3653

THE ENERGY-SAVING POTENTIAL OF PRECOOLING INCOMING OUTDOOR AIR BY INDIRECT EVAPORATIVE COOLING

P. Chen, Ph.D. **H. Qin, Ph.D.** **Y.J. Huang**
Member ASHRAE

H. Wu, Ph.D.
Member ASHRAE
C. Blumstein, Ph.D.

ABSTRACT

This paper investigates the energy-saving potential of using indirect evaporative coolers to precool incoming outdoor air as the first stage of a standard cooling system. For dry and moderately humid locations, either exhaust room air or outdoor air can be used as the secondary air to the indirect evaporative precooler with similar energy savings. Under these conditions, the use of outdoor air is recommended due to the simplicity in installing the duct system. For humid locations, the use of exhaust room air is recommended because the precooling capacity and energy savings, will be greatly increased. For locations with short cooling seasons, the use of indirect evaporative coolers for precooling may not be worthwhile.

The paper also gives some simplified indices for easily predicting the precooling capacity, energy savings, and water consumption of an indirect evaporative precooler. These indices can be used for cooling systems with continuous operation, but further work is needed to determine whether the same indices are also suitable for cooling systems with intermittent operations.

INTRODUCTION

The application of indirect evaporative cooling (IEC) in arid locations is well known (Watt 1986). The main advantages are the reduction of energy consumption and less use of mechanical refrigeration, thereby reducing the release of chlorofluorocarbons (CFCs). There is also interest in how to use IECs in humid and moderately humid locations where dehumidification of supply outdoor air is usually necessary, a function that cannot be performed by a simple indirect evaporative cooler. Some new applications have appeared, including precooling incoming outdoor air by IEC (Supple and Broughton 1985; Peterson and Hunn 1985), using desiccant systems with indirect and direct evaporative-cooling (Burns et al. 1985; Manley et al. 1985; Matsuki et al. 1983), and improving the COP of a mechanical refrigeration system by sending the leaving secondary air from an IEC to an air-cooled condenser. Precooling outdoor air by indirect evaporative cooling is comparatively easy as the first stage of an air-handling system. The size of the second stage (for example, a conventional air-handling installation) can then be reduced with the result that both the first cost and the energy consumption of the second stage are lower. Therefore, a detailed investigation of the effect of precooling by indirect evaporative cooling seems to be warranted.

THE ADVANTAGES OF PRECOOLING SUPPLY OUTDOOR AIR BY IEC

For nonresidential buildings or buildings in humid and moderately humid locations, indirect evaporative cooling with or without direct evaporative cooling (one-stage or two-stage) usually is not sufficient to meet cooling loads, but it can serve as a booster for mechanical refrigeration. In this case the role of the IEC is to precool the incoming outdoor air (Figure 1). The first stage is an IEC and the second stage is a conventional air-handling unit (an air-cooling coil, an air washer, etc.). The IEC cools the incoming outdoor air (primary air). The secondary air can be either outdoor air (Figure 1a) or exhaust room air (Figure 1b). When outdoor air is used, the dry-bulb temperature of the primary air will be lowered depending on the effectiveness of the IEC, but its humidity ratio will not be changed. This results in the reduction of the sensible, but not latent, cooling load of the second stage in comparison with the system without precooling. When exhaust room air is used as the secondary air, its temperature after direct evaporation sometimes may be lower than the dew-point temperature of the primary air. In such cases, there may be some dehumidification, i.e., condensation, of the primary air and reductions in both the sensible and latent loads of the

Peilin Chen and **Huimin Qin** are professors in the Department of Mechanical Engineering at Tongji University, China. **Yu Joe Huang** is a staff scientist in the Energy Analysis Program of the Energy and Environment Division at Lawrence Berkeley Laboratory, Berkeley, CA. **Hofu Wu** is a professor in the Department of Architecture at California State Polytechnical University, Pomona, CA. **Carl Blumstein** is a professor in the university-wide Energy Research Group at the University of California, Berkeley, CA.

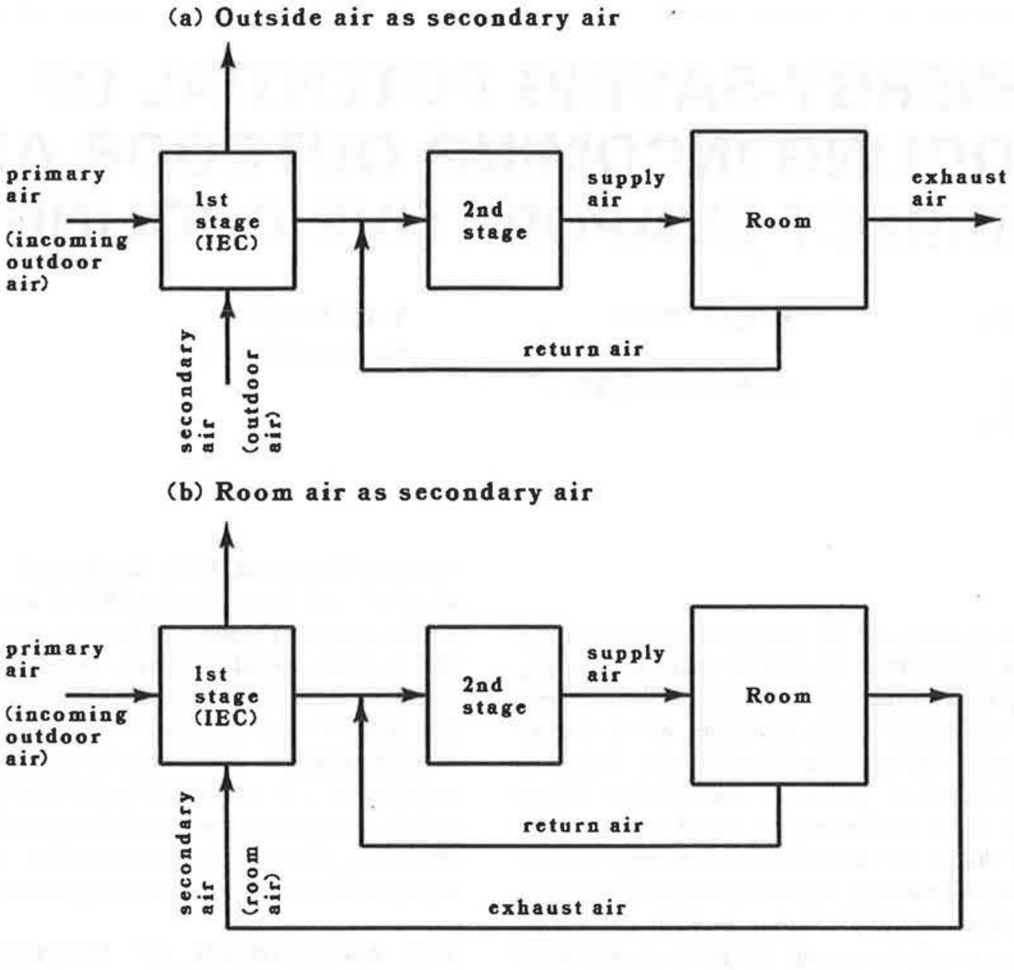

Figure 1 *Cooling systems using indirect evaporative precooling.*
(a) Outside air as secondary air.
(b) Room air as secondary air.

second stage. In a hybrid system, the IEC for precooling outdoor air reduces both peak demand and total energy consumption in the cooling season. According to our computer simulation, for example, if a tube-type IEC is installed to precool the outdoor air, it will provide a cooling capacity of more than 10 kW and a reduction of electric demand of the mechanical refrigeration greater than 3 kW per m³/s of primary air in the hottest summer days in Sacramento, California.

ESTIMATING THE ENERGY SAVINGS OF IEC PRECOOLING

To estimate the energy-saving potential of precooling by IEC, calculations are made for various U.S. and Chinese cities using the bin method and an analytical heat and mass transfer IEC model developed previously by the authors (Chen et al. 1991). As described in that paper, the calculated effectiveness of different IEC designs using the authors'model varied from 40% to 80%, depending on the primary airflow rate, and agreed well with manufacturers' data. The following locations and IEC precooling configurations have been investigated :

1. Sixteen California climatic zones as designated by the California Energy Commission, eight other American cities (Fort Worth, San Antonio, Lake Charles, Miami, Phoenix, Atlanta, New York, and Chicago), and two Chinese cities (Beijing and Shanghai) were selected for calculations. The climates in these cities range from very arid to very humid (see Table 1). Hourly weather data for these locations have been processed to derive the number of hours falling within each 5°C and 0.002 humidity ratio band. Calculations are then carried out using the authors' IEC model for the bin distribution of outdoor air parameters. Only those bins with temperatures greater than or equal to 25°C are taken into account.
2. Two kinds of IECs (tube-type and plate-type) were studied. Calculations were made for the total precooling capacity, the total energy saving in comparison with mechanical refrigeration, and the total water consumption for the cooling season.
3. Either outdoor air or exhaust room air is used as the secondary air. The flow rate of secondary air, L^2, is fixed at 0.378 m³/s (800 cfm) for a typical plate-type IEC or for one core of a typical tube-type IEC.

TABLE 1
Climatic Parameters for Climate Locations Covered in Analysis

Location	Cool. Deg. Days 65°F	Cool. Deg. Hrs/24 75°F	Lat. Enth. Hrs/24 *	Location	Cool. Deg. Days 65°F	Cool. Deg. Hrs/24 75°F	Lat Enth. Hrs/24 *
California cities				*Other U.S. cities*			
1 (Arcata)	1	0	0.0	Fort Worth	2453	1045	490.1
2 (Santa Rosa)	920	527	0.7	San Antonio	2811	1077	540.1
3 (Oakland)	80	10	0.0	Lake Charles	2631	849	816.9
4 (Sunnyvale)	198	50	1.4	Miami	4005	1194	1154.7
5 (Santa Maria)	88	51	0.0	Phoenix	3661	2145	96.8
6 (Long Beach)	89	194	9.7	Atlanta	1543	405	284.4
7 (San Diego)	657	55	12.6	New York	1005	256	118.2
8 (El Toro)	826	243	11.3	Chicago	969	272	120.7
9 (Pasadena)	1053	372	10.2				
10 (Riverside)	1313	676	3.9	*Chinese cities*			
11 (Red Bluff)	1925	958	3.0	Beijing	1583	607	303.8
12 (Sacramento)	1166	540	1.5	Shanghai	1880	663	738.3
13 (Fresno)	916	916	2.0				
14 (China Lake)	2773	1672	0.0				
15 (El Centro)	4239	2610	60.4				
16 (Mt. Shasta)	552	212	2.2				

* Latent enthalpy hours calculated at base temperature of 75°F and humidity ratio of 0.0116; units are in Btu·hour/pound air)

4. The primary airflow rate, L^1, is not fixed. The ratio of L^1/L^2 is in the range of 0.625 - 2.5 (for the tube type) or 0.8 - 6.0 (for the plate type).
5. When exhaust room air is used as the secondary air, it is assumed to be always at a dry-bulb temperature of 25°C and a relative humidity of 50%.

RESULTS

The purpose of our calculations was to determine the effect of precooling by IEC on energy and water consumption. Previous research by the authors has shown that IEC performance varied with the primary and secondary airflows and the wet-bulb temperature of the entering air (L^1, L^2, and *twb*, Chen et al. 1991). Since $L2$ is fixed by the manufacturer and *twb* is determined by the climate, the primary design determinant of IEC effectiveness becomes the ratio of the primary to the secondary airflow rates (L^1/L^2). Three results were obtained for various L^1/L^2 ratios during the cooling season:

1. Total precooling capacity Q, which is the sum of the precooling capacity of each bin considered. The average hourly cooling capacity per m^3/s of primary air can be calculated as follows:

$$q = \frac{Q}{\text{bin hours} \times L_1} \qquad (1)$$

where

q	=	average hourly precooling capacity per m^3/s of primary air, kWh/h (m^3/s);
Q	=	total cooling capacity in cooling season, kWh;
bin hours	=	total hours with outdoorair dry-bulb temperature t ≥ 25 °C, h;
L_1	=	primary airflow rate, m^3/s.

2. Total water consumption W, which is the sum of the water consumption due to evaporation in the secondary

airflow for each bin considered. The average hourly water consumption per m^3/s of primary air can be calculated by Equation 2:

$$w = \frac{W}{\text{bin hours} \times L_1} \quad (2)$$

where

w = average hourly water consumption for every 1 m^3/s of primary air, kg/h (m^3/s); and
W = total water consumption in cooling season, kg.

3. Total energy-saving *ES*, which is the sum of energy saving due to precooling outdoor air by IEC in comparison with a conventional water chiller used to provide the equivalent precooling capacity for each bin considered. Power for fans and pumps in the refrigeration installation are taken into account using the following regression equations based on Warren (1985), assuming a chilled-water temperature of 7°C:

$$COP = 3.517/(0.4719 + 0.2232 \cdot t_{cond}) \quad (3)$$

where

t_{cond} = condenser water temperature, °C.

If t_{cond} is taken to be t_{wbo} + 5.0, the equation then becomes

$$COP = 3.517/(0.4719 + 0.2232 \cdot (t_{wbo} + 5.0)) . \quad (4)$$

The average hourly energy saving per m^3/s of primary air is

$$es = \frac{ES}{\text{bin hours} \times L_1} \quad (5)$$

where

es = average hourly energy saving per m^3/s of primary air, kWh/h(m^3/s) and
ES = total energy saving in cooling season, kWh.

In calculating Equation 5, the additional energy consumption due to the increased air resistance on both the primary and secondary sides is included. The values for q, es, and w are listed in Tables 2 through 5. The designs considered include typical tube-type IECs with L_1/L_2 ratios of 2.5 or 1.25 and typical plate-type IECs with L_1/L_2 ratios of 4.74 or 2.37. Results for other L_1/L_2 values are not listed in the tables, but they are evident in the subsequent figures.

Tables 2 and 3 assume that outside air is used as the secondary air, while Tables 4 and 5 assume that exhaust room air is used. For the latter case, the cooling process of the primary air may include dehumidification because the dry-bulb temperature of the secondary air after evaporation may be lower than the dew-point temperature of the primary air. Lastly, when the enthalpy of the outdoor air is below that of indoor air, we use outdoor air instead of exhaust room air in our calculations.

For the arid California cities, when outdoor air is used as the secondary air the relationship between the average hourly energy saving (*es*) and water consumption (*w*) is nearly linear (Figure 2). For the non-California cities, the situation is different except for Phoenix, which is very dry. When the secondary air is the exhaust room air, the distribution is quite divergent and the relation between es and w cannot be expressed by any kind of curve.

The curves in Figure 2 have limited utility because for any city other than those cities calculated neither the abscissa nor the ordinate can be known without calculation in detail in advance. If we can find indices that are appropriate for all cities, these indices would be useful in predicting energy and water consumption for precooling outdoor air by IEC. Our results suggest that such indices do exist.

SIMPLIFIED INDICES OF IEC PRECOOLING POTENTIAL

For the case of outdoor air used as secondary air, let q, es, and w be divided by the average wet-bulb temperature depression of the outdoor air of the bins considered, and the new parameters q_o, es_o, and w_o are obtained:

$$q_o = \frac{q}{\Delta t_m} , \quad (6)$$

$$es_o = \frac{es}{\Delta t_m} , \quad (7)$$

where

Δt_m = average wet-bulb temperature depression of outdoor air of the bins considered, °C;
q_o = precooling capacity index, kWh/h·°C (m^3/s);
es_o = energy saving index, kWh/h·°C (m^3/s); and
w_o = water consumption index, kg/h·°C (m^3/s).

Here the subscript "o" means outdoor air applied as secondary air.

As shown in Table 3, the resultant "precooling indices," i.e., q_o, es_o, and w_o, are essentially the same for all California cities for the same values of L_1/L_2. Their average values are shown in Table 6.

Figures 3 and 4 show two sets of curves by plotting the indices q_o, es_o, and w_o against the flow rate ratios L_1/L_2 for the tube- and plate-type precoolers. These curves can be applied to all California cities. The indices for non-California cities have very similar values.

TABLE 2
Precooling Capacities for Different Indirect Evaporative Designs with Outside Air as Secondary Air

City	Bin hrs	Precooling capacity by IEC type and L_1/L_2* (kWh/h per m^3/s air) tube 1.25	tube 2.50	plate 2.37	plate 4.74	Energy savings by IEC type and L_1/L_2* (kWh/h per m^3/s air) tube 1.25	tube 2.50	plate 2.37	plate 4.74	Water use by IEC type and L_1/L_2* (kg/h per m^3/s air) tube 1.25	tube 2.50	plate 2.37	plate 4.74
California climate locations													
Arcata (1)	72	5.26	4.00	5.60	3.93	1.35	0.84	1.42	0.85	14.3	8.5	11.6	6.9
China Lake (2)	3515	9.06	6.98	9.79	6.97	2.34	1.62	2.52	1.65	24.9	14.9	20.2	12.2
El Centro (3)	5004	9.51	7.37	10.36	7.44	2.55	1.79	2.77	1.84	26.0	15.7	21.3	13.0
El Toro (4)	1433	5.47	4.20	5.89	4.16	1.44	0.91	1.53	0.93	14.9	8.9	12.1	7.2
Fresno (5)	3446	7.84	6.07	8.53	6.10	2.12	1.45	2.29	1.49	21.4	12.9	17.5	10.6
Long Beach (6)	1966	5.40	4.15	5.82	4.11	1.43	0.90	1.52	0.92	14.7	8.8	11.9	7.2
Mount Shasta (7)	1787	6.69	5.14	7.21	5.11	1.76	1.16	1.88	1.18	18.3	11.0	14.8	8.9
Oakland (8)	441	5.26	4.01	5.61	3.93	1.35	0.84	1.42	0.84	14.3	8.5	11.6	6.9
Pasadena (9)	2186	6.29	4.84	6.78	4.79	1.66	1.08	1.77	1.10	17.1	10.3	13.9	8.4
Red Bluff (10)	2500	7.75	5.97	8.38	5.96	2.04	1.39	2.20	1.42	21.2	12.7	17.2	10.4
Riverside (11)	3230	7.45	5.76	8.08	5.75	1.98	1.34	2.14	1.37	20.4	12.2	16.6	10.1
Sacramento (12)	2086	6.96	5.38	7.55	5.37	1.88	1.26	2.02	1.29	19.0	11.4	15.5	9.4
San Diego (13)	3128	4.57	3.51	4.92	3.46	1.20	0.73	1.28	0.75	12.4	7.4	10.1	6.0
Santa Maria (14)	382	6.18	4.71	6.60	4.64	1.58	1.01	1.67	1.02	16.9	10.1	13.6	8.1
Santa Rosa (15)	1570	7.06	5.45	7.64	5.42	1.88	1.26	2.02	1.29	19.3	11.6	15.7	9.5
Sunnyvale (16)	1617	5.15	3.96	5.54	3.90	1.36	0.85	1.44	0.86	14.0	8.4	11.4	6.8
Other U.S. locations													
Fort Worth	3658	4.01	3.15	4.42	3.19	1.14	0.71	1.25	0.75	10.8	6.6	8.9	5.5
San Antonio	4737	4.40	3.45	4.84	3.50	1.25	0.80	1.36	0.84	11.9	7.2	9.8	6.0
Lake Charles LA	4737	3.18	2.50	3.52	2.54	0.92	0.53	1.00	0.58	8.5	5.2	7.1	4.3
Miami	7735	4.18	3.29	4.62	3.34	1.22	0.77	1.34	0.82	11.2	6.8	9.3	5.7
Phoenix	4316	7.78	6.06	8.49	6.11	2.11	1.46	2.29	1.51	21.3	12.9	17.4	10.6
Atlanta	2514	3.24	2.53	3.55	2.55	0.90	0.52	0.97	0.55	8.7	5.3	7.2	4.4
New York	1702	4.05	3.14	4.40	3.13	1.10	0.66	1.18	0.69	10.9	6.6	8.9	5.4
Chicago	1666	4.32	3.35	4.70	3.35	1.18	0.73	1.27	0.76	11.7	7.1	9.6	5.8
Chinese locations													
Beijing	2574	4.31	3.34	4.69	3.34	1.17	0.72	1.26	0.75	11.7	7.0	9.5	5.8
Shanghai	3191	3.16	2.48	3.44	2.49	0.91	0.53	0.98	0.56	8.4	5.2	6.9	4.2

*L_1/L_2 is the ratio of the primary and secondary air flow rates.

For the case when exhaust room air is used as secondary air, the indices can be obtained by following equations:

$$q_r = \frac{q}{t_{dbm} - t_{wbr}}, \qquad (9)$$

$$es_r = \frac{es}{t_{dbm} - t_{wbr}}, \qquad (10)$$

$$w_r = \frac{w}{t_{dbm} - t_{wbr}}, \qquad (11)$$

where

t_{dbm} = average dry-bulb temperature of outdoor air of the bins considered, °C;
t_{wbr} = room air wet-bulb temperature assumed to be constant over cooling season, °C;
q_r = precooling capacity index, kWh/h°C (m^3/s);
es_r = energy savings index, kWh/h°C (m^3/s); and
w_r = water consumption index, kg/h°C (m^3/s).

Here, the subscript "r" means exhaust room air applied as secondary air.

TABLE 3
Precooling Indices for Different Indirect Evaporative Designs with Outside Air as Secondary Air

City	Avg. dry bulb temp. depres.	Precooling index by IEC type and L_1/L_2* (kWh/h°C per m³/s air)				Energy savings index by IEC type and L_1/L_2* (kWh/h°C per m³/s air)				Water use index by IEC type and L_1/L_2* (kg/h°C per m³/s air)			
		tube 1.25	tube 2.50	plate 2.37	plate 4.74	tube 1.25	tube 2.50	plate 2.37	plate 4.74	tube 1.25	tube 2.50	plate 2.37	plate 4.74
California climate locations													
Arcata (1)	8.59	0.61	0.47	0.65	0.46	0.16	0.10	0.17	0.10	1.67	0.99	1.35	0.80
China Lake (2)	15.21	0.60	0.46	0.64	0.46	0.15	0.11	0.17	0.11	1.64	0.98	1.33	0.80
El Centro (3)	15.64	0.61	0.47	0.66	0.48	0.16	0.11	0.18	0.12	1.67	1.00	1.36	0.83
El Toro (4)	8.90	0.61	0.47	0.66	0.47	0.16	0.10	0.17	0.10	1.67	1.00	1.36	0.82
Fresno (5)	12.81	0.61	0.47	0.67	0.48	0.17	0.11	0.18	0.12	1.67	1.01	1.37	0.83
Long Beach (6)	8.76	0.62	0.47	0.66	0.47	0.16	0.10	0.17	0.11	1.68	1.00	1.36	0.82
Mount Shasta (7)	11.14	0.60	0.46	0.65	0.46	0.16	0.10	0.17	0.11	1.64	0.98	1.33	0.80
Oakland (8)	8.60	0.61	0.47	0.65	0.46	0.16	0.10	0.17	0.10	1.66	0.99	1.35	0.80
Pasadena (9)	10.30	0.61	0.47	0.66	0.47	0.16	0.11	0.17	0.11	1.67	1.00	1.35	0.81
Red Bluff (10)	12.74	0.61	0.47	0.66	0.47	0.16	0.11	0.17	0.11	1.66	1.00	1.35	0.82
Riverside (11)	12.24	0.61	0.47	0.66	0.47	0.16	0.11	0.18	0.11	1.66	1.00	1.36	0.82
Sacramento (12)	11.29	0.62	0.48	0.67	0.48	0.17	0.11	0.18	0.11	1.68	1.01	1.37	0.83
San Diego (13)	7.37	0.62	0.48	0.67	0.47	0.16	0.10	0.17	0.10	1.68	1.00	1.36	0.82
Santa Maria (14)	10.18	0.61	0.46	0.65	0.46	0.16	0.10	0.16	0.10	1.66	0.99	1.34	0.80
Santa Rosa (15)	11.52	0.61	0.47	0.66	0.47	0.16	0.11	0.18	0.11	1.67	1.00	1.36	0.82
Sunnyvale (16)	8.35	0.62	0.47	0.66	0.47	0.16	0.10	0.17	0.11	1.68	1.00	1.36	0.82
Other U.S. locations													
Fort Worth	6.38	0.63	0.49	0.69	0.50	0.18	0.11	0.20	0.12	1.69	1.03	1.40	0.86
San Antonio	7.07	0.62	0.49	0.69	0.50	0.18	0.11	0.19	0.12	1.68	1.02	1.39	0.85
Lake Charles LA	4.92	0.65	0.51	0.72	0.52	0.19	0.11	0.20	0.12	1.73	1.06	1.44	0.88
Miami	6.49	0.64	0.51	0.71	0.52	0.19	0.12	0.21	0.13	1.73	1.06	1.44	0.88
Phoenix	13.04	0.60	0.47	0.65	0.47	0.16	0.11	0.18	0.12	1.63	0.99	1.33	0.82
Atlanta	5.23	0.62	0.49	0.68	0.49	0.17	0.10	0.19	0.11	1.67	1.01	1.37	0.84
New York	6.53	0.62	0.48	0.67	0.48	0.17	0.10	0.18	0.11	1.68	1.01	1.37	0.83
Chicago	6.96	0.62	0.48	0.68	0.48	0.17	0.11	0.18	0.11	1.68	1.01	1.37	0.83
Chinese locations													
Beijing	6.96	0.62	0.48	0.67	0.48	0.17	0.10	0.18	0.11	1.68	1.01	1.37	0.83
Shanghai	4.91	0.64	0.51	0.70	0.51	0.19	0.11	0.20	0.11	1.72	1.05	1.41	0.87

*L_1/L_2 is the ratio of the primary and secondary air flow rates.

These "precooling indices," i.e., q_r, es_r, and w_r, are shown in Table 5. As in the case when outdoor air is used, the indices using room air as the secondary air are constant for different locations, except that the water use indices (w) are comparatively discrepant. The reason for this is that outdoor air is used as secondary air when the outdoor air enthalpy is less than that of room air and the fraction of outdoor air used varies from site to site. The average values of the indices for California cities are listed in Table 7.

Figures 5 and 6 show the curves of indices plotted against the flow rate ratios L_1/L_2. From the numbers in Tables 3 and 5, it also can be seen that the indices for non-California cities are close to the average values for California cities in the table above.

CONCLUSIONS

1. For California cities, using either outdoor air or exhaust room air as the secondary air provides nearly equal precooling capacity q (Tables 2 and 4). Slightly larger energy savings will be obtained if the secondary air is exhaust room air. But sometimes, especially for tube-type IECs, water consumption will also be larger, so there will be a trade-off between energy consumption and water consumption. In California, outdoor air is probably preferable as the secondary air for precooling ventilation air in nonresidential buildings because installation of the duct system will cost less. In humid locations like Miami, San Antonio, Lake Charles, Fort Worth, or Shanghai, using exhaust room air as the secondary air gives much greater precooling capacity and energy savings. Although water consumption will be much greater, exhaust room air should still be preferable to outdoor air because in such locations water consumption is usually relatively inexpensive.

2. The indices presented in this paper are useful because they are suitable for all of California and the only

TABLE 4
Precooling Capacities for Different Indirect Evaporative Designs with Room Air as Secondary Air

City	Bin hrs	Precooling capacity by IEC type and L_1/L_2* (kWh/h per m^3/s air) tube 1.25	tube 2.50	plate 2.37	plate 4.74	Energy savings by IEC type and L_1/L_2* (kWh/h per m^3/s air) tube 1.25	tube 2.50	plate 2.37	plate 4.74	Water use by IEC type and L_1/L_2* (kg/h per m^3/s air) tube 1.25	tube 2.50	plate 2.37	plate 4.74
California climate locations													
Arcata (1)	72	5.29	4.04	5.68	4.00	1.36	0.85	1.44	0.87	14.1	8.5	11.5	6.8
China Lake (2)	3515	9.11	7.03	9.97	7.11	2.36	1.64	2.57	1.69	23.9	14.4	18.6	11.5
El Centro (3)	5004	10.12	7.84	11.16	8.01	2.75	1.94	3.02	2.02	29.5	17.5	19.4	12.2
El Toro (4)	1433	6.06	4.65	6.57	4.64	1.61	1.05	1.73	1.07	25.3	14.2	12.5	7.6
Fresno (5)	3446	8.27	6.40	9.09	6.50	2.24	1.55	2.46	1.61	24.4	14.5	15.8	9.9
Long Beach (6)	1966	5.94	4.57	6.45	4.56	1.59	1.03	1.71	1.06	26.4	14.8	12.4	7.5
Mount Shasta (7)	1787	6.94	5.34	7.56	5.37	1.83	1.22	1.98	1.26	20.3	12.0	14.1	8.6
Oakland (8)	441	5.36	4.10	5.76	4.05	1.38	0.86	1.46	0.88	17.7	10.2	11.7	6.9
Pasadena (9)	2186	6.80	5.23	7.40	5.24	1.81	1.20	1.95	1.24	24.8	14.2	13.7	8.4
Red Bluff (10)	2500	8.02	6.19	8.78	6.25	2.13	1.45	2.32	1.50	24.9	14.6	16.0	9.9
Riverside (11)	3230	7.86	6.07	8.61	6.14	2.11	1.44	2.29	1.49	27.3	15.8	15.4	9.6
Sacramento (12)	2086	7.38	5.71	9.05	5.77	2.00	1.36	2.18	1.41	19.9	12.0	14.0	8.7
San Diego (13)	3128	5.26	4.03	5.69	4.01	1.41	0.89	1.50	0.91	29.3	16.0	11.2	6.7
Santa Maria (14)	382	6.19	4.73	6.67	4.70	1.58	1.02	1.69	1.04	18.9	11.1	13.5	8.1
Santa Rosa (15)	1570	7.39	5.70	8.08	5.75	1.98	1.30	2.15	1.38	21.2	12.6	14.6	9.0
Sunnyvale (16)	1617	5.47	4.20	5.92	4.18	1.45	0.92	1.55	0.95	21.6	12.2	11.5	6.9
Other U.S. locations													
Fort Worth	3658	7.20	5.25	7.40	5.11	2.14	1.37	2.18	1.36	25.0	14.1	12.5	7.7
San Antonio	4737	7.36	5.43	7.66	5.32	2.18	1.42	2.24	1.41	26.9	15.2	12.8	7.9
Lake Charles LA	4737	8.05	5.49	7.65	5.08	2.48	1.49	2.32	1.39	22.2	12.5	12.7	7.6
Miami	7735	8.04	5.69	8.02	5.57	2.46	1.54	2.42	1.53	19.1	11.2	13.0	8.0
Phoenix	4316	8.64	6.70	9.50	6.82	2.38	1.66	2.60	1.73	26.4	15.6	16.1	10.2
Atlanta	2514	6.31	4.50	6.30	4.27	1.85	1.13	1.82	1.08	22.2	12.4	11.3	6.8
New York	1702	5.91	4.37	6.13	4.28	1.67	1.04	1.71	1.04	20.8	11.8	11.4	6.9
Chicago	1666	6.16	4.60	6.46	4.53	1.75	1.11	1.81	1.13	20.3	11.7	11.7	7.1
Chinese locations													
Beijing	2574	7.50	5.37	7.52	5.07	2.19	1.37	2.16	1.30	29.4	16.3	13.1	7.8
Shanghai	3191	9.06	6.08	8.48	5.43	2.85	1.71	2.64	1.53	19.8	11.4	13.6	8.0

*L_1/L_2 is the ratio of the primary and secondary air flow rates.

variable is the flow rate ratio L_1/L_2, which is always given in advance for any kind of calculation. So, it is easy to use them to predict the total precooling capacity, energy savings, and water consumption during the cooling season. The method of prediction is as follows:

When outdoor air is used as the secondary air,

$$Q_o = q_o \times \Sigma hours \times \Delta t_m , \qquad (12)$$

$$ES_o = es_o \times \Sigma hours \times \Delta t_m , \qquad (13)$$

$$W_o = w_o \times \Sigma hours \times \Delta t_m , \qquad (14)$$

where

- Q_o = total precooling capacity, kWh;
- ES_o = total energy saving, kWh;
- W_o = total water consumption, kg;
- q_o, es_o, w_o = indices;
- $\Sigma hours$ = bin hours with outdoor air temperature ≥25°C; and
- Δt_m = average outdoor air wet-bulb temperature depression for the bins considered, °C.

When exhaust room air is used as the secondary air,

$$Q_r = q_r \times \Sigma hours \times (t_{dbm} - t_{wbr}) , \qquad (15)$$

$$ES_r = es_r \times \Sigma hours \times (t_{dbm} - t_{wbr}) , \qquad (16)$$

$$W_r = w_r \times \Sigma hours \times (t_{dbm} - t_{wbr}) , \qquad (17)$$

TABLE 5
Precooling Indices for Different Indirect Evaporative Designs with Room Air as Secondary Air

City	Avg. dry bulb temp. depres.	Precooling index by IEC type and L_1/L_2* (kWh/h°C per m³/s air)				Energy savings index by IEC type and L_1/L_2* (kWh/h°C per m³/s air)				Water use index by IEC type and L_1/L_2* (kg/h°C per m³/s air)			
		tube 1.25	tube 2.50	plate 2.37	plate 4.74	tube 1.25	tube 2.50	plate 2.37	plate 4.74	tube 1.25	tube 2.50	plate 2.37	plate 4.74
California climate locations													
Arcata (1)	25.28	0.71	0.54	0.76	0.54	0.18	0.11	0.19	0.12	1.9	1.1	1.5	0.9
China Lake (2)	30.80	0.70	0.54	0.77	0.55	0.18	0.13	0.20	0.13	1.8	1.1	1.4	0.9
El Centro (3)	32.72	0.68	0.53	0.75	0.54	0.19	0.13	0.20	0.14	2.0	1.2	1.3	0.8
El Toro (4)	26.77	0.68	0.52	0.74	0.52	0.18	0.12	0.19	0.12	2.8	1.6	1.4	0.9
Fresno (5)	30.41	0.66	0.51	0.72	0.52	0.18	0.12	0.20	0.13	1.9	1.1	1.3	0.8
Long Beach (6)	26.75	0.67	0.51	0.72	0.51	0.18	0.12	0.19	0.12	3.0	1.6	1.4	0.8
Mount Shasta (7)	27.99	0.68	0.52	0.74	0.53	0.18	0.12	0.19	0.12	2.0	1.2	1.4	0.8
Oakland (8)	25.41	0.71	0.54	0.76	0.53	0.18	0.11	0.19	0.12	2.3	1.4	1.5	0.9
Pasadena (9)	27.78	0.68	0.52	0.74	0.53	0.18	0.12	0.20	0.12	2.5	1.4	1.4	0.8
Red Bluff (10)	29.56	0.68	0.53	0.75	0.53	0.18	0.12	0.20	0.13	2.1	1.2	1.4	0.8
Riverside (11)	29.56	0.67	0.52	0.73	0.52	0.18	0.12	0.20	0.13	2.3	1.4	1.3	0.8
Sacramento (12)	29.13	0.65	0.51	0.72	0.51	0.18	0.12	0.19	0.13	1.8	1.1	1.2	0.8
San Diego (13)	25.78	0.66	0.51	0.72	0.51	0.18	0.11	0.19	0.11	3.7	2.0	1.4	0.9
Santa Maria (14)	26.35	0.73	0.56	0.78	0.55	0.19	0.12	0.20	0.12	2.2	1.3	1.6	0.9
Santa Rosa (15)	28.91	0.67	0.51	0.73	0.52	0.18	0.12	0.19	0.13	1.9	1.1	1.3	0.8
Sunnyvale (16)	26.08	0.66	0.51	0.72	0.51	0.18	0.11	0.19	0.12	2.6	1.5	1.4	0.8
Other U.S. locations													
Fort Worth	28.25	0.69	0.50	0.71	0.49	0.20	0.13	0.21	0.13	2.4	1.4	1.2	0.7
San Antonio	28.67	0.67	0.50	0.70	0.49	0.20	0.13	0.21	0.13	2.5	1.4	1.2	0.7
Lake Charles LA	28.06	0.79	0.54	0.75	0.50	0.24	0.15	0.23	0.14	2.2	1.2	1.2	0.7
Miami	29.26	0.70	0.50	0.70	0.49	0.22	0.14	0.21	0.13	1.7	1.0	1.1	0.7
Phoenix	31.11	0.65	0.50	0.71	0.51	0.18	0.12	0.19	0.13	2.0	1.2	1.2	0.8
Atlanta	26.69	0.70	0.50	0.70	0.48	0.21	0.13	0.20	0.12	2.5	1.4	1.3	0.8
New York	26.51	0.68	0.50	0.70	0.49	0.19	0.12	0.20	0.12	2.4	1.4	1.3	0.8
Chicago	27.08	0.66	0.50	0.69	0.49	0.19	0.12	0.19	0.12	2.2	1.3	1.3	0.8
Chinese locations													
Beijing	27.66	0.76	0.54	0.76	0.51	0.22	0.14	0.22	0.13	3.0	1.6	1.3	0.8
Shanghai	28.42	0.85	0.57	0.80	0.51	0.27	0.16	0.25	0.14	1.9	1.1	1.3	0.8

*L_1/L_2 is the ratio of the primary and secondary air flow rates.

where

Q_r	=	total precooling capacity, kWh;
ES_r	=	total energy saving, kWh;
W_r	=	total water consumption, kg;
q_r, es_r, w_r	=	indices;
t_{dbm}	=	average outdoor air dry-bulb temperature of the bins considered, °C;
t_{wbr}	=	room air wet-bulb temperature assumed to be constant over the cooling season, °C.

All the indices can be obtained according to the L_1/L_2 value from Figures 3 through 6. Of course, Σhours, Δtm, and t_{dbm} depend on the site, but these are not difficult to calculate if the bin distributions of outdoor air parameters are available.

3. Since the indices are constant for a particular flow rate ratio L_1/L_2, the more bin hours there are, the greater is the precooling effect. For cities with a short cooling season (such as Oakland, Arcata, Santa Maria, etc.), it may be not worthwhile to apply IEC as a first stage to precool incoming outdoor air. For cities with longer cooling seasons (such as El Centro, Fresno, Riverside, China Lake, etc.), precooling has a very good energy-saving effect.

4. The reduction of peak electric demand by precooling is apparent. Table 8 shows some typical numbers for a typical tube-type IEC with outdoor air used as the secondary air: If the secondary air is exhaust room air, the peak demand reduction for cities with comparatively long cooling seasons lies in the range of 3.5-5.5 kW.

TABLE 6

Type	L_1/L_2	q_o	es_o	w_o	Type	L_1/L_2	q_o	es_o	w_o
Tube	2.50	0.47	0.105	1.00	Plate	5.92	0.41	0.08	0.69
	1.25	0.61	0.16	1.67		4.74	0.47	0.11	0.82
	0.625	0.74	0.20	2.88		2.37	0.66	0.17	1.35
						1.00	0.86	0.23	2.53

TABLE 7

Type	L_1/L_2	q_r	es_r	w_r	Type	L_1/L_2	q_r	es_r	w_r
Tube	2.50	0.52	0.12	1.34	Plate	5.92	0.46	0.10	0.71
	1.25	0.68	0.18	2.31		4.74	0.53	0.12	0.84
	0.625	0.83	0.23	4.14		2.37	0.74	0.20	1.37
						1.00	0.97	0.26	2.50

TABLE 8

City	L_1/L_2	Peak Demand Reduction, kW, per 1 m^2/s of primary air
El Centro	2.5	3.0
	1.25	3.8
Oakland	2.5	1.2
	1.25	1.7
Pasadena	2.5	2.2
	1.25	3.0
Sacramento	2.5	3.2
	1.25	4.0
Sunnyvale	2.5	3.2
	1.25	4.2

5. The indices presented are for cooling systems with continuous operation. Further work is needed to determine whether the indices are suitable for cooling systems with intermittent operation.

REFERENCES

Burns, P.R., J.W. Mitchell, and W.A. Beckman. 1985. Hybrid desiccant cooling systems in supermarket applications. *ASHRAE Transactions* 91(1).

Chen, P.L., H.M. Qin, Y.J. Huang, and H.F. Wu. 1991. A heat and mass transfer model for thermal and hydraulic calculations of indirect evaporative cooler performance. Lawrence Berkeley Laboratory Report 28201. *ASHRAE Transactions* 97(2).

Manley, D.L., K.L. Bowlen, and B.M. Cohen. 1985. Evaluation of gas-fired desiccant-based space conditioning for supermarkets. *ASHRAE Transactions* 91(1).

Matsuki, K., M. Tatsuoka, and T. Tonomura 1983. A prototype solar desiccant air-conditioner. *Proceedings of the 18th Intersociety Energy Conservation Engineering Conference*, vol 4.

Peterson, J.L., and B.D. Hunn. 1985. The use of indirect evaporative cooling to reduce peak electric demand in new office buildings. *ASHRAE Transactions* 91(1B).

Supple, R.G., and D.R. Broughton. 1985. Indirect evaporative cooling — mechanical cooling design. *ASHRAE Transactions* 91(1B).

Turner, R.H., and F.C. Chen. 1987. Research requirements in the evaporative cooling field. *ASHRAE Transactions* 93(1).

Warren, M. 1985. *Conventional cooling systems*. Unpublished. Lawrence Berkeley Laboratory report.

Watt, J.R. 1986. *Evaporative air conditioning handbook*, 2nd ed. Chapman and Hall.

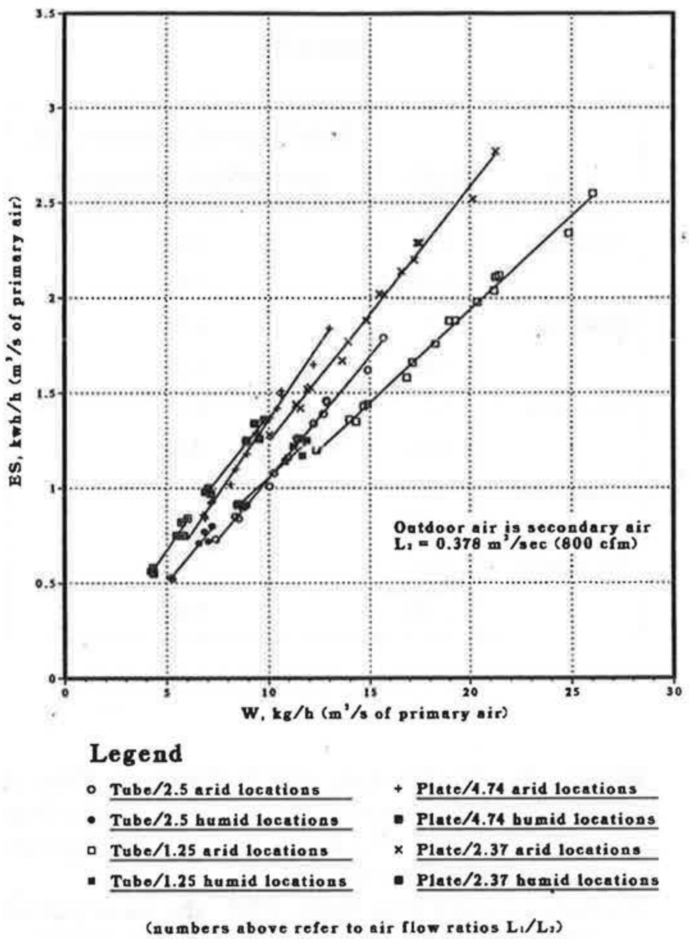

Figure 2 *Relationship between water consumption (W) and energy savings (ES).*

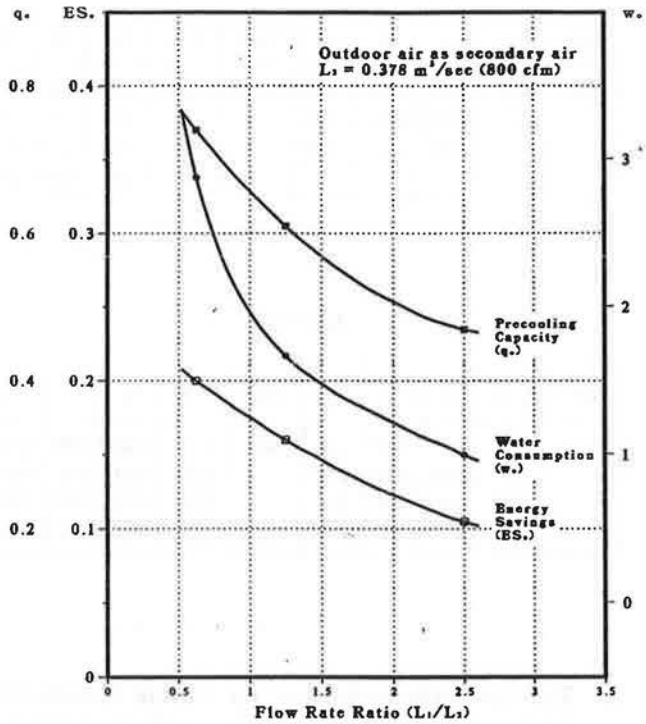

Figure 3 *Curves of indices vs. flowrate ratio (L_1/L_2) for tybe type evaporative cooler with outdoor air as secondary air.*

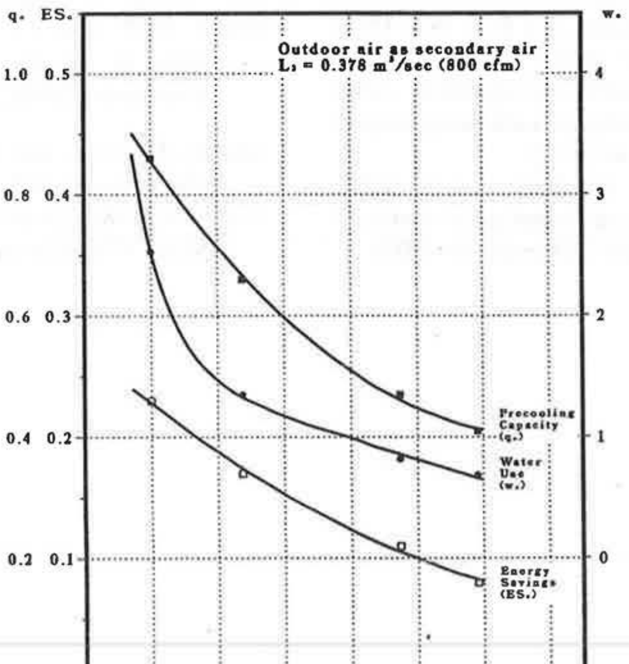

Figure 4 *Curves of indices vs. flowrate ratio (L_1/L_2) for plate type evaporative cooler with outdoor air as secondary air.*

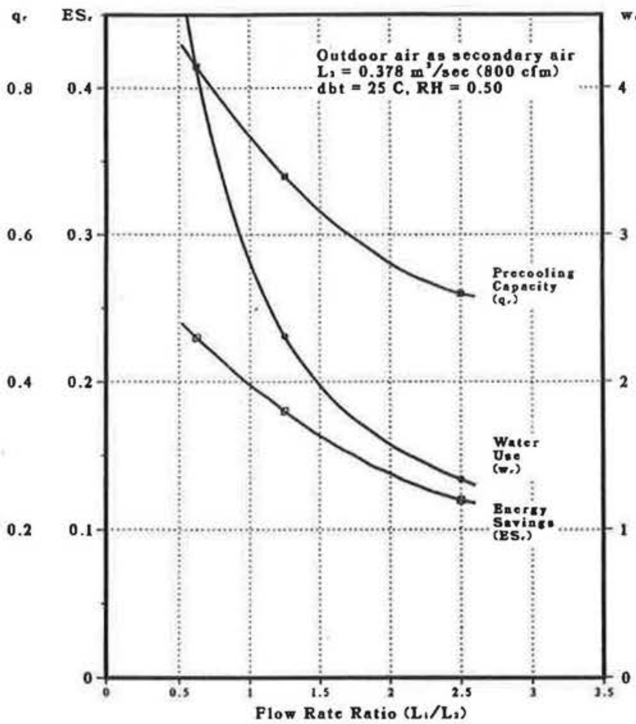

Figure 5 *Curves of indices vs. flowrate ratio (L_1/L_2) for tube type evaporative cooler with room air as secondary air.*

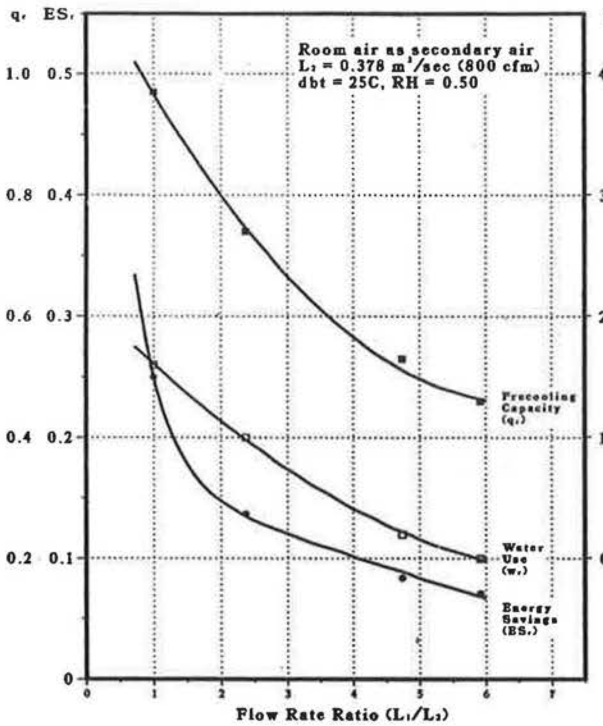

Figure 6 *Curves of indices vs. flowrate ratio (L_1/L_2) for plate type evaporative cooler with room air as secondary air.*