Journal of Thermophysics and Heat Transfer
Vol. 20, No. 3, July–September 2006

Technical Notes

TECHNICAL NOTES are short manuscripts describing new developments or important results of a preliminary nature. These Notes should not exceed 2500 words (where a figure or table counts as 200 words). Following informal review by the Editors, they may be published within a few months of the date of receipt. Style requirements are the same as for regular contributions (see inside back cover).

Heat and Mass Transfer on Surfaces of Cooling Coils

M. S. Söylemez*
University of Gaziantep, 27310 Gaziantep, Turkey

Nomenclature

A = air side area of the cooling coil, m^2
A_d = air side area for dry surface of the cooling coil where condensation does not occur, m^2
A_i = refrigerant side heat transfer area of the cooling coil, m^2
C_{pa} = specific heat of air flowing over the cooling coil surface, $kJ/(kg_{da} \cdot K)$
C_{pm} = mean specific heat of air flowing over the cooling coil surface, $kJ/(kg_{da} \cdot K)$
C_{pv} = specific heat of water vapor, $kJ/(kg_w \cdot K)$
h_c = outside convective heat transfer coefficient of air over the wetted cooling coil surface, $W/(m^2 \cdot K)$
h_{fg} = latent heat of evapouration of water vapor, kJ/kg_w
h_r = inside film heat transfer coefficient between refrigerant and heat and mass transfer surface, $W/(m^2 \cdot K)$
i_a = average enthalpy of flowing air, kJ/kg_{da}
i_i = enthalpy of saturated air at wetted interface surface temperature, kJ/kg_{da}
i_v = enthalpy of water vapor, kJ/kg_v
i_1, i_2 = enthalpy of flowing air at inlet and at outlet sections, kJ/kg_{da}
m = mass flow rate of dry air, kg_{da}/s
P = ambient pressure, kPa
P_i = dew point pressure of saturated water vapor at mean interface temperature, kPa
q = heat transfer rate, W
R = defined parameter in Eq. (3), $kg_{da} \cdot K/kJ$
t_a = mean dry bulb temperature of flowing air, °C
t_c = dry bulb temperature of flowing air at the point where condensation starts, °C
t_d = dew point temperature of air at the inlet section, °C
t_i = mean interface temperature of the wetted cooling coil surface, °C
t_r = mean refrigerant temperature, °C
t_1, t_2 = dry bulb temperature of inlet air and of air at the exit, °C
W_a = humidity ratio of air at the inlet section, kg_w/kg_{da}
W_i = humidity ratio of saturated air at the mean interface temperature of the wetted surface, kg_w/kg_{da}
Δi_a = difference between enthalpy values of air at inlet and exit sections, kJ/kg_{da}

Subscripts

da = dry air
v = water vapor

I. Introduction

An interface surface occurs between air and water on the cooling coils of air conditioning units as a result of condensation of water vapor on these cooled surfaces. The interface temperatures of cooling coil surfaces are necessary to calculate the required heat and mass transfer area for designing the size of this cooling equipment.

Ratio of inside to outside heat transfer coefficients, ratio of outside to inside areas of heat transfer, and mean temperature of refrigerant are important parameters, in determining the mean thermohydraulic operating conditions of a cooling unit. Mean interface temperature t_i can be determined by a combined thermal and hydraulic analysis of cooling coil surfaces, where time-consuming iterative methods are needed. These iterative techniques can be replaced by some time-conscious calculation techniques for more efficient calculations. The numerical calculation technique presented in this Note, based on a short-cut method for computing the interface temperature, is put forth to replace the tedious and time-consuming iterations. The formula developed for calculating the interface temperature is original. Once the interface temperature is known, heat and mass transfer area on which the required cooling will occur can be calculated.

A thorough search of the current literature shows that there have been many studies[1–9] on the heat and mass transfer analyses on cooling coil surfaces. Except for the work of Stoecker,[10] there have been no previous detailed studies on the speedy estimation of the interface temperature for cooling coil surfaces that include the local elevation (ambient pressure) and mean operating temperature of cooling coils. This work presents an alternative practical numerical method developed for determining all of the temperatures of various regions on the cooling coil surface, as shown schematically in Fig. 1. Original interestingly accurate results are presented. Fixed and variable parameters used in formulating the mean interface temperature of the wetted surface and interface area of the cooling coil are the thermophysical properties of water vapor and air, the ambient pressure, the mean refrigerant temperature, the required exit enthalpy of the air or the heat transfer rate, the film heat transfer coefficients, and the ratio of outside fin area to inside area of heat transfer. The outputs can be calculated easily with the help of practical formulas instead of going through time-consuming iterative procedures.

The solution method offered by Stoecker[10] needs a multinodal numerical integration procedure that is based on approximations, and the calculation time is 10–20 times (depending on the number of nodes selected) greater than that required for the method

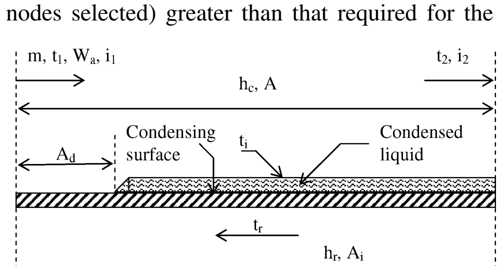

Fig. 1 Schematic of heat and mass transfer surface of typical cooling coil.

Received 8 February 2005; revision received 30 November 2005; accepted for publication 11 December 2005.

*Associate Professor, Department of Mechanical Engineering; sait@gantep.edu.tr.

presented here, which considerably improves the computational efficiency. Stoecker[10] claims that the enthalpy of air is a second-order polynomial function of the refrigerant temperature, where the coefficients are all temperature dependent. Because the coefficients are dependent on temperature, iteration is unavoidable. The formulation presented here appears to be easier when compared to the method presented by Stoecker.[10]

II. Mathematical Formulation

The heat transfer rate through a cooling coil can be estimated by the following equation[10]:

$$q = (h_c A / C_{pm})(i_a - i_i) = m \Delta i_a \tag{1}$$

Alternatively, heat transfer between the refrigerant and the inner cooling coil surface can be calculated as

$$q = h_r A_i (t_i - t_r) \tag{2}$$

Equating Eq. (1) to Eq. (2) yields

$$(t_i - t_r)/(i_a - i_i) = h_c (A/A_i)/C_{pm} h_r = R \tag{3}$$

Solving Eq. (3) for t_i yields

$$t_i = t_r + R[i_a - (C_{pa} t_i + W_i h_{fg})] \tag{4}$$

where the enthalpy of saturated air at the interface temperature is[11]

$$i_i = C_{pa} t_i + W_i h_{fg} \tag{5}$$

With the help of the following equations, W_i can be determined to simplify the calculation procedure[12]:

$$W_i = \frac{0.622 P_i}{P - P_i} = \frac{0.622 \exp\{18.6 - [5206.9/(273.15 + t_i)]\}}{P - \exp\{18.6 - [5206.9/(273.15 + t_i)]\}} \tag{6}$$

where P_i is[12]

$$P_i = \exp\{18.6 - [5206.9/(273.15 + t_i)]\} \tag{7}$$

Equation (4) can be expanded to the following form:

$$t_i = t_r + R\left[i_a - \left(C_{pa} t_i + \frac{0.622 h_{fg} \exp\{18.6 - [5206.9/(273.15 + t_i)]\}}{P - \exp\{18.6 - [5206.9/(273.15 + t_i)]\}}\right)\right] \tag{8}$$

Heat and mass transfer area can be extracted from Eq. (1) as

$$A = \frac{m \Delta i_a C_{pm}}{h_c (i_a - i_i)} \tag{9}$$

The following heat balance equation may be used for approximately obtaining the exit temperature of the air[10]:

$$m C_{pm} (t_1 - t_2) = A h_c \{[(t_1 + t_2)/2] - t_i\} \tag{10}$$

Rearranging to solve for t_2,

$$t_2 = \frac{[m C_{pm} - (h_c A/2)] t_1 + h_c A t_i}{[m C_{pm} + (h_c A/2)]} \tag{11}$$

The value of the mean specific heat of the wet air can be determined by means of the following equality[10]:

$$C_{pm} = C_{pa} + W_a i_v \tag{12}$$

On the other hand, the following equation can be used to determine dew point temperature of the inlet air[10]:

$$t_d = \frac{5206.9}{18.6 - \ln[W_a P/(0.622 + W_a)]} \tag{13}$$

In some cases, the dew point temperature of the air at the inlet may be lower than the temperature of the wetted interface surface, and so condensation may begin after a distance (that varies due to the inlet air and mean refrigerant temperatures) from the inlet section, as soon as the temperature drops to the dew point temperature of the inlet air. For this case, the temperature of the air at which condensation just begins to occur can be estimated by the following simple heat balance equation[10]:

$$t_c = t_d + [h_r (t_d - t_r)/h_c (A/A_i)] \tag{14}$$

The area swept by the air where condensation does not occur can be determined using the overall heat transfer procedure in the heat balance equation for the point at which condensation just starts as in the following equation[10]:

$$A_d = \frac{m C_{pm} (t_1 - t_c)}{\{[(t_1 + t_c)/2] - t_r\}} \left[\frac{1}{h_c} + \frac{(A/A_i)}{h_r}\right] \tag{15}$$

Equation (15) is given for the air side heat and mass transfer area, and the A/A_i term is used in this equation for the refrigerant side convective heat transfer resistance.

III. Results and Discussion

For a typical cooling coil problem,[10] it is assumed that $C_{pm} = 1.025$ kJ/(kg · K), $C_{pa} = 1.006$ kJ/(kg · K), $h_{fg} = 2500$ kJ/kg, $t_1 = 27°$C, $t_r = 10°$C, $P = 101.325$ kPa, $m = 2.78$ kg/s, $i_1 = 60.5$ kJ/kg, $i_2 = 40$ kJ/kg, $A/A_i = 16$, $h_c = 55$ W/(m^2 · K), and $h_r = 3000$ W/(m^2 · K). The value of t_i is calculated by using Eq. (8) as 13.55°C after two successive iterations. This value is calculated by Stoecker[10] as 13.43°C by the two nodal point numerical integration technique by using a second-order polynomial enthalpy function depending on temperature. The two results are in good agreement. The small amount of difference between the results is mainly a result of the selection of coefficients of the polynomial function at the mean refrigerant temperature instead of the actual interface temperature that was calculated finally at the end in the method of Stoecker.[10] However, if the coefficients were changed after each iteration, the number of iterations would be greater as compared to that for the present method, and the solution technique would still not be as simple as the present one. There is no need to use tables and correlating coefficients for each interface temperature in the presented technique as with the method of Stoecker.[10] The heat and mass transfer area and the exit air dry bulb temperature are calculated as 81 m^2 and 14.58°C by the present method, whereas they were found by Stoecker's work[10] as 87 m^2 and 15.1°C, respectively, by selecting two nodes for the numerical integration. The area results as 87.8 m^2 and the exit air dry bulb temperature as 14.5°C for a single node. The deviation between the results is not surprising because of the differences between the mean interface and refrigerant temperatures that were used in calculating the enthalpy of the air at the interface surface in the numerical integration technique of Stoecker.[10]

On the other hand, the accuracy of the numerical integration proposed by Stoecker[10] strongly depends on the number of nodes selected. As the number of nodes is increased, more sensitive and reliable results are achieved, but computation time increases exponentially. The values of interface temperature, exiting air-dry bulb temperature, and heat and mass transfer area are all are plotted in Figs. 2–5 for various R values and refrigerant temperatures. The values are presented numerically in Tables 1 and 2. It can be deduced

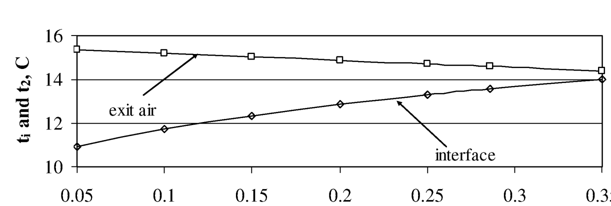

Fig. 2 Temperatures vs R values for $t_r = 10°$C.

Table 1 Variation of temperatures and area of heat and mass transfer vs R

	R, kg · K/kJ						
Variables	0.05	0.10	0.15	0.2	0.25	0.286	0.35
t_i	10.94	11.71	12.31	12.84	13.3	13.55	13.98
t_2	15.36	15.19	15.03	14.86	14.69	14.58	14.38
A	56.32	62.36	68.19	74.47	81	85.2	93.53

Table 2 Variation of temperatures and area of heat and mass transfer vs refrigerant temperature

	t_r, °C						
Variables	−5	−2.5	0	2.5	5	7.5	10
t_i	4.35	5.96	7.42	9.09	10.61	12.15	13.55
t_2	15.82	15.80	15.75	15.62	15.42	15.08	14.58
A	32.42	35.86	39.87	46.06	54.11	66.52	85.2

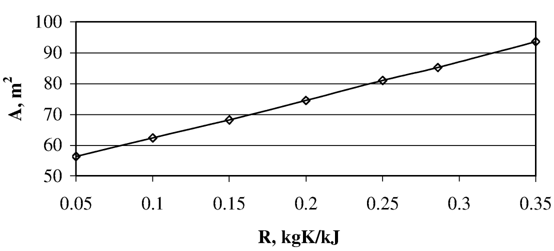

Fig. 3 Area requirement for $t_r = 10°$C.

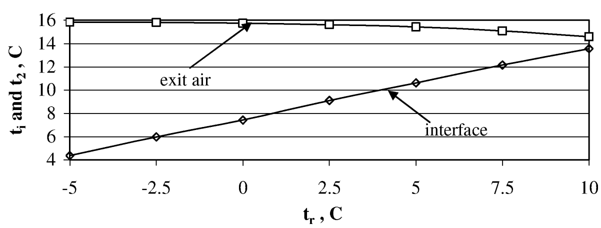

Fig. 4 Temperatures vs t_r values for $R = 0.286$ kg · K/kJ.

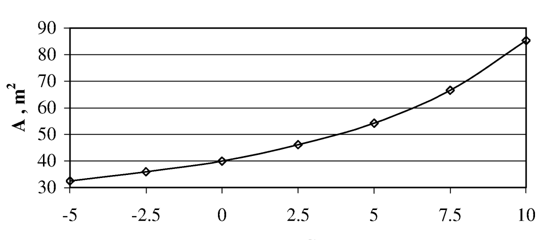

Fig. 5 Area requirement for $R = 0.286$ kg · K/kJ.

that, as the parameter R or the refrigerant temperature increases, the mean interface temperature and the required heat transfer area increases, but the exit air temperature decreases. The maximum deviation between the most sensitive result of Stoecker[10] in which iteration is not performed and the mean interface temperature is calculated instead of the mean refrigerant temperature and the technique presented in this Note is almost 6.5% in the calculated results of the area of heat and mass transfer.

Furthermore, to check the validity of the present work, mean interface and mean dry bulb temperatures are used to calculate the mean condensation heat transfer coefficient with the exact (analytical) film condensation heat transfer coefficient for the plate available by Holman.[13] The exact solution of Holman[13] can be used to check the validity of the latent heat transfer part of the sample problem during condensation, as presented at the beginning of this section. The value of latent heat transfer is calculated by multiplying the latent enthalpy change that can be read from a psychrometric chart by the mass rate of flow of air. Then this value is divided into the calculated heat and mass transfer area to evaluate the rate of latent heat flux. The latent heat transfer that occurs during condensation of water vapor on the cooling coil per unit heat and mass transfer area (latent heat flux) is divided into the mean temperature difference between the mean dry bulb temperature of the flowing air and the mean interface temperature to calculate the mean condensation heat transfer coefficient. This value is calculated for the sample problem and compared with the analytical result of Holman.[13] The deviation between the two mean condensation heat transfer coefficients is within 6.6%. The same calculation procedure was performed by using the result of Stoecker[10] to compare his results with the exact solution of Holman[13] and a deviation of 14.2% was found. This indicates that the result of present study is closer to the analytical solutions of Holman[13] in comparison with the result of Stoecker.[10] This clearly validates the present formulation.

Additionally, the results of the present formulation, that is, Eqs. (13) and (14), are compared with the results of a typical partially wetted cooling coil surface problem available by Stoecker.[10] Available data for the sample problem are $C_{pm} = 1.025$ kJ/(kg · K), $C_{pa} = 1.006$ kJ/(kg · K), $h_{fg} = 2500$ kJ/kg, $t_1 = 30°$C, $t_r = 10°$C, $P = 101.325$ kPa, $m = 0.278$ kg/s, $i_1 = 56.6$ kJ/kg, $A/A_i = 18$, $h_c = 60$ W/(m^2 · K), and $h_r = 2250$ W/(m^2 · K). Dew point temperature and the temperature at the beginning of condensation are calculated, using Eqs. (13) and (14), as 14.7 and 24.5°C, respectively. These are calculated as 14.5 and 23.5°C, respectively, by Stoecker.[10] On the other hand, the dry portion heat and mass transfer area is calculated by Eq. (15) as 2.24 m^2, and it was given by Stoecker[10] as 2.53 m^2. Results generally agree with each other. The deviation between the results is possibly a result of numerical approximation of the present formulation technique and the error in reading the data from a psychrometric chart in the solution offered by Stoecker.[10]

IV. Conclusions

The solution method presented here is not exact. However, it is shown to be reliable and accurate enough because the mean condensation heat transfer coefficient obtained from this technique by using the mean interface temperature and the mean dry bulb temperature of the flowing air was in good fit with the exact solution of Holman[13] for a specific example problem. It is clear that the mean interface temperature can be determined easily by the present calculation method numerically for each local elevation, that is, for given barometric pressure and inlet air conditions, in a fast manner. The validity of the optimization formulation was checked through numerical examples. The required heat and mass transfer surface area for the cooling coils can also be determined with an accuracy level of about 6%. The present formulation may be helpful for cooling unit designers and manufacturers.

A fast method for calculating the interface temperature of cooling coil surfaces is presented yielding simple algebraic formulas. The present method can be used for calculating heat and mass transfer area for interface surfaces of cooling coils, that is, for evaporators and fan coil units. A simplified method was used in the present study for deriving psychrometric properties of moist air based on a numerical approximation method, and it was used in the rapid thermal analysis of wetted cooling coil surfaces.

References

[1]Chen, Y. M., and Sun, C. Y., "Experimental Study on the Heat and Mass Transfer of a Combined Absorber–Evapourator Exchanger," *International Journal of Heat and Mass Transfer*, Vol. 40, No. 4, 1997, pp. 961–971.

[2]Bourouni, K., Martin, R., Tadrist, L., and Tadrist, L., "Modeling of Heat and Mass Transfer in a Horizontal-Tube Falling-Film Evapourator for Water Desalination," *Desalination*, Vol. 116, No. 2–3, 1998, pp. 165–183.

[3]Lutiğan, J., Cvengro, J., and Micov, M., "Heat and Mass Transfer in the Evapourating Film of a Molecular Evapourator," *Chemical Engineering Journal*, Vol. 85, No. 2–3, 2002, pp. 225–234.

[4]Conlisk, A. T., "Analytical Solutions for the Heat and Mass Transfer in a Falling Film Absorber," *Chemical Engineering Science*, Vol. 50, No. 4, 1995, pp. 651–660.

[5]Boulama, K., Galanis, N., and Orfi, J., "Heat and Mass Transfer between Gas and Liquid Streams in Direct Contact," *International Journal of Heat and Mass Transfer*, Vol. 47, No. 17–18, 2004, pp. 3669–3681.

[6]Seker, D., Karatas, H., and Egrican, N., "Frost Formation on Fin-and-Tube Heat Exchangers," *International Journal of Refrigeration*, Vol. 27, No. 4, 2004, pp. 367–374.

[7]Selim, A. M., and Elsayed, M. M., "Interfacial Mass Transfer and Mass Transfer Coefficient in Aqua Ammonia Packed Bed Absorber," *International Journal of Refrigeration*, Vol. 22, No. 4, 1999, pp. 263–274.

[8]Ganzevles, F. L. A., and van der Geld, C. W. M., "Heat and Mass Transfer from Internal Flows to Hemispheres and Flat Parts in Between," *International Journal of Heat and Mass Transfer*, Vol. 41, No. 2–3, 1998, pp. 3705–3718.

[9]Dai, Y. J., and Sumathy, K., "Theoretical Study on a Cross-Flow Direct Evapourative Cooling Coil Using Honeycomb Paper as Packing Material," *Applied Thermal Engineering*, Vol. 22, No. 13, 2002, pp. 1417–1430.

[10]Stoecker, W. F., *Air Conditioning, Heating and Ventilating*, Illinois Univ. Press, IL, Chicago, 1967, pp. 239–256.

[11]McQuiston, F. C., and Parker, J. D., *Heating Ventilating and Air Conditioning*, Wiley, Singapore, 1994, pp. 55–58.

[12]Stoecker, W. F., *Design of Thermal Systems*, McGraw–Hill, Singapore, 1989, pp. 75–78.

[13]Holman, J. P., *Heat Transfer*, McGraw–Hill, Tokyo, 1983, pp. 409–412.