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SUBJECT: Investigation of Evaporative Pads for Greenhouse Heating and Cooling with Low Temperature Waste Heat, Part III

AUTHOR: V.G. Nehemiah, J.S. Law, and J.P. Andre

Consultants: W.K. Furlong and L.V. Wilson

ABSTRACT

The low temperature waste heat rejected from stationary power plants can perhaps be used to heat and cool commercial greenhouses. This study was conducted to obtain design relations for the simultaneous heat and mass transfer between air and water in a cross flow evaporative pad system. Local gas and liquid film transfer coefficients were determined for aspen fiber and polyurethane pads located in a prototype greenhouse. By conducting adiabatic water cooling and isothermal experiments the effect of air and water flow rates and film temperatures on the transfer coefficients was studied. The cross flow humidification system was modeled numerically on the computer, and a preliminary parametric investigation was conducted to design greenhouses for summer and winter conditions.

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Contents

Page

1. Summary 4
2. Introduction 5
3. Theory 5
   3.1 Basic Equations 5
   3.2 Adiabatic-Isothermal Case 7
   3.3 Adiabatic Cooling Case 8
4. Apparatus 8
5. Experimental Procedures 10
   5.1 Determination of Heating Effect Due to Fan 10
   5.2 Adiabatic-Isothermal Case 10
   5.3 Adiabatic Cooling Case 11
   5.4 Pressure Drop Investigation 11
6. Results and Discussion of Results 11
   6.1 General Correlations for Aspen Pads 11
   6.2 Adiabatic Cooling Runs - Aspen Pad 14
   6.3 Adiabatic-Isothermal Runs - Polyurethane Pad 18
   6.4 Pressure Drop and Flooding Investigation - Polyurethane Pad 22
      6.4.1 Pressure Drop 22
      6.4.2 Flooding 22
      6.4.3 Wetting Characteristics of Polyurethane Pad 22
   6.5 Comparison of Aspen and Polyurethane Pads 24
   6.6 Computer Simulation 24
   6.7 Sensitivity Analysis 25
7. Conclusions 25
8. Recommendations 25
9. Acknowledgements 25
10. Appendix 26
   10.1 Computer Program 26
   10.2 Experimental Errors and Sensitivity Analysis 45
   10.3 Sample Calculations 51
   10.4 Heating Effect Due to Fan 52
   10.5 Data Tabulation 52
   10.6 Comparison of Two Pads 57
   10.7 Simulation of Experimental Runs 57
   10.8 Parametric Study for Design Purposes 58
   10.9 Nomenclature 60
   10.10 Location of Original Data 61
   10.11 Literature References 61

1. SUMMARY

One of the applications of low temperature waste heat rejected from stationary power plants is to control the ambient air conditions in greenhouses using an evaporative pad system. In the cross flow evaporative pad system, water flows vertically down through the pad and the air horizontally across it.

This study was designed to characterize aspen and polyurethane evaporative pads using local transfer coefficients. Gas film transfer coefficients were determined from adiabatic-isothermal runs and correlated by the equations:

$$h_g a_H = 1.05\ G'^{0.99}\ L'^{-0.025}\ e^{0.01\ T_{GF}} \text{ (aspen)}$$

$$k_g a_M = 4.27\ G'^{1.0}\ L'^{-0.015}\ e^{0.009\ T_{GF}} \text{ (aspen)}$$

$$h_g a_H = 9.0\ G'^{0.74}\ L'^{-0.22} \text{ (polyurethane)}$$

$$k_g a_M = 48.2\ G'^{0.67}\ L'^{-0.24} \text{ (polyurethane)}$$

The temperature functionality for the polyurethane pad could not be quantized due to the scatter in the data.

A computer simulation of the cross flow pad system was developed and actual operation was simulated to within 2%. The local liquid film heat transfer coefficients obtained with the simulation for the aspen pad system were correlated:

$$h_L a_H = 2.46\ G'^{0.4}\ L'^{0.15}\ e^{0.01\ T_{LF}} \text{ (aspen)}$$

The computer simulation has been used to conduct a preliminary parametric study for designing greenhouses during summer and winter conditions. A preliminary comparison of the two pad materials shows the aspen pad to be more effective in cooling the air and water than the polyurethane pad. The pressure drop across the aspen pad was considerably lower than across the polyurethane pad.

It is recommended that better water distribution techniques be found for the polyurethane pad before the transfer coefficient correlations are determined. It is also suggested that a detailed parametric study be conducted for the proposed greenhouse.

2. INTRODUCTION

The fastest growing of the principal energy conversions, the generation of electrical energy, has begun to make a significant environmental impact by the discharge of low temperature waste heat. For every kilowatt of power produced, about two kilowatts of waste heat are discarded to the environment which leads to thermal pollution. One of the many suggestions for using the waste heat is to heat and cool commercial greenhouses (2, 3). Rural and suburban applications become more attractive if the food growing structures can be located in the nuclear reactor exclusion area. The greenhouse would then serve as a closed cycle cooling tower.

Greenhouses have been heated by conventional heat exchangers or cooled by evaporative pads for many years. It has been shown that a forced convection system utilizing finned heat exchangers and evaporative pads can be utilized for summer cooling and winter heating of greenhouses with low temperature waste heat (12).

A prototype greenhouse has been constructed to obtain quantitative design relationships for evaporative pad systems. Recently Mutsakis et al. (9) and Foulis et al. (4) have attempted to obtain design relations by conducting experiments with the model greenhouse. Mutsakis recommended several equipment modifications. Foulis found the functionalities of the local gas phase heat and mass transfer coefficients with water and air flow rates, but the temperature dependence could not be quantized due to the large scatter in the data. Foulis suggested that accurate temperature measurements were necessary to predict the overall correlations.

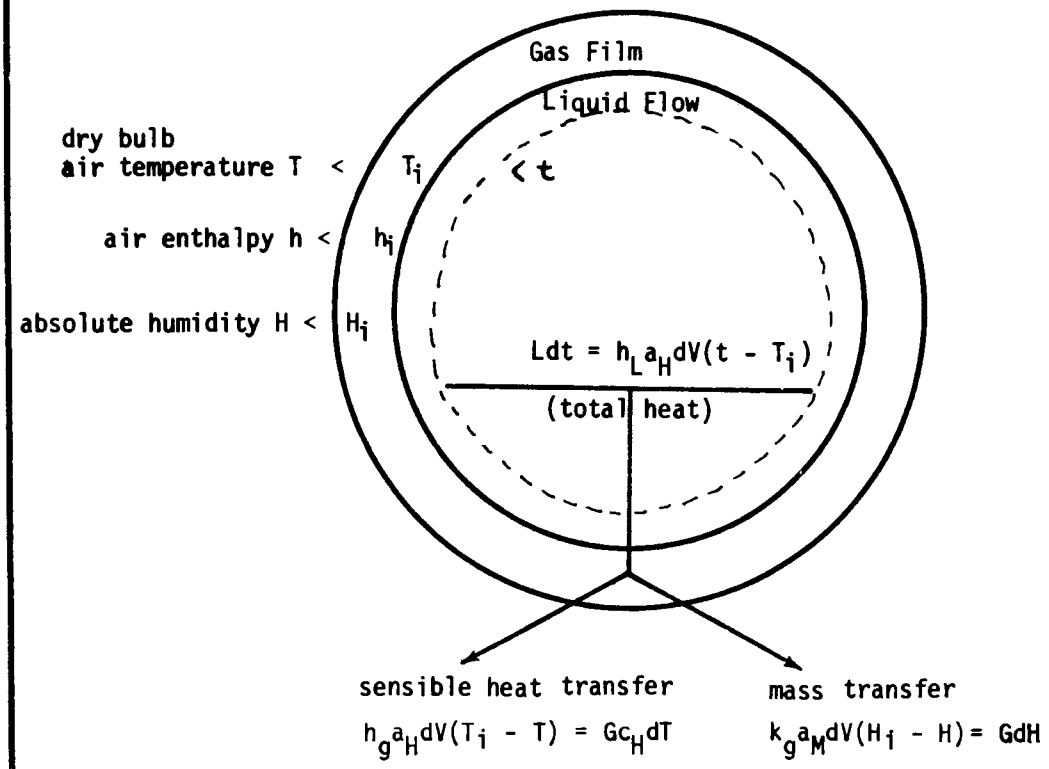
This study was conducted to: (1) test aspen and polyurethane pads and (2) use the quantitative functionalities in a computer program to simulate the cross flow system.

3. THEORY

3.1 Basic Equations

Evaporative pad systems are characterized by simultaneous interphase heat and mass transfer. The heat transferred is in the form of sensible and/or latent heat. The basic equations for such a system can be developed from the two film theory.

If an evaporating water droplet in an airstream is considered, as shown in Fig. 1, the mass transfer occurs due to evaporation from the interface which is at equilibrium. It must be remembered that the heat and mass transfer within the pad may occur from the droplets or wetted surfaces of the fibers; therefore the area of heat transfer, a_H, and mass transfer, a_M, cannot be separated from the experimental heat and mass transfer coefficients.



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HEAT AND MASS TRANSFER RELATIONSHIP BETWEEN WATER, INTERFACIAL FILM, AND AIR (1)

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Since the specific heat capacity of water is assumed to be unity, it has been omitted in the equations. The total heat transfer from water to the interface is

$$Ldt = h_L a_H dV(t - T_i) \tag{1}$$

The sensible heat transfer from the interface to the bulk air is

$$h_g a_H dV(T_i - T) = Gc_H dT \tag{2}$$

Mass transfer due to evaporation from the interface to bulk air can be represented in terms of the absolute humidity driving force.

$$k_g a_H dV(H_i - H) = GdH \tag{3}$$

A further basic definition is the Lewis number which is defined as the ratio, $h_g/k_g c_H$, and for an air-water system it equals one (10).

The overall enthalpy coefficient is related to the individual transfer coefficients as follows:

$$\frac{1}{K_g a_M} = \frac{1}{k_g a_M} + \frac{m}{h_L a_H} \tag{4}$$

Finally, the number of overall transfer units which is a measure of the degree of difficulty for the transfer process is given by

$$NTU = \frac{K_g V}{L} \tag{5}$$

The basic equations can be combined and integrated to determine the various coefficients.

3.2 Adiabatic-Isothermal Case

If the water is introduced at the wet bulb temperature (the adiabatic saturation temperature) of the entering air, the air is cooled and humidified along the adiabatic saturation curve. Since the water temperature is constant, no sensible heat transfer between the bulk water and the interface

occurs, and the interface is at the wet bulb temperature. The sensible heat transferred from bulk air to the interface equals the latent heat for the vaporization of water transferred to the gas film, and integration of Eqs. (2) and (3) yields

$$\frac{h_g a_H V}{G c_H} = \ln\left(\frac{T_{in} - T_{wb}}{T_{out} - T_{wb}}\right) \quad \text{(where } c_H \text{ is assumed constant)} \qquad (6)$$

$$\frac{k_g a_M V}{G} = \ln\left(\frac{H_{sat} - H_{in}}{H_{sat} - H_{out}}\right) \quad \text{(where } H_i = H_{sat} \text{ at } T_{wb}\text{)} \qquad (7)$$

From Eqs. (6) and (7) one can obtain the local gas film heat and mass transfer coefficients.

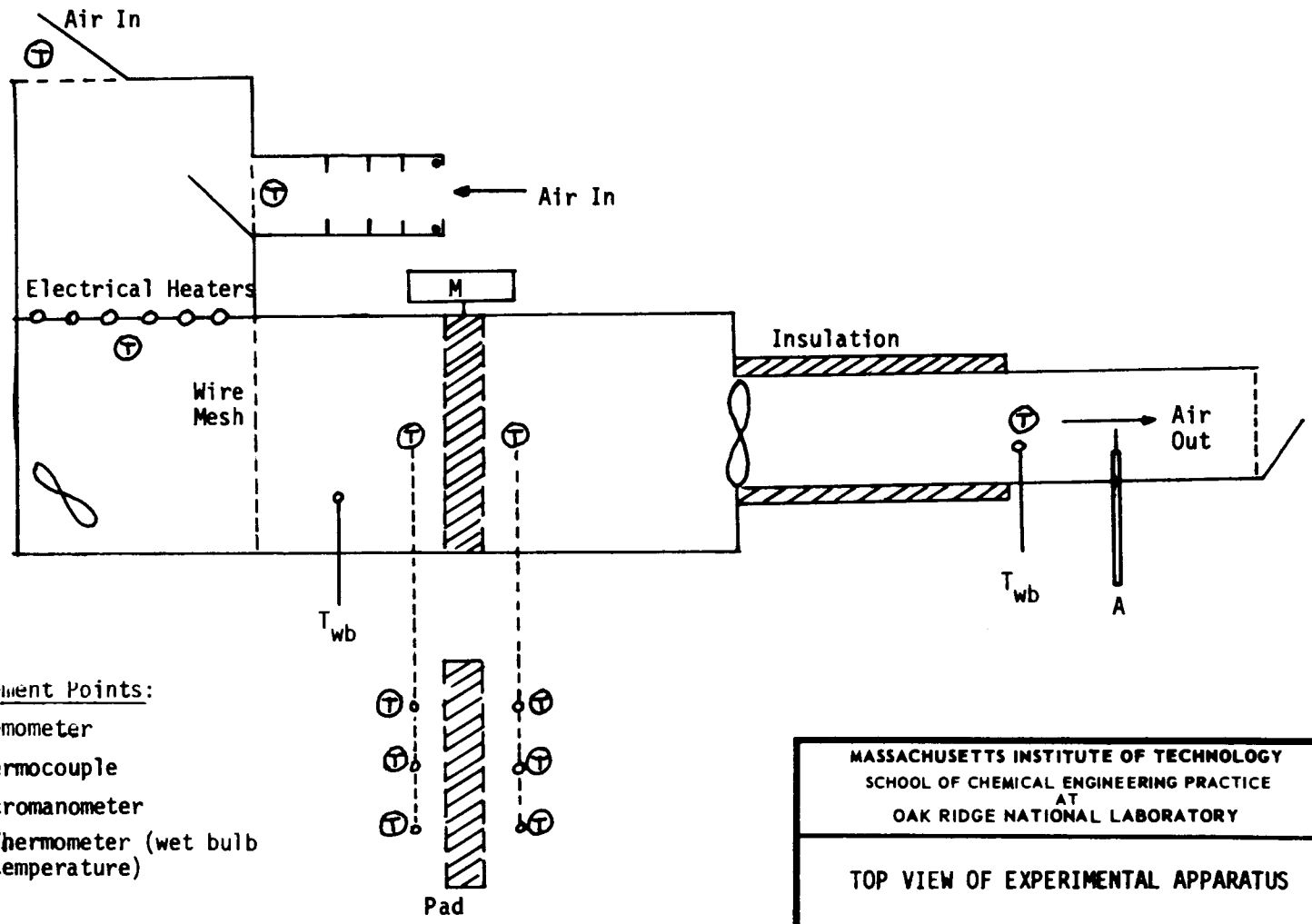
3.3 Adiabatic Cooling Case

In a real cross flow humidification system the conditions of water temperature and air enthalpy vary both horizontally and vertically in the pad. To obtain the liquid film heat transfer coefficient, the adiabatic cooling case has to be considered. The basic equations given in Sect. 3.1 cannot be combined and integrated due to the unknown interface or driving force conditions. The cross flow humidification system has been modeled numerically by considering the pad to be a number of incremental units with each small unit operating as a counter flow contactor. A computer code has been written to simulate this system (Details are given in Appendix 10.1).

4. APPARATUS

The experimental apparatus is as shown in Fig. 2. The dimensions of the apparatus were approximately 50-ft long x 8-ft wide x 10-ft high. The main tunnel section of the apparatus was constructed of plywood. Circular ducts made of galvanized steel were used for the inlet and exhaust of the air. The inlet duct was 30 in. in diameter and was fitted with a steam manifold to allow adjustment of the humidity and wet bulb temperature of the incoming air. The entering air passed through vertical baffles to obtain a well mixed air stream and over a bank of six, vertical, finned, heaters (8-kw each) provided to raise the temperature. A small room fan, wire mesh, and masking tape baffles were used to obtain flat velocity and temperature profiles entering the evaporative pad.

Two different pads were studied: a 6-lb aspen fiber pad (84 in. high, 33 in. wide, and 2 in. thick) manufactured by Acme Engineering and Mfg. Co.



Measurement Points:

A = Anemometer

T = Thermocouple

M = Micromanometer

T_{wb} = Thermometer (wet bulb temperature)

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TOP VIEW OF EXPERIMENTAL APPARATUS

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and a polyurethane pad (88 in. high, 33 in. wide, and 1 in. thick) manufactured by Scott Industrial Foam. The polyurethane pad provides a large non-wetting surface area and a three-dimensional skeletal structure with a constant 97% void volume. Both pads were encased in a galvanized wire mesh to provide support and shape.

A constant-speed, 1/2-hp fan was used to draw the air through the apparatus and into the exit duct. The air flow rate was measured downstream of the fan by use of an Alnor, type 8500, thermoanemometer with a range of 10 to 1600 ft/min. Vertical and horizontal velocity profiles were determined at two-inch intervals and integrated over the area of the duct to obtain the air flow rate. Flow straighteners were provided to obtain a uniform velocity profile before the velocity measurements. The air flow was controlled by varying the inlet duct aperture and/or the 8-ft-high inlet door at the entrance port. The pressure drop across the pad was measured with a micromanometer.

The water flow through the pad was monitored with two rotameters connected in parallel to enable easy adjustment to low or high flow rates. The inlet water temperature to the pad was maintained constant by a thermostatically-controlled mixing valve.

The exhaust air duct was insulated with glass wool,and the air temperatures were measured using calibrated chromel-alumel thermocouples. The wet bulb temperature was measured with precision thermometers fitted with a wetted wick before the pad and after the fan. All the points of measurement are noted on Fig. 2.

5. EXPERIMENTAL PROCEDURE

5.1 Determination of Heating Effect Due to Fan

The difference in temperature before and after the fan was measured using differential thermocouples as a function of gas flow rate. The rise in temperature due to the heat input of the fan was also determined with precision glass-mercury thermometers.

5.2 Adiabatic-Isothermal Case

The water was introduced at the top of the pad at or near the wet bulb temperature of the entering air. The water was recirculated at the desired flow rate until steady state conditions were reached. Steady state was attained when the temperature of the entering and exiting water was constant and equal to the wet bulb temperature of the entering air. Cold makeup water was added to the recirculating water to compensate for the heating effect of the recirculating pump and for the water loss due to evaporation and overflow. Temperature and flow measurements were then made. The

experiment was repeated for different air and water flow rates and different wet bulb temperatures.

5.3 Adiabatic Cooling Case

The adiabatic cooling experiments are conducted similar to the isothermal case; however, the entering water was not at the wet bulb temperature of the entering air, and the water was not recirculated. Warm water was introduced at the top of the pad, and a temperature gradient in the water flowing down the pad resulted. The experiment was repeated for different air and water flow rates and for different entering water temperatures.

5.4 Pressure Drop Investigation

The pressure drop across the pad with different air and water flow rates was also measured. First the air inlet door was placed in a fixed position, and water flow rate varied. At each water flow rate the air flow rate was measured (as in Sect. 5.2), and the pressure drop across the pad determined. The experiment was then repeated for different air flow rates by varying the position of the air inlet door but keeping the water flow rate constant.

6. RESULTS AND DISCUSSION OF RESULTS

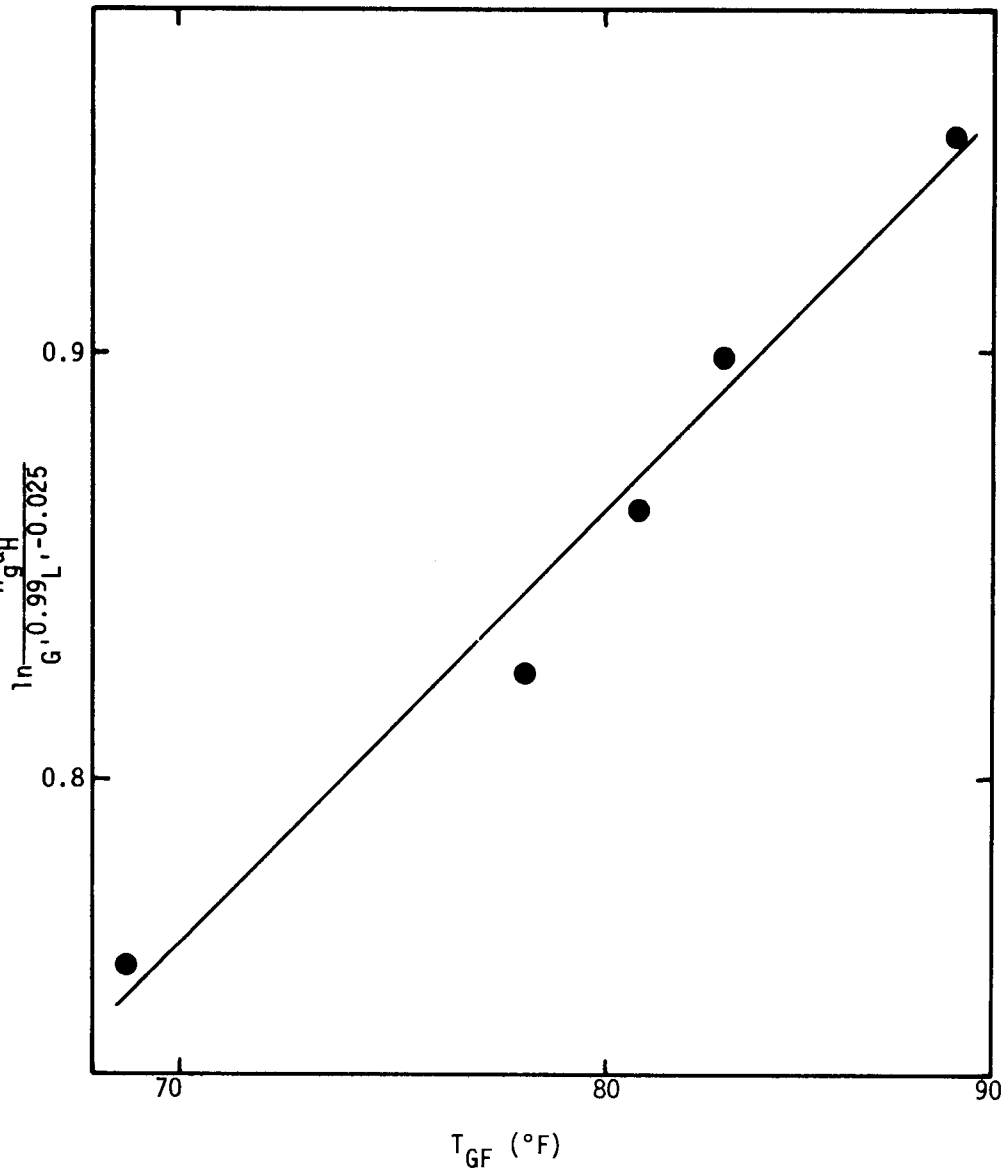
6.1 General Correlations for Aspen Pads

Foulis et al. (4) conducted adiabatic-isothermal runs on the 6-lb aspen pad and obtained the functionality of the transfer coefficients with air and water flow rates. The relations were:

$$h_g a_H \;\alpha\; G'^{0.99}\, L'^{-0.025}$$

$$k_g a_M \;\alpha\; G'^{1.0}\, L'^{-0.015}$$

Adiabatic-isothermal runs were conducted for constant G' and L' with varying gas film temperature, T_{GF} [calculated in the same manner as McAdams (7); see computer program listing for equations, Appendix 10.1]. The results are plotted in Figs. 3 and 4 and tabulated in Appendix 10.5. All exit temperature measurements were corrected for the heating effect of the fan as explained in Appendix 10.4. The figures show the gas film coefficients normalized by air and water superficial velocities. The power



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GAS FILM HEAT TRANSFER COEFFICIENT
AS A FUNCTION OF FILM TEMPERATURE

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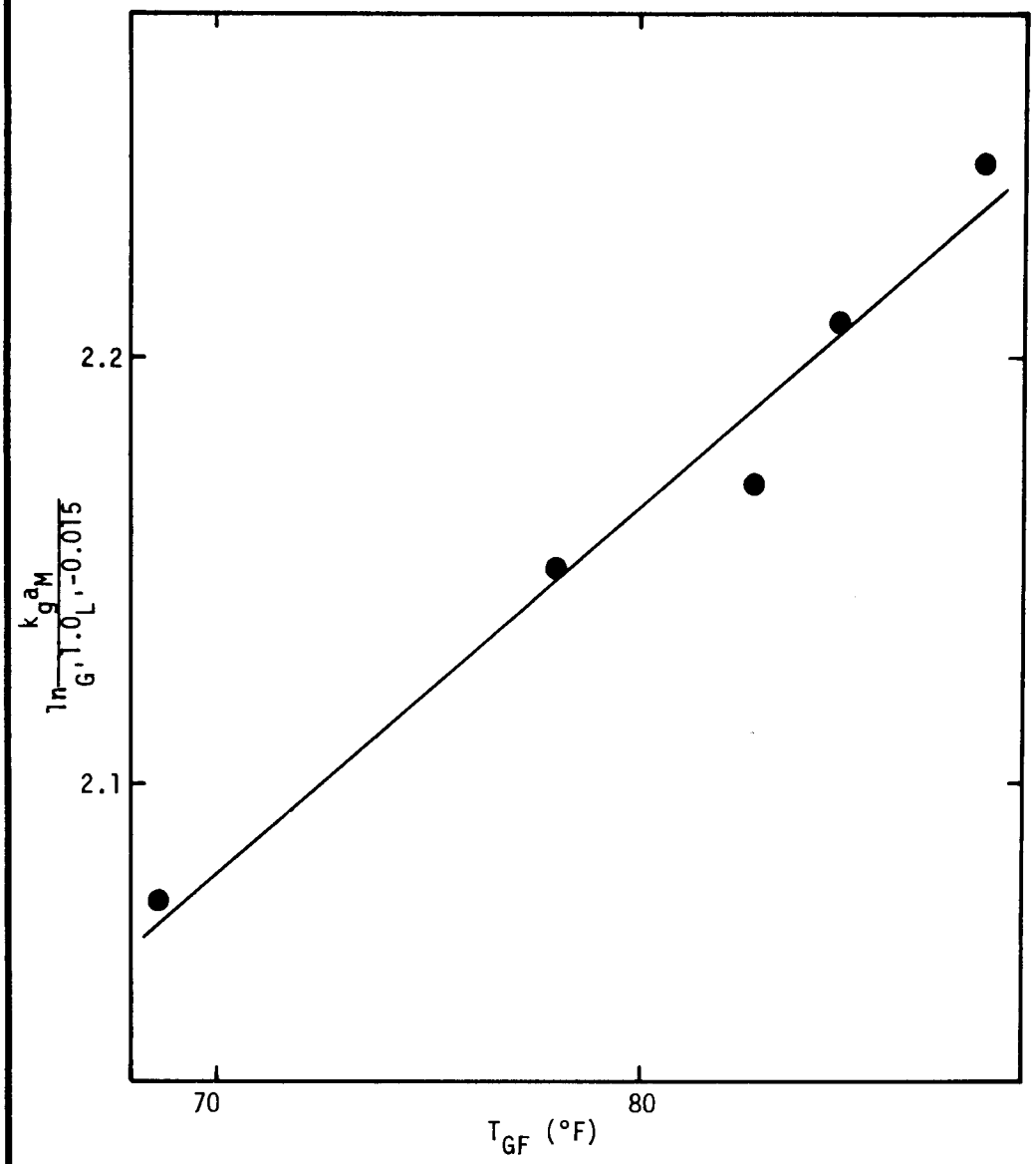

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GAS FILM MASS TRANSFER COEFFICIENT
AS A FUNCTION OF FILM TEMPERATURE

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dependence obtained by Foulis was assumed to be accurate. From a linear least square fit of the data, the following correlations were obtained:

$$h_g a_H = 1.05\ G'^{0.99}\ L'^{-0.025}\ \exp(0.01\ T_{GF}) \tag{8}$$

$$k_g a_M = 4.27\ G'^{1.0}\ L'^{-0.015}\ \exp(0.009\ T_{GF}) \tag{9}$$

The correlations are valid for the following range of variables: superficial water velocity from 54 to 164 lb/min-ft^2, superficial air velocity from 5 to 14 lb/min-ft^2, and the film temperature from 65 to 110°F.

The dependence of both transfer coefficients on the gas film temperature is small and about the same order of magnitude as that obtained by other investigators (7). The small dependence correlates with the exponents found for flow of saturated air in packed beds or across cylinders (4).

The ratio of Eqs. (8) and (9) should give the Lewis relation (see Sect. 3.1):

$$Le = \frac{1}{c_H}\left[\frac{h_g a_H}{k_g a_M}\right] = \frac{1}{c_H}\left[\frac{1.05\ G'^{0.99}\ L'^{-0.025}\ \exp(0.01\ T_{GF})}{4.27\ G'\ L'^{-0.015}\ \exp(0.009\ T_{GF})}\right] \approx \frac{0.246}{c_H} \tag{10}$$

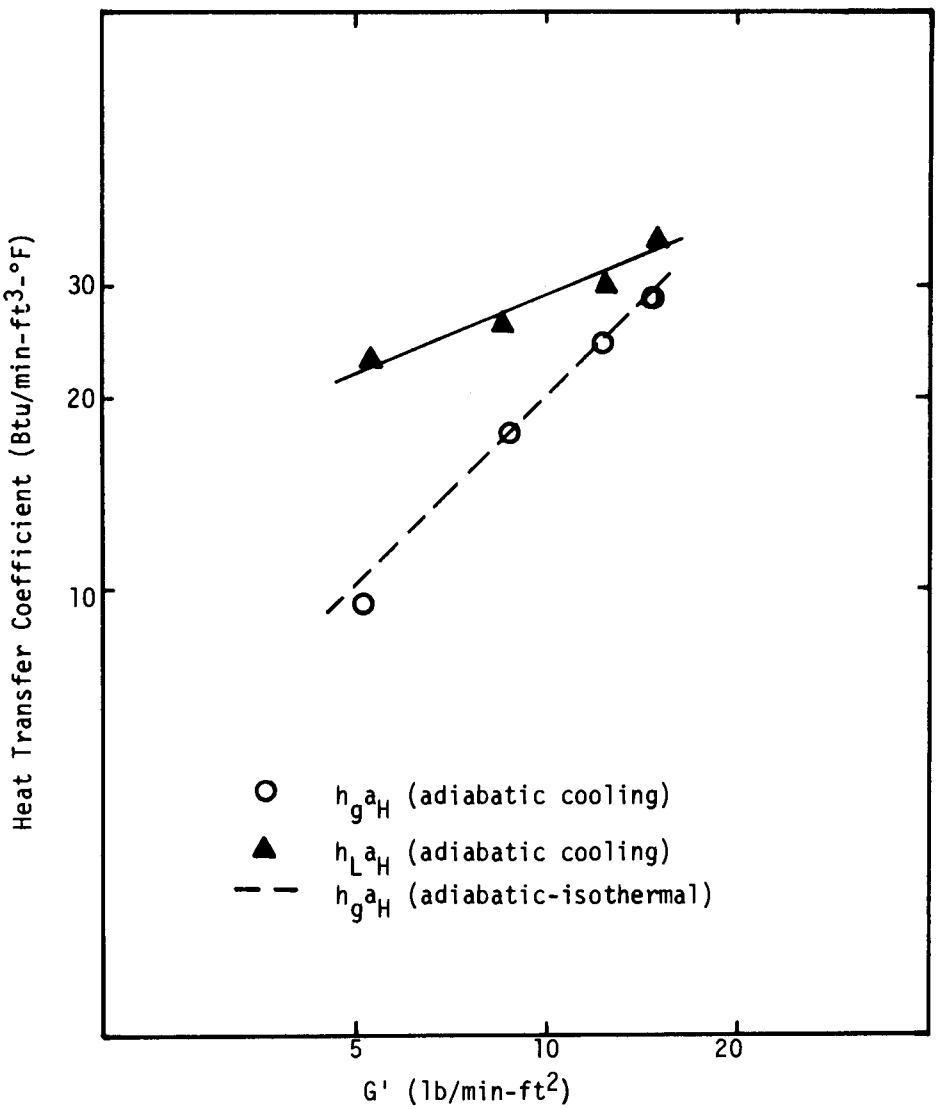
where the heat and mass transfer are assumed equal. The humid heat capacity, c_H, is given by:

$$c_H = 0.24 + 0.45\ (H_{mean}) \tag{11}$$

which makes Le $\simeq$ 1.0.

6.2 Adiabatic Cooling Runs - Aspen Pad

A computer model which simulates the cross flow humidification system was used to analyze the data. The Mickley (8) approach was followed to determine the transfer coefficients using a constant tieline slope ($=-h_L a_H/k_g a_M$). The data and results of the adiabatic cooling runs are summarized in Appendix 10.5. The data are plotted in Figs. 5, 6, and 7. Figure 5 is a plot of gas and liquid film heat transfer coefficients as a function of the superficial air velocity. The dashed line in the plot represents Eq. (8) or the experimentally determined relationship from adiabatic-isothermal runs. The values of $h_g a_H$ computed by the model from data of adiabatic cooling runs agree well with the results of the isothermal runs. Similarly, Figs.



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INFLUENCE OF THE GAS SUPERFICIAL VELOCITY ON THE HEAT TRANSFER COEFFICIENTS			
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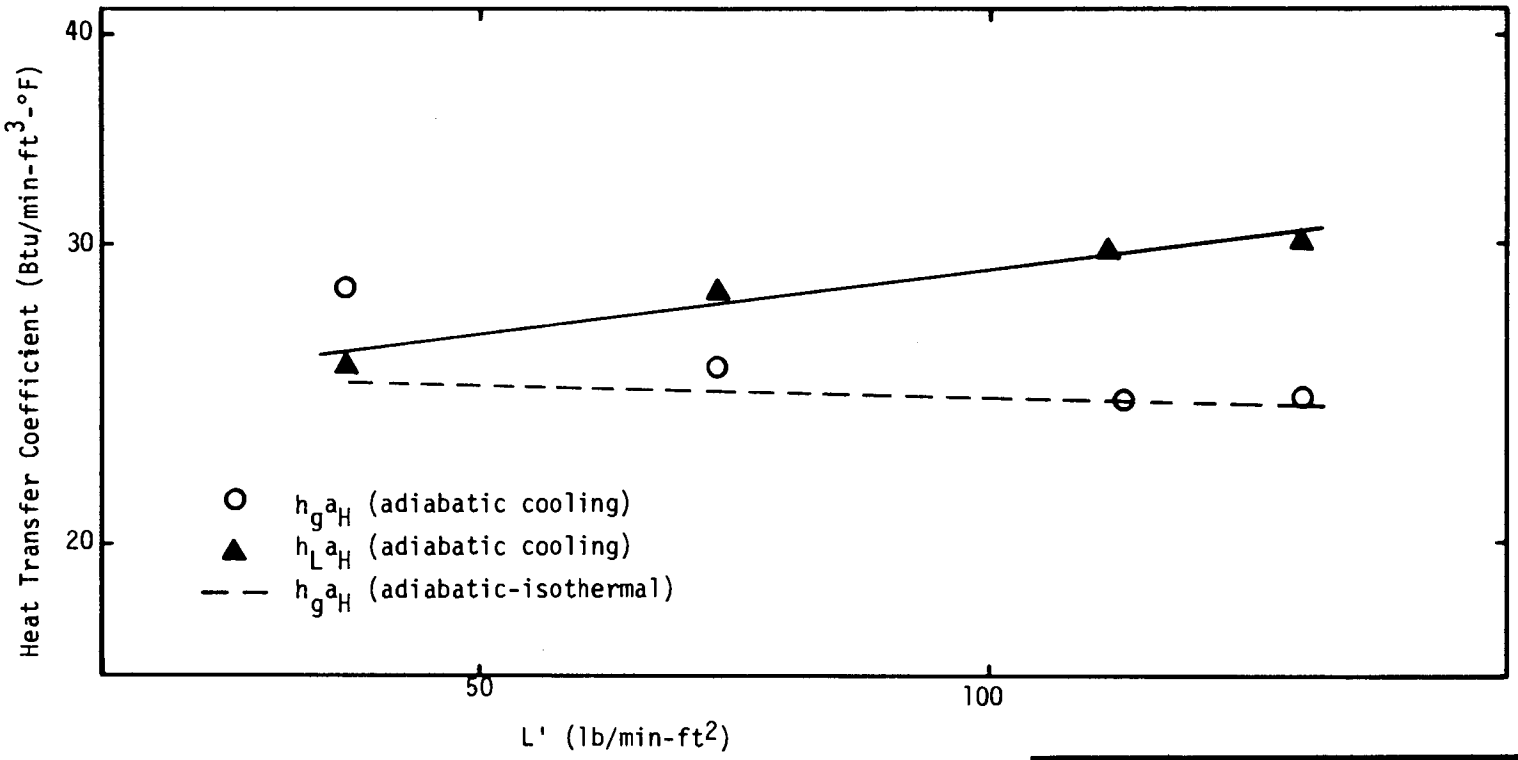

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INFLUENCE OF THE LIQUID SUPERFICIAL VELOCITY ON THE HEAT TRANSFER COEFFICIENTS

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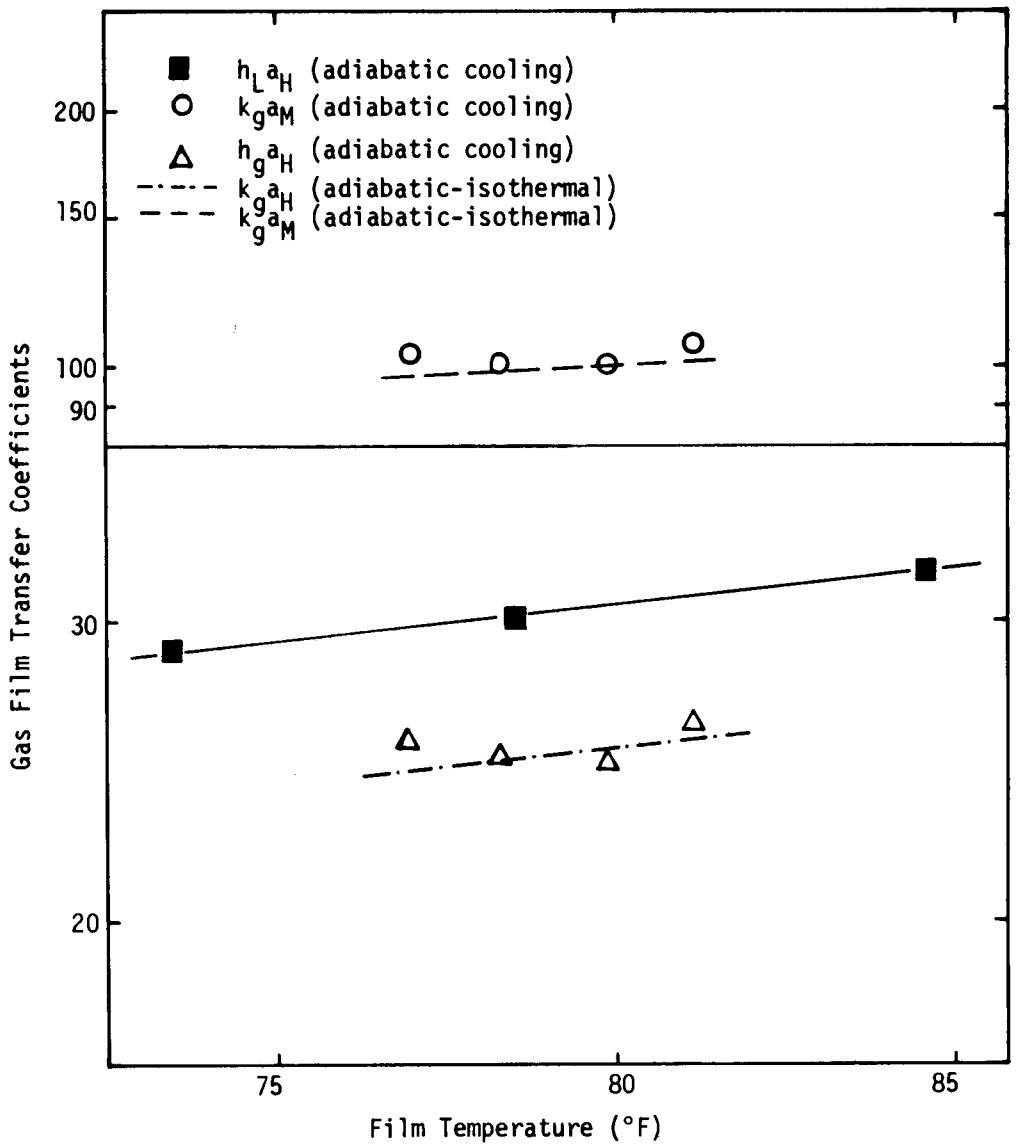

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INFLUENCE OF FILM TEMPERATURE ON GAS FILM TRANSFER COEFFICIENTS

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4-19-72	JPA	CEPS-X-148	7

6 and 7 are plots of gas and liquid film transfer coefficients as a function of superficial liquid velocity and film temperature. The agreement is again good between results of adiabatic isothermal runs and adiabatic cooling runs using the computer model.

The plots of liquid film heat transfer coefficient were correlated by a linear least square fit to obtain:

$$h_L a_H = 2.46\ G'^{0.4}\ L'^{0.15}\ \exp(0.01\ T_{LF}) \tag{12}$$

The correlation shows that the liquid film coefficient does not have a very strong dependency on the liquid flow rate which perhaps results from the laminar flow of water in a thin film along the fiber. The water flow rates were perhaps not sufficient to promote mixing or turbulence to decrease liquid film resistance substantially.

The liquid film resistance is seen to decrease with an increase in the air flow rate; however, the effect is much lower on $h_L a_H$ than on $k_g a_M$ or $h_g a_H$. The dependency on the air rate is high and is probably due to surface renewal or wavy film formation caused by the turbulent air flow. The liquid film heat transfer coefficient is not a strong function of film temperature.

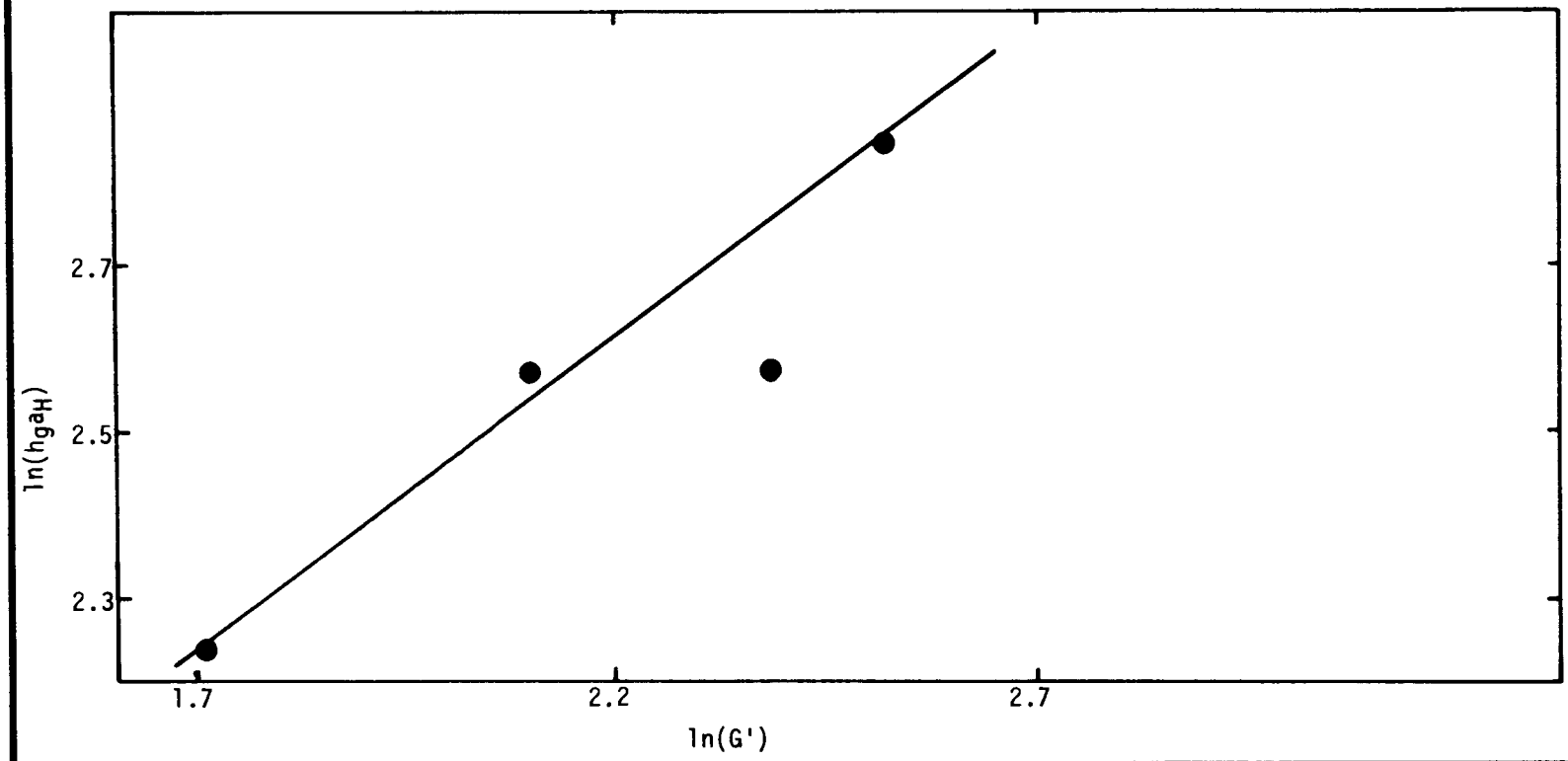
The tieline slope ($=-h_L a_H/k_g a_M$) can be obtained from the ratio of Eqs. (12) and (9):

$$-\frac{h_L a_H}{k_g a_M} \approx 0.6\ G'^{-0.6}\ L'^{0.2}\ \exp[0.01(T_{LF} - T_{GF})] \tag{13}$$

The maximum difference between the liquid and gas film temperatures in all the runs was 10°F which gives a maximum value of 1.1 for the temperature functionality of the tieline slope {i.e., $\exp[0.01(T_{LF} - T_{GF})] = 1.1$}. This shows that the assumption of a constant tieline slope (see Sect. 6.2) with temperature was reasonable for this range of experimental conditions.

6.3 Adiabatic-Isothermal Runs - Polyurethane Pad

The analysis of the data was identical to that used for the aspen pad (see Sect. 6.1). The data and the calculated values are summarized in Appendix 10.5. Figures 8 and 9 are plots of the gas phase heat and mass transfer coefficients versus superficial air velocity. Figure 10 is a plot of gas film heat and mass transfer coefficients (normalized by the superficial air velocity) versus superficial water velocity. Due to some scatter in the determination of the functionality of liquid velocity, the line was drawn through the averaged points. From the slope of the plots the following correlations are obtained:



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GAS FILM TRANSFER COEFFICIENT AS A FUNCTION OF SUPERFICIAL AIR VELOCITY FOR THE POLYURETHANE PAD

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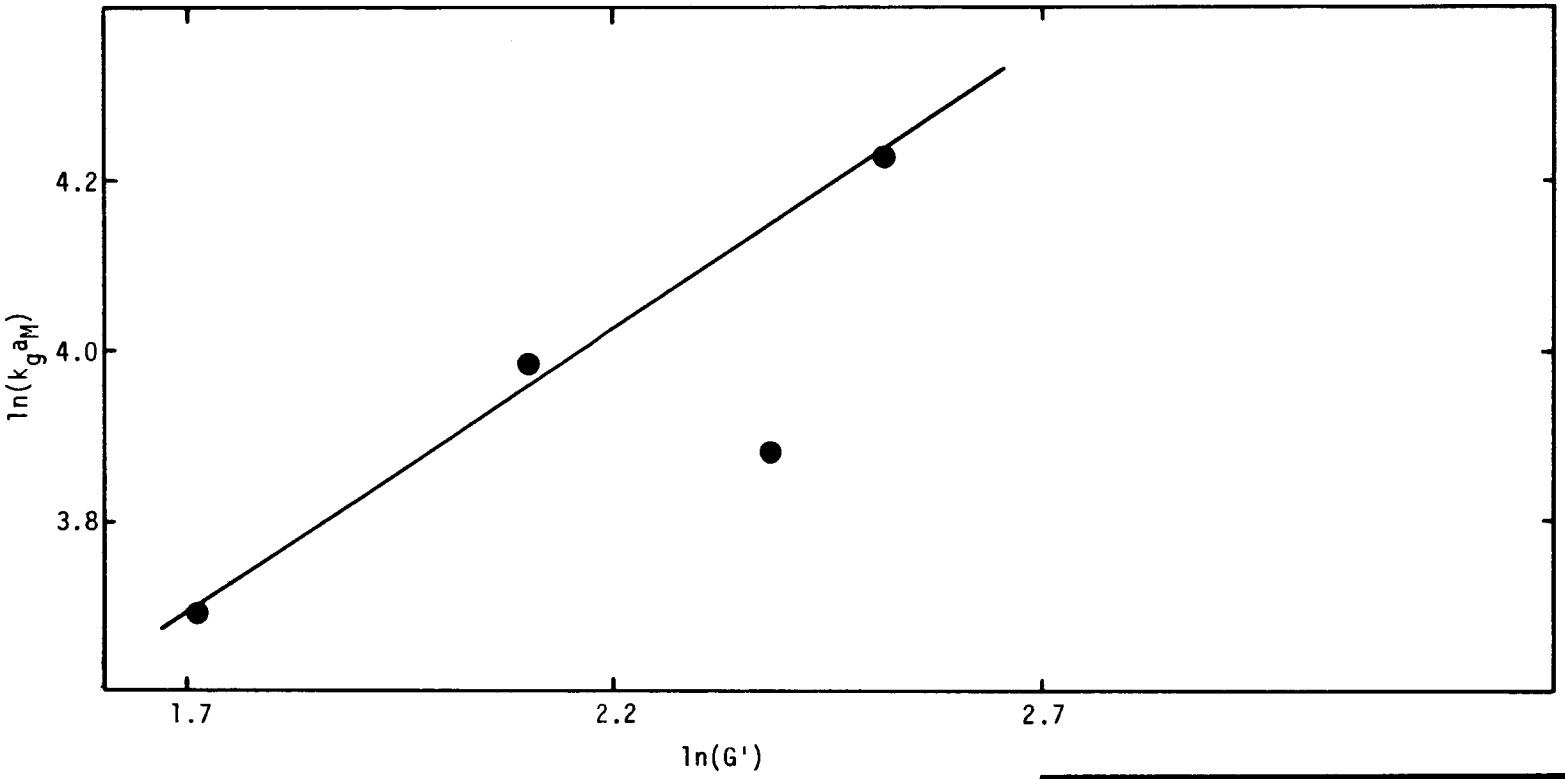

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GAS FILM MASS TRANSFER COEFFICIENT AS A FUNCTION OF SUPERFICIAL AIR VELOCITY FOR THE POLYURETHANE PAD

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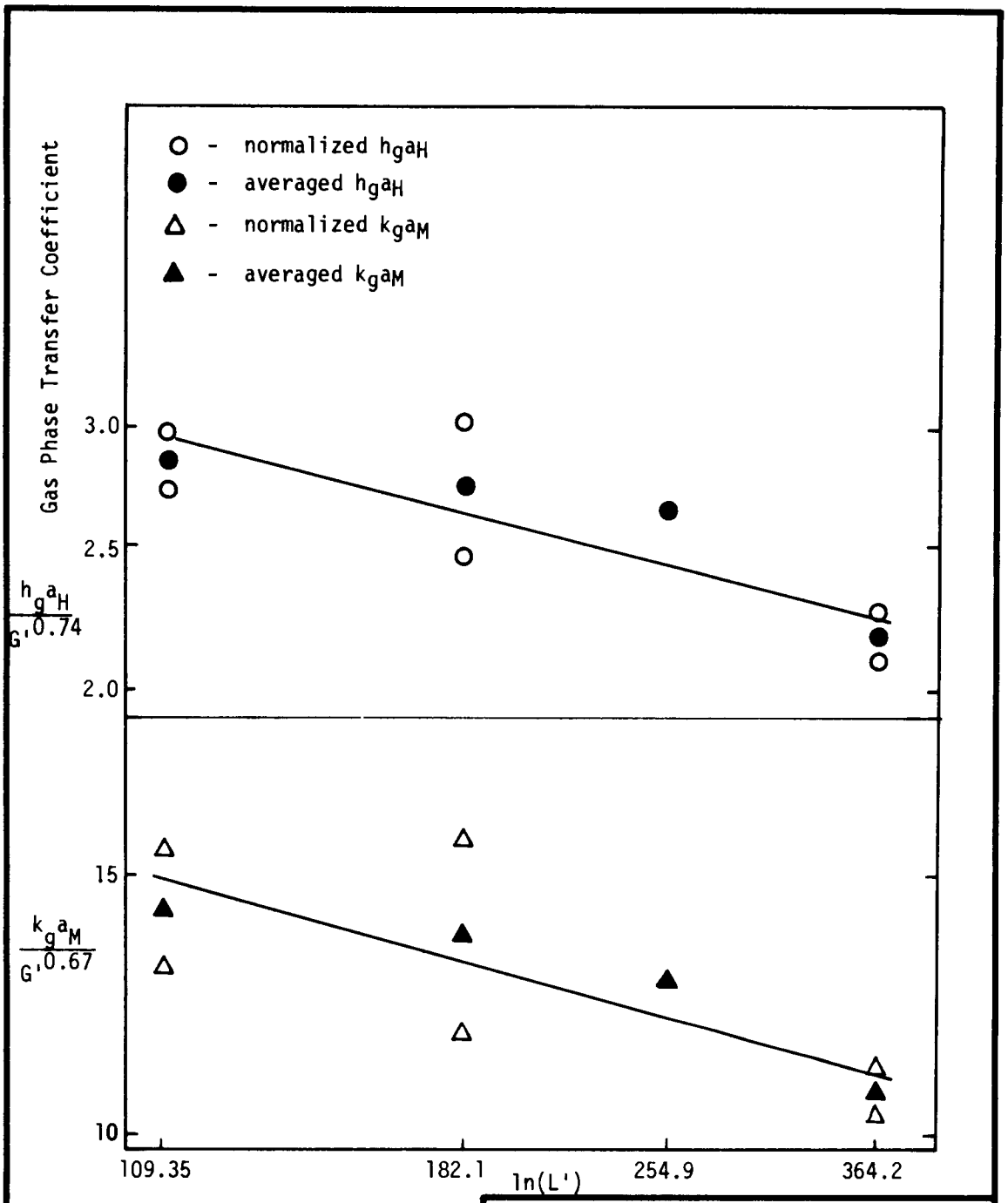

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GAS FILM TRANSFER COEFFICIENTS AS A FUNCTION OF SUPERFICIAL LIQUID VELOCITY

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$$h_g a_H = 9.0\ G'^{0.74}\ L'^{-0.22} \tag{14}$$

$$k_g a_M = 48.2\ G'^{0.67}\ L'^{0.24} \tag{15}$$

Unfortunately due to scatter in the data the film temperature function could not be determined. One can infer qualitatively from the data (see Appendix 10.5) that the transfer coefficients have a very small dependence on film temperature.

From the correlation it can be seen that both transfer coefficients are a strong function of air rate and increase rapidly with air velocity. However, the functionality is lower than that for the aspen pad. It is also seen that liquid velocity has an unaccountably strong negative effect on the transfer coefficients, much larger than that for the aspen pad.

No correlation for the liquid film heat transfer coefficient was obtained since only one adiabatic cooling run was conducted to obtain a preliminary comparison between the two types of pads.

6.4 Pressure Drop and Flooding Investigation - Polyurethane Pad

6.4.1 Pressure Drop

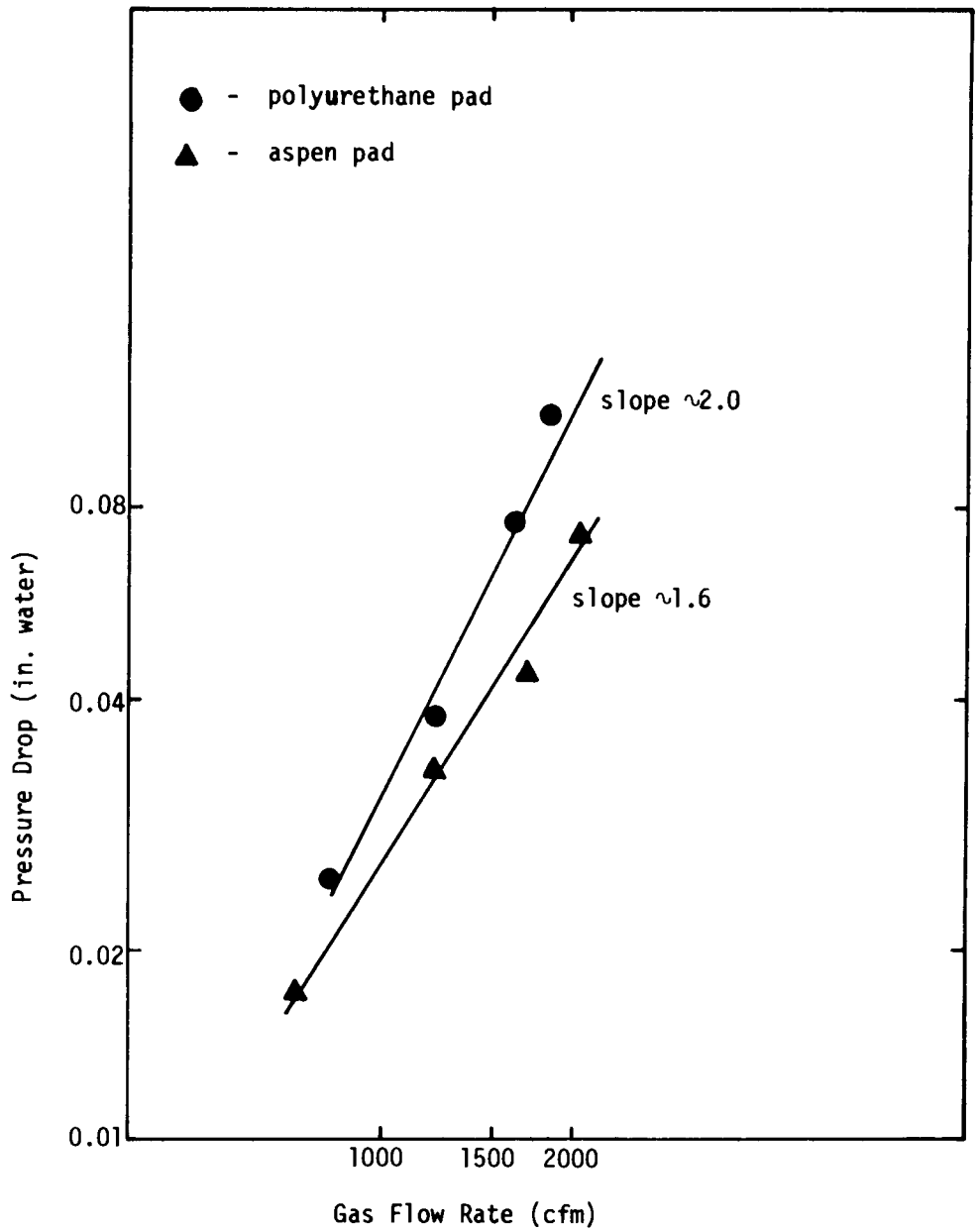
For a constant liquid rate of 7 gpm, the pressure drop across the polyurethane pad was measured and plotted as a function of gas flow rate (see Fig. 11). Similar pressure drop data for the aspen pad are presented for comparison. The pressure drop for both pads increases with increasing air flow rate. However, the pressure drop across the polyurethane pad is greater than that across the aspen pad.

6.4.2 Flooding

A qualitative study was made to study flooding in the polyurethane pad. For a maximum air rate of ∿4500 cfm the liquid rate was increased until flooding occurred. Flooding was defined by water flowing along the pad sides rather than through the pad itself, and it occurred at a liquid rate of ∿12 gpm [compared to 9 gpm for aspen fiber (9)]. To achieve higher water flow rates the distributor would have to be redesigned.

6.4.3 Wetting Characteristics of Polyurethane Pad

The pad does not seem to be wetted by the water. Even after soaking the pad for several hours, the pad became dry in certain sections when the air flow was started. This uneven water distribution through the pad gives a smaller area for the transfer operations.



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PRESSURE DROP ACROSS PADS AS A FUNCTION OF GAS FLOW
(Liquid rate = 7 gpm)

DATE	DRAWN BY	FILE NO.	FIG.
4-19-72	JPA	CEPS-X-148	11

6.5 Comparison of Aspen and Polyurethane Pads

The important features of both pads are summarized in Table 1. The transfer coefficients are compared for identical operating conditions of temperatures and flow rates.

Table 1. Comparison of Aspen and Polyurethane Pads

	Aspen Pad (Run 34)	Polyurethane Pad (Run 56)
Type of pad	long fibers	three-dimensional network
Thickness	2 in.	1 in.
Pressure Drop (from Fig. 11)	0.054 in. water	0.14 in. water
$h_g a_H$ (Btu/min-ft^3-°F)	23	24
$h_L a_H$ (Btu/min-ft^3-°F)	30	22
$k_g a_M$ (lb/min-ft^3)	99	94

The polyurethane pad flooded at a considerably higher water flow rate (∿436 lb/min-ft^2) than the aspen pad (∿175 lb/min-ft^2). This seems to be the only advantage of the polyurethane pad over the aspen pad. The polyurethane pad has a greater pressure drop, and the resistances to heat and mass transfer are greater which indicates that the polyurethane pad is less efficient in cooling the water and air streams. The lower transfer coefficients for the polyurethane pad may be due to a smaller effective transfer area caused by uneven water distribution (see Sect. 6.4.3).

6.6 Computer Simulation

The experimental transfer coefficient correlations for the aspen pad were used in a computer model of the cross flow humidification system to simulate some of the experimental runs. The design program, LANSIM, predicts the outlet air and water temperatures and humidity for a given set of inlet water and air conditions. The simulation predicted the outlet air temperature within 0.4% and water temperature within 0.5% which shows the validity of the assumption of a constant tieline slope. The results of the simulation are given in Appendix 10.7.

LANSIM was used to conduct a parametric study to obtain data for designing and operating a greenhouse. The parameters were varied from the limiting winter to the limiting summer conditions, as described by Furlong (5, 6). The results of this study are tabulated in Appendix 10.8.

6.7 Sensitivity Analysis

A sensitivity analysis was conducted using the computer simulation. The greatest source of possible error in measurement was the inlet wet bulb temperature since requisite gas velocities for measurement were not always obtained. The maximum estimated error was about 0.6°F which yields a 7% error in the transfer coefficient. The complete results of the analysis are shown in Appendix 10.2.

7. CONCLUSIONS

1. For the aspen pad, local gas and liquid phase transfer coefficients have been correlated as a function of superficial air and water velocities and film temperature.

2. For the polyurethane pad the gas film transfer coefficients have been related to the superficial air and water velocities.

3. The aspen pad was more effective than the polyurethane pad.

4. The experimental runs were simulated with a computer code, and a partial parametric study for designing a greenhouse using the computer simulation, and the experimental correlations has been made.

5. The main inaccuracy in the experimental measurements was in the inlet wet bulb temperature.

8. RECOMMENDATIONS

1. Different packing materials and densities should be studied.

2. Parametric studies of the proposed greenhouse should be conducted with the computer simulation.

3. Variations in pad orientation should be investigated.

4. To study the polyurethane pad, a better water distribution system must be implemented.

9. ACKNOWLEDGEMENTS

We wish to express our gratitude to our consultants, L.V. Wilson and W.K. Furlong, for their valuable help and suggestions during the project. We also thank E.M. Lees for his prompt attention in our experimental equipment requirements.

10. APPENDIX

10.1 Computer Program Description

Constant Tieline Slope

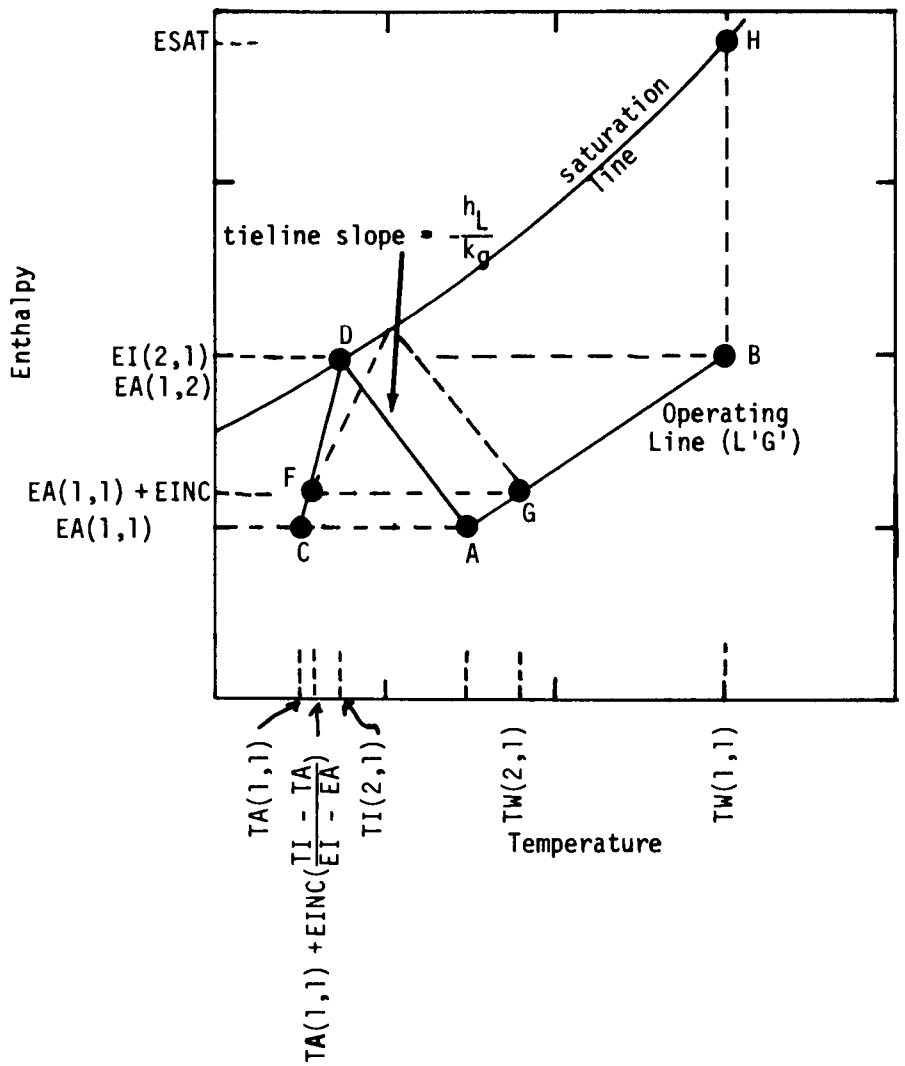
If the evaporative pad is used in a cross flow mode, the heat and mass transfer coefficients can be calculated by approximating the pad as a 40x40 grid, with each box of the grid being treated as a counterflow tower. With small enough boxes in the grid, the cross flow and counterflow modes are indistinguishable.

A computer program was developed by Foulis et al. (4) to calculate heat and mass transfer coefficients with the Mickley approach (8) by assuming the slope of the tieline between the operating line and the air enthalpy saturation curve to be constant with respect to film temperature throughout the pad. The subroutines used to calculate overall NTU, water temperature, and air enthalpy profiles were first described by Park and Vance (11). A detailed description follows.

The data to be read in for each run includes the water flow rate, entering water temperature, exiting water temperature, gas flow rate, entering gas temperature (wet and dry bulb), exiting gas temperature (dry bulb), and pressure in the pad. The volume of the pad is constant at 3.21 ft^3 for the aspen pad, and the slope of the operating line is the quotient of the flow rates (L/G).

Given the input data, subroutine PROFIL assumes an overall number of transfer units (NTU) equal to the water temperature difference between the inlet air and saturated air at the water temperature. Then subroutine TOWER is called to determine the air enthalpy and water temperature profile by incrementing the enthalpy and temperature in each grid by that fraction of the NTU times the enthalpy driving force. The average water exit temperature from the pad is calculated and returned to PROFIL. This value is compared with the experimental value, and if the difference is less than the given tolerance, the guessed NTU is taken as the correct value. If not, the NTU is corrected by a fraction of the error, and TOWER is called again. If the tolerance condition is not met in ten iterations, the program exits and begins another run with new data, and the message, the NTU does not converge, is printed.

In the next part of the program an estimate is made of the slope of the tieline between the operating line and the enthalpy saturation curve (initially -0.5); the air temperature and interface enthalpy and temperature profiles are determined; and the slope is adjusted until the calculated air exit temperature matches the experimental value. For each small box subroutine MKLY divides the air enthalpy interval into five increments. The procedure is started at the bottom of the box (EA, TW-1) (see Fig. 12), and subroutine SATFLM is called to determine interface conditions based on water temperature, air enthalpy, and tieline slope. A Newton Raphson



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DETERMINATION OF INTERFACE CONDITIONS AND AIR TEMPERATURE (FOR INCREMENTAL COUNTERFLOW UNIT)

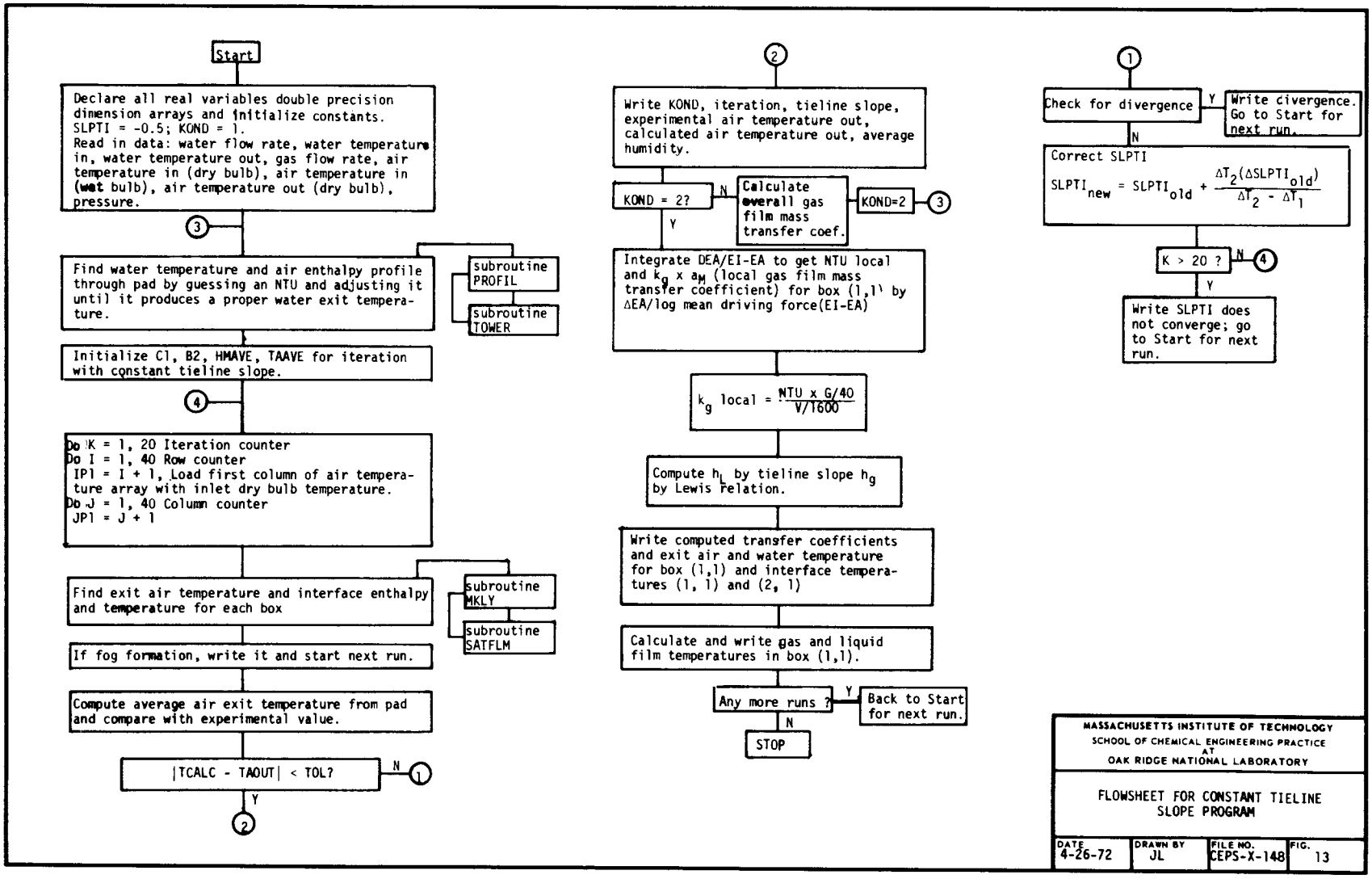
DATE	DRAWN BY	FILE NO.	FIG.
4-19-72	JL	CEPS-X-148	12

iteration is used to solve the equation for the intersection of the saturation curve and the tieline. After a return to MKLY with interface conditions (point D), the program determines the next point of air enthalpy and temperature. Air enthalpy is incremented and point F is found by moving up along line CD by the incremental amount of enthalpy. The new water temperature (point G) is found from the operating line at the given enthalpy. This process of finding interface conditions and air temperature is repeated until the exit enthalpy from each box in the grid is reached. If at any time Newton's iteration converges to a point above the enthalpy saturation curve, fog formation is noted, and the program returns to start the next run. For a detailed account of the theory, refer to Ref. (5).

After calculating the complete air temperature and interface temperature and enthalpy profiles in this manner, the program calculates an average air exit temperature from the pad. If the difference between the calculated value and the experimental value is less than the tolerance, then the correct value of the tieline slope has been found. Otherwise, the tieline slope is corrected by a fraction of the error. If the tieline slope diverges or if twenty iterations have been made and convergence has not been achieved, then the computer passes on to the next run.

After initial determination of all the profiles, an overall gas film mass transfer coefficient is calculated from the overall NTU using Eq. (5). Then the new interface conditions are used to improve the water temperature and air enthalpy profiles, and these improved profiles are then used to perform an integration of d(EA)/EI-EA on the first box in the grid to get a local NTU and thereby a local gas film mass transfer coefficient (k_g). The liquid film heat transfer coefficient (h_L) is calculated from k_g, and the tieline slope ($-h_L/k_G$) and the gas film heat transfer coefficient is determined from k_g by the Lewis relation ($h_g/k_g = c_H$). The liquid and gas film temperatures are then calculated taking a log mean of inlet and outlet conditions. The calculated transfer coefficients and temperatures are printed out. If more data are to be run, the program returns to start on new data; otherwise the program stops.

An overall flow diagram of the program is given in Fig. 13. The flow diagram of the subroutines is given in Fig. 14. The listing of the program follows.



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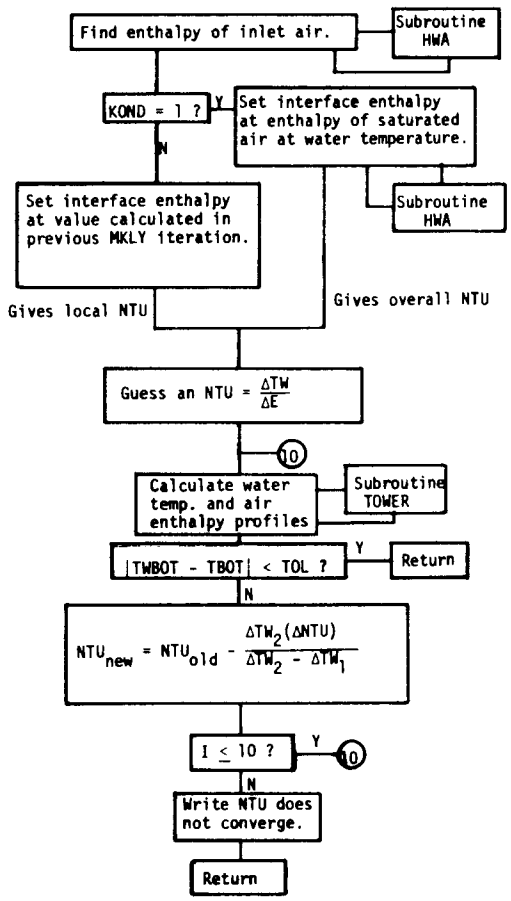
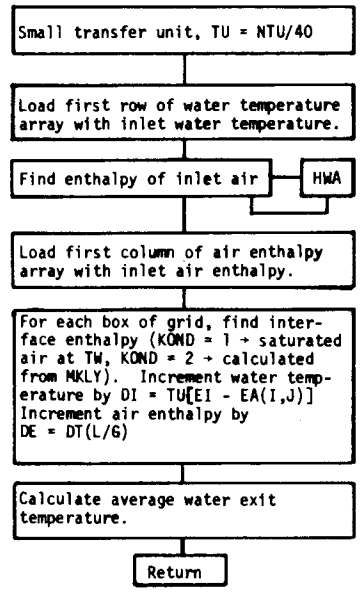
FLOWSHEET FOR CONSTANT TIELINE SLOPE PROGRAM

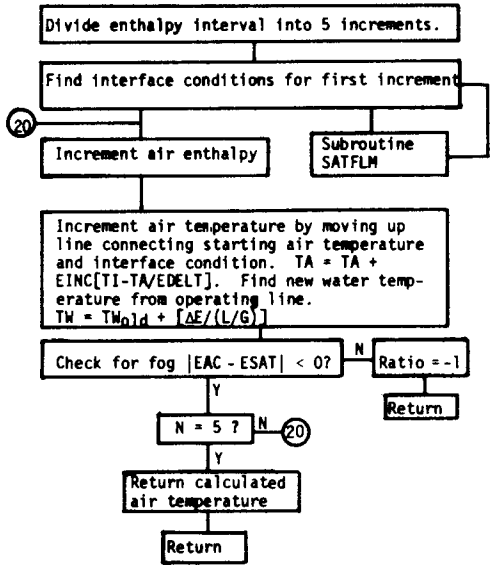
DATE	DRAWN BY	FILE NO.	FIG.
4-26-72	JL	CEPS-X-148	13



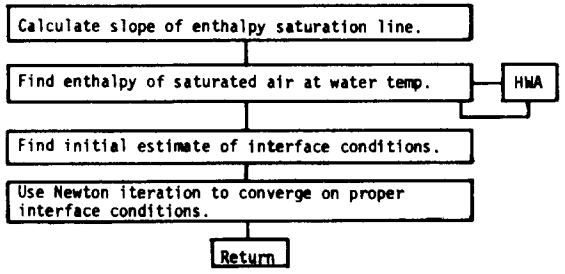
Subroutine PROFIL



Subroutine TOWER



Subroutine MKLY



Subroutine SATFLM

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FLOWSHEET FOR SUBROUTINES USED IN COMPUTER MODELS

DATE	DRAWN BY	FILE NO.	FIG.
4-26-72	JL	CEPS-X-148	14

Constant Tieline Slope Program to Evaluate Transfer Coefficients

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LEVEL 20.1 (AUG 71)                                  OS/360  FORTRAN H

          COMPILER OPTIONS - NAME=  MAIN,OPT=02,LINECNT=60,SIZE=0000K,
                             SOURCE,EBCDIC,NOLIST,NODECK,LOAD,MAP,NOEDIT,NOID,NOXREF
                C     PROGRAM TO CALCULATE LOCAL HEAT AND MASS TRANSFER COEFFICIENTS FOR A
                C     CROSS-FLOW COOLING TOWER (E. G. GREENHOUSE EVAPORATOR PADS).
                C     BY J. FOULIS, K. WEISEL, A. LAMOTTE (FEB. 1972)
                C     MIT CHEMICAL ENGINEERING PRACTICE SCHOOL, CEPS-X-143
                C      MODIFIED BY V. NEHEMIAH, J. LAW, J. ANDRE (APRIL 1972)
                C     MIT CHEMICAL ENGINEERING PRACTICE SCHOOL, CEPS-X-148
                C     ALGORITHM SUGGESTED BY DR. H. S. MICKLEY (1949)
                C     APPROXIMATES SMALL SECTIONS OF CROSS-FLOW TOWER AS BEING LITTLE COUNTER-
                C     FLOW TOWERS.
  ISN 0002            IMPLICIT REAL*8(A-H,O-Z)
  ISN 0003            DIMENSION EIIN(40,40),TW(41,40),EA(40,41),TITLE(10),TIIN(40,40),TA
                     1(40,41),TI(2,2)
                C     EIIN=INTERFACE ENTHALPY ARRAY.
                C     TW=WATER TEMPERATURE ARRAY.
                C     EA=AIR ENTHALPY ARRAY.
                C     TIIN=INTERFACE TEMPERATURE ARRAY.
                C     TA=AIR TEMPERATURE ARRAY.
  ISN 0004      100    FORMAT(I3)
  ISN 0005      101    FORMAT (10A8/8F10.3)
  ISN 0006      200    FORMAT(1H1,10A8/10X,'LIQUID FLOW RATE =',F8.3,' GALLONS/MINUTE'/
                      110X,'WATER TEMP. IN =',F7.1,' DEGREES F'/
                      210X,'WATER TEMP. OUT =',F6.1,' DEGREES F'/
                      310X,'AIR FLOW RATE =',F6.0,' CUBIC FEET/MINUTE'/
                      410X,'AIR TEMP. IN (DRY BULB) =',F7.1,' DEGREES F'/
                      410X,'AIR TEMP. IN (WET BULB) =',F7.1,' DEGREES F'/
                      610X,'AIR TEMP. OUT (DRY BULB) =',F6.1,' DEGREES F'/
                      710X,'PAD VOLUME =',F5.2,' CUBIC FEET'/
                      810X,'PRESSURE =',F6.2,'PSIA')
  ISN 0007       201  FORMAT(1H0,'KOND=',I1,4X,'ITERATION=',I3,4X,'TIE-LINE SLOPE VALUE=
                     1',E13.5,5X,'TAOUT=',F6.1,'    TCALC=',F6.1,'  AVE HUMDTY=',E12.4)
  ISN 0008      202    FORMAT(' KOND=',I1,5X,'TIE-LINE SLOPE DOES NOT CONVERGE')
  ISN 0009       203   FORMAT('1AIR FLOW RATE',F6.0,5X,'LIQUID FLOW',F8.3,5X,'WATER TEMP.',
                      1',F6.1/'0HL*AH =',D12.4/'0HG*AH =',D12.4/'0KG*AM =',D12.4/'1 ')
  ISN 0010      204    FORMAT(3(1H ,3(F5.1,10X)/))
  ISN 0011      205    FORMAT(' OVERALL KG*AM = ',D12.4)
  ISN 0012      206    FORMAT(4(5X,F5.1/))
  ISN 0013      207    FORMAT(' TLF=',F6.1,10X,' TGF='F6.1)
                C     NINC=NUMBER OF SLICES TAKEN PERPENDICULAR TO LIQUID FLOW AND TO GAS FLOW.
  ISN 0014            NINC=40
  ISN 0015            DFNINC=DFLOAT(NINC)
                C     VOLUME=PAD VOLUME (CUBIC FEET).
  ISN 0016            VOLUME=3.21D0
  ISN 0017            ZERO=0.0D0
                C     UNIT=CONVERSION FACTOR FROM GALLONS WATER/CUBIC FOOT AIR TO POUNDS/POUND.
  ISN 0018            UNIT=8.331D0/.695D-1
                C     SLPTI, THE TIE LINE SLOPE, IS -(HL*AH/KG*AM).
                C     MAKE AN INITIAL ESTIMATE OF SLPTI (THE TIE-LINE SLOPE).
  ISN 0019            SLPTI=-.5D0
                C     KARDS=NUMBER OF RUNS FED TO COMPUTER.
  ISN 0020            READ(5,100) KARDS
  ISN 0021            DO 5 NUMRUN=1,KARDS
                C     AL=WATER FLOW RATE (GALLONS/MINUTE).
                C     TWTOP=ENTERING WATER TEMPERATURE (DEGREES F).
                C     TWBOT=EXITING WATER TEMPERATURE (DEGREES F).
                C     G=GAS FLOW RATE (CFM).
                C     TAIN=ENTERING GAS DRY-BULB TEMPERATURE (DEGREES F).
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            C     TAWB=ENTERING GAS WET-BULB TEMPERATURE (DEGREES F).
            C     TAOUT=EXITING GAS DRY-BULB TEMPERATURE (DEGREES F).
            C     PRSSR=ATMOSPHERIC PRESSURE IN PAD (MILLIMETERS OF HG).
ISN 0022            READ(5,101)TITLE,AL,TWTOP,TWBOT,G,TAIN,TAWB,TAOUT,PRSSR
            C     CONVERT PRSSR TO PSIA.
ISN 0023            PRSSR=PRSSR*.19337D-1
ISN 0024            WRITE(6,200)TITLE,AL,TWTOP,TWBOT,G,TAIN,TAWB,TAOUT,VOLUME,PRSSR
            C     KOND=CONDITION. KOND=1 FOR FIRST APPROXIMATION OF TEMPERATURE AND ENTHALPY
            C      PROFILES, AND KOND=2 DURING FINAL DETERMINATION.
ISN 0025            KOND=1
            C     RATIO=L/G (LBS. WATER/LB. AIR). IT IS ALSO THE SLOPE OF THE OPERATING LINE.
ISN 0026            RATIO=AL*UNIT/G
            C     PROFIL FINDS A WATER TEMPERATURE AND AIR ENTHALPY PROFILE THROUGH THE PAD
            C      BY GUESSING AN NTU AND ADJUSTING IT UNTIL IT PRODUCES A PROPER WATER EXIT
            C      TEMPERATURE.
ISN 0027    7       CALL PROFIL(NINC,RATIO,TWTOP,TWBOT,TAWB,TAIN,PRSSR,KOND,EIIN,TW,EA
                   1,DFNINC,EAIN,ANTU)
ISN 0028            C1=.1D4
ISN 0029            B2=SLPTI
ISN 0030            DO 4 K=1,20
ISN 0031            HMAVE=ZERO
ISN 0032            TAAVE=ZERO
ISN 0033            DO 1 I=1,NINC
ISN 0034            IP1=I+1
            C     LOAD THE FIRST COLUMN OF THE AIR TEMPERATURE ARRAY WITH THE INLET DRY-BULB
            C      TEMPERATURE.
ISN 0035    6       TA(I,1)=TAIN
ISN 0036            DO 2 J=1,NINC
ISN 0037            JP1=J+1
            C     MKLY FINDS (BY SUCCESSIVE ALTERATIONS OF SLPTI UNTIL IT PREDICTS THE PROPER
            C      AIR EXIT TEMPERATURE) THE VALUE OF SLPTI (=MINUS THE RATIO OF HL*AH TO
            C      KG*AM). IT ALSO GIVES PROFILES OF INTERFACE TEMPERATURE AND ENTHALPY.
ISN 0038            CALL MKLY(PRSSR,SLPTI,TW(I,J),TW(IP1,J),EA(I,J),EA(I,JP1),EIIN(I,J),TIIN(I
                   1),TIIN(I,J),TA(I,J),TA(I,JP1),RATIO,EIIN(I,J),I,J)
ISN 0039            IF(RATIO .EQ. -.1D1) GO TO 9
ISN 0041    2       CONTINUE
ISN 0042            TAAVE=TAAVE+TA(I,JP1)
            C     HUMDTY=AIR HUMIDITY
ISN 0043            HUMDTY=(EA(I,JP1)-.24D0*(TA(I,JP1)-.32D2))/(.45D0*(TA(I,JP1)-.32D2)+
                   1)+.10717D4)
ISN 0044            HMAVE=HMAVE+HUMDTY
ISN 0045    1       CONTINUE
            C        TAAVE=AVE OUTLET AIR TEMP
ISN 0046            TAAVE=TAAVE/DFNINC
            C     HMAVE=AVERAGE HUMIDITY OF OUTLET AIR.
ISN 0047            HMAVE=HMAVE/DFNINC
            C     TCALC=PREDICTED AIR EXIT DRY-BULB TEMPERATURE.
ISN 0048            TCALC=TAAVE
ISN 0049            C2=TAOUT-TCALC
ISN 0050            IF(DABS(C2) .LT. .5D-1) GO TO 3
ISN 0052            WRITE(6,201)KOND,K,SLPTI,TAOUT,TCALC
ISN 0053            IF(DABS(C2) .GE. DABS(C1) .AND. K .GT. 3) GO TO 10
            C     CORRECT SLPTI.
ISN 0055            CHANGE=DABS(1.D0+C2/(TAIN-TAOUT))
ISN 0056            SLPTI=SLPTI*DSQRT(CHANGE)
ISN 0057            IF (K .GT. 1)SLPTI=B2+C2*(B1-B2)/(C2-C1)
ISN 0059            B1=B2
ISN 0060            C1=C2
```

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ISN 0061          B2=SLPTI
ISN 0062     4    CONTINUE
ISN 0063          WRITE(6,202)KOND
ISN 0064          GO TO 5
ISN 0065    10    KOND=KOND+2
ISN 0066          WRITE(6,202) KOND
ISN 0067          SLPTI=-.5D0
ISN 0068          GO TO 5
ISN 0069     3    CONTINUE
ISN 0070          WRITE(6,201)KOND,K,SLPTI,TACUT,TCALC,HMAVE
ISN 0071          IF(KOND .EQ. 2) GO TO 8
ISN 0073          KOND=2
            C    AKGOVR IS THE OVERALL GAS FILM MASS TRANSFER COEFFICIENT
ISN 0074          AKGOVR=ANTU*AL*8.331D0/VOLUME
ISN 0075          WRITE(6,205)AKGOVR
ISN 0076          GO TO 7
ISN 0077     8    CONTINUE
            C    INTEGRATE DEA/(EI-EA) TO GET NTU (=KG*AM*VOL/G) AND KG*AM.
ISN 0078          DELONE=EIIN(2,1)-EA(1,1)
ISN 0079          DELTWO=EIIN(1,1)-EA(1,2)
ISN 0080          DLMEAN=(DELONE-DELTWO)/DLOG(DELONE/DELTWO)
ISN 0081          ANTU=(EA(1,2)-EA(1,1))/DLMEAN
ISN 0082          AKGAM=ANTU*G*.6950D-1*DENINC/VOLUME
            C    AKGAM=THE GAS FILM MASS TRANSFER COEFFICIENT * AREA OF MASS TRANSFER.
            C    HLAH=THE LIQUID FILM HEAT TRANSFER COEFFICIENT * AREA OF HEAT TRANSFER.
            C    HGAH=THE GAS FILM HEAT TRANSFER COEFFICIENT * AREA OF HEAT TRANSFER.
            C    LIQUID MASS TRANSFER COEFFICIENT IS ASSUMED INFINITE.
ISN 0083          HLAH=-AKGAM*SLPTI
ISN 0084          T=(TA(1,2)+TA(1,1))/.2D1
ISN 0085          HUMDTY=((EA(1,2)+EA(1,1) )*.5D0-.24D0*(T-.32D2))/(.45D0*(T-.32D2)+
                 1.10717D4 )
            C    USE LEWIS CORRELATION TO FIND HG*AH.
ISN 0086          HGAH=AKGAM*(.24D0+.45D0*HUMDTY)
ISN 0087          WRITE(6,203) G,AL,TWTOP,HLAH,HGAH,AKGAM
ISN 0088          WRITE(6,206)TIIN(1,1),TIIN(2,1),TW(2,1),TA(1,2)
            C    TGF IS THE AVERAGE GAS FILM TEMPERATURE IN THE SLICE (1,1)
            C    TLF IS THE AVERAGE LIQUID FILM TEMPERATURE IN SLICE (1,1)
            C    FIND THEM.
ISN 0089          TI(1,1)=TIIN(1,1)
ISN 0090          TI(2,1)=TIIN(2,1)
ISN 0091          IF((TI(1,1)-TI(2,1))*(TA(1,1)-TA(1,2))) 21,22,22
ISN 0092    21    IF((TA(1,1)-TI(1,1))*(TA(1,2)-TI(2,1))) 23,23,24
ISN 0093    22    IF((TA(1,1)-TI(2,1))*(TA(1,2)-TI(1,1))) 23,23,25
ISN 0094    23    TGF=(TI(1,1)+TI(2,1)+TA(1,1)+TA(1,2))/.4D1
ISN 0095          GO TO 26
ISN 0096    24    TGF=     ((TA(1,1)-TI(1,1)+TI(2,1)-TA(1,2))/DLOG((TA(1,1)-TI(1,1))/
                 1(TA(1,2)-TI(2,1)))+TI(1,1)+TI(2,1))*.5D0
ISN 0097          GO TO 26
ISN 0098    25    TGF=     ((TA(1,1)-TI(2,1)+TI(1,1)-TA(1,2))/DLOG((TA(1,1)-TI(2,1))/
                 1(TA(1,2)-TI(1,1)))+TI(1,1)+TI(2,1)) *.5D0
ISN 0099    26    TLF=     ((TW(1,1)-TI(1,1)+TI(2,1)-TW(2,1))/DLOG((TW(1,1)-TI(1,1))/
                 1(TW(2,1)-TI(2,1)))+TI(1,1)+TI(2,1))*.5D0
ISN 0100          WRITE(6,207) TLF,TGF
ISN 0101     9    CONTINUE
ISN 0102     5    CONTINUE
ISN 0103          STOP
ISN 0104          END
```

Subroutine Profile

```
LEVEL 20.1 (AUG 71)                                  OS/360  FORTRAN H

          COMPILER OPTIONS - NAME=  MAIN,OPT=02,LINECNT=60,SIZE=0000K,
                             SOURCE,EBCDIC,NOLIST,NODECK,LOAD,MAP,NOEDIT,NOID,NOXREF
 ISN 0002            SUBROUTINE PROFIL(NINC,RATIO,TWTOP,TWBOT,TAWB,TAIN,PRSSR,KOND,EIIN
                    1,TW,EA,DFNINC,EAIN,ANTU)
 ISN 0003            IMPLICIT REAL*8  (A-H,O-Z)
               C SUBROUTINE BASED ON A PROGRAM BY DR. JAMES PARK
               C    PROFIL FINDS A WATER TEMPERATURE AND AIR ENTHALPY PROFILE THROUGH THE PAD
               C     BY GUESSING AN NTU AND ADJUSTING IT UNTIL IT PRODUCES A PROPER WATER EXIT
               C     TEMPERATURE.
               C    NINC=NUMBER OF SLICES TAKEN PERPENDICULAR TO LIQUID FLOW AND TO GAS FLOW.
               C    RATIO=L/G (LBS. WATER/LB. AIR). IT IS ALSO THE SLOPE OF THE OPERATING LINE.
               C    TWTOP=ENTERING WATER TEMPERATURE (DEGREES F).
               C    TWBOT=EXITING WATER TEMPERATURE (DEGREES F).
               C    TAWB=ENTERING GAS WET-BULB TEMPERATURE (DEGREES F).
               C    TAIN=ENTERING GAS DRY-BULB TEMPERATURE (DEGREES F).
               C    PRSSR=ATMOSPHERIC PRESSURE IN PAD (MILLIMETERS OF HG).
               C    KOND=CONDITION. KOND=1 FOR FIRST APPROXIMATION OF TEMPERATURE AND ENTHALPY
               C     PROFILES, AND KOND=2 DURING FINAL DETERMINATION.
               C    EIIN=INTERFACE ENTHALPY ARRAY.
               C    TW=WATER TEMPERATURE ARRAY.
               C    EA=AIR ENTHALPY ARRAY.
               C    EAIN=INLET AIR ENTHALPY
 ISN 0004            DIMENSION EIIN(40,40),TW(41,40),EA(40,41)
 ISN 0005      300   FORMAT(' NTU DOES NOT CONVERGE')
 ISN 0006      301   FORMAT(' NTU=',D11.4,3X,' KOND=',I1,3X,' TWBOT=',F5.1,3X,' TBOT=',F5.
                    1F5.1)
 ISN 0007            ITRCNT=10
 ISN 0008            TOLER=5.0D-2
 ISN 0009            TWDELT=TWTOP-TWBOT
               C    HWA CALCULATES THE ENTHALPY OF HUMID AIR GIVEN THE PRESSURE, WET-BULB, AND
               C     DRY-BULB TEMPERATURES.
 ISN 0010            CALL HWA(TAWB,TAIN,PRSSR,EAIN,HUM)
 ISN 0011            IF(KOND.EQ. 1) GO TO 1
 ISN 0013            EI=EIIN(1,1)
 ISN 0014            GO TO 2
               C    EI=INTERFACE ENTHALPY.
               C    IF KOND=1, THE PROGRAM HAS NOT YET SET A VALUE OF EI. WE APPROXIMATE IT VIA
               C     HWA, BY ASSUMING THE INTERFACE IS SATURATED AIR AT WATER TEMPERATURE. WHEN
               C     KOND=2, THE PROGRAM HAS A BETTER VALUE OF EI.
 ISN 0015      1     CALL HWA(TWTOP,TWTOP,PRSSR,EI,HUM)
 ISN 0016      2     CONTINUE
               C    ANTU=NUMBER OF TRANSFER UNITS.
               C    MAKE AN INITIAL ESTIMATE OF ANTU.
 ISN 0017            ANTU=TWDELT/(EI-EAIN)
 ISN 0018            B2=ANTU
               C    ITERATE CALLING TOWER, ADJUSTING ANTU, UNTIL IT PREDICTS THE PROPER EXIT
               C     WATER TEMPERATURE.
 ISN 0019            DO 102 I=1,ITRCNT
               C    TOWER INCREMENTS WATER TEMPERATURES FOR A FRACTION OF THE PAD BY THAT
               C     FRACTION OF NTU * THE DRIVING FORCE. AIR ENTHALPY IS INCREMENTED BY THE
               C     WATER TEMPERATURE INCREMENT * AL/G.
               C    TBOT=CALCULATED WATER OUTLET TEMPERATURE.
 ISN 0020            CALL TOWER(TWTOP,TBOT,TAWB,TAIN,PRSSR,RATIO,ANTU,KOND,EIIN,NINC,TW
                    1,EA,DFNINC,EAIN)
 ISN 0021            C2=TWBOT-TBOT
 ISN 0022            IF(DABS(C2) .LT. TOLER) GO TO 100
 ISN 0024            ANTU=ANTU*(1.0D0-C2/TWDELT)
 ISN 0025            IF (I .GT. 1) ANTU=B2-C2*(B2-B1)/(C2-C1)
```

```
ISN 0027            B1=B2
ISN 0028            C1=C2
ISN 0029            B2=ANTU
ISN 0030            WRITE(6,301) ANTU,KOND,TWBOT,TBOT
ISN 0031      102   CONTINUE
ISN 0032            WRITE(6,300)
ISN 0033      100   CONTINUE
ISN 0034            WRITE(6,301) ANTU,KOND,TWBOT,TBOT
ISN 0035            RETURN
ISN 0036            END
```

Subroutine Tower

```
LEVEL 20.1 (AUG 71)                          OS/360  FORTRAN H

          COMPILER OPTIONS - NAME=  MAIN,OPT=02,LINECNT=60,SIZE=0000K,
                             SOURCE,EBCDIC,NOLIST,NODECK,LOAD,MAP,NOEDIT,NOID,NOXREF
  ISN 0002            SUBROUTINE TOWER(TWTOP,TBOT,TAWB,TAIN,PRSSR,RATIO,ANTU,KOND,EIIN,N
                     1INC,TW,EA,DFNINC,EAIN)
  ISN 0003            IMPLICIT REAL*8  (A-H,O-Z)
                C SUBROUTINE BASED ON A PROGRAM BY DR. JAMES PARK
                C    TOWER INCREMENTS WATER TEMPERATURES FOR A FRACTION OF THE PAD BY THAT
                C     FRACTION OF NTU * THE DRIVING FORCE. AIR ENTHALPY IS INCREMENTED BY THE
                C     WATER TEMPERATURE INCREMENT * AL/G.
                C    TWTOP=ENTERING WATER TEMPERATURE (DEGREES F).
                C    TBOT=CALCULATED WATER OUTLET TEMPERATURE.
                C    TAWB=ENTERING GAS WET-BULB TEMPERATURE (DEGREES F).
                C    TAIN=ENTERING GAS DRY-BULB TEMPERATURE (DEGREES F).
                C    PRSSR=ATMOSPHERIC PRESSURE IN PAD (MILLIMETERS OF HG).
                C    RATIO=L/G (LBS. WATER/LB. AIR). IT IS ALSO THE SLOPE OF THE OPERATING LINE.
                C    ANTU=NUMBER OF TRANSFER UNITS.
                C    KOND=CONDITION. KOND=1 FOR FIRST APPROXIMATION OF TEMPERATURE AND ENTHALPY
                C     PROFILES, AND KOND=2 DURING FINAL DETERMINATION.
                C    EIIN=INTERFACE ENTHALPY ARRAY.
                C    NINC=NUMBER OF SLICES TAKEN PERPENDICULAR TO LIQUID FLOW AND TO GAS FLOW.
                C    TW=WATER TEMPERATURE ARRAY.
                C    EA=AIR ENTHALPY ARRAY.
                C    EAIN=ENTHALPY OF AIR ENTERING PAD.
  ISN 0004            DIMENSION EIIN(40,40),TW(41,40),EA(40,41)
  ISN 0005            TU=ANTU/DFNINC
  ISN 0006            DO 5 J=1,NINC
  ISN 0007      5     TW(1,J)=TWTOP
  ISN 0008            CALL HWA(TAWB,TAIN,PRSSR,EAIN,HUM)
  ISN 0009            DO 6 I=1,NINC
  ISN 0010            IP1=I+1
  ISN 0011            EA(I,1)=EAIN
  ISN 0012            DO 7 J=1,NINC
  ISN 0013            JP1=J+1
  ISN 0014            IF (KOND .EQ. 1) GO TO 1
                C    EI=INTERFACE ENTHALPY.
  ISN 0016            EI=EIIN(I,J)
  ISN 0017            GO TO 2
                C    HWA CALCULATES THE ENTHALPY OF HUMID AIR GIVEN THE PRESSURE, WET-BULB, AND
                C     DRY-BULB TEMPERATURES.
  ISN 0018      1     CALL HWA(TW(I,J),TW(I,J),PRSSR,EI,HUM)
  ISN 0019      2     CONTINUE
                C    INCREMENT TEMPERATURE AND ENTHALPY.
                C    DT=WATER TEMPERATURE CHANGE OVER A SLICE.
                C    DE=AIR ENTHALPY CHANGE OVER A SLICE.
  ISN 0020      9     DT=TU*(EI-EA(I,J))
  ISN 0021            DE=DT*RATIO
  ISN 0022            EA(I,JP1)=EA(I,J)+DE
  ISN 0023            TW(IP1,J)=TW(I,J)-DT
  ISN 0024        7   CONTINUE
  ISN 0025      6     CONTINUE
                C    CALCULATE TBOT.
  ISN 0026            TBOT=0.0D0
  ISN 0027            DO 8 J=1,NINC
  ISN 0028      8     TBOT=TBOT+TW(IP1,J)
  ISN 0029            TBOT=TBOT/DFNINC
  ISN 0030            RETURN
  ISN 0031            END
```

Subroutine MKLY

```
LEVEL 20.1 (AUG 71)                                  OS/360  FORTRAN H

          COMPILER OPTIONS - NAME=  MAIN,OPT=02,LINECNT=60,SIZE=0000K,
                             SOURCE,EBCDIC,NOLIST,NODECK,LOAD,MAP,NOEDIT,NOID,NOXREF
  ISN 0002          SUBROUTINE MKLY(PRSSR,SLPTI,TW,TW1,EA,EA1,EI,TI,TA,TAOUT,RATIO,EIIN,I,J)
                   1N,I,J)
  ISN 0003          IMPLICIT REAL*8  (A-H,O-Z)
              C    MKLY FINDS (BY SUCCESSIVE ALTERATIONS OF SLPTI UNTIL IT PREDICTS THE PROPER
              C     AIR EXIT TEMPERATURE) THE VALUE OF SLPTI (=MINUS THE RATIO OF HL*AH TO
              C     KG*AM). IT ALSO GIVES PROFILES OF INTERFACE TEMPERATURE AND ENTHALPY.
              C    MKLY APPROXIMATES ONE SLICE OF PAD AS A COUNTER-FLOW COLUMN.
              C     AUTHOR- J. FOULIS, K. WEISEL, A. LAMOTTE (FEB. 1972)
              C     MIT CHEMICAL ENGINEERING PRACTICE SCHOOL, CEPS-X-143
              C    PRSSR=ATMOSPHERIC PRESSURE IN PAD (MILLIMETERS OF HG).
              C    SLPTI, THE TIE LINE SLOPE, IS -(HL*AH/KG*AM).
              C    TW IS WATER TEMPERATURE AT TOP OF COLUMN.
              C    TW1 IS WATER TEMPERATURE AT BOTTOM OF COLUMN.
              C    EA=AIR ENTHALPY AT BOTTOM OF COLUMN.
              C    EA1=AIR ENTHALPY AT TOP OF COLUMN.
              C    EI=INTERFACE ENTHALPY.
              C    TI=INTERFACE TEMPERATURE.
              C    TA=AIR TEMPERATURE AT BOTTOM OF COLUMN.
              C    TAOUT=AIR TEMPERATURE AT TOP OF COLUMN.
              C    RATIO=L/G (LBS. WATER/LB. AIR). IT IS ALSO THE SLOPE OF THE OPERATING LINE.
              C    EIIN=INTERFACE ENTHALPY ARRAY.
              C    I AND J ARE SLICE CO-ORDINATES.
  ISN 0004    500   FORMAT (' FOG FORMATION IN ROW',I3,', COLUMN',I3)
              C    TAC=AIR TEMPERATURE
              C    EAC=AIR ENTHALPY
  ISN 0005          TAC=TA
  ISN 0006          EAC=EA
  ISN 0007          N=1
              C    INCE=THE NUMBER OF AIR ENTHAPY INCREMENTS.
  ISN 0008          INCE=5
              C    SATFLM FINDS THE CONDITIONS OF THE INTERFACE CORRESPONDING TO A GIVEN WATER
              C     TEMPERATURE, AIR ENTHALPY, FOR A GIVEN TIE-LINE SLOPE.
  ISN 0009          CALL SATFLM(PRSSR,TW1,EA,SLPTI,EI,TI,I,J,N)
  ISN 0010          EDELT=EI-EA
  ISN 0011          EIIN=EI
              C    EINC IS THE INCREMENT OF AIR ENTHALPY.
  ISN 0012          EINC=(EA1-EA)/DFLOAT(INCE)
  ISN 0013          IF (EDELT .EQ. 0.) GO TO 4
              C    INCREMENT AIR ENTHALPY AND TEMPERATURE.
  ISN 0015          TAC=TAC+EINC*(TI-TAC)/EDELT
  ISN 0016          EAC=EAC+EINC
  ISN 0017          DO 3 N=2,INCE
              C    FIND NEW WATER TEMPERATURE FROM OPERATING LINE.
              C    TWC=WATER TEMPERATURE
  ISN 0018          TWC=TW1+(EAC-EA)/RATIO
              C    FIND NEW INTERFACE CONDITIONS.
  ISN 0019          CALL SATFLM(PRSSR,TWC,EAC,SLPTI,EI,TI,I,J,N)
  ISN 0020          EDELT=EI-EAC
  ISN 0021          IF (EDELT .EQ. 0.) GO TO 4
              C    INCREMENT AIR CONDITIONS.
  ISN 0023          TAC=TAC+EINC*(TI-TAC)/EDELT
  ISN 0024          EAC=EAC+EINC
              C    HWA CALCULATES THE ENTHALPY OF HUMID AIR GIVEN THE PRESSURE, WET-BULB, AND
              C     DRY-BULB TEMPERATURES.
              C    ESAT IS THE ENTHALPY OF SATURATED AIR AT TEMPERATURE TAC.
  ISN 0025          CALL HWA(TAC,TAC,PRSSR,ESAT,HUM)
```

```
              C     CHECK FOR FOG.
ISN 0026             FOG=EAC-ESAT
ISN 0027             IF(FOG)3,4,5
ISN 0028      5      EAC=ESAT
ISN 0029      4      WRITE(6,500)I,J
ISN 0030             RATIO=-.1D1
ISN 0031             RETURN
ISN 0032      3      CONTINUE
ISN 0033             TAOUT=TAC
ISN 0034             RETURN
ISN 0035             END
```

Subroutine SATFLM

```
LEVEL 20.1 (AUG 71)                                  OS/360  FORTRAN H

          COMPILER OPTIONS - NAME=  MAIN,OPT=02,LINECNT=60,SIZE=0000K,
                             SOURCE,EBCDIC,NOLIST,NODECK,LOAD,MAP,NOEDIT,NOID,NOXREF
 ISN 0002            SUBROUTINE SATFLM(PRSSR,TW,EA,SLPTI,EI,TI,I,J,N)
 ISN 0003            IMPLICIT REAL*8  (A-H,O-Z)
               C    SATFLM FINDS THE CONDITIONS OF THE INTERFACE CORRESPONDING TO A GIVEN WATER
               C     TEMPERATURE, AIR ENTHALPY, FOR A GIVEN TIE-LINE SLOPE.
               C     AUTHOR- J. FOULIS, K. WEISEL, A. LAMOTTE (FEB. 1972)
               C     MIT CHEMICAL ENGINEERING PRACTICE SCHOOL, CEPS-X-143
               C    PRSSR=ATMOSPHERIC PRESSURE IN PAD (MILLIMETERS OF HG).
               C    TW=WATER TEMPERATURE.
               C    EA=AIR ENTHALPY.
               C    SLPTI, THE TIE LINE SLOPE, IS -(HI*AH/KG*AM).
               C    EI=INTERFACE ENTHALPY.
               C    TI=INTERFACE TEMPERATURE.
 ISN 0004      600   FORMAT(' SATFLM DOES NOT CONVERGE AT POINT',I4,' OF (' I3,',',I3,'
                    1)')
 ISN 0005            SL=(4.31811D-4*TW-3.93844D-2)*TW+1.46021D0
               C    HWA CALCULATES THE ENTHALPY OF HUMID AIR GIVEN THE PRESSURE, WET-BULB, AND
               C     DRY-BULB TEMPERATURES.
 ISN 0006            CALL HWA(TW,TW,PRSSR,ESAT,HUM)
 ISN 0007            T2=TW+(EA-ESAT)/(SL-SLPTI)
 ISN 0008            CALL HWA(T2,T2,PRSSR,EG1,HUM)
 ISN 0009            EG2=EA+(T2-TW)*SLPTI
 ISN 0010            F1=EG1-EG2
 ISN 0011            SL=(4.31811D-4*T2-3.93844D-2)*T2+1.46021D0
 ISN 0012            TI=(EA-EG1-SLPTI*TW+SL*T2)/(SL-SLPTI)
               C    ITERATE TO ACHIEVE CONVERGENCE.
 ISN 0013            DO 1 L=1,30
 ISN 0014            CALL HWA(TI,TI,PRSSR,EG1,HUM)
 ISN 0015            EG2=EA+(TI-TW)*SLPTI
 ISN 0016            F2=EG1-EG2
 ISN 0017            IF(DABS(F2/EG1).LT..2D-2) GO TO 2
 ISN 0019            T1=T2
 ISN 0020            T2=TI
 ISN 0021            TI=T2+(T1-T2)*F2/(F2-F1)
 ISN 0022            F1=F2
 ISN 0023      1     CONTINUE
 ISN 0024            WRITE(6,600) N,I,J
 ISN 0025      2     EI=(EG1+EG2)/2.D0
 ISN 0026            TI=TW+(EI-EA)/SLPTI
 ISN 0027            RETURN
 ISN 0028            END
```

Subroutine HWA

```
LEVEL 20.1 (AUG 71)                                OS/360  FORTRAN H

          COMPILER OPTIONS - NAME=  MAIN,OPT=02,LINECNT=60,SIZE=0000K,
                             SOURCE,EBCDIC,NOLIST,NODECK,LOAD,MAP,NOEDIT,NOID,NOXREF
 ISN 0002            SUBROUTINE  HWA (TA,TB,PB,H,HS)
                C
                C    AUTHOR - DR. R. E. FUNDERLIC, FORMERLY COMPUTING TECHNOLOGY CENTER
                C                                (CIRCA 1958)
                C    CALCULATES THE ENTHALPY, SPECIFIC HEAT, VAPOR PRESSURE AND
                C        SPECIFIC HUMIDITY OF HUMID AIR.  ENTHALPY HAS A ZERO  VALUE AT
                C        32 DEG. F.
                C        TA - AIR WET BULB, DEG. F.
                C        TB - AIR DRY BULB, DEG. F.
                C        PB - BAROMETRIC PRESSURE, PSIA.
                C        H  - ENTHALPY, BTU/POUND OF HUMID AIR.
                C        HS - SPECIFIC HUMIDITY, POUNDS OF WATER/POUND OF DRY AIR.
                C
 ISN 0003            IMPLICIT REAL*8  (A-H,O-Z)
 ISN 0004            CA = 459.69D0
 ISN 0005            CB = 32.0D0
 ISN 0006            ONE = 1.0D0
 ISN 0007            TAR = TA - CB
 ISN 0008            TAB = TA + CA
 ISN 0009            HV = 1075.16 + 1.7822*TAB - .66948D-3*(TAB**2 - 2.41759D+5)
                    1     - 3.3793D+2*DLOG(TAB/491.69)
 ISN 0010            PU = 671.69/TAB
 ISN 0011            PPV = -18.1973D0*(PU - ONE) + 5.02808D0*DLOG(PU) - 3.18125D-7
                    1       *(DEXP(26.1205*(ONE - ONE/PU)) - ONE)
                    2    + .18726D-1*(DEXP(-8.039453D0*(PU - ONE)) - ONE)
 ISN 0012            PV = DEXP(PPV)*14.696D0
 ISN 0013            HS = 0.62197D0*PV/(PB - PV)
 ISN 0014            XI = HS*HV
 ISN 0015            TBR=TA-CB
 ISN 0016            HA = TBR*(.24006D0 + 3.590D-6*TBR)
 ISN 0017            H = HA + XI
 ISN 0018            RETURN
 ISN 0019            END
```

Computer Nomenclature

AKGAM	product of gas film mass transfer coefficient and area, lb/min-ft^3
AKGLOG	log of AKGAM
AKGOVR	product of overall gas film mass transfer coefficient and area, lb/min-ft^3
AL	liquid flow rate (gal/min)
ANTU	overall or local mass transfer unit = $\frac{K_g a_M V}{G}$ or $\frac{k_g a_M V}{G}$
ANULL	dummy variable to fit read statement
B1	previous iteration value
B2	value of iterated variable
C1	previous iteration value of difference between guessed and calculated variable
C2	value of difference between guess and calculated variable presently
CONLOG	log of basis for tieline slope variation (SLPTI * (TFZERO)$^{0.00916}$)
CONST	basis for tieline slope variation (SLPTI * (TFZERO)$^{0.00916}$)
DE	air enthalpy change over a small increment in TOWER
DELONE	enthalpy driving force at bottom of column, Btu/lb dry air
DELTWO	enthalpy driving force at top of column, Btu/lb dry air
DFNINC	number of slices in grid (real constant 40)
DLMEAN	log mean of enthalpy driving force through one box
DT	water temperature change over a small increment in TOWER, °F
EA	air enthalpy array
EA1	outlet air enthalpy from a small increment, Btu/lb dry air
EAC	air enthalpy in MKLY, Btu/lb dry air
EAIN	inlet air enthalpy, Btu/lb dry air
EDELT	enthalpy driving force, Btu/lb dry air
EG1	enthalpy at interface via HWA, Btu/lb dry air

EG2	enthalpy at interface via Newton iteration, Btu/lb dry air
EI	interface enthalpy, Btu/lb dry air
EIIN	interface enthalpy array, Btu/lb dry air
EINC	increment of air enthalpy, Btu/lb dry air
ESAT	enthalpy of saturated air at water temperature, Btu/lb dry air
F1	difference in air enthalpies at interface - first iteration, Btu/lb dry air
F2	difference in air enthalpies at interface - second iteration, Btu/lb dry air
FLMTMP	subroutine to calculate liquid and gas film temperatures
G	air flow rate, ft^3/min
GPRIME	superficial gas velocity, lb/min-ft^2
HGAH	product of gas film heat transfer coefficient and area, Btu/ft^3-min-°F
HLAH	product of liquid film heat transfer coefficient and area, Btu/ft^3-min-°F
HMAVE	average air exit humidity, lb water/lb dry air
HUMDTY	air humidity, lb water/lb dry air
HWA	subroutine to calculate enthalpy of saturated air at a given temperature
INCE	number of enthalpy increments for each box
KARDS	number of runs fed to computer
KOND	indicator of initial approximation or final determination
MKLY	subroutine to determine air temperature array and interface conditions
NINC	number of slices in grid, integer variable = 40
PDHITE	pad height, ft
PRIMEL	superficial liquid velocity, lb/min-ft^2
PROFIL	subroutine to calculate air enthalpy and water temperature profiles and overall NTU

PRSSR	pressure in pad, mm Hg
RATIO	liquid flow rate/gas flow rate (also slope of operating line)
SATFLM	subroutine to determine interface conditions
SL	slope of enthalpy saturation line, Btu/lb-°F
SLPTI	slope of tieline ($-h_L/k_g$), Btu/min-ft^2-°F)/(lb/min-ft^2)
SLPTIA	absolute value of tieline slope, (Btu/min-ft^2-°F)/(lb/min-ft^2)
SLPTIO	basis of tieline slope variation
T1	intermediate temperature in Newton iteration, °F
T2	intermediate temperature in Newton iteration, °F
TA	air temperature array, °F
TA1	air temperature out of one increment, °F
TAAVE	calculated average air exit temperature from pad, °F
TAC	air temperature in MKLY, °F
TAIN	inlet air temperature (dry bulb), °F
TAOUT	exit air temperature (dry bulb), °F
TAWB	inlet air temperature (wet bulb), °F
TBOT	calculated exit water temperature, °F
TCALC	calculated average at exit temperature from pad, °F
TFZERO	base film temperature in tieline slope variation, °F
TGF	gas film temperature, °F
TI	interface temperature, °F
TIIN	interface temperature array, °F
TI1	interface temperature at bottom of column, °F
TLF	liquid film temperature, °F
TOLER	tolerance for convergence
TOWER	subroutine to determine air enthalpy and water temperature profile

TU	overall transfer unit for one increment (= NTU/40)
TW	water temperature array, °F
TW1	water temperature at bottom of column (each increment), °F
TWBAR	average water temperature in increment, °F
TWBOT	experimental water exit temperature, °F
TWC	water temperature in MKLY, °F
TWDELT	difference between experimental and calculated water exit temperature, °F
TWTOP	inlet water temperature, °F
UNIT	conversion factor from gpm/cfm to lb water/lb air
VOLUME	pad volume, ft^3

Design Program - LANSIM*

After obtaining correlations for the local heat and mass transfer coefficients for a given pad, the response of outlet conditions with varying inlet conditions can be determined for a wide range of values. Performance charts could then be drawn to help facilitate greenhouse design.

The range of parameters to be studied should be the limiting winter and summer conditions as described by Furlong (5). The inlet air conditions in the summer would be taken as 91°F dry bulb and 79.2°F wet bulb; in winter as saturated air at either 50 or 60°F depending on the required ΔT for proper pad operation. Water inlet temperature would be 110°F in summer and 70°F in winter. The pad would be taken as multiples of 34 in. width and 2 in. thickness with variable height, say 4, 6, and 8 ft. The liquid flow rate would be kept slightly below flooding. The gas flow rate would be variable (say 100, 150, and 200 cfm/ft^2 pad area). Thus using these values of parameters, 27 cases could be studied (3 inlet temperatures, 3 pad heights, 3 air flow rates). In all cases the response variables would be the water exit temperature (or ΔT for water in and out), and the air exit temperature and humidity.

The input data are as follows: liquid flow rate, inlet water temperature, air flow rate, inlet air temperature (dry and wet bulb), pad height, and pressure. The pad volume, slope of the operating line, and superficial liquid and gas velocities are then calculated.

During the initial approximation, interface temperatures have not yet been determined, so that film temperatures cannot be calculated. Thus, the local transfer coefficients are calculated from the correlation using only the superficial gas and liquid velocity dependence. Calculated tieline slope and overall NTU are used in subroutines TOWER and MKLY to determine

*Law, Andre, and Nehemiah simulation.

water temperature, air enthalpy and temperature, and interface enthalpy and temperature arrays. From this initial approximation, film temperatures for each box in the pad can be calculated.

In the final determination the calculated film temperatures are used in the correlation to give values of local liquid heat transfer and gas mass transfer coefficients for each box. The tieline slope given from the correlation is then used by subroutines TOWER and MKLY to determine the temperature and enthalpy profiles through the pad.

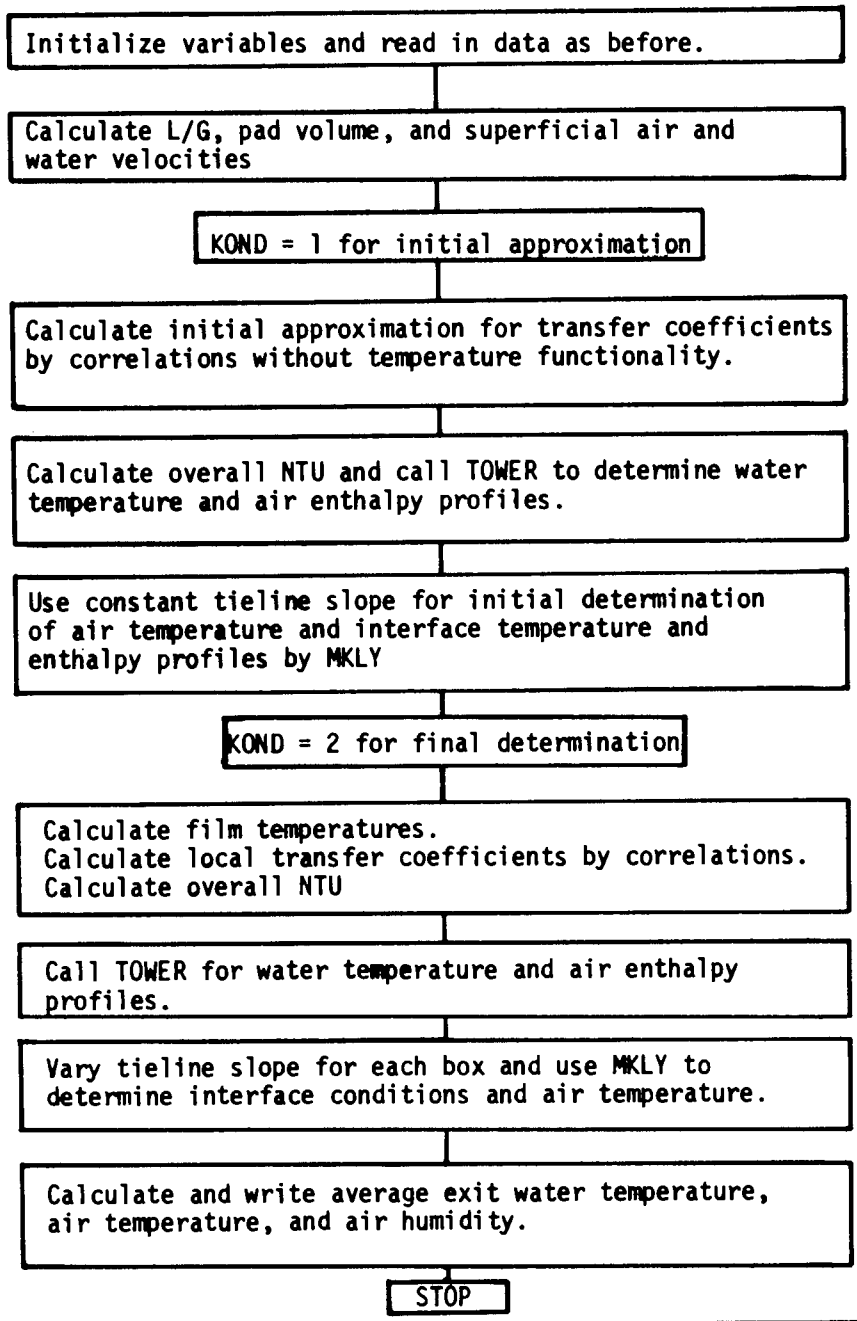
An average exit water temperature and air exit temperature and humidity are calculated and printed out as response variables. The value of the tieline slope has thus been varied according to film temperature and gives more accurate profiles of temperature and enthalpy than the constant tieline assumption. If desired, the program could go through one more iteration to improve the calculated profiles and film temperatures. The logic flowsheet of the program is given in Fig. 15, and the listing of the program follows.

10.2 Experimental Errors and Sensitivity Analysis

Estimates of the error in measurement of the various experimental parameters were made: $L = \pm 2.5\%$, $t_{in} = \pm 0.2°F$, $t_{out} = \pm 0.2°F$, $G = \pm 3\%$, $T_{in} = \pm 0.4°F$, $T_{wbin} = \pm 0.6°F$, $T_{out} = \pm 0.2°F$, and $P = \pm 20$ mm Hg. A sensitivity analysis was conducted on a typical run, successively varying each parameter by the estimated error. The computer model was used to calculate the new heat and mass transfer coefficients for the perturbation. The results of the analysis are summarized in Table 2.

Table 2. Sensitivity Analysis

		% Variation			
Parameter	Perturbation	$k_{g\ av}$	h_L	h_g	k_g
L	$\pm$ 2.5%	+2.9	+3.2	+2.4	+2.4
t_{in}	$\pm$ 0.2°F	+0.9	+1.1	+1.1	+1.1
t_{out}	+ 0.2°F	-2.7	-3.5	-0.7	-0.8
G	+ 3%	-0.4	-1.2	+1.6	+1.6
T_{in}	+ 0.4°F	0.0	-0.6	+2.1	+2.2
T_{wbin}	+ 0.6°F	+4.5	+4.5	+7.3	+7.2
T_{out}	+ 0.2°F	0.0	+1.0	-3.0	-3.0
P	+ 20 mm Hg	+2.4	+0.5	+1.4	+1.5



MASSACHUSETTS INSTITUTE OF TECHNOLOGY
SCHOOL OF CHEMICAL ENGINEERING PRACTICE
AT
OAK RIDGE NATIONAL LABORATORY

FLOWSHEET FOR DESIGN PROGRAM LANSIM

DATE	DRAWN BY	FILE NO.	FIG.
4-26-72	JL	CEPS-X-148	15

Design Program - LANSIM

```
LEVEL 20.1 (AUG 71)                                    OS/360  FORTRAN H

          COMPILER OPTIONS - NAME=  MAIN,OPT=02,LINECNT=60,SIZE=0000K,
                             SOURCE,EBCDIC,NOLIST,NODECK,LOAD,MAP,NOEDIT,NOID,NOXREF
                   C      PROGRAM TO CALCULATE EXIT AIR AND WATER CONDITIONS FOR A CROSS
                   C      FLOW COOLING TOWER (E.G. GREENHOUSE EVAPORATOR PADS), GIVEN INLET
                   C      CONDITIONS FOR AIR AND WATER AND CORRELATIONS FOR LOCAL HEAT AND MASS
                   C      MASS TRANSFER COEFFICIENTS
                   C      BY V. NEHEMIAH, J. LAW, J. ANDRE (APRIL 1972)
                   C      MIT CHEMICAL ENGINEERING PRACTICE SCHOOL, CEPS-X-148
                   C     ALGORITHM SUGGESTED BY DR. H. S. MICKLEY (1949)
                   C     APPROXIMATES SMALL SECTIONS OF CROSS-FLOW TOWER AS BEING LITTLE COUNTER-
                   C     FLOW TOWERS.
  ISN 0002          IMPLICIT REAL*8(A-H,O-Z)
  ISN 0003          DIMENSION EIIN(41,40),TW(41,40),EA(40,41),TITLE(10),TIIN(41,40),TA
                   1(40,41)
                   C     EIIN=INTERFACE ENTHALPY ARRAY.
                   C     TW=WATER TEMPERATURE ARRAY.
                   C     EA=AIR ENTHALPY ARRAY.
                   C     TIIN=INTERFACE TEMPERATURE ARRAY.
                   C     TA=AIR TEMPERATURE ARRAY.
  ISN 0004         100   FORMAT(I3)
  ISN 0005         101   FORMAT (10A8/8F10.3)
  ISN 0006         200   FORMAT(1H1,10A8/10X,'LIQUID FLOW RATE =',F8.3,' GALLONS/MINUTE'/
                        110X,'WATER TEMP. IN =',F7.1,' DEGREES F'/
                        210X,'AIR FLOW RATE =',F6.0,' CUBIC FEET/MINUTE'/
                        310X,'AIR TEMP. IN (DRY BULB) =',F7.1,' DEGREES F'/
                        410X,'AIR TEMP. IN (WET BULB) =',F7.1,' DEGREES F'/
                        510X,'PAD HEIGHT=',F5.2,' FEET'/
                        610X,'PRESSURE =',F6.2,'PSIA')
  ISN 0007          201  FORMAT(1H0,'KOND=',I1,4X,'   NUMRUN=',I3,4X,'TIE-LINE SLOPE VALUE=
                        1',E13.5,5X,'TAOUT=',F6.1,'  AVE HUMIDITY=',E12.4)
  ISN 0008         206   FORMAT(4(5X,F5.1/))
  ISN 0009         207   FORMAT(' TLF=',F6.1,10X,' TGF=',F6.1)
  ISN 0010          208  FORMAT(' VOL=',D12.4,'LPRIME=',D12.4,'GPRIME=',D12.4,'HL*AH=',D12.4
                        14,'AKGAM',D12.4/' SL=',D12.4,'KGOV=',D12.4,'NTU=',D12.4)
  ISN 0011          210  FORMAT(1X,'EXIT WATER TEMPERATURE=',F10.3)
                   C     NINC=NUMBER OF SLICES TAKEN PERPENDICULAR TO LIQUID FLOW AND TO GAS FLOW.
  ISN 0012          NINC=40
  ISN 0013          DFNINC=DFLOAT(NINC)
  ISN 0014          ZERO=0.0D0
                   C     UNIT=CONVERSION FACTOR FROM GALLONS WATER/CUBIC FOOT AIR TO POUNDS/POUND.
  ISN 0015          UNIT=8.331D0/.6950D-1
                   C     KARDS=NUMBER OF RUNS FED TO COMPUTER.
  ISN 0016          READ(5,100) KARDS
  ISN 0017          DO 5 NUMRUN=1,KARDS
  ISN 0018          IFLG=1
  ISN 0019          KOUNT=1
  ISN 0020          KEY=1
                   C     AL=WATER FLOW RATE (GALLONS/MINUTE).
                   C     TWTOP=ENTERING WATER TEMPERATURE (DEGREES F).
                   C     G=GAS FLOW RATE (CFM).
                   C     TAIN=ENTERING GAS DRY-BULB TEMPERATURE (DEGREES F).
                   C     TAWB=ENTERING GAS WET-BULB TEMPERATURE (DEGREES F).
                   C      PDHITE=PAD HEIGHT (FEET)
                   C     PRSSR=ATMOSPHERIC PRESSURE IN PAD (MILLIMETERS OF HG).
  ISN 0021          READ (5,101) TITLE,AL,TWTOP,G,TAIN,TAWB,PDHITE,PRSSR,ANULL
                   C     CONVERT PRSSR TO PSIA.
  ISN 0022          PRSSR=PRSSR*.19337D-1
  ISN 0023          WRITE(6,200 ) TITLE,AL,TWTOP,G,TAIN,TAWB,PDHITE,PRSSR
```

```
            C     KOND=CONDITION. KOND=1 FOR FIRST APPROXIMATION OF TEMPERATURE AND ENTHALPY
            C      PROFILES, AND KOND=2 DURING FINAL DETERMINATION.
ISN 0024           KOND=1
            C     RATIO=L/G (LBS. WATER/LB. AIR). IT IS ALSO THE SLOPE OF THE OPERATING LINE.
ISN 0025           RATIO=AL*UNIT/G
            C     VOLUME=PAD VOLUME (CUBIC FEET).
ISN 0026           VOLUME=2.D0*3.4D1*PDHITE/1.44D2
            C     PRIMEL=SUPERFICIAL LIQUID VELOCITY (LB/MIN*FT2)
ISN 0027           PRIMEL=AL*8.331D0*1.44D2/(2.D0*3.4D1)
            C      GPRIME=SUPERFICIAL GAS VELOCITY (LB/MIN*FT2)
ISN 0028           GPRIME=G*.6950D-1*1.2D1/(3.4D1*PDHITE)
            C      HLAH= THE LIQUID FILM HEAT TRANSFER COEFF.* AREA OF HEAT TRANSFER
            C      AKGAM=THE GAS FILM MASS TRANSFER COEFF.* AREA OF MASS TRANSFER
            C      IF KOND=1, ASSUME NO TEMPERATURE DEPENDENCE OF HL*AH AND KG*AM.
ISN 0029           HLLOG=DLOG(2.457D0)+.399D0*DLOG(GPRIME)+.147D0*DLOG(PRIMEL)
ISN 0030           HLAH=DEXP(HLLOG)
ISN 0031           AKGLOG=DLOG(4.2685D0)+DLOG(GPRIME)-.146D-1*DLOG(PRIMEL)
ISN 0032           AKGAM=DEXP(AKGLOG)
ISN 0033           SLPTI=-HLAH/AKGAM
ISN 0034           TGF=(TAIN+TAWB)*.5D0
ISN 0035      7    CONTINUE
            C      SL=APPROX. SLOPE OF THE AIR ENTHALPY SATURATION CURVE
ISN 0036           SL=(4.31811D-4*TGF-3.93844D-2)*TGF+1.46021D0
            C      AKGOVR= OVERALL GAS FILM MASS TRANSFER COEFFICIENT
            C      CALCULATED FROM: 1/AKGOVR=1/AKGAM+SL/HLAH
ISN 0037           AKGOVR=AKGAM*HLAH/(HLAH+SL*AKGAM)
            C      ANTU=NUMBER OF TRANSFER UNITS (KG*AM*V/L)
ISN 0038           ANTU=AKGOVR*VOLUME/(AL*8.331D0)
ISN 0039           WRITE(6,208) VOLUME,PRIMEL,GPRIME,HLAH,AKGAM,SL,AKGOVR,ANTU
            C     TOWER INCREMENTS WATER TEMPERATURES FOR A FRACTION OF THE PAD BY THAT
            C      FRACTION OF NTU * THE DRIVING FORCE. AIR ENTHALPY IS INCREMENTED BY THE
            C      WATER TEMPERATURE INCREMENT * AL/G.
            C     TBOT=CALCULATED WATER OUTLET TEMPERATURE.
ISN 0040           CALL TOWER(TWTOP,TBOT,TAWB,TAIN,PRSSP,RATIO,ANTU,IFLG,EIIN,NINC,TW
                  1,EA,DFNINC,EAIN)
ISN 0041           WRITE(6,210) TBOT
            C      TAAVE=AVE. AIR EXIT TEMPERATURE
            C      HMAVE=AVE. EXIT HUMIDITY
ISN 0042           TAAVE=ZERO
ISN 0043           HMAVE=ZERO
ISN 0044           DO 1 I=1,NINC
ISN 0045           IP1=I+1
            C      FIRST COLUMN OF AIR TEMP ARRAY= INLET DRY BULB TEMP
ISN 0046           TA(I,1)=TAIN
ISN 0047           DO 2 J=1,NINC
ISN 0048           JP1=J+1
            C       IF KOND=1, CONSTANT TIE-LINE SLOPE IS ASSUMED IN ORDER TO CALCULATE INTER
            C      FACE TEMPERATURE AND ENTHALPY. IF KOND=2, THESE INTERFACE CONDITIONS ARE U
            C      USED TO CALCULATE FILM TEMPERATURES FOR USE IN CORRELATIONS FOR HL*AH AND
            C      KG*AM
ISN 0049           IF (KOND .EQ. 1) GO TO 4
            C      SUBROUTINE FLMTMP CALCULATES FILM TEMPS BY LOG MEAN OF INLET,OUTLET,
            C      AND INTERFACE CONDITIONS
ISN 0051           CALL FLMTMP(TA(I,J),TA(I,JP1),TIIN(I,J),TIIN(IP1,J),TW(I,J),TW(IP1
                  1,J),TGF,TLF)
ISN 0052           HLLOG=DLOG(2.457D0)+.399D0*DLOG(GPRIME)+.147D0*DLOG(PRIMEL)+TLF*
                  1.1D-1
ISN 0053           HLAH=DEXP(HLLOG)
```

```
ISN 0054          AKGLOG=DLOG(4.2685D0)+DLOG(GPRIME)-.146D-1*DLOG(PRIMEL)+.902D-2*
                 1TGF
ISN 0055          AKGAM=DEXP(AKGLOG)
            C    SLPTI, THE TIE LINE SLOPE, IS -(HL*AH/KG*AM).
ISN 0056          SLPTI=-HLAH/AKGAM
ISN 0057     4    CONTINUE
            C     MKLY FINDS AN INTERFACE TEMP. AND ENTHALPY AND AIR EXIT TEMP GIVEN TIELINE
ISN 0058          CALL MKLY(PRSSR,SLPTI,TW(I,J),TW(IP1,J),EA(I,J),EA(I,JP1),EIIN(I,J),TIIN(I
                 1),TIIN(I,J),TA(I,J),TA(I,JP1),RATIO,EIIN(I,J),I,J)
ISN 0059          CALL SATFLM(PRSSR,TW(IP1,J),EA(I,J),SLPTI,EIIN(IP1,J),TIIN(IP1,J),
                 1I,J,1)
ISN 0060          IF (RATIO .EQ. -.1D1) GO TO 9
ISN 0062     2    CONTINUE
            C    HUMDTY=AIR HUMIDITY
ISN 0063          HUMDTY=(EA(I,JP1)-.24D0*(TA(I,JP1)-.32D2))/(.45D0*(TA(I,JP1)-.32D2)+
                 1)+.10717D4)
ISN 0064          TAAVE=TAAVE+TA(I,JP1)
ISN 0065          HMAVE=HMAVE+HUMDTY
ISN 0066     1    CONTINUE
            C      TAAVE=AVE OUTLET AIR TEMP
ISN 0067          TAAVE=TAAVE/DFNINC
            C    HMAVE=AVERAGE HUMIDITY OF OUTLET AIR.
ISN 0068          HMAVE=HMAVE/DFNINC
ISN 0069          WRITE (6,201) KOND,NUMRUN,SLPTI,TAAVE,HMAVE
ISN 0070          IF (KOND .EQ. 2) GO TO 8
ISN 0072    82    CONTINUE
ISN 0073          KOND=2
            C     SUBROUTINE FLMTMP CALCULATES FILM TEMPS BY LOG MEAN OF INLET,OUTLET,
            C     AND INTERFACE CONDITIONS
ISN 0074          CALL FLMTMP(TA(1,1),TA(1,2),TIIN(1,1),TIIN(2,1),TW(1,1),TW(2,1),
                 1TGF,TLF)
ISN 0075          WRITE(6,207) TLF,TGF
ISN 0076          HLLOG=DLOG(2.457D0)+.399D0*DLOG(GPRIME)+.147D0*DLOG(PRIMEL)+TLF*
                 1.1D-1
ISN 0077          HLAH=DEXP(HLLOG)
ISN 0078          AKGLOG=DLOG(4.2685D0)+DLOG(GPRIME)-.146D-1*DLOG(PRIMEL)+.902D-2*
                 1TGF
ISN 0079          AKGAM=DEXP(AKGLOG)
ISN 0080          GO TO 7
ISN 0081     8    CONTINUE
ISN 0082          IF (KOUNT .EQ. 2) GO TO 81
            C     SECOND ITERATION
ISN 0084          KOUNT=2
ISN 0085          GO TO 82
ISN 0086    81    CONTINUE
ISN 0087          IF (KEY .EQ. 2) GO TO 71
            C     THIRD ITERATION
ISN 0089          KEY=2
ISN 0090          GO TO 82
ISN 0091    71    CONTINUE
ISN 0092     9    CONTINUE
ISN 0093     5    CONTINUE
ISN 0094          STOP
ISN 0095          END
```

```
LEVEL 20.1 (AUG 71)                                  OS/360  FORTRAN H

          COMPILER OPTIONS - NAME=  MAIN,OPT=02,LINECNT=60,SIZE=0000K,
                             SOURCE,EBCDIC,NOLIST,NODECK,LOAD,MAP,NOEDIT,NOID,NOXREF
  ISN 0002            SUBROUTINE FLMTMP(TA,TA1,TI,TI1,TW,TW1,TGF,TLF)
  ISN 0003            IMPLICIT REAL*8 (A-H,O-Z)
                 C    SUBROUTINE FLMTMP CALCULATES FILM TEMPS BY LOG MEAN OF INLET,OUTLET,
                 C    AND INTERFACE CONDITIONS
                 C TGF IS THE AVERAGE GAS FILM TEMPERATURE IN THE SLICE (I,J)
                 C TLF IS THE AVERAGE LIQUID FILM TEMPERATURE IN SLICE (I,J)
                 C FIND THEM
  ISN 0004            IF((TI-TI1)*(TA-TA1)) 21,22,22
  ISN 0005       21   IF((TA-TI)*(TA1-TI1)) 23,23,24
  ISN 0006       22   IF((TA-TI1)*(TA1-TI)) 23,23,25
  ISN 0007       23   TGF=(TI+TI1+TA+TA1)/.4D1
  ISN 0008            GO TO 26
  ISN 0009       24   TGF=((TA-TI+TI1-TA1)/DLOG((TA-TI)/(TA1-TI1))+TI+TI1)*.5D0
  ISN 0010            GO TO 26
  ISN 0011       25   TGF=((TA-TI1+TI-TA1)/DLOG((TA-TI1)/(TA1-TI))+TI+TI1)*.5D0
  ISN 0012       26   TLF=((TW-TI+TI1-TW1)/DLOG((TW-TI)/(TW1-TI1))+TI+TI1)*.5D0
  ISN 0013            RETURN
  ISN 0014            END
```

Subroutines - TOWER, MKLY, SATFLM, and HWA from constant tieline slope program used to evaluate transfer coefficients.

10.3 Sample Calculations

A sample calculation for Run 48 (adiabatic-isothermal run) is as follows:

Data: $L' = 364.16$ (lb/min-ft^2)

$G' = 11.4$ (lb/min-ft^2)

$T_{wbin} = 64.85°F \simeq$ average water temperature

$T_{in} = 84.19°F$

$T_{out} = 78.03°F$

$H_{in} = 0.0087$ (lb water/lb dry air)

$H_{out} = 0.0101$ (lb water/lb dry air)

$H_{sat} = 0.0131$ (lb water/lb dry air)

For the polyurethane pad,

$$\text{area for water flow} = \frac{(33 \text{ in.})(1 \text{ in.})}{(144 \text{ in.}^2/\text{ft}^2)} = 0.229 \text{ ft}^2$$

$$\text{area for air flow} = \frac{(33 \text{ in.})(84 \text{ in.})}{(144 \text{ in.}^2/\text{ft}^2)} = 19.25 \text{ ft}^2$$

$\therefore \quad L = 364.16(0.229) = 83.39$ lb/min

and $\quad G = 11.4(19.25) = 219.4$ lb dry air/min

$V = 1.6 \text{ ft}^3$

<u>Humid Heat Capacity</u>

$$H_{mean} = \frac{H_{in} + H_{out}}{2} = 0.0094 \text{ (lb water/lb dry air)}$$

$$c_H = 0.24 + 0.45\, H_{mean} = 0.244 \text{ Btu/lb dry air-°F}$$

Transfer Coefficients

$$h_g a_H = \frac{Gc_H}{V} \ln\left(\frac{T_{in} - T_{wb}}{T_{out} - T_{wb}}\right) = \frac{(219.4)(0.244)}{1.6} \ln\left(\frac{84.19 - 64.85}{78.03 - 64.85}\right)$$

$$= 12.83 \text{ Btu/min-°F/ft}^3 \qquad (6)$$

$$k_g a_M = \frac{G}{V} \ln\left(\frac{H_{sat} - H_{in}}{H_{sat} - H_{out}}\right) = \frac{219.4}{1.6} \ln\left(\frac{0.0131 - 0.0087}{0.0131 - 0.0101}\right)$$

$$= 52.53 \text{ lb/min-ft}^3 \qquad (7)$$

Lewis Number

$$Le = \frac{h_g a_H}{(k_g a_M) c_H} = \frac{12.83}{(52.53)(0.244)} = 1.0$$

Gas Film Temperature

$$T_{GF} = T_{wb} + \frac{1}{2} \frac{T_{in} - T_{out}}{\ln\left(\frac{T_{in} - T_{wb}}{T_{out} - T_{wb}}\right)} = 64.85 + \frac{1}{2} \frac{84.19 - 78.03}{\ln\left(\frac{84.19 - 64.85}{78.03 - 64.85}\right)}$$

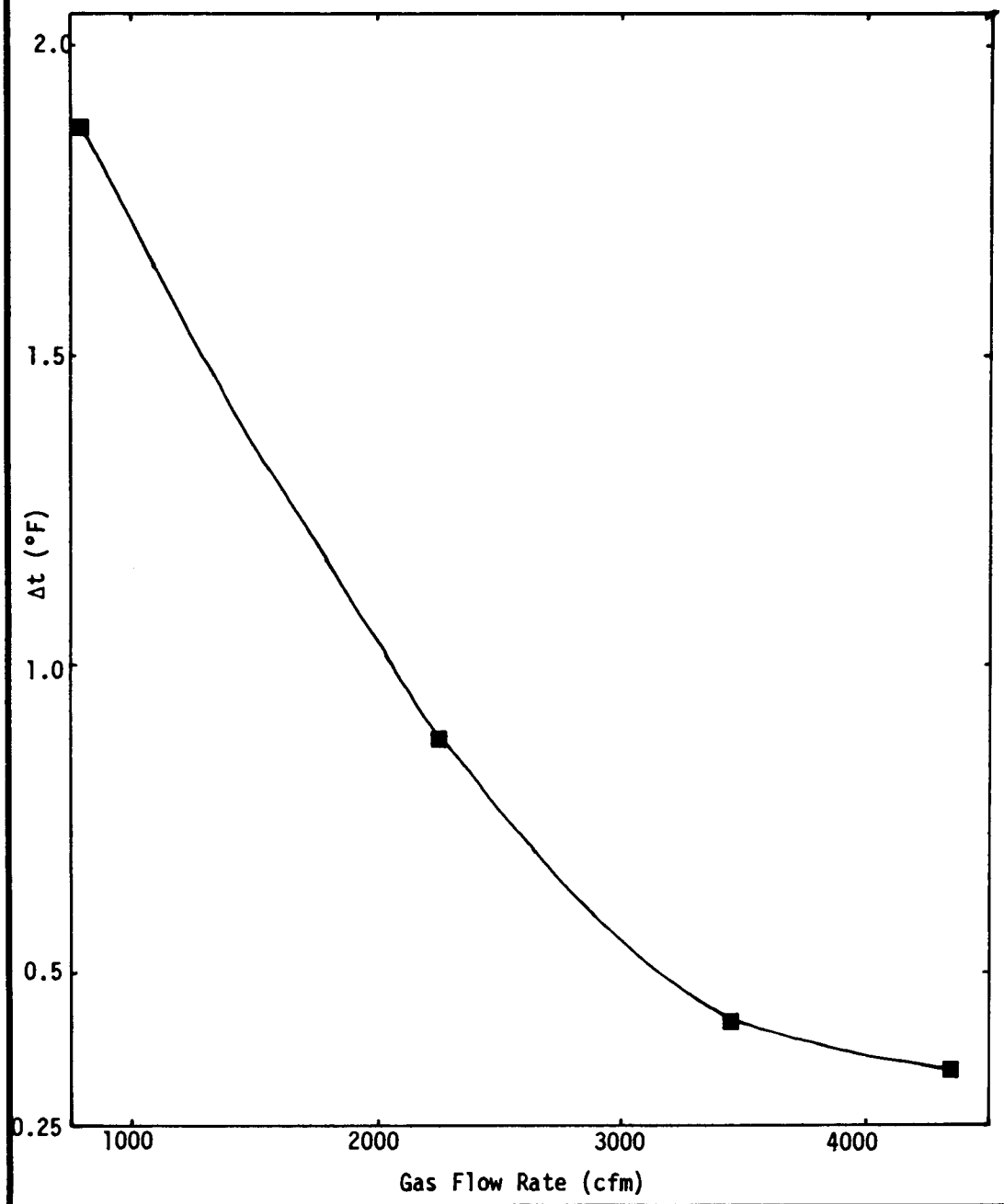
$$= 72.88°F$$

10.4 Heating Effect Due to Fan

The rise in temperature due to the heat input of the fan is plotted as a function of the volumetric gas flow rate and is shown in Fig. 16. All exit bulk air temperatures were corrected using this plot.

10.5 Data Tabulation

The data and results for the aspen and polyurethane pads are listed in Tables 3, 4, and 5.



MASSACHUSETTS INSTITUTE OF TECHNOLOGY
SCHOOL OF CHEMICAL ENGINEERING PRACTICE
AT
OAK RIDGE NATIONAL LABORATORY

TEMPERATURE EFFECT OF FAN VS AIR FLOW RATE

DATE	DRAWN BY	FILE NO.	FIG.
4-18-72	JPA	CEPS-X-148	16

Table 3. Adiabatic Isothermal Runs (Aspen Pad)

	Run 20	Run 21	Run 22	Run 23	Run 41
L'	127.46	127.46	127.46	127.46	127.46
G'	12.35	12.1	12.03	11.75	11.69
t_{in}	62.28	68.55	72.14	75.74	76.814
t_{out}	62.32	68.35	71.93	75.2	76.393
T_{wbin}	66.23	68.54	72.31	75.89	77.04
T_{in}	84.97	103.1	103.66	117.14	119.82
T_{out}*	68.69	77.55	80.11	85.67	86.45
H_{in}	0.00678	0.00703	0.00983	0.00978	0.01018
H_{out}	0.01051	0.0129	0.01525	0.01702	0.01785
$h_g a_H$	22.75	23.89	24.67	24.97	26.16
$k_g a_M$	91.54	96.82	98.23	99.63	102.87
c_H	0.244	0.245	0.246	0.246	0.246
T_{GF}	68.7	78.04	80.78	82.69	88.06
Le	1.02	1.01	1.02	1.02	1.03

*Bulk mean air outlet temperature (corrected for effect of fan heating, see Appendix 10.4).

Table 4. Adiabatic Cooling Runs (Aspen Pad)

	Run 29	Run 30	Run 31	Run 32	Run 33	Run 34	Run 35	Run 36	Run 37	Run 38	Run 39	Run 40
L'	54.63	91.05	127.46	163.88	127.46	127.46	127.46	127.46	127.46	127.46	127.46	127.46
G'	12.2	12.2	12.2	12.2	8.77	12.35	14.93	5.27	12.2	12.2	12.2	12.2
t_{in}	88.57	88.66	88.57	88.75	88.48	88.84	88.85	89.03	77.18	85.52	94.95	103.91
t_{out}[1]	67.05	70.28	72.89	75.14	75.85	74.24	73.17	79.63	71.01	74.05	77.29	79.81
T_{in}	83.67	83.75	83.68	83.46	81.83	82.99	83.89	83.61	84.48	84.81	85.28	85.69
T_{out}[2]	71.39	72.8	73.6	74.05	75.12	74.64	74.9	78.71	73.59	75.30	77.35	78.77
T_{wbin}	63.62	63.17	62.99	62.99	65.5	65.41	65.86	68.00	67.28	67.28	67.55	67.55
T_{wbout}	66.35	67.43	68.14	68.86	72.27	70.12	70.4	76.14	69.41	71.0	72.81	74.52
T_{GF}	76.2	76.6	76.7	76.7	77.1	77.2	77.8	77.9	77.0	78.3	79.9	81.2
T_{LF}	78.4	78.9	79.1	79.3	80.3	80.1	80.2	82.6	73.4	78.6	84.6	90.1
$h_g a_H$	28.39	25.46	24.35	23.98	17.47	24.28	28.58	9.44	25.54	25.06	24.85	26.28
$k_g a_M$	116.4	104.5	99.93	98.41	71.44	99.33	116.9	38.48	104.3	102.3	101.4	107.3
$h_L a_H$	25.67	28.11	29.68	29.97	26.14	30.22	35.44	22.91	28.89	30.04	32.11	34.12
K_g	20.01	21.13	21.77	21.82	17.82	21.57	25.28	12.86	22.98	21.91	21.03	20.56
NTU	2.57	1.63	1.2	0.93	0.98	1.19	1.39	0.71	1.27	1.21	1.16	1.13
$\frac{h_L a_H}{k_g a_M}$	0.22	0.27	0.3	0.3	0.37	0.3	0.3	0.6	0.28	0.29	0.32	0.32

All temperature measurements were taken to 3 decimal places but have been reported to 2 places only.

[1]Bulk mean water temperature.

[2]Bulk mean air outlet temperature (corrected for effect of fan heating).

Table 5. Adiabatic Isothermal Runs (Polyurethane Pad)

	Run 42	Run 43	Run 44	Run 45	Run 46	Run 48	Run 49	Run 50	Run 53	Run 54	Run 55	Run 47	Run 51	Run 52
L'	254.91	254.91	254.91	254.91	109.25	364.16	182.08	254.91	109.25	364.16	182.08	254.91	254.91	254.91
G'	10.82	10.197	10.09	10.68	11.56	11.40	11.19	12.41	11.60	10.5	11.6	10.87	8.16	5.54
t_{in}	69.45	80.22	84.89	73.58	65.33	64.79	63.89	65.51	65.6	66.2	65.84	65.69	65.51	66.31
t_{out}	69.39	79.99	84.84	73.34	65.26	64.90	63.82	65.44	65.6	66.2	65.84	65.61	65.44	66.24
T_{wbin}	69.33	80.09	84.48	73.32	65.29	64.85	63.86	65.49	65.58	66.2	65.84	73.26	73.77	74.02
T_{in}	87.11	121.32	123.44	94.94	83.42	84.19	84.67	86.49	87.04	83.86	83.79	84.19	86.68	86.61
T_{out}*	79.77	106.03	109.3	88.42	76.32	78.03	77.08	78.58	77.76	77.79	76.19	77.95	77.85	77.7
H_{in}	0.0113	0.0126	0.0167	0.0127	0.0092	0.0087	0.0079	0.0086	0.0088	0.0101	0.00975	0.0092	0.00855	0.00855
H_{out}	0.013	0.0163	0.02	0.0142	0.0108	0.0101	0.0096	0.01035	0.0109	0.0114	0.0115	0.0105	0.0106	0.0107
$h_g a_H$	16.94	14.00	13.53	11.31	16.81	12.83	14.86	17.2	18.39	12.98	18.76	13.09	13.06	9.39
$k_g a_M$	69.53	58.00	57.37	45.71	66.5	52.53	58.69	68.42	80.07	53.49	81.45	48.3	53.87	40.07
c_H	0.245	0.247	0.248	0.246	0.244	0.244	0.244	0.244	0.244	0.245	0.245	0.244	0.244	0.244
T_{GF}	76.22	96.53	93.78	82.4	72.43	72.88	72.22	73.86	73.66	73.41	72.74	73.26	73.77	74.02
Le	0.99	0.98	0.95	1.00	1.04	1.00	1.04	1.03	0.94	0.99	0.94	1.11	0.99	0.96

*Bulk mean air outlet temperature (corrected for effect of fan heating).

One adiabatic cooling run: Run 56 (see Sect. 10.6).

10.6 Comparison of the Two Pads

To compare the two pads, runs were performed at identical conditions. The data from the two runs were analyzed by the computer model (Sect. 10.1). The data and results are summarized in Table 6.

Table 6. Comparison of the Two Pads

Variable	Aspen Pad Run 34	Polyurethane Pad Run 56
L' (lb/min-ft^2)	127.46	254.92
G' (lb/min-ft^2)	12.35	12.35
T_{in} (°F)	82.99	83.24
T_{out} (°F)	74.64	77.24
T_{wbin} (°F)	65.41	65.81
T_{wbout} (°F)	70.12	68.36
t_{in} (°F)	88.84	88.52
t_{out} (°F)	74.24	80.17
$h_g a_H$ (Btu/min-ft^3)	24.28	24
$h_\ell a_H$ (Btu/min-ft^3)	30.22	22
$k_g a_M$ (lb/min-ft^3)	99.33	94
Length of pad (in.)	84	83
Width of pad (in.)	33	33
Thickness of pad (in.)	2	1
V (ft^3)	3.21	1.60

10.7 Simulation of Experimental Runs

The experimental transfer coefficient correlations [Eqs. (8), (9), and (12)] for the aspen pad were used in the design program LANSIM (Appendix 10.1) to simulate some of the experimental runs. The results of the

simulation are summarized in Table 7. These simulations are seen to predict the outlet air temperature within 0.4% and water temperature to within 0.6% of the experimentally measured values.

Table 7. Simulation of Experimental Runs

Run No.	Basis	Exit Water Temperature t_{out} Comptd	Exit Water Temperature t_{out} Expermtl	Percent Deviation of t_{out}	Exit Air Temperature T_{out} Comptd	Exit Air Temperature T_{out} Expermtl	Percent Deviation of T_{out}
34	standard	73.83	74.23	-0.54	74.4	74.64	-0.32
37	low t_{in}	71.20	71.01	+0.27	74.1	73.59	+0.69
40	high t_{in}	78.67	79.81	-1.43	78.4	78.77	-0.47
33	low G'	75.42	75.85	-0.57	75.0	75.12	-0.16
35	high G'	73.21	73.17	+0.05	74.4	74.90	-0.67
30	low L'	69.97	70.28	-0.44	72.9	72.80	+0.14
32	high L'	74.54	75.14	-0.80	74.0	74.05	-0.07

10.8 Parametric Study for Design Purposes

The simulation program (LANSIM) has been shown to predict the proper exit air and water temperature with the experimental correlations for the local transfer coefficients and inlet conditions for the aspen pad (see Table 7). A preliminary parametric study was performed to bracket actual operating conditions (5, 6) for a greenhouse in summer and winter. Table 8 summarizes the variation of parameters. The liquid rate was kept fixed near the flooding rate. Width and thickness were taken as standard for a pad (multiple thicknesses or widths could also be studied), pressure was taken as constant at atmospheric pressure. Variables in this study were the pad height (3 sizes), air flow rate (3 superficial velocities),and inlet conditions (3 values).

Table 8. Variation of Design Parameters for Aspen Pad

Constants: liquid flow rate = 3 gpm/ft pad width (8.5 gpm for a 34 in. pad width)
width of pad = 34 in.
thickness of pad = 2 in.
pressure = 745 mm Hg

Height (ft)	Air Flow Rate In (cfm/ft^2 pad area)	Inlet Conditions (°F) t_{in}	T_{in}	T_{wb}
4	100	70	50	50
6	150	70	60	60
8	200	110	91	79

The results of the study are shown in Table 9. The computer design is quite flexible and will simulate almost any desired operating condition without much modification. However, the winter conditions were not adequately characterized. The details of the program are given in App. 10.1.

Table 9. Parametric Study Results

(a) Summer Conditions: water temperature in = t_{in} = 110°F
air temperature in = T_{in} = 91°F
wet bulb temperature = T_{wb} = 79°F

Total Air Flow Rate (cfm/ft^2 pad area)	Height (ft)	t_{out} (°F)	T_{out} (°F)	Humidity Out (lb/lb)
100	4.0	97.3	90.3	0.0298
150	4.0	94.85	88.6	0.0281
200	4.0	93.12	87.5	0.0270
100	6.0	93.34	89.3	0.0288
150	6.0	90.72	87.6	0.0271
200	6.0	88.93	86.6	0.0260
100	8.0	90.37	88.5	0.0279
150	8.0	87.76	86.9	0.0263
200	8.0	86.08	85.9	0.0253

(b) Winter Conditions: All runs were made at the same variation of air flow rate and height of pad but inlet conditions of t_{in} = 70°F, T_{in} = 60 or 50°F, and T_{wb} = 60 or 50°F gave fog formation in the first box (1,1) of the grid.

10.9 Nomenclature

a_H	area of water interface per unit volume for heat transfer, ft^2/ft^3
a_M	area of water interface per unit volume for mass transfer, ft^2/ft^3
c_H	humid heat capacity of moist air, Btu/°F-lb dry air
G	total gas mass flow, lb/min
G'	superficial gas mass velocity, lb/min-ft^2
h	air enthalpy at temperature T, Btu/lb dry air
h_g	local gas film heat transfer coefficient, Btu/min-ft^2-°F
h_i	enthalpy of saturated air at interface conditions T_i, Btu/lb dry air
h_L	local liquid film heat transfer coefficient, Btu/min-ft^2-°F
H	absolute humidity of air, lb vapor/lb dry air
H_i	absolute humidity of air saturated at interface temperature T_i, lb vapor/lb dry air
H_{sat}	absolute humidity of saturated air, lb vapor/lb dry air
k_g	local gas film mass transfer coefficient, lb/min-ft^2
K_g	overall mass transfer coefficient, lb/min-ft^2
L	water flow rate, lb/min
L'	superficial water velocity, lb/min-ft^2
Le	Lewis number, $h_g a_H/(k_g a_M)c_H$
m	slope of enthalpy saturation line (at point of operation), Btu/lb-°F
NTU	overall number of transfer units, dimensionless
P	atmospheric pressure, mm Hg
t	water temperature, °F
T	dry bulb air temperature, °F
T_{GF}	gas film temperature, °F
T_i	interface temperature, °F

T_{LF} liquid film temperature °F

T_{out} outlet dry bulb air temperature, °F

T_{wb} wet bulb temperature of inlet air, °F

V pad volume, ft

10.10 Location of Original Data

The original data are located in ORNL Databook Nos. A5578-G, pp. 59-100, A-5511-G, pp. 79-100, A-5103-G, pp. 75-100, and A-5258-G, pp. 81-92, on file at the M.I.T. School of Chemical Engineering Practice, Bldg. 1000, ORNL.

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