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A HEAT AND MASS TRANSFER MODEL FOR THERMAL AND HYDRAULIC CALCULATIONS OF INDIRECT EVAPORATIVE COOLER PERFORMANCE

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ABSTRACT

A new heat and mass transfer model based on basic principles has been developed for thermal and hydraulic calculations of indirect evaporative cooler performance. The features of this model are: 1) some simplifications have been incorporated to make it more user-friendly, and 2) it is universal and can be used to analyze different indirect evaporative cooler designs and conditions, such as tube- or plate-types, those that use outdoor or room air as the secondary air, and either sensible cooling or cooling and dehumidification of the primary air. Variations of the basic calculation procedure for these differing conditions are provided. The paper also compares results from sample calculations for several designs to data from other sources for validation. This model can be used for energy analyses as well as for system or product optimization.

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INTRODUCTION

The use of Indirect Evaporative Cooling (IEC) has a high potential of meeting air conditioning needs at low energy costs. An IEC process is characterized by having two distinct air passages, one termed the primary and the other the secondary air passage. The primary air is usually outdoor air that is supplied to the room after it has been cooled by air in the secondary air passages through heat transfer. The secondary air can be either outdoor air or room exhaust air. The surface of the secondary air passages is wetted by circulating water, so that heat and mass transfer takes place between the wet surface and the secondary air. As a result, the temperatures of both are decreased.

When outdoor air is used as the secondary air the average wet surface temperature, or effective surface temperature, must be higher than the wet bulb temperature of entering secondary air but lower than dry bulb temperatures of the primary and secondary air. The enthalpy of the saturated air at the effective surface temperature should be greater than that of both the primary air and secondary air leaving the evaporative cooler, and the humidity ratio of the air at the effective surface must be greater than that of the secondary air. When room exhaust air is used as the secondary air and the primary air is both cooled and dehumidified, the situation will be different. In this case, the effective surface air enthalpy will be greater than that of secondary air and less than the enthalpy of primary air.

Indirect evaporative coolers can be either tube-type (Figure 1a) or plate-type (Figure 1b). For the tube-type, the primary air flows through the tubes while the secondary air passes around the tubes in a cross flow. Water dripping onto the outside surface of the tubes keeps them wet and cools the secondary air by direct evaporation. For the plate-type, a number of plates form flat passages of primary and secondary air one after another. The primary air flows through the dry passages horizontally, while the secondary air flows upwards through the wet passages as water droplets flow downwards.

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NOMENCLATURE

A = area, m^2
B = barometric pressure, mbar
C_o = local loss coefficient, dimensionless
COP = coefficient of performance
c_p = specific heat of air, kJ/kg°C
D = mass transfer coefficient, kg/m^2s
d = diameter or hydraulic diameter, m
E = effectiveness, dimensionless
f = friction loss factor, dimensionless
g = mass flowrate, kg/s
h = enthalpy, kJ/kg
L = length, m, or volumetric flowrate, m^3/s
Q = cooling capacity, kW
RH = relative humidity, dimensionless
s = tube distance, m
t = temperature, °C
v = velocity, m/s
w = humidity ratio, kg/kg
W = required fan power, W
α = convective heat transfer coefficient, $W/m^{2o}C$
ΔP = total pressure drop, Pa
ΔP_f = friction loss, Pa
ΔP_l = local loss, Pa
ρ = density, kg/m^3

Subscripts

db =	dry bulb	p =	primary	1 =	entering
wb =	wet bulb	s =	secondary	2 =	leaving
i =	inside	r =	room	w =	of effective surface
o =	outside				

THERMAL CALCULATIONS WHEN OUTDOOR AIR IS USED AS THE SECONDARY AIR

The heat and mass transfer processes inside an indirect evaporative air cooler are complicated. For practical purposes, the following simplifying assumptions have been made:

1) The water film temperature over all tubes and plates is assumed to be uniform and called the effective surface temperature.
2) The thermal resistance of the tube wall (for the tube-type) or of the plates (for the plate-type) are assumed to be negligible compared with the resistances of the air films, and is ignored in the model. This means that the temperatures of the tubes or plates is assumed to be uniform in thickness.
3) The heat and mass transfer effects between the water droplets and the air in the secondary air side are assumed to be negligible and ignored in the calculations.

Heat and mass transfer between the secondary air and the wet surface

For a differential heat transfer area dA_o (Figure 2), the following equation can be written:

$$g_s dh_s = D_2 (h_w - h_s) dA_o \qquad (1)$$

where g_s = secondary air flow rate, kg/s
D_2 = mass transfer coefficient, kg/m^{2}s
h_w = enthalpy of saturated air with effective surface temperature t_w, kJ/kg
h_s = enthalpy of secondary air, kJ/kg

Substituting the Lewis relationship $D_2 = \alpha_o/1000c_p$ into Equation 1 gives:

$$g_s dh_s = \frac{\alpha_o}{1000c_p}(h_w - h_s)\, dA_o \tag{2}$$

where c_p = specific heat of air, kJ/kg°C
α_o = convective heat transfer coefficient, W/m$^2\cdot$°C

Integrating Equation 2 results in:

$$\frac{h_w - h_{s1}}{h_w - h_{s2}} = \exp\left(\frac{\alpha_o A_o}{1000 g_s c_p}\right) \tag{3}$$

where h_{s1} = enthalpy of entering secondary air, kJ/kg
h_{s2} = enthalpy of leaving secondary air, kJ/kg
A_o = heat transfer area at secondary air side, m^2

From Equation 3 we can obtain the equation for calculating h_{s2} :

$$h_{s2} = h_w - \frac{h_w - h_{s1}}{\exp\left(\frac{\alpha_o A_o}{1000 g_s c_p}\right)} \tag{4}$$

Heat balance of primary air

When the cooling process of the primary air is sensible its heat balance equation is (Figure 3):

$$1000 g_p c_p (t_{dbp1} - t_{dbp2}) = \alpha_i \frac{t_{dbp1} - t_{dbp2}}{\ln\left(\frac{t_{dbp1} - t_w}{t_{dbp2} - t_w}\right)} A_i \tag{5}$$

After rearranging Equation 5 the following form can be obtained:

$$t_{dbp2} = t_w + \frac{t_{dbp1} - t_w}{\exp\left(\frac{\alpha_i A_i}{1000 g_p c_p}\right)} \tag{6}$$

where t_{dbp1} = dry bulb temperature of entering primary air, °C
t_{dbp2} = dry bulb temperature of leaving primary air, °C
t_w = effective surface temperature, °C
α_i = convective heat transfer coefficient, W/m^{2}°C
g_p = primary air flow rate, kg/s
A_i = heat transfer area at the primary air side, m^2

The heat transfer effectiveness, E, can then be calculated as follows:

$$E = \frac{t_{dbp1} - t_{dbp2}}{t_{dbp1} - t_{wbs1}} = \frac{t_{dbp1} - t_{dbp2}}{t_{dbp1} - t_{wbp1}} \tag{7}$$

where t_{wbp1}, t_{wbs1} = wet bulb temperatures of entering primary and secondary air, °C

Heat balance between primary and secondary air

The heat loss of the primary air must equal the heat gain of the secondary air. Therefore:

$$g_s\,(h_{s2} - h_{s1}) = g_p c_p\,(t_{dbp1} - t_{dbp2}) \tag{8}$$

From here,

$$h_{s2} = h_{s1} + \frac{g_p}{g_s}\,c_p\,(t_{dbp1} - t_{dbp2}) \tag{9}$$

Substituting Equation 5 into Equation 9 gives:

$$h_{s2} = h_{s1} + \frac{g_1}{g_2} c_p (t_{dbp1} - t_w)\left[1 - \frac{1}{\exp(\frac{\alpha_i A_i}{1000 g_1 c_p})}\right] \tag{10}$$

Although we can theoretically use Equations 4 and 10 to solve for t_w, this is mathematically difficult. A better method is to solve for it through trial and error starting with an initial guess for t_w. Two values for h_{s2} can then be calculated using Equations 4 and 9, and compared to each other. If they are not equal, another value of t_w is guessed until the calculated h_{s2}s equal each other. This is easily done on a computer. Once t_w is obtained, t_{dbp2} can be calculated using Equation 6.

The equation to calculate the humidity ratio of the leaving secondary air has the same form as Equation 4, i.e.,

$$w_{s2} = w_w - \frac{w_w - w_{s1}}{\exp\,(\frac{\alpha_o A_o}{1000 g_s c_p})} \tag{11}$$

where w_{s1} = humidity ratio of entering secondary air, kg/kg
w_{s2} = humidity ratio of leaving secondary air, kg/kg
w_w = humidity ratio of saturated air with temperature of t_w, kg/kg

If only sensible cooling takes place, the humidity ratio of the primary air will not be changed, i.e., $w_{p2} = w_{p1}$, where w_{p1} and w_{p2} are the humidity ratios of the entering and leaving primary air. The enthalpy of the leaving primary air is therefore:

$$h_{p2} = 1.01 t_{dbp2} + w_{p2} (2500 + 1.84 t_{dbp2}) \quad (12)$$

The cooling capacity of the indirect evaporative cooler is

$$Q = g_p (h_{p1} - h_{p2}) \quad (13)$$

and the evaporation rate of the wet surface is

$$\Delta W = g_s (w_{s2} - w_{s1}) \quad (14)$$

The formulas for calculating α_i and α_o are given in Appendix 1.

CALCULATION OF AIR PRESSURE DROPS AND COPs

There are two kinds of air pressure losses, friction loss and local loss. The friction loss can be calculated using Equation 15:

$$\Delta P_f = \frac{f}{d} l \frac{v^2 \rho}{2} \quad (15)$$

where ΔP_f = friction loss, Pa
f = friction loss factor, dimensionless
l = length of passages, m
d = diameter or hydraulic diameter, m
v = air velocity, m/s
ρ = air density, kg/m^3

Colebrook's formula can be used to calculate the friction loss factor f (see Appendix 2).

The local loss can be calculated using Equation 16

$$\Delta P_l = \Sigma C_o \frac{v^2 \rho}{2} \quad (16)$$

where ΔP_l = local loss, Pa
ΣC_o = sum of local loss coefficients, dimensionless

The total pressure drop is then

$$\Delta P = \Delta P_f + \Delta P_l \tag{17}$$

and the required fan power is

$$W = \frac{L_p \Delta P_p}{\eta_p} + \frac{L_s \Delta P_s}{\eta_s} \tag{18}$$

where W = required fan power, W
L_p, L_s = primary and secondary air flow rates, m^3/s
$\Delta P_p, \Delta P_s$ = primary and secondary air pressure drops, Pa
η_p, η_s = efficiencies of primary and secondary air fans, dimensionless

Because of the very low exhaust velocity of the secondary air, its dynamic losses are negligible.

When the cooling capacity Q[kW] and required fan power W are obtained, the Coefficient of Performance (COP) of the IEC can be calculated as follows:

$$COP = \frac{Q}{1000W} \tag{19}$$

To facilitate computation, a number of computer programs and subroutines have been written based on the above methodology. To assess the seasonal performance of IECs, these subroutines have also been incorporated into the Residential Systems (RESYS) portion of the DOE-2.1D program. This effort is described in a companion technical paper evaluating the performance of several IEC designs in typical residential buildings in California climates (Wu et al. 1989).

VALIDATION

As a limited validation of the model, the performance of several typical IEC designs were calculated using the described methodology and compared to manufacturers' data, field measurements and data from other sources.

Tube-type IEC

The geometrical parameters used for a typical tube-type indirect evaporative cooler are given in the following table:

Tube inside diameter (d_i)	25.4 mm
Tube outside diameter (d_o)	27.4 mm
Tube length (L)	1.365m
Number of tubes	160
Number of tubes in one row	6
Number of rows	15
Form of tube banks	staggered
Tube distance in a row (s_1)	0.0527m (center to center)
Row distance (s_2)	0.04564m (center to center)
Secondary air inlet height	0.316m

Outdoor air is used as secondary air, with dry and wet bulb temperatures (t_{dbo} and t_{wbo}) ranging from 30 to 42 °C and 17 to 35 °C, respectively. The primary air flow rate is varied from 0.236 to 0.944 m^3/s (500 to 2000 cfm), while the secondary air flow rate is fixed at 0.378 m^3/s (800 cfm). A barometrical pressure of 101325 Pa (1atm) is used.

For the hydraulic calculation, a plastic tube roughness of 0.025 mm is used, and local loss coefficients for the primary air of 0.5 at the entrances and 0.46 at the exits of the tubes. The local loss coefficient for secondary air through the staggered tube bank is calculated as follows :

$$C_o = 0.25 + \frac{0.1175}{(\frac{s_1}{d_o} - 1)^{1.08} Re^{0.16}} \tag{20}$$

where Re is the Reynolds Number. The fan efficiency is assumed to be 0.8.

The results from the calculations are listed in Table 1. The following observations can be made:

1) The primary air flow rate, L_p, and outdoor air wet bulb temperature, t_{wbo}, have a great effect on the effectiveness, E, of the cooler .

2) The outside dry bulb temperature, t_{dbo}, does not greatly affect cooler effectiveness, E, in the range covered by the calculations. Under the same L_p and t_{wbo}, different t_{dbo}s give nearly the same E. For example, for t_{wbo}=26°C and L_p=0.944m^3/s when t_{dbo} are 42,38,34, or 26°C, the effectiveness E will be 0.46,0.45,0.45, or 0.45 respectively. This means that in this range of dry bulb temperatures we have nearly the same effectiveness values.

Figure 4 shows a set of curves obtained by plotting average effectiveness, E, against the average mass flow rates of the primary air, g_p, for various outside wet bulb temperatures as indicated in Table 1. The linear line, A, on the figure is taken from the manufacturer's data, and seems to be an approximate value representing an average effectiveness, E, for different wet bulb temperatures and primary air flow rates.

Figure 5 shows the relation between the primary air pressure drop and its flow rates. The lower line is obtained using the model, while the top line is taken from manufacturer's data. The two lines are nearly coincident.

The above comparisons indicate that the model described conforms to manufacturer's data for tube-type IECs and seems to be dependable.

To understand the sensitivity of barometric pressure B on cooler effectiveness, E, calculations have been made at 3 different pressures (100000, 90000 and 80000 Pa) with an outdoor air dry bulb temperature of 42 °C and wet bulb temperatures from 26 to 35 °C. The results are plotted in Figure 6 where the abscissa is the primary air mass flow rate, (kg/s), and the ordinate is the IEC effectiveness, E. It can be seen that all points with the same wet bulb temperature but different B lie on smooth curves. That indicates that barometric pressure has no effect on effectiveness provided that it is expressed as a function of the mass, and not volume, flow rate of primary air.

Plate-type

The geometrical parameters for a typical plate-type indirect evaporative cooler are shown in the following table :

Width of primary air passages	0.0048m
Width of secondary air passages	0.004m
Height of primary air passages	0.48m (fin thickness deducted)
Length of primary air passages	0.267m
Cross section of secondary air passages	0.004m by 0.267m
Length of secondary air passages	0.535m
Number of passages for primary air	101
Number of passages for secondary air	100

Calculations were done for dry and wet bulb temperature ranges of primary air from 34 to 42 °C and from 20 to 35 °C, respectively. The primary air flow rate is varied from 0.2 to 2.3 m^3/s, while the secondary air flow rate is fixed at 0.38 m^3/s. The barometric pressure is likewise fixed at 101325 Pa.

For the hydraulic calculations, a wall surface roughness of 0.03 mm for primary air passages is used. For secondary air passages, the wall surface roughness is assumed to be 0.9 mm because of the velvety surface. The local loss coefficient at the entrances and exits of both the primary and secondary air passages are assumed to be 8.0 and 1.03, respectively. For the secondary air flow, a local loss coefficient of 5.5 is used to account for the 90° change in the air stream direction.

The results of the calculations are shown in Table 2 and plotted in Figure 7. It is evident from the plot that the general conclusions for the tube-type also apply to the plate-type IEC. Figure 8 compares plate-type effectivenesses calculated by the model to that from a manufacturer's data. The top band represents model results calculated using an outdoor air wet bulb temperature 21.7°C and dry bulb temperatures from 34 to 42 °C. The lower band represents data supplied to the author's by a manufacturer. It can be seen that the two bands are close to each other, with the maximum discrepancy occurring at a moderate mass flow rate of about 1.5 (kg/s). [Wu 1989] obtained an average effectiveness of 0.54 from field measurements of a plate-type IEC similar to the abovementioned one, with the following average values: t_{dbo} = 30.4 °C, t_{wbo} = 18.6 °C, L_1 = 0.92 m^3/s, and L_2 = 0.33 - 0.378 m^3/s .

The calculated air pressure drops also agree with manufacturer's data.

THERMAL CALCULATIONS WHEN ROOM AIR IS USED AS SECONDARY AIR

Sometimes IEC can be used as a first stage of the air handling system to reduce the cooling load of conventional mechanical refrigeration. In such instances, it is preferable to use room exhaust air as the secondary air for precooling the outdoor, or primary, air. This IEC application is particularly attractive in humid areas where the air-conditioned room temperature is usually much lower than the outside air temperature during the summer.

There are two cases for such an application:

Case1. The temperature difference between the room and outside air is so large that the secondary air temperature is lower than the dew point temperature of the primary air. In this case, the cooling process will result in dehumidification of the primary air.

Case2. The secondary air temperature is not low enough to cause condensation in the primary air flow. In this case, the cooling process will be only sensible.

Since the cooling process in Case 2 can be calculated using the method described in Section 2, only the calculation method for Case 1 will be discussed here.

For the secondary air, the heat and mass transfer processes are the same as shown in Figure 2. Therefore, Equations 3 and 4 can still be used. The processes for the primary air, however, will be different. Due to condensation, there is simultaneous heat and mass transfer. For a differential area dA_i (Figure 9) we can write;

$$g_p dh_p = D_1 (h_p - h_w) dA_i \tag{21}$$

Substituting the Lewis relationship $D_1 = \alpha_i/(1000c_p)$ into Equation 21 gives:

$$g_p dh_p = \frac{\alpha_i}{1000c_p} (h_p - h_w) dA_i \tag{22}$$

After integrating Equation 22, we obtain

$$\frac{h_{p2} - h_w}{h_{p1} - h_w} = \exp\left(- \frac{\alpha_i A_i}{1000 g_p c_p}\right) \tag{23}$$

$$h_{p2} = h_w + (h_{p1} - h_w) \exp\left(- \frac{\alpha_i A_i}{1000 g_p c_p}\right) \tag{24}$$

Since the heat balance equation between primary and secondary air is

$$g_s (h_{s2} - h_{s1}) = g_p (h_{p1} - h_{p2}) \tag{25}$$

we can substitute Equation 24 into Equation 25 to produce:

$$h_{s2} = h_{s1} + \frac{g_p}{g_s} \tag{26}$$

Since the right sides of Equations 4 and 26 are equal to each other, we can obtain:

$$h_w = \frac{\left[1 - \exp(-\frac{\alpha_o A_o}{1000 g_2 c_p})\right] h_{s1} + \frac{g_1}{g_2}\left[1 - \exp(-\frac{\alpha_i A_i}{1000 g_1 c_p})\right] h_{p1}}{\left[1 - \exp(-\frac{\alpha_o A_o}{1000 g_2 c_p})\right] + \frac{g_1}{g_2}\left[1 - \exp(-\frac{\alpha_i A_i}{1000 g_1 c_p})\right]} \tag{27}$$

After h_w is solved by Equation 27, h_{s2} can be calculated by Equations 4 or 26, and h_{p2} by Equation 24. Since

$$\frac{w_{p2} - w_w}{w_{p1} - w_w} = \exp(-\frac{\alpha_i A_i}{1000 g_p c_p}) \tag{28}$$

we get:

$$w_{p2} = w_w + (w_{p1} - w_w)\exp(-\frac{\alpha_i A_i}{1000 g_p c_p}) \tag{29}$$

The leaving primary air t_{dbp2}, humidity ratio reduction Δw_p , enthalpy h_{p2}, and the IEC cooling capacity, Q, can be calculated as follows:

$$t_{dbp2} = \frac{h_{p2} - 2500 w_{p2}}{1.01 + 1.84 w_{p2}} \tag{30}$$

$$\Delta w_p = g_p\,(w_{p1} - w_{p2}) \tag{31}$$

$$h_{p2} = 1.01 t_{dbp2} + w_{p2}\,(2500 + 1.84 t_{dbp2}) \tag{32}$$

$$Q = g_p\,(h_{p1} - h_{p2}) \tag{33}$$

For cooling with dehumidification, the enthalpy effectiveness can be calculated as follows:

$$E_h = \frac{h_{p1} - h_{p2}}{h_{p1} - h_{s1}} = \frac{h_o - h_{p2}}{h_o - h_r} \tag{34}$$

where h_o = enthalpy of outdoor air, kJ/kg
h_r =enthalpy of room air, kJ/kg

Since there is no general definition of effectiveness for cooling with dehumidification, i.e., Case 1 conditions, we have defined two forms of effectiveness :

$$E_{t_o} = \frac{t_{dbp1} - t_{dbp2}}{t_{dbp1} - t_{dbs1}} \tag{35}$$

or,

$$E_{t_o} = \frac{t_{dbo} - t_{dbp2}}{t_{dbo} - t_{dbr}} \tag{36}$$

where t_{dbo} = outdoor air dry bulb temperature, °C
t_{dbr} = room air dry bulb temperature, °C

$$E_t = \frac{t_{dbo} - t_{dbp2}}{t_{dbo} - t_{wbr}} \tag{37}$$

where t_{wbr} = wet bulb temperature of room air, °C

Example. Thermal calculations are made for the tube- and plate-type IECs described in Sections 4.1 and 4.2 using room air as the secondary air. The room air conditions are:

t_{dbr}	= 27.0 °C	h_r	= 61.73 kJ/kg
t_{wbr}	= 21.27 °C	w_r	= 0.01352 kg/kg
RH_r	= 0.60		

The outdoor air parameters are assumed to be the same as in Sections 4.1 and 4.2. The secondary air flow rates also have fixed values. The results of calculations are listed in Tables 3 and 4. From the tables we can see:

1) Cooling of the primary air can be either sensible or with dehumidification depending on the dry and wet bulb temperatures of the outdoor air and the primary air flow rate when the room air conditions are fixed. Our program can distinguish which process actually takes place and determine which method of calculation should be used. If Tables 3 and 4 show no E_h for certain outdoor conditions, that means under those conditions only sensible cooling of the primary air takes place.

2) The cooling capacities and COPs are much bigger than those when outdoor air is used as secondary air.

3) For cooling with dehumidification, enthalpy effectivenesses versus primary air mass flow rates are plotted in Figure 10 for different outdoor air parameters. It is obvious that enthalpy effectiveness is a function only of the primary air mass flow rate and not of outdoor air parameters.

For sensible cooling, the temperature effectiveness may be greater than 1, which means that the leaving primary air temperature is lower than that of room air.

CONCLUSIONS

The heat and mass transfer processes of an IEC are very complicated, making them difficult and time-consuming to calculate by standard mathematical methods. Our method using some simplifications makes the calculations convenient, while maintaining dependable results. We have written several computer programs and subroutines using this methodology that are being used for energy analyses as well as system design or product optimization.

The inputs for these programs are the outdoor and room air parameters, air flow rates and the geometrical parameters of the IEC. For a given IEC with given air flow rates, the only required inputs are the air parameters. The outputs of the programs are all the relevant parameters of an IEC, i.e., the leaving primary and secondary air parameters, the water evaporation rate, air pressure drops, and the cooling capacity, COP, effectiveness, and power demand of the unit.

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Table 1. Calculations for a Tube-type Indirect Evaporative Cooler
(Secondary air is outside air, flowrate is 0.378 m^3/s)

Outdoor air (°C)		Primary air flowrate	Effectiveness	COP	Water Evaporation rate (g/s)	Air pressure drop (Pa)	
Dry bulb	Wet bulb	(m^3/s)				primary	secondary
42	35	0.944	0.50	17.5	2.23	188	8
	32		0.49	23.9	3.10	189	8
	29		0.47	29.8	3.92	190	8
	26		0.46	35.2	4.68	191	8
	35	0.472	0.60	68.7	1.77	52	8
	32		0.59	95.1	2.47	52	8
	29		0.58	119.8	3.15	52	8
	26		0.56	142.8	3.79	53	8
	35	0.236	0.69	167.6	1.46	15	8
	32		0.68	233.9	2.05	15	8
	29		0.67	296.6	2.63	15	8
	26		0.66	356.4	3.18	15	8
38	32	0.944	0.49	14.4	1.86	190	8
	29		0.47	20.6	2.72	191	8
	26		0.45	26.4	3.52	192	8
	23		0.44	31.6	4.26	193	8
	32	0.472	0.59	57.1	1.50	52	8
	29		0.58	83.1	2.20	53	8
	26		0.56	107.3	2.86	53	8
	23		0.55	129.8	3.49	53	8
	32	0.236	0.68	140.5	1.25	15	8
	29		0.67	205.9	1.84	15	8
	26		0.66	268.0	2.41	15	8
	23		0.65	326.8	2.96	15	8
34	29	0.944	0.47	11.5	1.51	192	8
	26		0.45	17.5	2.35	193	8
	23		0.44	23.1	3.13	194	8
	20		0.42	28.1	3.85	195	8
	29	0.472	0.57	46.2	1.23	53	8
	26		0.56	71.5	1.93	53	8
	23		0.55	95.2	2.58	53	8
	20		0.53	117.9	3.18	54	8
	29	0.236	0.67	114.5	1.03	15	8
	26		0.66	178.9	1.62	15	8
	23		0.65	240.2	2.19	15	8
	20		0.64	298.2	2.74	15	8
30	26	0.944	0.45	8.8	1.17	194	8
	23		0.43	14.7	1.99	195	8
	20		0.42	20.0	2.75	196	8
	17		0.40	24.9	3.46	197	8
	26	0.472	0.56	35.7	0.97	53	8
	23		0.54	60.5	1.66	54	8
	20		0.53	84.4	2.28	54	8
	17		0.52	106.1	2.89	54	8
	26	0.236	0.65	89.3	0.83	15	8
	23		0.64	153.0	1.41	15	8
	20		0.63	213.3	1.98	15	8
	17		0.62	270.6	2.52	15	8

Table 2. Calculations for a Plate-type Indirect Evaporative Cooler
(Secondary air is outside air, flowrate is 0.38 m^3/s)

Outdoor air (°C) Dry bulb	Outdoor air (°C) Wet bulb	Primary air flowrate (m^3/s)	Effectiveness	COP	Water Evaporation rate (g/s)	Air pressure drop (Pa) primary	Air pressure drop (Pa) secondary
42	35	2.3	0.50	13.2	4.19	233	96
	32		0.47	17.8	5.74	234	96
	29		0.45	21.8	7.14	235	96
	26		0.43	25.3	8.41	236	97
	23		0.41	28.5	9.54	237	97
	20		0.39	31.3	10.57	238	98
	35	1.6	0.58	27.3	3.66	122	81
	32		0.56	37.0	5.03	122	82
	29		0.53	45.7	6.29	123	82
	26		0.51	53.4	7.45	123	82
	23		0.49	60.3	8.50	124	83
	20		0.47	66.4	9.46	124	83
	35	0.9	0.71	62.9	2.91	45	71
	32		0.69	86.1	4.04	45	72
	29		0.66	107.1	5.10	45	72
	26		0.64	126.4	6.08	46	72
	23		0.62	143.8	7.00	46	73
	20		0.60	159.5	7.84	46	73
	35	0.2	0.90	46.2	1.74	4	66
	32		0.89	64.5	2.45	5	66
	29		0.87	81.9	3.15	5	67
	26		0.86	98.6	3.83	5	67
	23		0.85	114.5	4.49	5	67
	20		0.84	129.6	5.13	5	68
38	32	2.3	0.47	10.6	3.41	236	97
	29		0.44	14.9	4.89	237	97
	26		0.42	18.8	6.23	238	98
	23		0.40	22.1	7.45	239	98
	20		0.38	25.1	8.54	239	98
	32	1.6	0.55	22.2	3.00	123	82
	29		0.53	31.5	4.33	124	83
	26		0.51	39.8	5.55	124	83
	23		0.48	47.2	6.66	124	83
	20		0.46	53.8	7.67	125	84
	32	0.9	0.68	51.5	2.44	45	72
	29		0.66	73.9	3.55	46	73
	26		0.64	94.5	4.57	46	73
	23		0.61	113.0	5.54	46	73
	20		0.59	130.4	6.40	46	74
	32	0.2	0.89	38.8	1.49	5	67
	29		0.87	56.9	2.21	5	67
	26		0.86	74.2	2.90	5	67
	23		0.85	90.6	3.58	5	68
	20		0.84	106.3	4.23	5	68

Table 2. Calculations for a Plate-type Indirect Evaporative Cooler (continued)
(Secondary air is outside air, flowrate is 0.38 m^3/s)

Outdoor air (°C)		Primary air flowrate (m^3/s)	Effectiveness	COP	Water Evaporation rate (g/s)	Air pressure drop (Pa)	
Dry bulb	Wet bulb					primary	secondary
34	29	2.3	0.43	8.2	2.69	238	98
	26		0.41	12.3	4.11	239	99
	23		0.39	16.0	5.40	240	99
	20		0.37	19.2	6.56	241	99
	29	1.6	0.52	17.4	2.40	124	83
	26		0.50	26.3	3.68	125	84
	23		0.48	34.2	4.85	125	84
	20		0.45	41.3	5.92	126	85
	29	0.9	0.65	40.9	1.98	46	73
	26		0.63	62.8	3.06	46	74
	23		0.61	82.5	4.07	46	74
	20		0.59	100.9	4.98	46	74
	29	0.2	0.87	31.6	1.25	5	67
	26		0.86	49.6	1.96	5	68
	23		0.85	66.7	2.65	5	68
	20		0.84	82.9	3.33	5	69

Table 3. Calculations for a Tube-type Indirect Evaporative Cooler
(Secondary air is return air, flowrate is 0.38 m^3/s)

Outdoor air (°C)		Primary air flowrate (m^3/s)	Effectiveness			COP	Water Evaporation rate (g/s)
Dry bulb	Wet bulb		E_{to}	E_t	E_h *		
42	35	0.944	0.46	0.34	0.22	67.7	5.40
	32		0.56	0.41	0.22	48.6	4.08
	29		0.62	0.45		45.2	3.74
	26		0.62	0.45		44.8	3.74
	35	0.472	0.59	0.42	0.34	342.1	4.33
	32		0.68	0.49	0.34	245.9	3.32
	29		0.79	0.57		188.4	2.65
	26		0.79	0.57		186.6	2.66
	35	0.236	0.73	0.53	0.47	1006.6	3.25
	32		0.80	0.58	0.47	724.5	2.56
	29		0.87	0.63	0.47	479.5	1.97
	26		0.95	0.69		473.9	1.92
38	32	0.944	0.53	0.35	0.22	49.1	4.14
	29		0.68	0.45		36.3	3.15
	26		0.68	0.45		36.0	3.16
	23		0.68	0.45		35.6	3.16
	32	0.472	0.67	0.44	0.34	248.6	3.36
	29		0.78	0.52	0.34	164.8	2.50
	26		0.86	0.57		150.8	2.30
	23		0.86	0.57		149.5	2.31
	32	0.236	0.82	0.54	0.47	734.8	2.59
	29		0.92	0.60		487.6	2.00
	26		1.04	0.69		384.9	1.71
	23		1.04	0.69		381.8	1.71
34	31	0.994	0.54	0.30	0.22	43.7	3.78
	28		0.74	0.41	0.22	27.9	2.70
	25		0.80	0.44		27.1	2.59
	22		0.80	0.44		26.9	2.59
	31	0.472	0.71	0.39	0.33	220.0	3.10
	28		0.89	0.49	0.33	141.6	2.27
	25		1.03	0.57		114.6	1.95
	22		1.03	0.57		113.7	1.95
	31	0.236	0.92	0.51	0.47	658.5	2.41
	28		1.06	0.58	0.47	420.5	1.84
	25		1.24	0.68		294.0	1.50
	22		1.24	0.68		292.0	1.50
30	26	0.944	1.17	0.40	0.21	18.9	2.09
	23		1.27	0.44		18.5	2.02
	20		1.27	0.44		18.3	2.03
	17		1.27	0.44		18.2	2.03
	26	0.472	1.42	0.49	0.33	96.1	1.81
	23		1.64	0.57		78.6	1.60
	20		1.64	0.57		78.0	1.60
	17		1.64	0.57		77.4	1.60
	26	0.236	1.70	0.58	0.47	286.3	1.52
	23		1.98	0.68		201.3	1.32
	20		1.98	0.68		200.0	1.32
	17		1.98	0.68		198.8	1.32

* E_h is not shown for sensible cooling.

Table 4. Calculations for a Plate-type Indirect Evaporative Cooler
(Secondary air is return air, flowrate is 0.38 m^3/s)

Outdoor air (°C)		Primary air flowrate (m^3/s)	Effectiveness			COP	Water Evaporation rate (g/s)
Dry bulb	Wet bulb		E_{to}	E_t	E_h *		
42	35	2.3	0.50	0.36	0.14	33.5	8.31
	32		0.56	0.40		31.4	7.51
	29		0.57	0.40		31.0	7.53
	26		0.56	0.40		30.7	7.54
	35	1.6	0.56	0.40	0.19	79.7	7.86
	32		0.67	0.48		66.3	6.39
	29		0.67	0.48		65.5	6.41
	26		0.66	0.48		64.8	6.41
	35	0.9	0.66	0.47	0.29	229.9	6.93
	32		0.80	0.58	0.29	165.5	5.23
	29		0.84	0.61		155.2	4.81
	26		0.84	0.61		153.7	4.82
	35	0.2	0.97	0.70	0.63	283.5	3.80
	32		1.05	0.76	0.63	204.2	3.02
	29		1.14	0.82	0.63	135.3	2.35
	26		1.16	0.84		121.3	2.17
38	32	2.3	0.58	0.38	0.14	24.3	6.29
	29		0.60	0.39		24.6	6.13
	26		0.60	0.39		24.3	6.13
	23		0.59	0.39		24.1	6.14
	32	1.6	0.64	0.42	0.19	57.9	5.96
	29		0.72	0.47		52.2	5.28
	26		0.72	0.47		51.6	5.29
	23		0.72	0.47		51.2	5.29
	32	0.9	0.75	0.49	0.29	167.5	5.29
	29		0.92	0.60		124.9	4.05
	26		0.92	0.60		123.7	4.06
	23		0.92	0.60		122.7	4.06
	32	0.2	1.09	0.72	0.63	207.6	3.05
	29		1.20	0.79	0.63	138.0	2.37
	26		1.27	0.83		98.4	1.98
	23		1.27	0.83		97.7	1.98

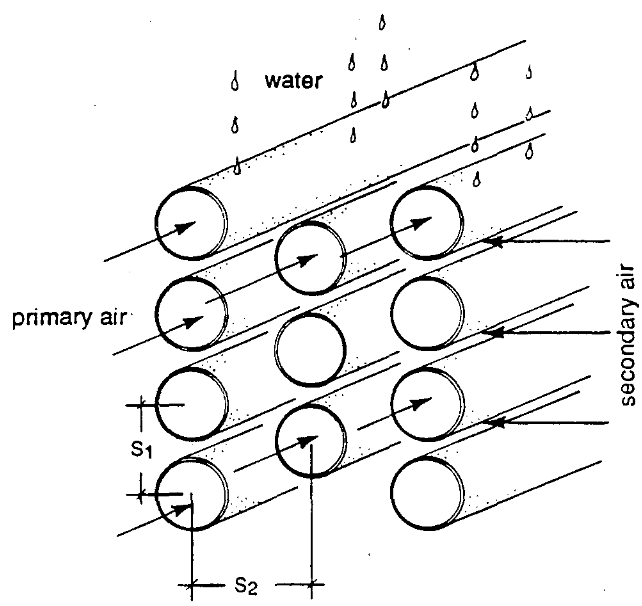
* E_h is not shown for sensible cooling.

Table 4.Calculations for a Plate-type Indirect Evaporative Cooler (continued)
(Secondary air is return air, flowrate is 0.38 m^3/s)

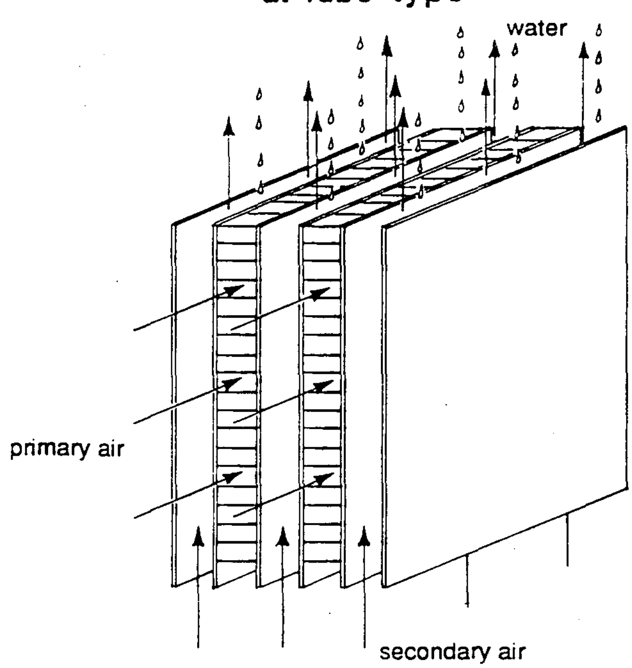
Outdoor air (°C)		Primary air flowrate (m^3/s)	Effectiveness			COP	Water Evaporation rate (g/s)
Dry bulb	Wet bulb		E_{to}	E_t	E_h *		
34	29	2.3	0.70	0.38		18.3	4.80
	26		0.69	0.38		18.1	4.80
	23		0.69	0.38		17.9	4.81
	20		0.69	0.38		17.8	4.81
	29	1.6	0.82	0.45	0.19	39.0	4.32
	26		0.84	0.46		38.8	4.20
	23		0.84	0.46		38.5	4.21
	20		0.84	0.46		38.2	4.21
	29	0.9	0.95	0.52	0.29	112.9	3.89
	26		1.09	0.60		94.0	3.31
	23		1.08	0.60		93.2	3.32
	20		1.08	0.60		92.5	3.32
	29	0.2	1.34	0.74	0.63	140.6	2.40
	26		1.50	0.83	0.63	79.3	1.81
	23		1.52	0.83		75.0	1.77
	20		1.52	0.83		74.6	1.77
30	26	2.3	1.08	0.37		12.2	3.55
	23		1.08	0.37		12.1	3.55
	20		1.08	0.37		12.0	3.55
	17		1.08	0.37		12.0	3.55
	26	1.6	1.32	0.46		26.4	3.17
	23		1.32	0.46		26.2	3.17
	20		1.32	0.45		25.9	3.17
	17		1.32	0.45		25.7	3.17
	26	0.9	1.67	0.57	0.29	65.0	2.67
	23		1.72	0.59		63.9	2.59
	20		1.72	0.59		63.4	2.59
	17		1.72	0.59		63.0	2.59
	26	0.2	2.24	0.77	0.63	81.4	1.83
	23		2.43	0.83		51.9	1.55
	20		2.43	0.83		51.7	1.55
	17		2.43	0.83		51.4	1.55

* E_h is not shown for sensible cooling.

Figure 1. Schematic Drawings of Indirect Evaporative Cooler Types



a. Tube-type



b. Plate-type

Figure 2. Heat and Mass Transfer of the Secondary Air

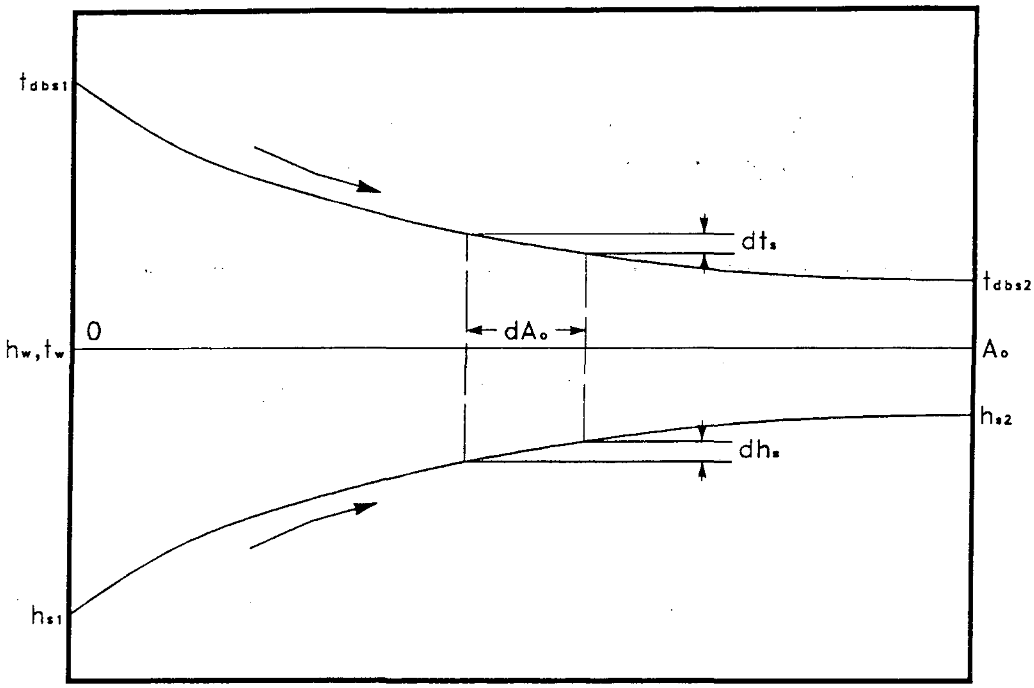


Figure 3. Heat Balance of the Primary Air
(sensible cooling)

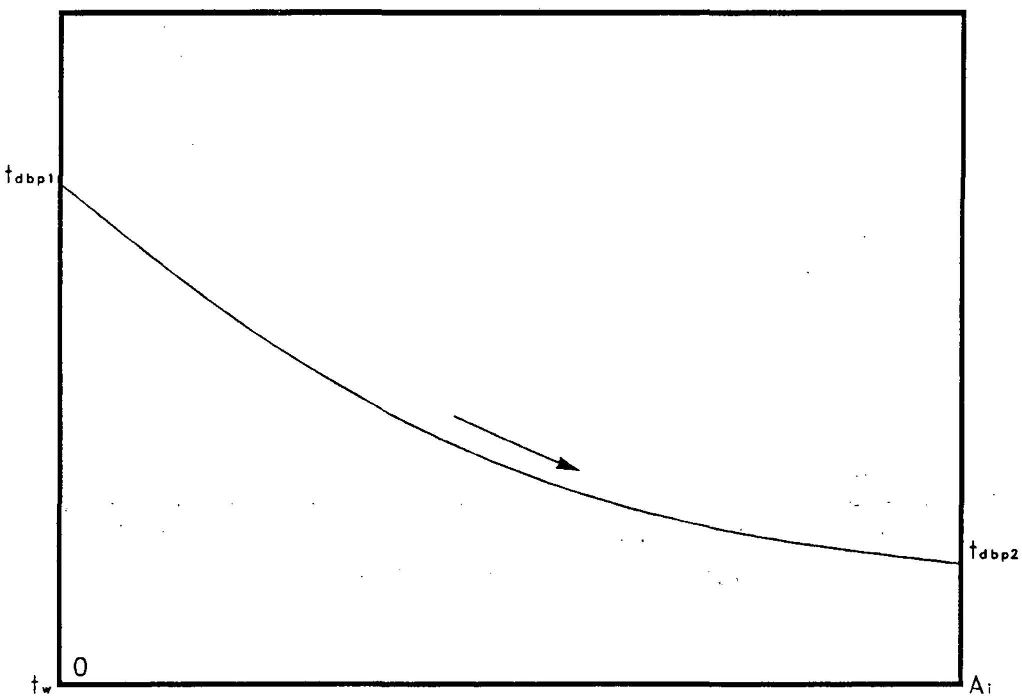


Figure 4. Effect of Air Flow Rate on Indirect Evaporative Cooler Effectiveness

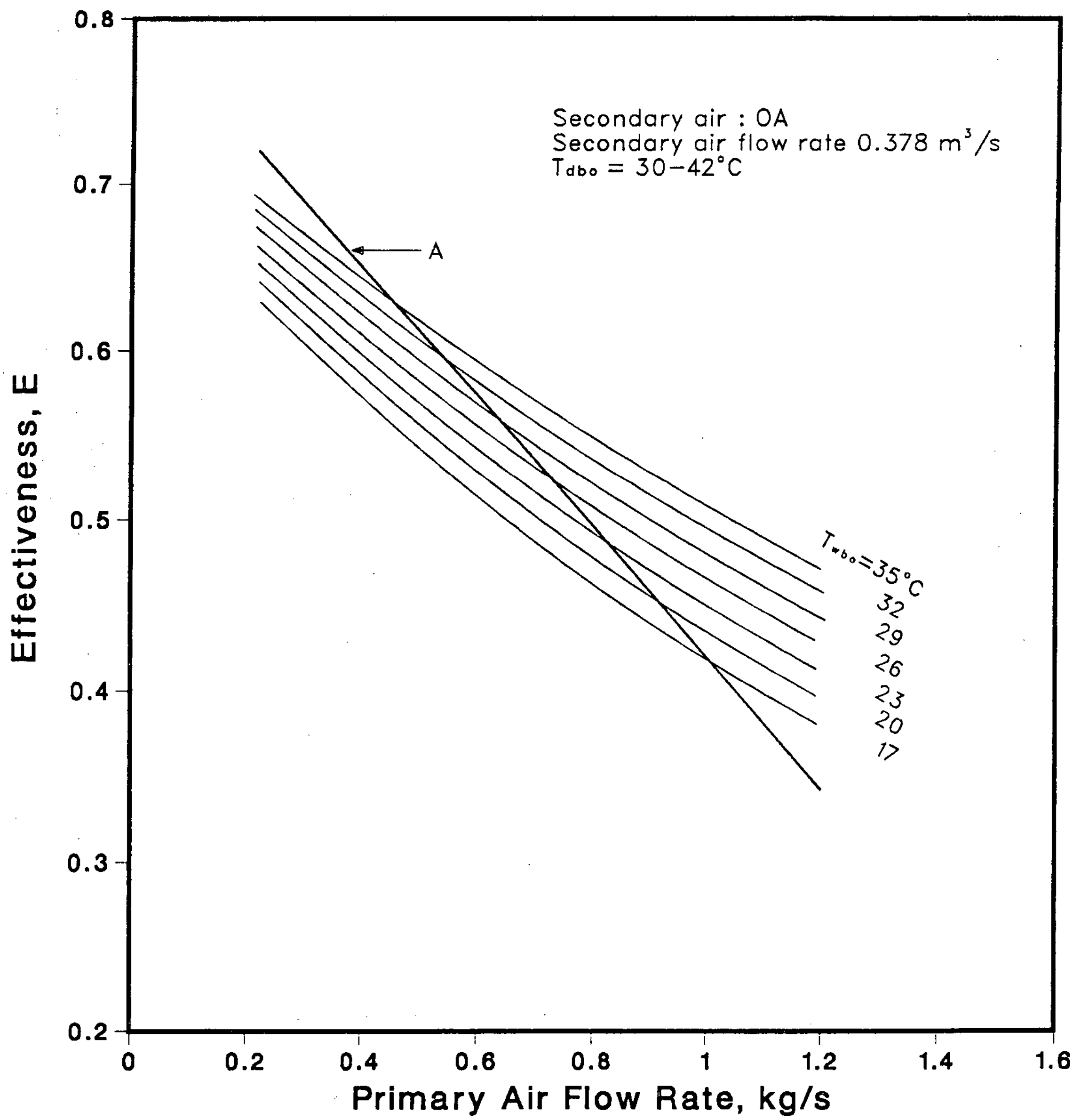


Figure 5. Primary Air Pressure Drop of a Tube Type Indirect Evaporative Cooler

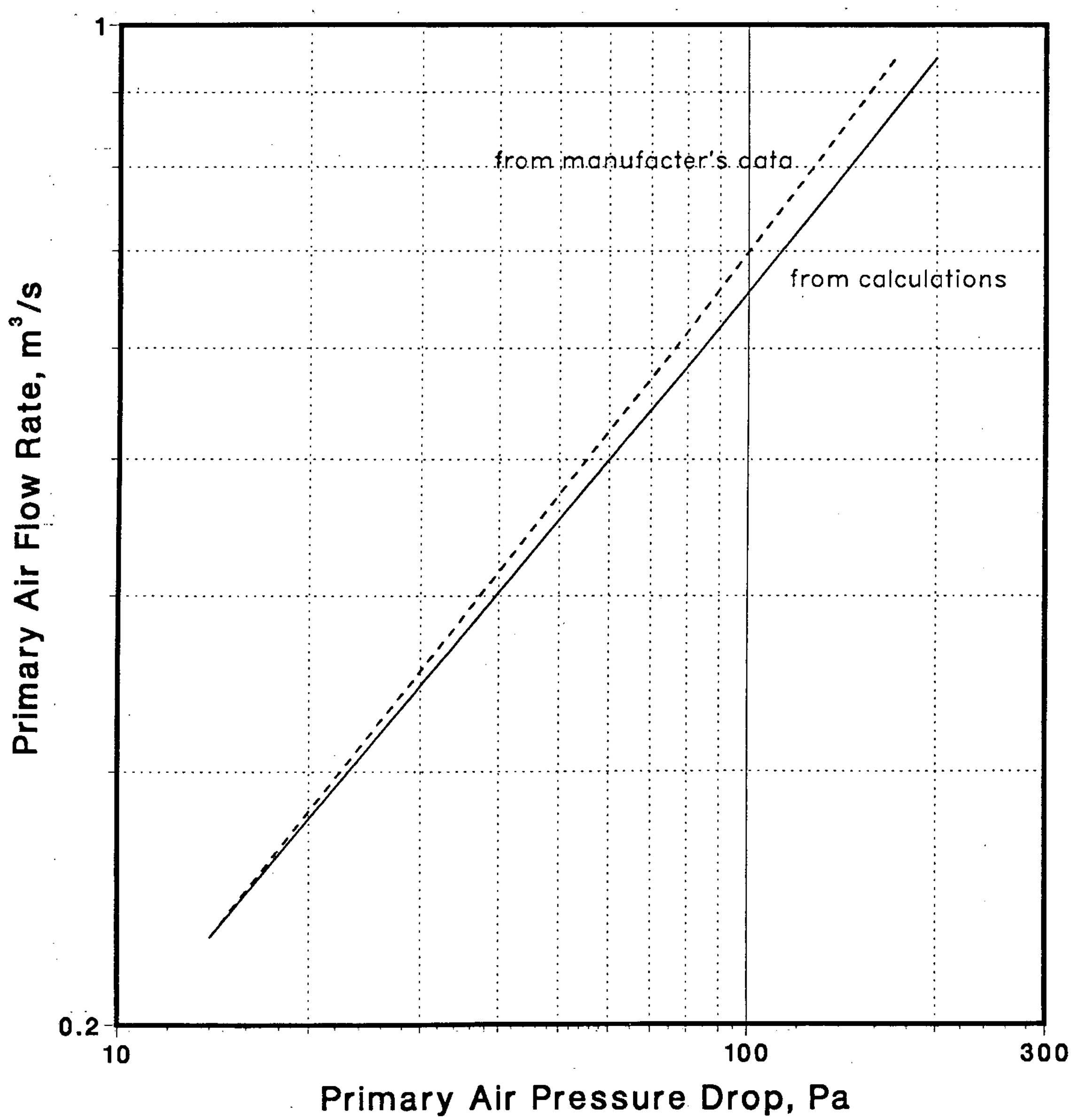


Figure 6. Effect of Barometric Pressure on Evaporative Cooler Effectiveness

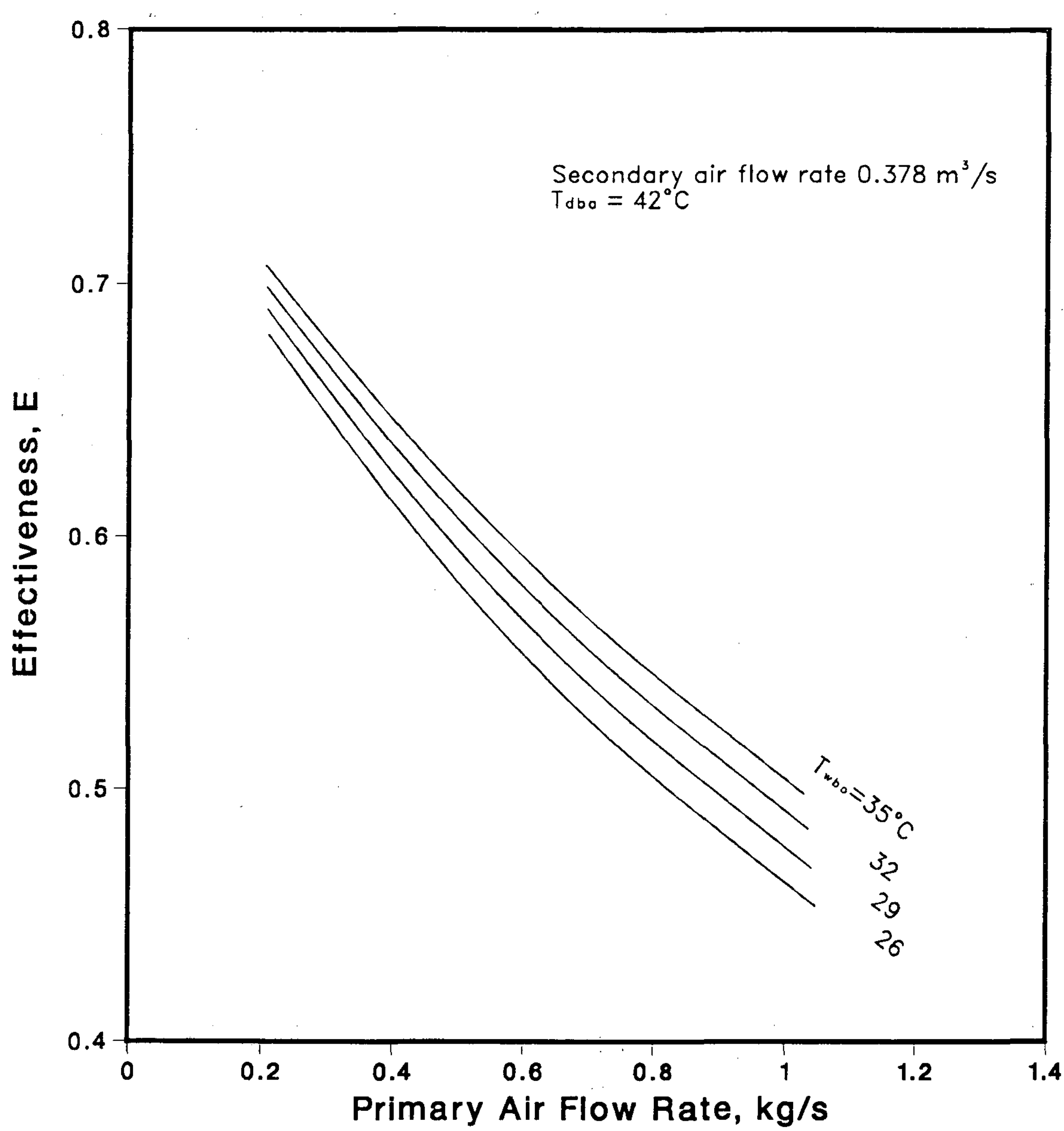


Figure 7. Effectiveness of a Plate Type Indirect Evaporative Cooler

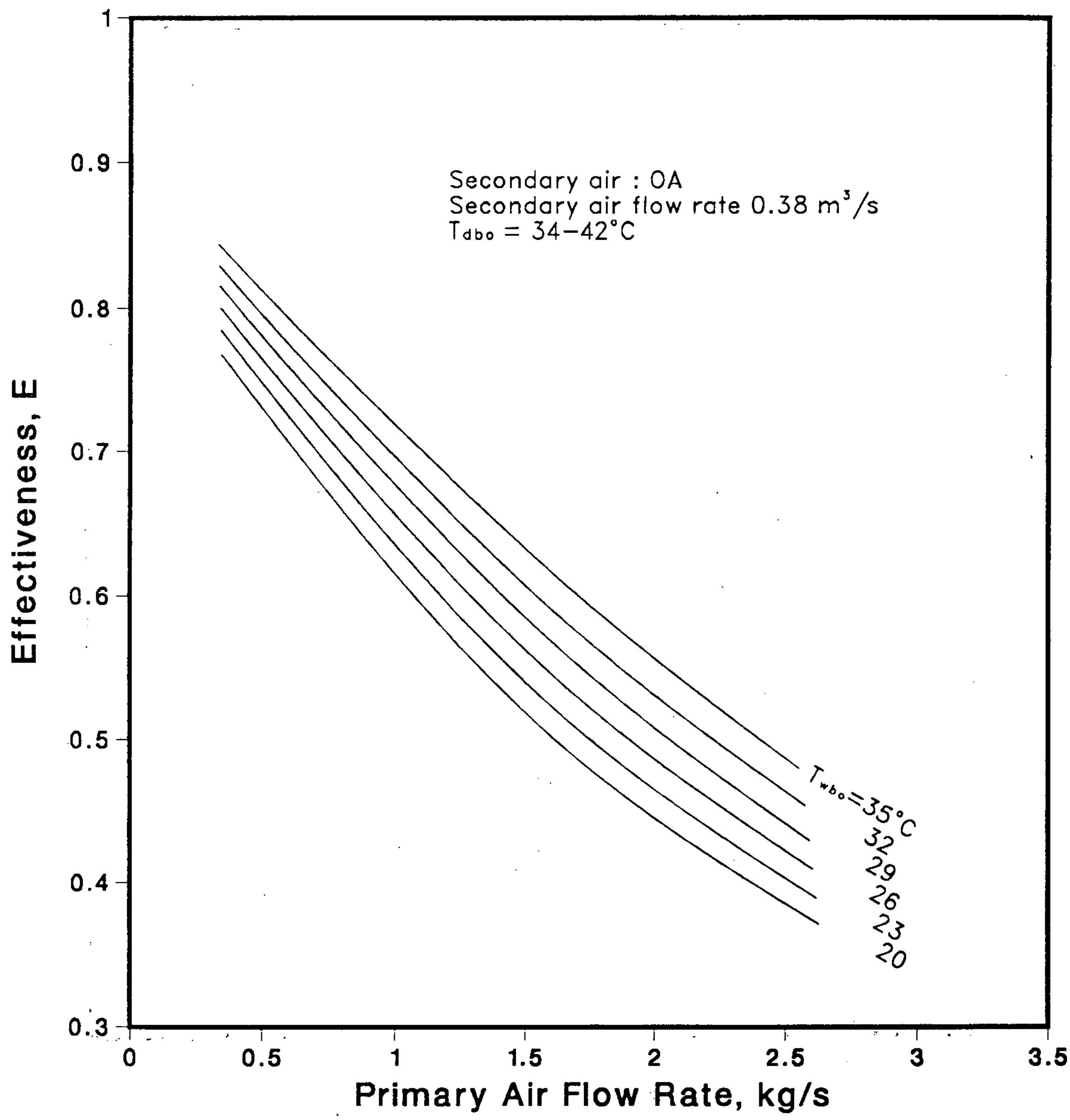


Figure 8. Comparison of Evaporative Cooler Effectiveness Calculated by Model to Manufacturer's Data

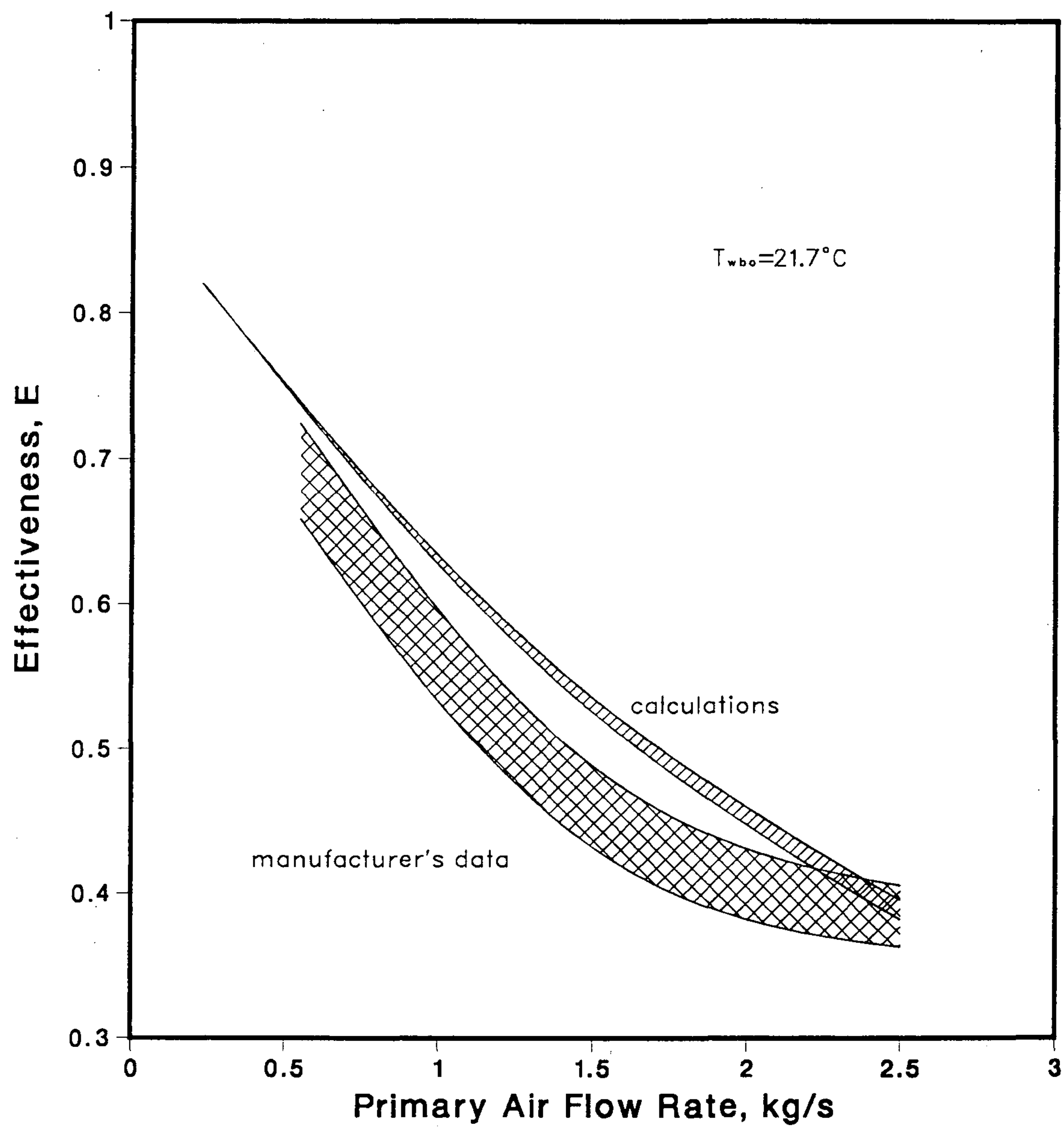


Figure 9. Heat and Mass Transfer of the Primary Air
(cooling with dehumidification)

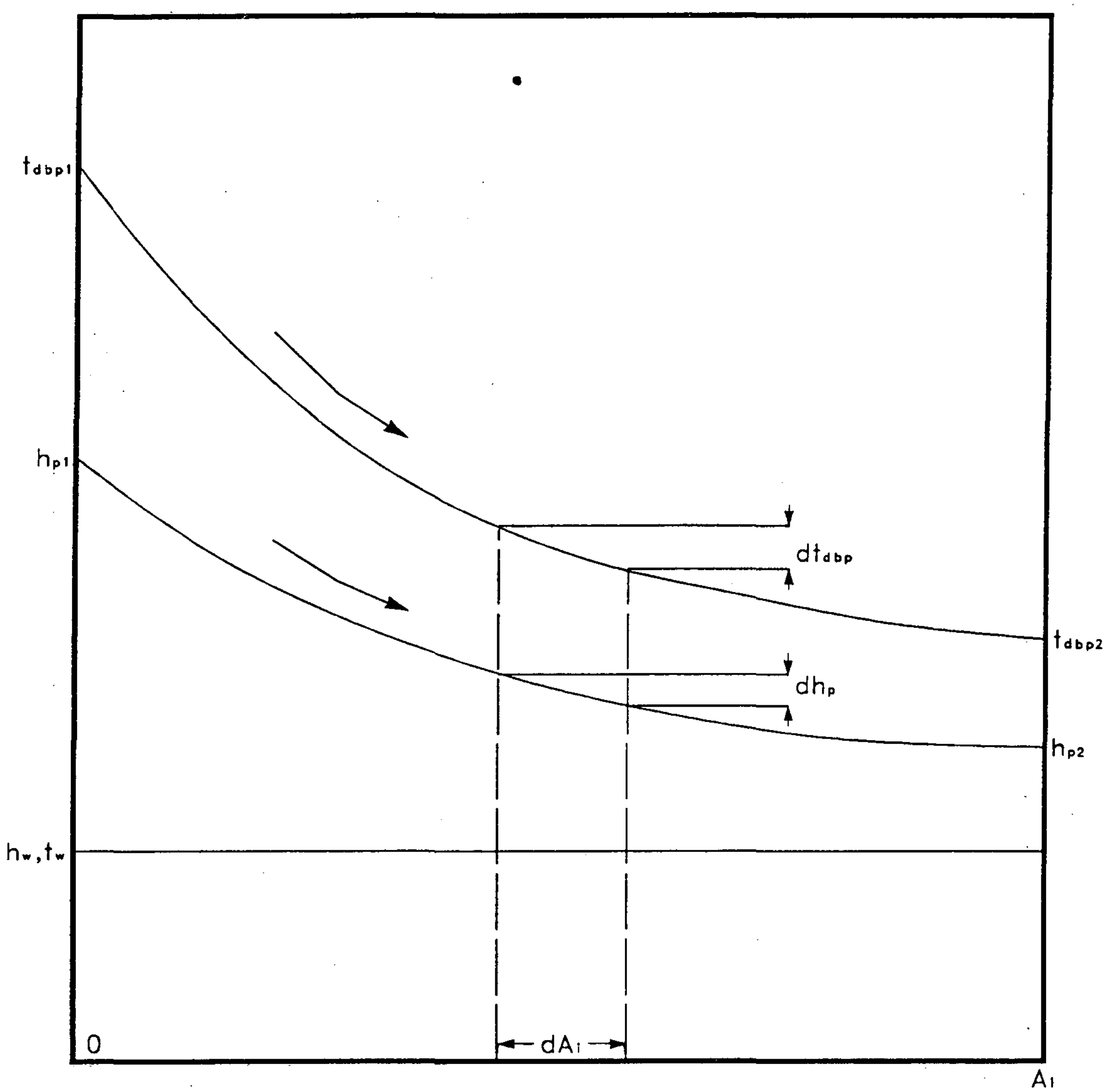
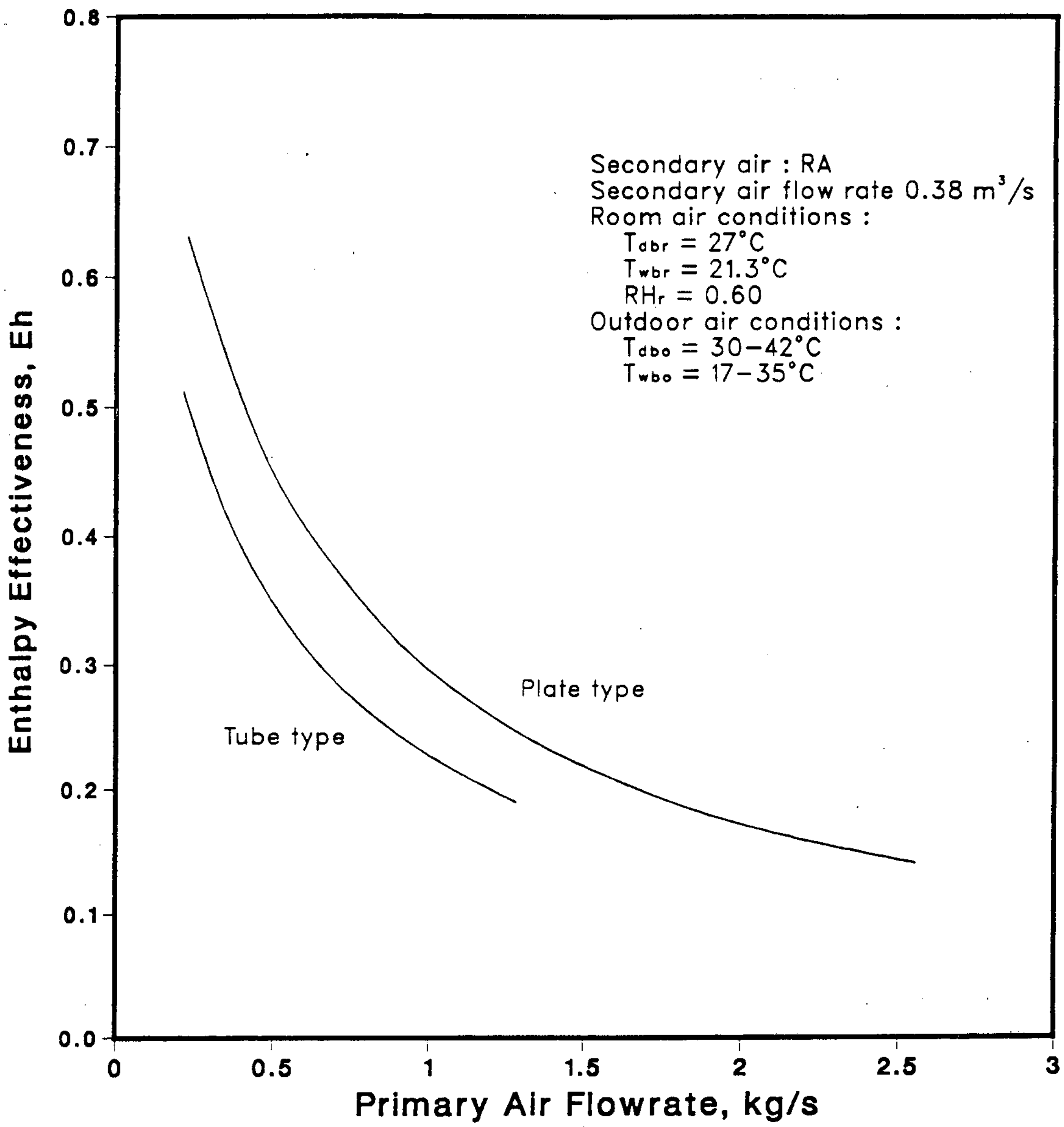


Fig.10 Enthalpy Effectiveness of the Dehumidifying Process in Tube and Plate Type Indirect Evaporative Coolers



APPENDIX 1. CALCULATION OF CONVECTIVE TRANSFER COEFFICIENTS

Nomenclature:

Nu = Nusselt Number, $Nu = \frac{\alpha d}{K}$

Re = Reynolds Number, $Re = \frac{v d \rho}{\mu}$

Pr = Prandtl Number, $Pr = \frac{1000 c_p \mu}{K}$

K = heat conductivity of air, W/m°C

μ = dynamic viscosity of air, kg/ms

ρ = air density, kg/m^3

v = air velocity, m/s

α_i, α_o = convective coefficients at primary and secondary air sides, W/m^2°C

t_{pm}, t_{sm} = average temperatures of primary and secondary air, °C

w_{pm}, w_{sm} = average humidity ratios of primary and secondary air, kg/kg

d = diameter or hydraulic diameter of the air passage, m

c_p = specific heat of air, kJ/kg°C

The following equations can be used to calculate K, μ and ρ :

$$K = 7.6916 \times 10^{-5} t + 0.024178 \tag{A.1}$$

$$\mu = 9.80665 \times 10^{-6} (1.712 + 0.0058t) \tag{A.2}$$

$$\rho = \frac{B(1+w)}{4.615(273.15+t)(0.62198+w)} \tag{A.3}$$

where t = air temperature, °C

B = barometric pressure, mbar

w = humidity ratio of air, kg/kg

1. Tube-type indirect evaporative coolers

The following equations are used for forced convection inside tubes under turbulent flow:

$$Nu_i = \frac{\alpha_i d_i}{K_i} = 0.23 Re_i^{0.8} Pr_i^{0.4} \left[1 + \left(\frac{d_i}{L}\right)^{0.7}\right] \tag{A.4}$$

$$Re_i = \frac{v_i d_i \rho_i}{\mu_i} \tag{A.5}$$

$$Pr_i = \frac{1000 c_p \mu_i}{K_i} \tag{A.6}$$

$$K_i = 7.6916\times10^{-5} t_{pm} + 0.024178 \tag{A.7}$$

$$\mu_i = 9.80665\times10^{-6}(1.712 + 0.0058 t_{pm}) \tag{A.8}$$

$$\rho_i = \frac{B(1+w_{pm})}{4.615(273.15+t_{pm})(0.62198+w_{pm})} \tag{A.9}$$

where $L =$ tube length, m
$d_i =$ inside tube diameter, m

From Equation A.4,

$$\alpha_i = \frac{K_i}{d_i} 0.23 Re_i^{0.8} Pr_i^{0.4} \left[1 + (\frac{d_i}{L})^{0.7}\right] \tag{A.10}$$

For forced convection when the secondary air flow is turbulent and normal to staggered tubes, row number is greater than 10, and $S_1/S_2 \leq 2$, the equation is:

$$Nu_o = \frac{\alpha_o d_o}{K_o} = 0.31 Re_o^{0.6} (\frac{S_1}{S_2})^{0.2} \tag{A.11}$$

Here,

$$Re_o = \frac{v_o d_o \rho_o}{\mu_o} \tag{A.12}$$

$$Pr_o = \frac{1000 c_p \mu_o}{K_o} \tag{A.13}$$

$$K_o = 7.6916\times10^{-5} t_{sm} + 0.024178 \tag{A.14}$$

$$\mu_o = 9.80665\times10^{-6}(1.712 + 0.0058 t_{sm}) \tag{A.15}$$

$$\rho_o = \frac{B(1+w_{sm})}{4.615(273.15+t_{sm})(0.62198+w_{sm})} \tag{A.16}$$

where $S_1 =$ tube distance (center to center) in a row, m
$S_2 =$ row distance (center to center), m
$d_o =$ outside tube diameter, m

From Equation A.11,

$$\alpha_o = \frac{K_o}{d_o} 0.31 Re_o^{0.6} (\frac{S_1}{S_2})^{0.2} \qquad (A.17)$$

2. Plate-type indirect evaporative coolers

For very narrow passages between plates, the following equations can be used:

a. When Re ≥ 1000,

$$Nu = \frac{\alpha d}{K} = 0.2 Re^{0.67} Pr^{0.4} (\frac{\mu}{\mu_w})^{0.1} \qquad (A.18)$$

b. When Re ≤ 10,

$$Nu = \frac{\alpha d}{K} = 1.68 (\frac{RePrd}{L})^{0.4} (\frac{\mu}{\mu_w})^{0.1} \qquad (A.19)$$

where d = hydraulic diameter of primary or secondary air passages, m
L = length of primary or secondary air passages, m
μ_w = dynamic viscosity of air with effective surface temperature t_w, kg/ms

In Equations A.18 and A.19 μ,K and ρ (needed to calculate Re) should be calculated using the average air temperature.

c. When 10<Re<1000, an interpolative method can be used.

APPENDIX 2. CALCULATION OF FRICTION LOSS FACTORS

Colebrook's formula is used to calculate the friction loss factor:

$$\frac{1}{\sqrt{f}} = 1.74 - 2\log(\frac{2k}{d} + \frac{18.7}{Re\sqrt{f}}) \qquad (A.20)$$

where f = friction loss factor, dimensionless
k = roughness of passage walls, m
d = hydraulic diameter, m
Re = Reynolds Number of the air flow, dimensionless

Since it is difficult to solve f directly by Equation A.20, an iterative method should be used.

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INFORMATION RESOURCES DEPARTMENT
1 CYCLOTRON ROAD
BERKELEY, CALIFORNIA 94720