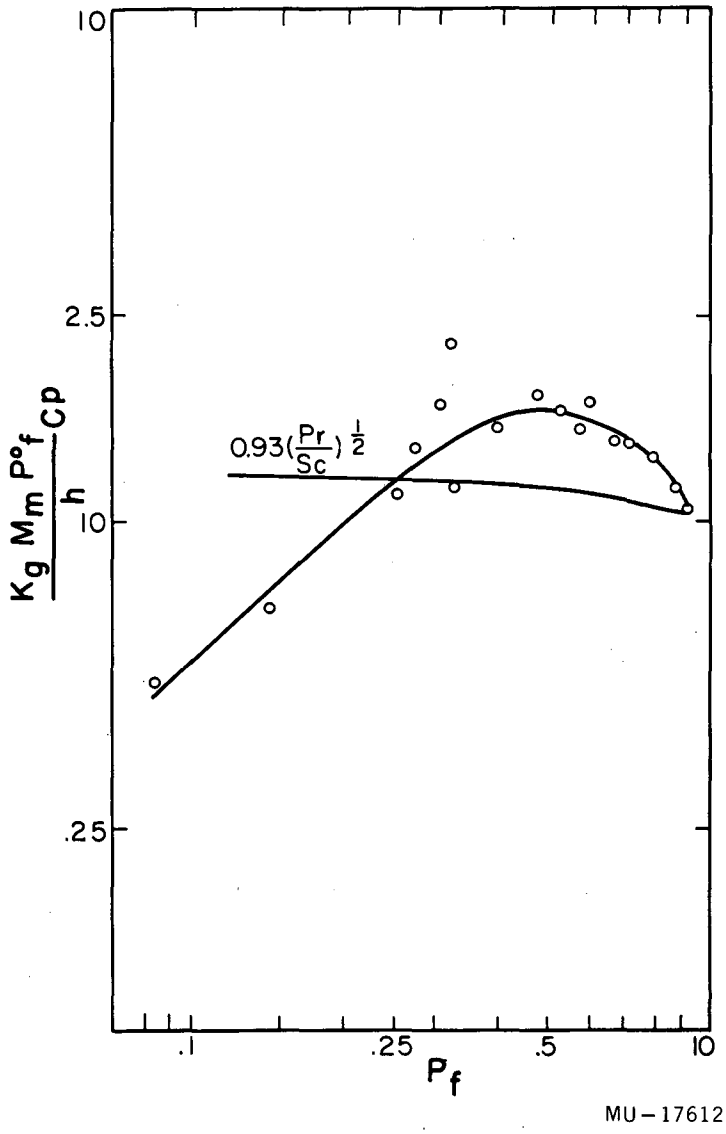
10
2.5
1.0
.25
$\frac{K_g M_m P^o_f C_p}{h}$
$0.93(\frac{Pr}{Sc})^{\frac{1}{2}}$
.1
.25
.5
10
P_f
MU-17612

Fig. 7.

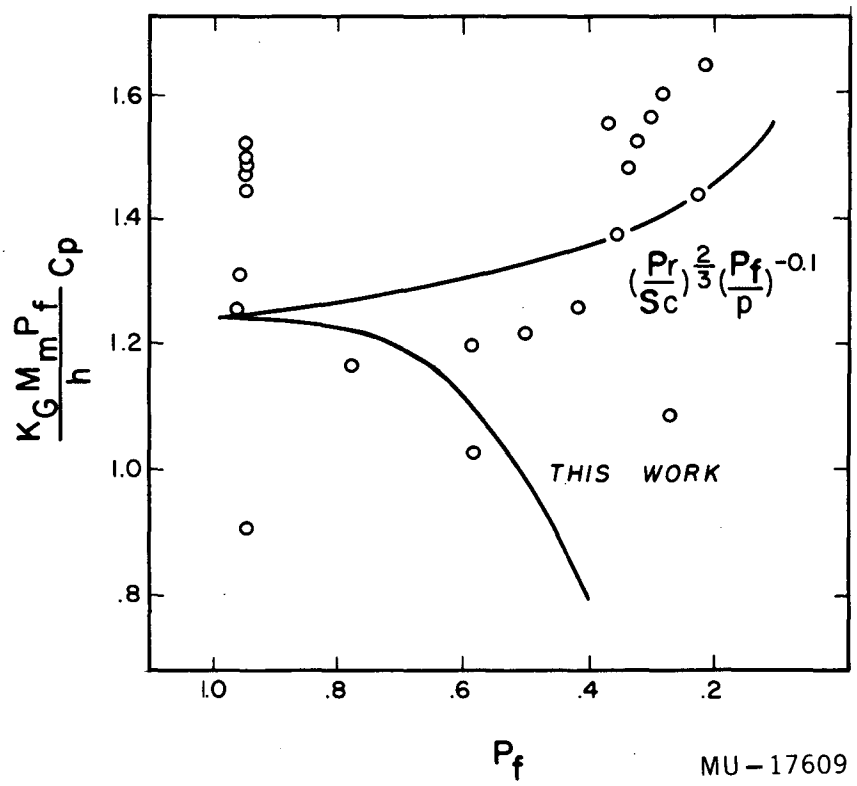
1.6
1.4
1.2
1.0
.8
1.0
.8
.6
.4
.2
$\frac{K_G M_m P_f}{h} C_p$
P_f
$(\frac{Pr}{Sc})^{\frac{2}{3}}(\frac{P_f}{p})^{-0.1}$
THIS WORK
MU-17609

Fig. 8.

and their radiation "correction," averaged at 11%, varies elevenfold from top to bottom. For the delicacy necessary in psychrometric work, their wetted-wall column is insensitive. They have numerous points in the region $p_f = 0.95$ where their log-mean-temperature driving force is comparatively low, 120°, and correspondingly more nearly approaching a true steady-state condition. These points have a spread of 50%. Because of their generous spread and lack of true correlations, no weight can be placed on their 0.10-power dependency.

The other team of researchers, Auberry and Griffiths, performed a great number of experiments with a wet-and-dry-bulb apparatus, determining their humidities by a dew-point analyzer.[5] Their reported data show relative humidities only. The several coupled temperature and absolute humidities given in their discussion do not agree with any water-vapor-pressure table known to this writer. These two points, with vapor pressure at freezing and boiling, were used as a basis of calculating their assumed vapor-pressure curve by the method of Birge.[9] The plot of their data as

$$\frac{k_g M_m p_f}{h} C_p \quad \text{versus} \quad p_f$$

is shown in Fig. 9 on rectangular coordinates so as to show more of their points. All eleven of their points with a partial-pressure driving force of less than 4 mm have been omitted because of extreme scatter. Indeed, some of their runs indicate a negative driving force, caused, no doubt, by the experimental inaccuracy of their dew-point apparatus.

The curve as determined in this work is superimposed on their data for comparison. While the data from Auberry and Griffiths are of the shotgun variety, they do show a general trend which is in agreement with this work. Theirs is the only comparable psychrometric study in the lower film-pressure-factor regions found by this author.

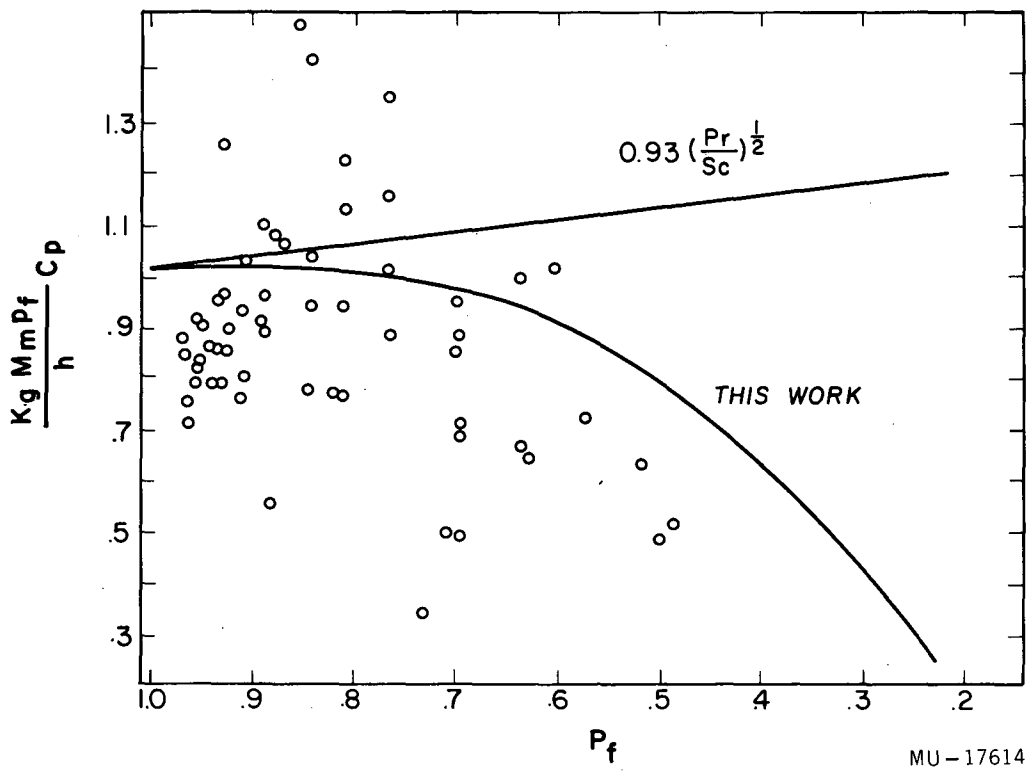
1.3
1.1
.9
.7
.5
.3
$\frac{K_g M_m P_f}{h} C_p$
$0.93 \left(\frac{Pr}{Sc}\right)^{\frac{1}{2}}$
THIS WORK
1.0
.9
.8
.7
.6
.5
.4
.3
.2
P_f
MU-17614

Fig. 9.

DISCUSSION AND CONCLUSIONS

The results of this investigation show that for the air – water-vapor system, the theoretical inclusion of the film-pressure factor to the first power is in error. The dependency is more nearly of the zero power, but no simple power function will explain the data. The experimental deviation from previous theory, however, is negligible in the region normally used for psychrometric determinations — namely, where the film-pressure factor differs from unity by less than 10%. It seems that determinations of psychrometric ratios at low film-pressure factors on other than the air – water-vapor system are necessary to establish a general dependency correlation. When the film-pressure factor differs seriously from unity, it is suggested that the psychrometric ratio from Fig. 7 be used.

The third objective of this investigation, the experimental determination of the effect of mass transfer on the heat-transfer coefficient, could not be satisfactorily achieved because of the serious deviation from previous theory of the results with p_f. It is also seen that to vary any corrective term for changing the heat-transfer coefficient with mass transfer sufficiently to indicate a trend would take an exceptionally high rate of mass transfer and corresponding high temperature differential between the vaporizing liquid and the main stream. This is normally impractical in wet-and-dry-bulb psychrometric determinations and necessitates forced mass transfer such as found in the work of Mickley.[26] It is felt that Mickley and his associates have adequately demonstrated the validity of the Θ correction as originally postulated by Ackermann.

Several of the corrective terms devised by other researchers have been calculated for the various determinations of this investigation. The γ function of Lynch and Wilke, as calculated for this investigation, varies from 0.943 at $p_f = 0.941$ to 0.989 at $p_f = 0.0865$. These deviate slightly from Θ as shown by Olander. Calculations of the modified γ for the various determinations of this investigation show no deviation from the value of Θ in the first three significant figures.

The β-corrective term of Beddingfield and Drew has been calculated for the determinations reported in this paper. The quantity β ranges from 1.011 to 1.144, in general increasing as film-pressure factor decreases. There is a lack of good experimental evidence for the validity of the β correction.

The Ackermann correction as extended by Mickley and associates, Θ, has excellent experimental verification over a wide range of rates of mass transfer. Values of ϕ as high as 2.7 with corresponding Θ of 0.2 were found by using forced mass transfer. By definition Θ varies between 1.0 and 0.0 only. The term γ, which is equal to $1 - \phi$, would have yielded a value of -1.7 at the above value of ϕ. The modified γ, equal to $1 - \phi/2$, would have yielded a value of -0.35. The Θ term thus is used in this paper.

It is suggested that the coefficient of 0.93 be used for the Sc, Pr correlation when the Θ corrective term is used instead of the value of 0.91 as suggested by Lynch and Wilke when they use their γ term. Recalculation of the data of Lynch and Wilke for the air — water-vapor system, using Θ instead of γ, gives excellent agreement with the value of 0.93.

The deviation of the results of this work from the theory of psychrometry using film-pressure factor to the first power is negligible when the film-pressure factor is near 1. Even for general psychrometric work at lower film-pressure factors, this deviation is not serious. If, however, precision work is desired at low film-pressure factors in determining concentrations of water vapor in air, the deviation must be taken into account. If, as in Run 164 with $p_f = 0.0865$, the wet- and dry-bulb measurements were used to determine mole fraction of water vapor in the gas stream on the basis of $0.93 \left(\frac{Pr}{Sc}\right)^{1/2}$, the result would be 0.923 instead of the actual value of 0.902. This 2% change would be 20% if based on air mole fraction. If the concentration were known, the expected wet-bulb temperature would be 207.60 °F instead of the actual 207.90°F. Similar calculations for Run 156 at $p_f = 0.569$ show a calculated mole fraction of water vapor of 0.426 instead of the actual 0.421 and a temperature of the wet bulb of 172.90°F instead of 173.34°F.

The major results of this work indicate that, since the heat-transfer coefficient has been corrected for simultaneous mass transfer by Θ and it should not vary with p_f, then $k_g p_f$ is not proportional to $(1/Sc)^{1/2}$ nor $(1/Sc)^{2/3}$. A closer approximation is $k_g p_f^0 \alpha (1/Sc)^{1/2}$, but no simple power function of p_f agrees with the data. Additionally, errors in determining absolute humidity are not great regardless of whether the theoretical Sc, Pr expression is used or whether Fig. 7 is used. Also, simple wet-bulb thermometry is not a reliable method of determining the role of p_f in mass transfer — extreme accuracy is required.

Schrage gives a theoretical approach to the departure from equilibrium conditions of a liquid surface undergoing mass transfer.[35] Calculations of the results reported in this paper were made by his method and, while a departure from equilibrium is shown, it is relatively small. Calculations of Run 164, used as a sample for calculations throughout this paper, show that the departure from equilibrium conditions at the wet bulb is only 0.28%, insufficient to influence the results to any great degree.

Studies of mass transfer from cylinders[23] and of heat, mass, and momentum transfer from drops[34] show that the majority of the transfer takes place in the region of the stagnation point and the turbulent wake, with much less taking place on the sides. This might well lead to temperature gradients around the periphery of a wet bulb. However, the basis for the heat, mass, and momentum analogy has been equal film thickness or resistance to transfer at a surface. The majority of experimental work has shown this to be substantially true, with some slight modification. Ranz, using 0.5-mil thermocouples to measure the temperature profile around an evaporating drop of water, found that as he approached the drop surface the isothermal profiles assumed more nearly the shape of the drop and equaled it at the drop surface.[34] He found also that the position of a thermoelement junction anywhere in the drop had no apparent effect on the indicated temperature and, since the thermal conductivity of liquid water is many times that of air, he assumed: (a) the temperature

at the surface of the drop was everywhere the measured value, and (b) the partial pressure of water vapor at the surface was everywhere equal to the saturation partial pressure of water vapor at that temperature. The diameter of the drops he worked with ranged from 0.05 to 0.11 cm. In lieu of other experimental work in this field, his assumptions are used in the work reported herein.

Since it was seen that extreme precision was needed in determining the true wet-bulb temperatures, because of the small partial-pressure driving forces involved, all determinations in this work in which the wet-bulb temperature fluctuated more than 0.02 C^{o} (difference) were discarded. The majority of the determinations reported had no fluctuation in temperature. The only other determinations discarded were several in which the stopcock seal on the sample bulb was broken during a run.

ACKNOWLEDGMENT

The author wishes to thank Professor C. R. Wilke for his long-standing patience and helpful suggestions in the supervision of this investigation, and the Department of Chemistry and Chemical Engineering and the Lawrence Radiation Laboratory for the use of their research facilities.

He wishes to additional thank Drs. Edward Lynch and Walter Dong for their assistance and friendly advice; Jane and Robert Waite; his wife, Carolyn; UCRL and Public Law 550.

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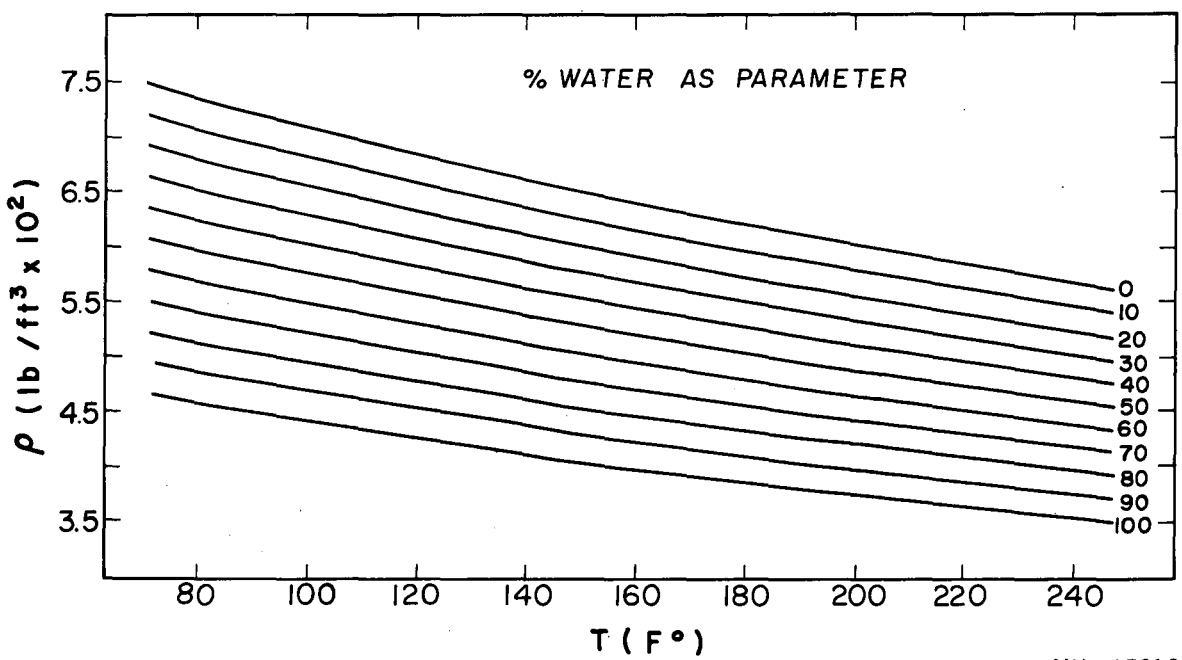
APPENDIX

PROPERTIES OF THE AIR – WATER VAPOR SYSTEM

Physical and thermodynamic properties of the air – water vapor system are summarized in Figs. 10 through 14 and Tables I through V.



MU-17616

Fig. 10

Table I

Calculated density of the air-water vapor system (in $lb/ft^2 x10^2$)											
% Water Vapor Temp. (°F)	0	10	20	30	40	50	60	70	80	90	100
68°	7.520	7.236	6.951	6.667	6.382	6.098	5.813	5.529	5.244	4.960	4.675
86°	7.272	6.997	6.722	6.447	6.172	5.897	5.621	5.346	5.071	4.796	4.521
104°	7.039	6.773	6.507	6.240	5.974	5.708	5.442	5.176	4.909	4.643	4.377
122°	6.820	6.562	6.304	6.046	5.788	5.531	5.273	5.015	4.757	4.499	4.241
140°	6.615	6.365	6.115	5.865	5.615	5.365	5.114	4.864	4.614	4.364	4.114
158°	6.422	6.179	5.936	5.694	5.451	5.208	4.965	4.722	4.480	4.237	3.994
176°	6.240	6.004	5.768	5.532	5.296	5.016	4.825	4.589	4.353	4.117	3.881
194°	6.068	5.838	5.609	5.380	5.150	4.921	4.692	4.462	4.233	4.003	3.774
212°	5.905	5.682	5.459	5.235	5.012	4.789	4.566	4.343	4.119	3.896	3.673
230°	5.751	5.534	5.316	5.099	4.881	4.664	4.447	4.229	4.012	3.794	3.577
248°	5.605	5.393	5.181	4.969	4.757	4.546	4.334	4.122	3.910	3.698	3.486

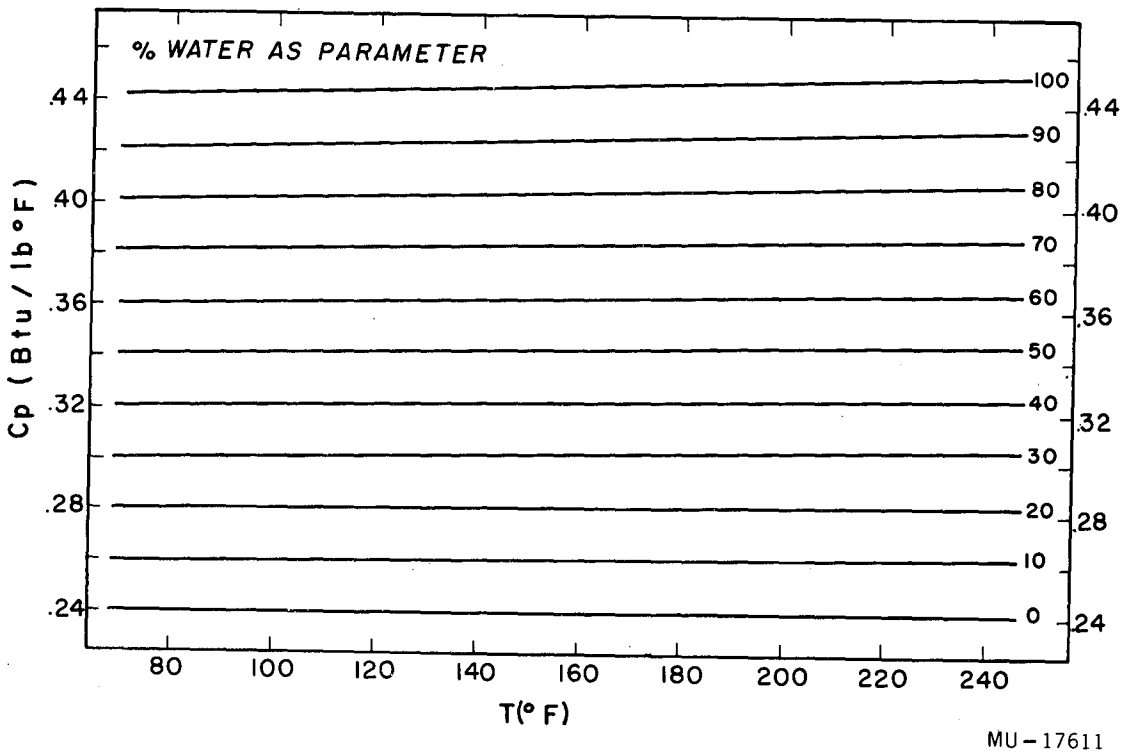

% WATER AS PARAMETER
Cp (Btu / lb °F)
.44
.40
.36
.32
.28
.24
100
90
80
70
60
50
40
30
20
10
0
80
100
120
140
160
180
200
220
240
T(° F)
MU-17611

Fig. 11.

Table II

Calculated heat capacity of the air-water vapor system (BTU/lb °F)											
% Water Vapor Temp. (°F)	0	10	20	30	40	50	60	70	80	90	100
68°	.2401	.2603	.2805	.3007	.3209	.3411	.3612	.3814	.4016	.4218	.4420
86°	.2402	.2605	.2808	.3010	.3212	.3416	.3619	.3822	.4024	.4227	.4430
104°	.2404	.2608	.2811	.3015	.3218	.3422	.3626	.3829	.4033	.4236	.4440
122°	.2406	.2610	.2815	.3019	.3224	.3428	.3632	.3837	.4041	.4246	.4450
140°	.2408	.2613	.2818	.3024	.3229	.3434	.3639	.3844	.4050	.4255	.4460
158°	.2410	.2616	.2833	.3028	.3234	.3440	.3646	.3852	.4058	.4264	.4470
176°	.2412	.2619	.2826	.3032	.3239	.3446	.3653	.3860	.4066	.4273	.4480
194°	.2414	.2622	.2829	.3037	.3244	.3452	.3660	.3867	.4075	.4282	.4490
212°	.2416	.2624	.2833	.3041	.3250	.3458	.3666	.3875	.4083	.4292	.4500
230°	.2418	.2627	.2836	.3046	.3250	.3464	.3673	.3882	.4092	.4301	.4510
248°	.2419	.2629	.2839	.3049	.3259	.3470	.3680	.3890	.4100	.4310	.4520

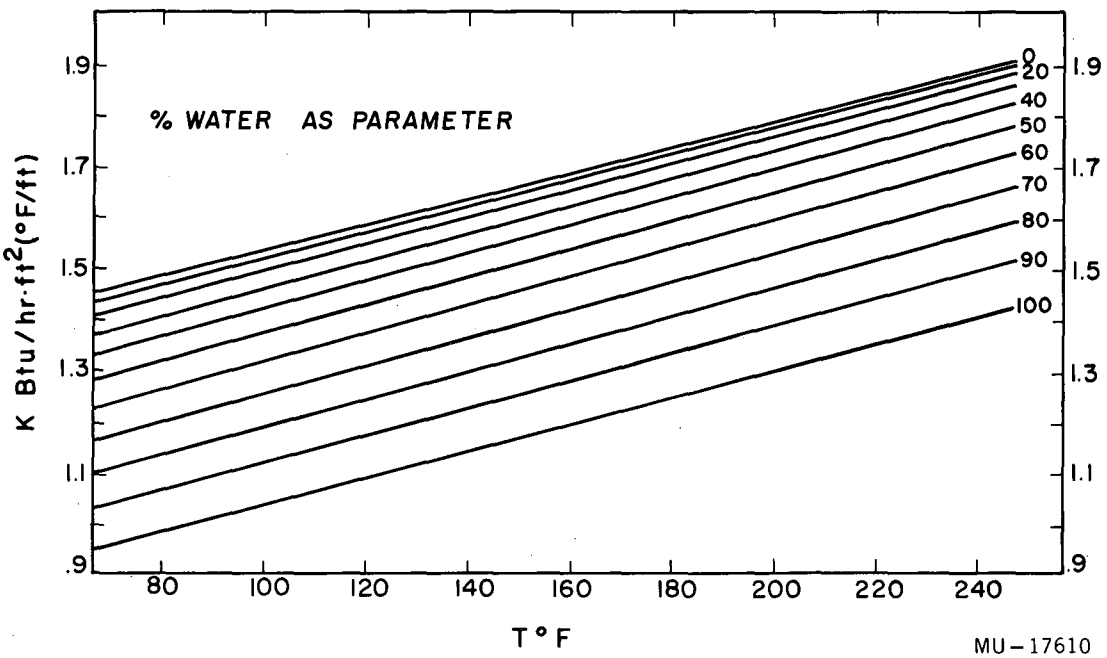

% WATER AS PARAMETER
K Btu/hr·ft²(°F/ft)
1.9
1.7
1.5
1.3
1.1
.9
80
100
120
140
160
180
200
220
240
T °F
0
20
40
50
60
70
80
90
100
MU-17610

Fig. 12.

Table III

Calculated thermal conductivity for the air-water vapor system
(BTU/hr ft^2 ($^\circ$F/ft)x10^2)

% Water Vapor Temp. ($^\circ$F)	0	10	20	30	40	50	60	70	80	90	100
68°	1.456	1.437	1.410	1.375	1.334	1.286	1.231	1.171	1.106	1.035	.9595
86°	1.504	1.486	1.460	1.426	1.385	1.338	1.283	1.223	1.157	1.085	1.008
104°	1.550	1.534	1.509	1.476	1.435	1.387	1.338	1.272	1.205	1.132	1.053
122°	1.596	1.581	1.557	1.525	1.485	1.437	1.382	1.321	1.235	1.179	1.099
140°	1.642	1.628	1.606	1.574	1.535	1.487	1.433	1.371	1.302	1.227	1.146
158°	1.687	1.675	1.653	1.623	1.584	1.537	1.482	1.420	1.351	1.274	1.192
176°	1.732	1.721	1.701	1.671	1.633	1.587	1.532	1.470	1.400	1.323	1.239
194°	1.776	1.767	1.748	1.719	1.682	1.636	1.582	1.519	1.449	1.371	1.286
212°	1.821	1.814	1.796	1.769	1.732	1.686	1.632	1.570	1.499	1.421	1.335
230°	1.863	1.857	1.841	1.815	1.779	1.734	1.680	1.618	1.548	1.469	1.382
248°	1.907	1.903	1.888	1.863	1.828	1.784	1.731	1.668	1.597	1.518	1.431

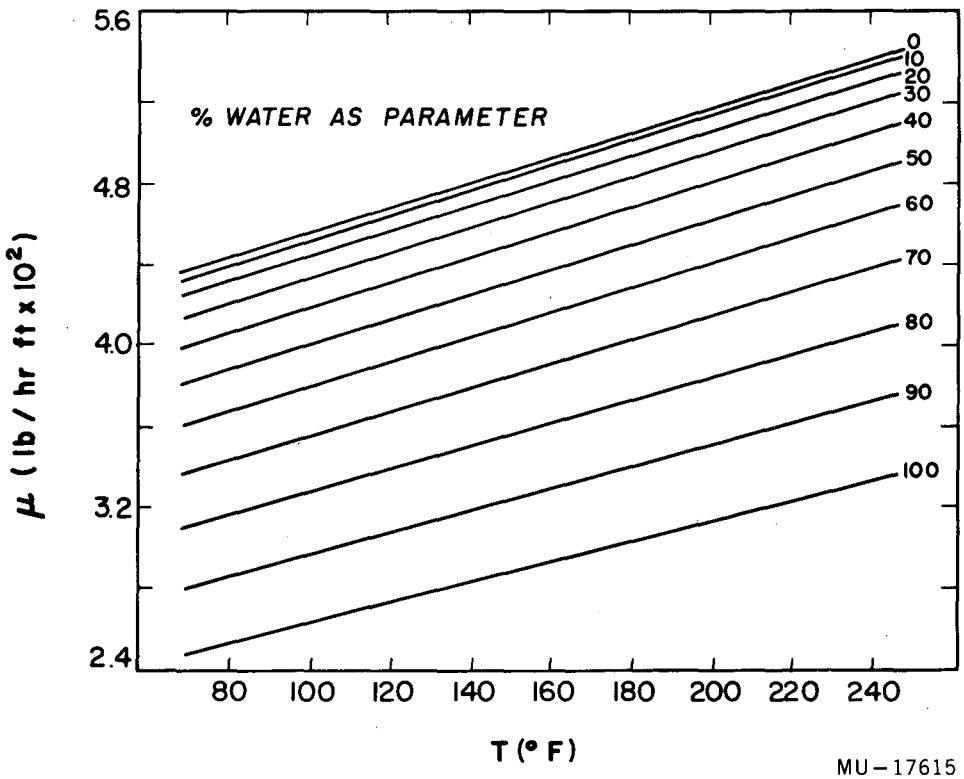

% WATER AS PARAMETER
0
10
20
30
40
50
60
70
80
90
100
5.6
4.8
4.0
3.2
2.4
μ (lb / hr ft x 10²)
80
100
120
140
160
180
200
220
240
T (° F)
MU-17615

Fig. 13.

Table IV

Calculated viscosity for the air-water vapor system (lb/ft hrx10^2)											
% Water Vapor Temp. (°F)	0	10	20	30	40	50	60	70	80	90	100
68°	4.356	4.311	4.232	4.120	3.977	3.802	3.597	3.360	3.093	2.794	2.464
86°	4.475	4.418	4.350	4.238	4.093	3.916	3.708	3.468	3.196	2.893	2.556
104°	4.588	4.535	4.469	4.358	4.214	4.036	3.825	3.581	3.303	2.993	2.647
122°	4.702	4.661	4.584	4.473	4.327	4.148	3.934	3.688	3.406	3.090	2.739
140°	4.813	4.774	4.699	4.588	4.442	4.262	4.047	3.797	3.511	3.190	2.831
158°	4.922	4.883	4.807	4.695	4.549	4.367	4.150	3.898	3.610	3.285	2.923
176°	5.031	4.994	4.919	4.808	4.661	4.479	4.260	4.005	3.712	3.380	3.013
194°	5.138	5.104	5.033	4.924	4.777	4.594	4.373	4.114	3.816	3.480	3.102
212°	5.247	5.212	5.139	5.029	4.881	4.697	4.474	4.214	3.914	3.575	3.194
230°	5.348	5.314	5.241	5.131	4.983	4.798	4.574	4.311	4.009	3.667	3.282
248°	5.452	5.423	5.354	5.247	5.100	4.914	4.689	4.423	4.115	3.765	3.371

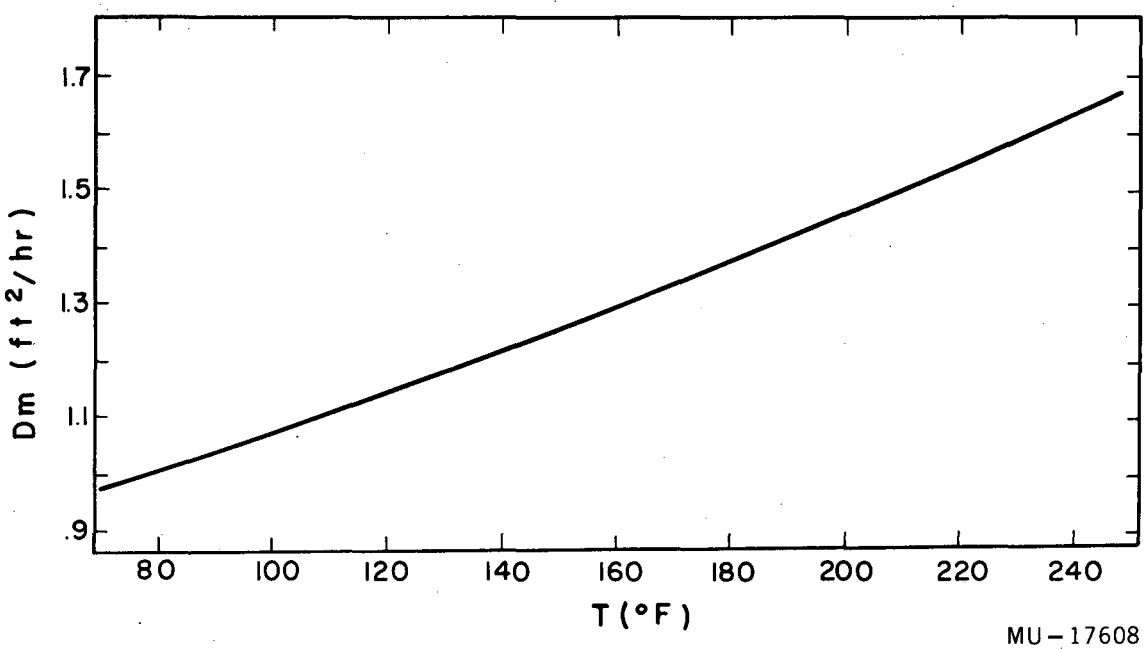

1.7
1.5
1.3
1.1
.9
Dm (ft 2/ hr)
80
100
120
140
160
180
200
220
240
T (°F)
MU-17608

Fig. 14.

Table V

Calculated diffusivity for the air-water vapor system(ft^2/hr)		
Temperature (^oF)	Calculated D	D from smoothed graph
68°	0.966	0.966
86°	1.027	1.025
104°	1.095	1.090
122°	1.160	1.155
140°	1.230	1.220
158°	1.295	1.290
176°	1.367	1.365
194°	1.444	1.440
212°	1.513	1.515
230°	1.585	1.590
248°	1.670	1.670

SAMPLE CALCULATIONS

A. Flow Rates

Calculation of the gas-flow rate was made by means of equations taken from the Fluid Meters Report.[2] The weight rate of flow through a nozzle is given as

$$w = 0.5250\ D_2^2\ Y\ C \sqrt{\rho_1}\ \Delta P_t \quad \text{lb/sec},$$

where

D_2 = nozzle diameter in inches,

Y = compressibility factor $1 - 0.41(p_1 - p_2)/p_1 \frac{C_p}{C_v}$,

C = discharge coefficient (see Fig. 9),

ρ_1 = density at upstream conditions,

ΔP_t = impact head minus pressure head at nozzle discharge in lb/in^2.

The values of C were obtained from the work of Bean, Buckingham, and Murphy.[7] For Run 164, which will be used as the sample throughout, the nozzle equation can be simplified to

$$w = 24{,}542\ C \sqrt{P_2\ \Delta P_t / T_1} \quad \text{lb/hr},$$

where $T_1 = {}^{\circ}K$,

$$w = 24{,}542 \times 0.985\ \frac{15.15 \times .03831}{400.0},$$

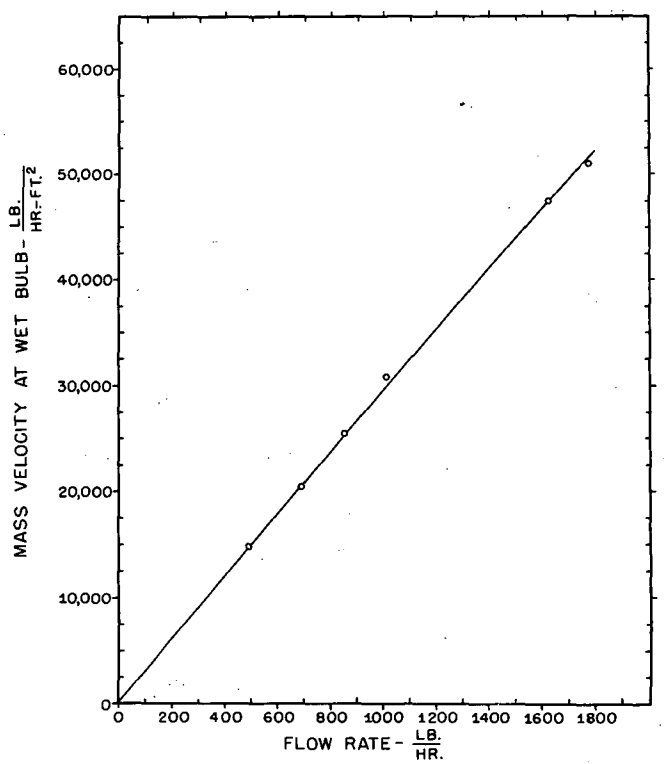
$$= 24{,}170 \times 0.03835 = 927 \text{ lb/hr}.$$

The gas mass velocity was obtained from a calibration chart prepared by Lynch (Fig. 15). The chart was made by making pitot traverses across the wet bulb and plotting the average against the mass velocity measured at the nozzle. For this run the value of G at the wet bulb was 27,200 lb/hr ft^2.

B. Heat-Transfer Corrections

Calculations of the heat-transfer coefficients for the two thermometers were made by the equation[29]

$$h_c = \frac{k}{d}\ 0.26\ Re^{0.6}\ Pr^{0.3},$$



MU-4857

Fig. 15. Average velocity at thermometer vs flow rate.

where d is the diameter of the thermometer. The diameter of the wet bulb was 0.29 inch and the dry bulb 0.23 inch. For the wet bulb,

$$Re = \frac{(0.0242)(27,200)}{(0.0360)} = 18,300, \quad Pr = \frac{(0.431)(0.0360)}{(0.0144)} = 1.077;$$

therefore

$$h_c = \frac{(0.0144)}{(0.0242)} (0.26)(18,300)^{0.6} (1.077)^{0.3}$$

$$= (0.155)(364)(1.022) = 56.4.$$

For the dry bulb $h_c = 60.9$. The radiation coefficient h_c was calculated for both thermometers by using the Stefan-Boltzman relationship with emissivities of 0.94 for the thermometer and 0.90 for the wetted wick. The radiation coefficients were 2.250 for the thermometer and 1.984 for the wick. By application of this correction the temperature of the gas was found to be 241.69°F (data value 241.57).

C. Psychrometric Determination

The weight of water entering the sample bulb at 22.54 °C was 465.0211 grams, corresponding to a volume of 466.107 cc. Barometric pressure was 749.8 mm and there was a water head of 5.9 inches in the flask after replacement, resulting in a pressure in the bulb of 738.8 mm. The calibrated volume of the sample bulb at this temperature was 507.314 cc, so that the remaining gas volume, composed of air and water vapor, was 41.207 cc. On the assumption that the air was saturated with water vapor, the volume of air in the bulb was 40.064 cc. The volume of sample gas in the bulb at 738.8 mm pressure, with correction for thermal expansion of the bulb at sampling temperature of 101.53°C, difference in sampling pressure, and gas-temperature correction assuming perfect gas law, is 407.07 cc. The mole fraction of water vapor in the sample is thus 0.90158.

The vapor pressure (Perry[29]) at the wet-bulb temperature corresponds to 700.13 mm Hg. The water-vapor pressure of the main stream, corrected for pressures at the wet bulb, is 682.59 mm. Thus the partial-pressure driving force is 17.54 mm or 0.02308 atmospheres. The latent heat of vaporization (Perry[30]) at 207.90°F is 972.75 Btu/lb. The

film-pressure factor is 0.0865 and the mean molecular weight is 18.95. The Θ corrective term is 0.995.

The psychrometric ratio is determined experimentally by the equation

$$\frac{k_g M_m p_f}{h} C_p = \frac{(t_g - t_w)\left\{1 + \frac{h_r}{h_c}\left[\frac{t_s - t_w}{t_g - t_w}\right]\right\}}{(p_w - p)\lambda} \quad \frac{M_m p_f \Theta}{M_w} C_p$$

$$= \frac{(241.69-207.90)\left\{1+\frac{1.984}{56.4}\left[\frac{238.32-207.90}{241.69-207.90}\right]\right\}}{(.02308)\ 972.75}\frac{(18.95)(.0865)(.995)}{18}(.431)$$

$$= 0.0416.$$

D. Dry-Gas Determination

The dry-gas determination to check the accuracy of the method of sample analysis was performed in the same fashion as a psychrometric determination with dry gas in the main gas stream and a drying tube filled with magnesium perchlorate in the sample line ahead of the sample bulb. Dry-gas run No. 5, using the same bulb as was used in run No. 164, will be used as a sample for calculations.

The weight of water entering the sample bulb at 22.68°C was 90.1124 grams, corresponding to a volume of 90.326 cc. Barometric pressure was 745.8 mm, and there was a water head of 2.8 inches on the flask after replacement, resulting in a pressure in the bulb of 740.6 mm. The remaining gas volume, now comprised of air and water vapor, was 416.988 cc. The bulb was shaken thoroughly and sufficient time was allowed for the remaining air in the bulb to become saturated with water vapor, so that the volume of water vapor in the bulb was 11.635 cc, leaving 405.353 cc for volume of air. The volume of sample gas in the calibrated bulb at 740.6 mm, with correction for thermal expansion of the bulb at sampling temperature of 100.43°C, difference in sampling pressure, and gas temperature correction assuming perfect gas law, is

$$(508.245)\left(\frac{745.8}{740.6}\right)\left(\frac{295.84}{373.59}\right) = 405.297 \text{ cc.}$$

This is a difference of 0.056 cc or an error of 0.014%.

E. Physical Properties

In the calculations of physical properties for the Schmidt, Prandtl, and Reynolds numbers, all properties were evaluated at log mean film conditions. Results are found in Figs. 10-14 and Tables I-V.

Average values of the heat capacity and density were used. The data for heat capacities came from Keenan and Keyes,[19] and the data for densities from Perry.[31] The thermal conductivities were calculated from the data of Keyes[20] by the method of Lindsay and Bromley.[22] Gas-mixture viscosities were calculated from the data of Keyes by the method of Buddenberg and Wilke.[10] The diffusivity as a function of temperature was calculated by using the average of D as reported by Lee and Wilke,[21] by the method of Buddenberg and Wilke, with temperature corrected by Hirschfelder et al.[18] For run No. 164:

$$Pr = \frac{(0.431)(0.0360)}{(0.0144)} = 1.077; \quad Sc = \frac{(0.0360)}{(0.0378)(1.57)} = 0.607,$$

$$0.93 \left(\frac{Pr}{Sc}\right)^{1/2} = 1.213.$$

EXPERIMENTAL AND CALCULATED RESULTS

Results of measurements and computations are shown in Table VI.

CALIBRATION

Calibration of the flow nozzle used is given in Fig. 16.

Table VI

Experimental and calculated results for wet-and-dry-bulb thermometer measurements

Run No.	Wet bulb (°F)	Dry bulb (°F)	Wall (°F)	G (lb/hr ft^2)	p (atmos)	p_f (atmos)	θ[a]	$\frac{k_g M_m p_f}{h} C_p$	$0.93(\frac{Pr}{Sc})^{1/2}$
137	129.20	233.29	229.21	30,500	.03658	.8755	.978	1.009	1.049
139	193.51	239.47	235.31	24,000	.01875	.3256	.991	0.380	1.173
140	194.86	238.72	234.30	23,600	.01247	.3032	.991	0.510	1.185
141	196.86	238.81	233.13	23,200	.01442	.2727	.991	0.379	1.191
144	145.74	235.05	230.00	29,400	.02847	.7868	.981	1.041	1.067
145	188.62	240.41	237.13	24,600	.01595	.3910	.988	0.598	1.160
146	198.61	241.74	238.01	23,000	.01837	.2490	.991	9.280	1.200
149	158.00	233.62	227.61	28,500	.02334	.7055	.983	0.998	1.089
152	163.36	234.32	227.73	27,800	.02178	.6617	.984	0.953	1.098
153	109.83	237.55	232.12	31,500	.04690	.9390	.973	0.986	1.027
155	178.11	238.63	233.10	26,200	.01701	.5204	.987	0.854	1.130
156	173.35	236.90	231.48	26,800	.01909	.5690	.987	0.863	1.122
157	182.07	238.52	235.62	25,600	.01509	.4699	.988	0.823	1.140
159	108.86	235.41	228.34	31,300	.04661	.9303	.973	0.978	1.027
161	108.25	237.15	231.84	31,300	.04592	.9412	.973	1.028	1.016
163	171.19	235.63	231.69	27,000	.01707	.5898	.986	1.008	1.114
164	207.90	241.69	238.32	27,200	.02308	.0865	.995	0.0416	1.240
165	204.89	236.78	234.05	28,300	.02266	.1406	.993	0.0952	1.224

a. θ = heat-transfer corrective term

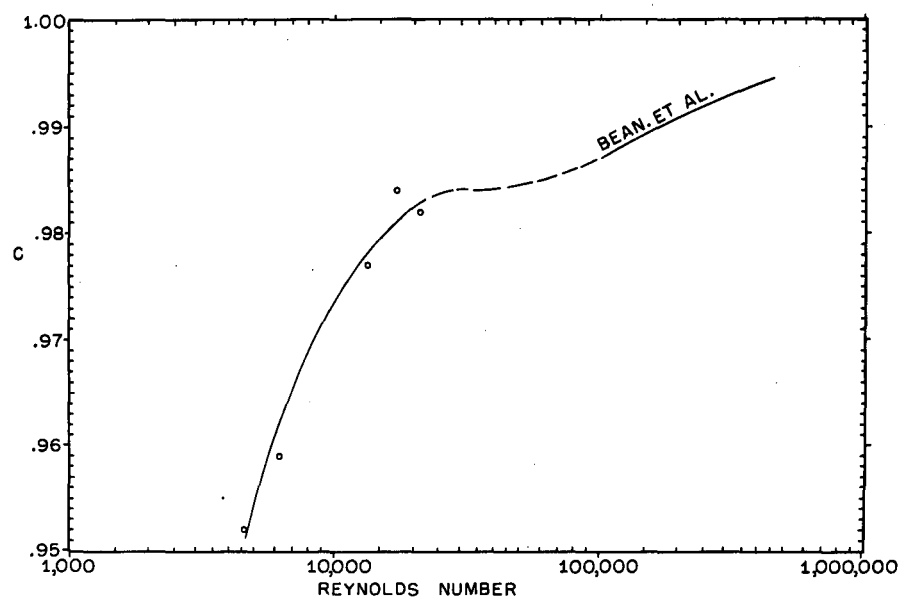


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Fig. 16. Discharge coefficient for flow nozzle A-1.

NOMENCLATURE

B_g	hypothetical film thickness--ft
C, C_o	coefficient of discharge
C_m	molal heat capacity at constant pressure--Btu/lb mole ^oF
C_p	heat capacity at constant pressure--Btu/lb ^oF
D	diffusivity--ft^2/sec
d	diameter--ft
D_m	molal diffusivity--lb moles/hr ft
D_2	nozzle throat diameter--inches
E_m	molal eddy diffusivity--lb moles/hr ft
G	mass velocity--lb/hr ft^2
g_c	universal gravity constant-- lb ft/lb(f) sec^2
h	heat transfer coefficient--Btu/hr ft^2 ^oF
j_d	mass-transfer factor
j_h	heat-transfer factor
k	thermal conductivity--Btu/hr ft^2 ^oF/ft
k_g	gas-film mass-transfer coefficient-- lb moles/hr ft^2 atmos
M_m	mean molecular weight--lb/lb mole
N	rate of transport of vapor--lb moles/hr ft^2
P	total pressure-atmos
p	partial pressure--atmos
p_f	film-pressure factor--atmos
q	heat transfer rate--Btu/hr ft^2
R	universal gas constant--atmos ft^3/lb mole ^oF

T	absolute temperature--^{o}R, ^{o}K
t	temperature--^{o}F, ^{o}C
u	velocity--ft/sec
u_m	average velocity based on total cross-section--ft/sec
w	weight rate of flow--lb/hr
x	length of diffusional path--ft
Y	compressibility factor

Greek Letters

α	the number of moles of gas A diffusing divided by the total number diffusing
β	heat-transfer corrective term defined by[7]
γ	heat-transfer factor defined by[24]
μ	viscosity--lb/ft^3
θ	heat-transfer corrective term defined by[25]
ϕ	heat-transfer corrective term defined by[25]

Dimensionless Groups

Re	Reynolds number--$du\rho/\mu$
Sc	Schmidt number--$\mu/\rho D$
Pr	Prandtl number--$C_p\mu/k$

Subscripts

A	refers to gas A
a	main stream
B	refers to gas B
c	convection
d	refers to mass transfer
G	refers to gas film
g	gas
h	refers to heat transfer
i	interface
m	mean
m	molal
r	radiation
s	surface
w	wet bulb

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