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A SIMPLE MODEL FOR COOLING AND DEHUMIDIFYING COILS FOR USE IN CALCULATING ENERGY REQUIREMENTS FOR BUILDINGS

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INTRODUCTION

Energy standards for air-conditioned buildings are starting to be expressed in terms of overall annual energy budgets rather than in the traditional prescriptive form so that designers can have maximum scope for ingenuity in designing energy efficient buildings. The prerequisite for this type of standard is the existence of an accurate yet reasonably simple procedure for predicting the energy consumption of buildings. One of the essential parts of such an energy analysis program for air-conditioned buildings is a calculation procedure to model the performance of air cooling coils over a broad range of operating conditions.

There are three possible conditions for the coil: "all wet," "all dry" and "partially wet-partially dry." Thus it is necessary to be able to deal with each of these possible conditions and to be able to determine which conditions will obtain with any given set of air and water inlet conditions.

This paper presents a set of relatively simple algorithms that can be used to predict the performance of a multirow coil with either circular or continuous flat fins. The results of this analytical method were compared with experimental measurements carried out for 8-row and 4-row coils. A brief summary of this comparison is given in this paper; a detailed description of the test facility, test procedures, data reduction techniques and a full report of the test results are presented in Ref 1.

ANALYSIS

Multirow finned-tube heat exchangers are usually arranged so that the moist air stream flows over the tube banks in a cross-counter flow fashion. Detailed analysis of such arrangement can be found in Ref 1. However for the purpose of the present paper, the cross-counter flow arrangement of a multirow heat exchanger is approximated by a counter flow passage between moist air and chilled water separated by metallic surface, as shown in Fig. 1.

The first step in calculating the performance of such a coil is to determine whether the coil surface is "all wet," "partially wet," or "all dry." To do this it is convenient to assume that the entire air side surface is wet (the right part of the coil arrangement shown in Fig. 1). Based on this assumption the outlet air condition from the cooling coil, as well as the air side surface temperatures at inlet and at exit, are calculated. Then, a comparison between the air dew-point temperature and the surface temperatures is made to determine the coil surface condition as follows:

1. If the coil surface temperature at the air inlet section is lower than the dew-point temperature at inlet to the coil, then the cooling coil surface is "all wet" (the right part of the coil).
2. If the surface temperature at the air outlet section is higher than the dew-point temperature of air at inlet, then the cooling coil surface is "all dry," (the left part of the coil as shown in Fig. 1).

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3. If any of the conditions in 1 or 2 is not satisfied, then the cooling coil is "partially wet-partially dry."

The following is a further study of the three possible cases of the coil surface.

COMPLETELY DRY SURFACE COIL

The cooling coil is considered to be operating under "all dry" surface condition when the dew-point temperature of air at inlet to the coil $t_{d,1}$ is lower than the temperature of the coil surface at any point along the air passage.

Consider the left part of the coil shown in Fig. 1, i.e., the dry portion, then at steady state condition, the heat transfer rate Q, the entering air and water temperatures $t_{a,1}$, $t_{w,2}$ respectively are related to the leaving air and water temperatures $t_{a,2}$, $t_{w,3}$ respectively, by the following relations:

$$Q = \frac{1}{X}\,(t_{a,1} - t_{a,2}) \tag{1}$$

$$Q = \frac{1}{Y}\,(t_{w,2} - t_{w,3}) \tag{2}$$

and

$$Q = U\,A_o\,\Delta t_{\ell,m} \tag{3}$$

where

$$X = \frac{1}{m_a c_p} \tag{4}$$

$$Y = \frac{-1}{m_w c_w} \tag{5}$$

and

$$\Delta t_{\ell,m} = \frac{(t_{a,1} - t_{w,3}) - (t_{a,2} - t_{w,2})}{\ell n\,\dfrac{t_{a,1} - t_{w,3}}{t_{a,2} - t_{w,2}}} \tag{6}$$

As the air and water inlet conditions and the overall heat transfer coefficient U, are known, the only unknown in Eq 1 to 3 are $t_{a,2}$, $t_{w,3}$ and Q. These equations can be rearranged in a matrix form as follows:

$$\begin{bmatrix} 1 & 0 & X \\ 0 & 1 & Y \\ Z & 1 & 0 \end{bmatrix} \cdot \begin{bmatrix} t_{a,2} \\ t_{w,3} \\ Q \end{bmatrix} = \begin{bmatrix} t_{a,1} \\ t_{w,2} \\ t_{a,1} + Zt_{w2} \end{bmatrix} \tag{7}$$

Solution of Eq 7 gives:

$$\begin{bmatrix} t_{a,2} \\ t_{w,3} \\ Q \end{bmatrix} = \frac{1}{W} \begin{bmatrix} Y & -X & X \\ -YZ & XZ & Y \\ Z & 1 & -1 \end{bmatrix} \cdot \begin{bmatrix} t_{a,1} \\ t_{w,2} \\ t_{a,1} + Zt_{w,2} \end{bmatrix} \tag{8}$$

where

$$Z = \text{Exp}\ [UA_o\ (X + Y)] \tag{9}$$

and

$$W = XZ + Y \tag{10}$$

Thus the expressions for $t_{a,2}$ and $t_{w,3}$ can be written as

$$t_{a,2} = t_{a,1} - K_1\ (t_{a,1} - t_{w,2}) \tag{11}$$

$$t_{w,3} = t_{a,1} - K_2\ (t_{a,1} - t_{w,2}) \tag{12}$$

where

$$K_1 = \frac{X(Z-1)}{W} \tag{13}$$

$$K_2 = \frac{(X+Y)Z}{W} \tag{14}$$

The evaluation of K_1 and K_2 is based on several parameters, one of which is the overall thermal resistance $\frac{1}{UA_o}$, where A_o is the total external surface area. The overall thermal resistance is calculated by: [1]

$$\frac{1}{UA_o} = \frac{1}{f_o A_o \eta_o} + \frac{\delta_t}{k_t A_i} + \frac{1}{f_i A_i} + \frac{F}{A_i} \tag{15}$$

where

$$\eta_o = 1 - \frac{A_s}{A_o}(1 - \eta_f) \qquad \text{(See Ref 3)} \tag{16}$$

A_s = the secondary surface area, and

η_f = the dry surface fin efficiency as calculated by the method given in Ref 7.

Calculation of the film heat transfer coefficient on the air side f_o, is given in Appendix A, whereas that on the water side f_i, is calculated using the expression:

$$f_i = 1429\ [1 + 0.0146\ t_{w,m}]\ V_w^{0.8}\ D_i^{-0.2} \tag{17}$$

where

$t_{w,m}$ = mean water temperature, C

V_w = average water velocity, mps

D_i = inside tube diameter, m

Eq 17 is recommended by ARI [2] and is valid where the water flow Reynolds number is greater than 3100.

As indicated earlier, for a completely dry surface coil, the air dew-point temperature $t_{d,1}$ has to be lower than the average external surface temperature of the coil $t_{s,2}$ at the exit section from the coil. This temperature, $t_{s,2}$, is calculated using the condition that at the steady state, the ratio of the potentials to heat flow is equal to that of the corresponding thermal resistances. Thus at any section in the coil, the surface temperature can be written as:

$$t_{s,2} = t_{w,2} + \frac{\Sigma R - R_o}{\Sigma R} (t_{a,2} - t_{w,2}) \tag{18}$$

where

$$\Sigma R = R_o + R_m + R_i + R_f = \frac{1}{UA_o} \tag{19}$$

$$R_o = \frac{1}{f_o A_o \eta_o} \tag{20}$$

$$R_m = \frac{\delta_t}{k_t A_i} \tag{21}$$

$$R_i = \frac{1}{f_i A_i} \tag{22}$$

and

$$R_f = \frac{F}{A_i} \tag{23}$$

COMPLETELY WET SURFACE COIL

The cooling coil is considered to be operating under "all wet" surface conditions when the dew-point temperature of the moist air at inlet to the coil is higher than the external surface temperature of the coil at any point along the air stream passage. In this case, simultaneous heat and mass transfer occurs, and therefore the enthalpy difference is the appropriate potential to be used in the heat transfer calculations. [3] In addition, to simplify the calculations, it is assumed that the saturated air enthalpy h_w, at temperature t_w, can be approximated by:

$$h_w = a + b\, t_w \tag{24}$$

where a and b are constants.

Eq 24 can be used over a relatively small range of temperature, which is the case of water temperature variation in a cooling coil, [3] without incurring a significant error.

Consider the right part of the coil shown in Fig. 1, i.e., the "all wet" coil. At steady state condition, the inlet air enthalpy $h_{a,2}$ and that calculated at the inlet water temperature $h_{w,1}$ are related to the outlet air and water enthalpies ($h_{a,3}$ and $h_{w,2}$ respectively) and the heat transfer rate Q, as follows:

$$Q = \frac{1}{X'} (h_{a,2} - h_{a,3}) \tag{25}$$

$$Q = \frac{1}{Y'} (h_{w,1} - h_{w,2}) \tag{26}$$

and the heat transfer equation based on enthalpy potential can be written as [3]

$$Q = U_c A_o \Delta h_{\ell,m} \tag{27}$$

where

$$X' = \frac{1}{m_a} \tag{28}$$

$$Y' = \frac{-b}{m_w c_w} \tag{29}$$

U_c = the overall heat transfer coefficient based on enthalpy potential

$\Delta h_{\ell,m}$ = the log-mean enthalpy difference.

The latter is defined as [1]

$$\Delta h_{\ell,m} = \frac{(h_{a,2} - h_{w,2}) - (h_{a,3} - h_{w,1})}{\ell n \dfrac{h_{a,2} - h_{w,2}}{h_{a,3} - h_{w,1}}} \tag{30}$$

Eq 25, 26 and 27 can be rearranged in a matrix form as:

$$\begin{bmatrix} 1 & 0 & X' \\ 0 & 1 & Y' \\ Z' & 1 & 0 \end{bmatrix} \cdot \begin{bmatrix} h_{a,3} \\ h_{w,2} \\ Q \end{bmatrix} = \begin{bmatrix} h_{a,2} \\ h_{w,1} \\ h_{a,2} + Z'h_{w,1} \end{bmatrix} \tag{31}$$

where

$$Z' = \text{Exp}\,[U_c A_o (X' + Y')] \tag{32}$$

Solution of Eq 31 gives

$$\begin{bmatrix} h_{a,3} \\ h_{w,2} \\ Q \end{bmatrix} = \frac{1}{W'} \begin{bmatrix} Y' & -X' & X' \\ Y'Z' & X'Z' & Y' \\ Z' & 1 & -1 \end{bmatrix} \cdot \begin{bmatrix} h_{a,2} \\ h_{w,1} \\ h_{a,2} + Z'h_{w,1} \end{bmatrix} \tag{33}$$

where

$$W' = X' Z' + Y' \tag{34}$$

Eq 33 can be rearranged to give $h_{a,3}$ and $t_{w,2}$ as follows:

$$h_{a,3} = K_3 h_{a,2} + K_4 h_{w,1} \tag{35}$$

and

$$t_{w,2} = K_5 t_{w,1} + K_6 (h_{a,2} - a) \tag{36}$$

where

$$K_3 = \frac{X' + Y'}{W'} \tag{37}$$

$$K_4 = \frac{X'Z' - X'}{W'} \tag{38}$$

$$K_5 = \frac{Z' (Y' + X')}{W'} \tag{39}$$

and

$$K_6 = \frac{Y' (1-Z')}{bW'} \tag{40}$$

The overall thermal resistance based on enthalpy potential and the total external heat transfer area A_o can be expressed as a sum of the individual thermal resistances as follows:

$$\frac{1}{U_c A_o} = b[R_m + R_i + R_f + R_{e,w} + \frac{c_p}{b} R_{a,w}] \tag{41}$$

where

$$R_{e,w} = \frac{c_p (1 - \eta_{o,w})}{b\eta_{o,w}} R_{a,w} , \tag{42}$$

$$R_{a,w} = \frac{1}{A_o f_{o,w}} \tag{43}$$

and

$$\eta_{o,w} = 1 - \frac{A_s}{A_o} (1 - \eta_{f,w}) \tag{44}$$

$f_{o,w}$ and $\eta_{f,w}$ are the average film heat transfer coefficient of a wet surface and the wet surface fin efficiency respectively. These are calculated by the procedure outlined in Appendix A.

The coil surface temperature at the air inlet to the wet section $t_{s,2}$, and at the exit $t_{s,3}$, which are used to determine the coil operating condition, are calculated by equating the ratios of thermal resistances to the corresponding enthalpy potentials at each section. This results in the following expressions:

$$t_{s,2} = \frac{1}{R_x+1} \left[t_{w,2} + \frac{R_x}{b} (h_{a,2} - a)\right] \tag{45}$$

and

$$t_{s,3} = \frac{1}{R_x+1} \left[t_{w,1} + \frac{R_x}{b} (h_{a,3} - a)\right] \tag{46}$$

where

$$R_x = \frac{R_m + R_i + R_f}{\frac{c_p}{b} \quad R_{a,w}+R_{e,w}} \tag{47}$$

Calculation of the air dry-bulb temperature at exit from the wet surface coil, $t_{a,3}$, is based on the method described in the ARI standard [2], and derived by Goodman.[4] Therefore

$$t_{a,3} = t_{s,3} + (t_{a,2} - t_{s,2}) \operatorname{Exp}(-N) \tag{48}$$

where

$$N = \frac{A_o \, f_{o,w} \, \eta_{o,w}}{c_p \, m_a} \tag{49}$$

PARTIALLY DRY - PARTIALLY WET SURFACE COIL

If the external surface of the cooling coil is not completely dry or completely wet, then the remaining possibility is that part of the coil surface is dry and the rest is wet. This is the case shown in Fig. 1. The boundary between the dry and wet portions of the coil is determined by locating the point on the external surface of the coil where its temperature is equal to the air dew-point temperature.

For the dry part of the coil

$$h_{a,2} - h_{a,1} = c_p (t_{a,2} - t_{a,1}) \tag{50}$$

Then, the water temperature at the boundary between the dry and wet portions of the coil $t_{w,2}$ can be found by solving Eq 11, 36 and 50. This results in

$$t_{w,2} = \frac{1}{1-K_1K_6c_p} [K_5 \, t_{w,1} + K_6 \, (h_{a,1} - c_p t_{a,1} \, K_1 - a)] \tag{51}$$

The air dry-bulb temperature at the boundary $t_{a,2}$, can be calculated by substituting Eq 51 into Eq 11, and finally, the coil surface temperature at the boundary $t_{s,2}$ is calculated by substituting Eq 51 into Eq 18. A combined interpolation and trial-and-error method is used to find the area of the wet and dry regions, due to the fact that the parameters K_1 through K_6 are dependent on the position of the boundary. First an initial estimate of the wet area is made, then it is adjusted assuming a direct linear relation between the difference in area of two successive iterations and the difference in the corresponding difference in the coil surface temperature, as follows:

$$\frac{A_{w,i+1} - A_{w,i-1}}{A_{w,i-1} - A_{w,i}} = \frac{t_{d,1} - t_{s,2,i-1}}{t_{s,2,i-1} - t_{s,2,i}} \tag{52}$$

Once the boundary between the dry and wet portions of the coil has been determined, the sensible and latent heat extraction, Q_s and Q_ℓ respectively, are calculated as follow:

$$Q_s = m_a \, c_p \, (t_{a,1} - t_{a,2}) \tag{53}$$

$$Q = m_a \, (h_{a,1} - h_{a,3}) \tag{54}$$

and

$$Q_\ell = Q - Q_s \tag{55}$$

where

Q is the total cooling load.

COMPARISON WITH EXPERIMENTAL RESULTS

The model presented in this paper to simulate cooling and dehumidifying coils is executed using a computer program that has been prepared to perform these calculations. [5] The input data to the program are the physical dimensions of the coil and the air and chilled water inlet conditions. The output is the sensible and latent cooling loads, outlet air and chilled water conditions, coil surface condition, and the external wet surface area as a percentage of the total external area.

Experimental work was conducted using 4-row and 8-row coils for the purpose of checking the proposed model. The tested coils consist of circular tubes with flat plate fins arranged in a staggered form with a square face area of 0.372 m^2 (see Fig. 2.) The two coils were tested over a range of inlet air conditions in order to obtain experimental results of all dry, all wet and partially dry-partially wet coil surface condition. The air face velocity and the inlet air psychrometric conditions were varied over the practical range used in conjunction with the HVAC systems. Details about the testing facility, coils description and the experimental results are given in Ref 1.

For the purpose of this paper, a summary of the test results together with the corresponding output of the present simulation program is given in Table 1 for the 4-row coil, and in Table 2 for the 8-row coil.

DISCUSSION

Tables 1 and 2 show the comparison between the experimentally determined quantities and the corresponding analytically calculated values. Agreement between these two sets of quantities is considered to be acceptable for the application in the overall HVAC system simulation program. From Tables 1 and 2, the maximum difference between the experimentally determined and the analytically predicted heat extraction does not exceed 4% in the average, for the tested coils during the dry and wet surface tests. In case of wet surface coil, this difference is also almost the same for the sensible cooling load. Details about the uncertainty analysis are given in Ref 1.

The simulation program results were checked at different loading conditions. Air face velocity varied between 0.8 to about 4.5 mps, for the 4-row coil, whereas in case of the 8-row coil the air face velocity varied between 1.5 and 4.5 mps. The inlet air condition to the cooling coil was selected to cover the part load and full load conditions as in general practice. Both coils were tested over air dry-bulb temperature range from about 18 C to about 38 C. The water flow rate as well as the inlet chilled water temperature were not varied, because the resistance to heat flow is very small on the water side relative to that on the air side of the coil. The present simulation program is applied only in case of multirow tube banks, which consist of four or more rows of circular tubes with circular or flat plate fins. However, the program can easily be modified to be used for other types of coils providing that the dry and wet surface fin efficiencies and the variation of the air side film heat transfer coefficient with the air flow Reynolds number are given.

CONCLUSION

A simple model to simulate cooling and dehumidifying coil is presented in this paper for the purpose of using it in the general computer programs for analysis of energy requirements in buildings. This model is capable of predicting the outlet air and chilled water conditions as well as the sensible and latent heat exchange in a coil. The range of air-face velocities and dry-bulb temperatures were to cover the part and full load operating conditions. The accuracy of this model is adequate for the purpose of building energy calculations, i.e., the experimental check indicates that the upper limit of the error is about 4% on basis of the total heat exchange.

APPENDIX A

DRY AND WET SURFACE COEFFICIENTS

The average convective film heat transfer coefficient f_o on the air side of a dry coil surface is calculated from the following expression: [6]

$$J = \frac{f_o}{G\, c_p} \cdot Pr^{2/3} \tag{A-1}$$

where G is the maximum air mass flux, and Pr is the Prandtl number.

The overall heat transfer parameter J is determined using the coil physical dimensions and the air flow Reynolds number, Re, as follows: [1]

$$J = C_1 \cdot R_e^{C_2} \tag{A-2}$$

where C_1 and C_2 are constants for a particular coil. Details of the evaluation of C_1 and C_2 are given in Ref 1.

For a wet surface coil, the average film heat transfer coefficient $f_{o,w}$ is calculated using the expression:

$$f_{o,w} = f_o \cdot C_f \tag{A-3}$$

where C_f is a correction factor and is a function of air flow Reynolds number (see Ref 1.)

The evaluation of the dry surface fin efficiency η_f is based on the work by Gardner [7] using f_o as an average film coefficient.

The wet surface fin efficiency $\eta_{f,w}$, is calculated by the method described in Ref 1.

The quantities C_1 and C_2 for the given coil are 0.101 and -.369 respectively as taken from Ref 1.

NOMENCLATURE

All quantities used in this paper are in SI units. For more details, see Ref 8.

a	Parameter given in Eq 24
A_i	Internal surface area
A_o	External surface area
A_s	Secondary surface area
A_w	Wet surface area
b	Parameter given in Eq 24
c_p	Specific heat of air at constant pressure
c_w	Specific heat of water
C_1 and C_2	Coil parameters given in Eq A-2 (dimensionless)
$C_{[illegible]}$	Correction factor (dimensionless)
D_i	Inside tube diameter
f_i	Convection film heat transfer coefficient on the water side
[illegible], $f_{o,w}$	Convective film heat transfer coefficient on the air side, for a dry and wet surface respectively
[illegible]	Fouling factor
[illegible]	Air flow rate per unit area
[illegible]	Enthalpy of moist air:
	i = 1, entering the dry part
	i = 2, leaving the dry part and entering the wet part
	i = 3, leaving the wet part of the coil
[illegible], $I_{w,j}$	Saturated air enthalpy at temperature t_w and $t_{w,j}$ respectively:
	j = 1, entering the wet part
	j = 2, leaving the wet part, and entering the dry part
	j = 3, leaving the wet part of the coil
[illegible]	Overall heat transfer J-factor defined by Eq A-1, (dimensionless).
[illegible]	Tube material thermal conductivity
[illegible], K_2, K_3 [illegible], K_5 & K_6	Parameters defined by Eq 13, 14, 37, 38, 39 and 40 respectively.
[illegible], m_w	Mass rate of flow of air and water respectively.
[illegible]	Parameter defined by Eq 49
[illegible]	Prandtl number
[illegible], [illegible], Q_ℓ	Heat transfer rate, total, sensible and latent respectively.
[illegible]	Thermal resistance
[illegible], [illegible], R_i, [illegible], R_{ew}, $R_{a,w}$ [illegible] R_x	Different thermal resistances defined by Eq 20, 21, 22, 23, 41, 42 and 43 respectively
[illegible]	Air dry-bulb temperature:
	i = 1, entering the dry part
	i = 2, leaving the dry part and entering the wet part
	i = 3, leaving the wet part of the coil
[illegible]	Air dew-point temperature at inlet to the coil
[illegible], $t_{s,2}$	External surface temperature at exit from the dry part and wet part of the coil respectively.

$t_{w,1}$	Water temperature,
	i = 1 entering the wet part
	i = 2 leaving the wet part and entering the dry part
	i = 3 leaving the dry part of the coil respectively
U	Overall heat transfer coefficient (temperature basis)
U_c	Overall heat transfer coefficient (enthalpy basis) in $kg/m^2 \cdot s$
W, W', X, X', Y, Y', Z & Z'	Parameters defined by Eq 10, 34, 4, 28, 5, 29, 9 and 32 respectively
η_f	Dry surface fin efficiency (dimensionless)
η_o, $\eta_{o,w}$	Overall extended surface efficiency of dry and wet surface respectively
δ_t	Tube wall thickness

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Table 1

Comparison between experimental and analytical quantities for 4-row coil

Test No.	Air Face Velocity fpm	Inlet Conditions			Δdbt F		ΔWT F		Q_{total} $\frac{Btuh}{1000}$		Q_{sen} $\frac{Btuh}{1000}$		Surface Condition
		dbt F	wbt F	WT F	Expt.	Anal.	Expt.	Anal.	Expt.	Anal.	Expt.	Anal.	
1	312	96.1	66.9	47.1	40.6	39.8	2.8	2.8	52.9	51.8	52.9	51.8	Dry
2	177	81.2	60.9	47.2	30.4	29.6	2.2	2.2	23.2	22.6	23.2	22.6	Dry
3	482	77.6	60.2	47.2	22.1	21.7	2.4	2.4	46.0	45.1	46.0	45.1	Dry
4	605	71.5	57.6	47.2	16.3	15.9	2.2	2.2	43.0	42.1	43.0	42.1	Dry
5	367	95.9	77.9	47.3	31.0	30.5	7.7	7.8	78.7	79.0	48.1	46.8	Wet
6	544	85.9	73.6	47.2	20.4	20.0	7.4	7.5	77.1	77.3	47.5	46.2	Wet
7	667	76.6	69.0	47.2	13.1	13.6	6.6	6.5	68.1	67.3	38.0	39.2	Wet
8	818	88.2	75.0	47.2	18.3	17.8	9.0	9.1	93.6	94.1	64.3	61.7	Wet
9	857	96.9	79.7	47.2	23.2	21.0	10.9	11.1	114.3	115.3	84.2	75.1	Dry-Wet
10	736	74.0	67.5	47.2	11.0	11.6	6.4	6.3	62.9	62.4	35.1	36.9	Wet
11	603	86.8	74.4	47.3	19.8	19.2	7.9	8.0	82.4	82.8	51.3	49.2	Wet
12	584	75.4	68.9	47.2	13.1	13.4	6.3	6.3	63.0	63.1	33.3	33.9	Wet

$C = (F - 32)/1.8$, $\Delta C = \Delta F/1.8$, $W = 0.29307$ Btuh and mps $= 0.00508 \times$ fpm

dbt = dry-bulb temperature, wbt = wet-bulb temperature

WT = Water temperature and Q_{sen} = sensible cooling

Δdbt = air dry-bulb temperature drop

ΔWT = Water temperature rise

Table 2

Comparison between experimental and analytical quantities for 8-row coil

Test No.	Air Face Velocity fpm	Inlet Conditions			Δdbt F		ΔWT F		Q_{total} $\frac{Btuh}{1000}$		Q_{sen} $\frac{Btuh}{1000}$		Surface Condition
		dbt F	wbt F	WT F	Expt.	Anal.	Expt.	Anal.	Expt.	Anal.	Expt.	Anal.	
1	310	77.5	60.6	47.0	28.3	28.3	2.2	2.2	39.5	39.4	39.5	39.4	Dry
2	475	77.2	60.3	46.9	27.5	27.6	3.2	3.2	56.5	56.5	56.5	56.5	Dry
3	618	82.4	61.7	46.7	29.7	29.9	5.3	5.1	82.9	83.5	82.9	83.5	Dry
4	381	93.6	66.1	46.7	42.9	42.4	4.8	4.5	74.1	70.2	74.1	70.2	Dry
5	250	97.7	75.9	47.3	47.2	47.8	4.6	4.7	84.3	85.4	50.3	50.9	Wet
6	286	93.1	77.4	46.8	41.1	41.4	6.6	6.5	102.5	100.9	50.8	51.1	Wet
7	470	78.7	69.5	46.8	24.3	24.0	6.4	6.2	100.3	97.2	52.1	51.6	Wet
8	285	86.6	71.0	47.2	36.3	36.8	4.2	4.2	75.0	74.6	44.0	44.6	Wet
9	445	99.0	73.4	47.2	45.0	45.4	6.3	6.4	115.3	117.4	83.0	84.3	Dry/Wet
10	590	84.4	68.0	47.6	29.5	30.3	5.9	5.9	106.4	106.2	75.0	76.2	Dry/Wet
11	675	70.1	62.3	47.2	16.9	17.9	4.5	4.5	79.8	80.1	50.0	52.7	Dry/Wet
12	819	84.7	68.2	47.3	27.5	27.5	7.1	7.3	127.6	130.7	96.3	96.0	Dry/Wet

C = (F - 32)/1.8, ΔC = ΔF/1.8, W = 0.29307 Btuh and mps = 0.00508 fpm

dbt = dry-bulb temperature, wbt = wet-bulb temperature

WT = Water temperature and Q_{sen} = sensible cooling

Δdbt = air dry-bulb temperature drop

ΔWT = Water temperature rise

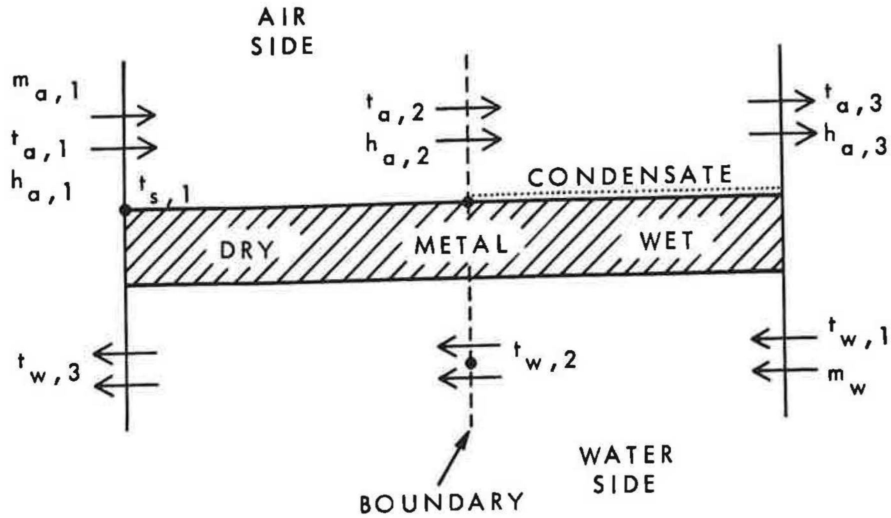

Fig. 1 Schematic diagram of the assumed cross-counter flow coil model

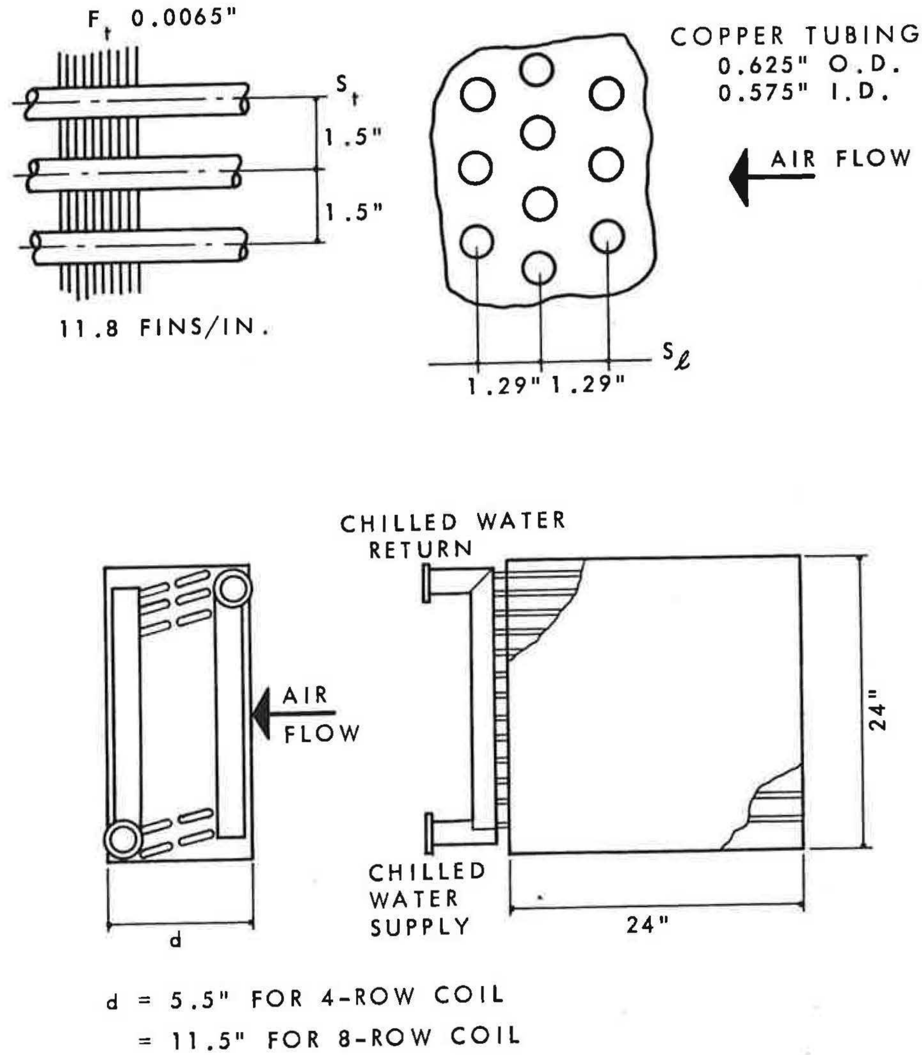

Fig. 2 The physical dimensions of the coil

DISCUSSION

CARL HILLER, Sandia Laboratories, Livermore, CA: Please comment on your methods of determining water and air side heat transfer coefficients, overall UA factor, and imput parameters required.

A.H. ELMAHDY: The average film heat transfer coefficient on the chilled water side (inside of tubes) is calculated using the turbulent flow relation described by McAdams (Heat Transmission, 1954):

$$Nu_i = 0.023\ Re_i^{0.8}\ Pr_i^{0.4}$$

where:

$$Nu_i = \frac{f_i d}{k}$$

This relation may underestimate f_i for the first row by about 7% since the flow is not fully developed. However, this does not result in a serious error since the water side resistance varied between 16 and 25% of the total thermal resistance.

Eq 15 in the paper gives the general expression to evaluate the overall U factor. In this equation, the average film heat transfer coefficient on the water side is calculated as described above, whereas that on the air side is evaluated using the method summarized in Appendix A. Details about this method are given in Ref 1 and will be presented as a technical paper at the next ASHRAE meeting.