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ASSESSMENT OF THE RADIATIVE COOLING POTENTIAL OF A COLLECTOR USING HOURLY WEATHER DATA

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Abstract—The radiative cooling potential is determined by the ambient temperature, relative humidity, wind velocity, and cloudiness. Previous assessments of the radiative cooling potential in Greece have been based on mean weather data. In this paper, 12 years of hourly weather data are used to assess the radiative cooling potential in Athens. The performance results for a simple radiator are also presented. The radiative cooling potential for Athens is promising and simple radiators can be used to estimate the cooling potential.

1. INTRODUCTION

Energy consumption for cooling in buildings has been increasing constantly during the last decade, especially in Southern European countries where hot summer conditions require cooling from May through September. Cooling requirements have been primarily satisfied by using active air-conditioning systems (mainly packaged units). In Greece, imports of conventional air-conditioning systems have increased by 900% from 1987 to 1990.[1] The need for energy economy and ecological constraints have oriented research toward passive and hybrid cooling techniques, which have been under investigation in the U.S. and in Israel for more than 15 years. In Europe, similar research has only recently been initiated.[2,3] Radiative cooling is a natural heat-dissipation technique which may be mechanically assisted by using a fan.[4] Parts of the building envelope or other appropriate devices are cooled by emitting infrared radiation to the sky which acts as a low-temperature heat sink.

The potential for radiative cooling for the northern part of the Mediterranean basin, using mean monthly weather data, has been presented in our previous paper.[5] In this study, we calculated the radiative cooling potential in Athens using measured hourly data over 12 years. This method allows a detailed evaluation of the feasibility of radiative cooling since frequency distributions of parameters characterising the radiative cooling potential can be obtained.

2. MODEL PRESENTATION

2.1. *Sky emissivity*

Every object on the surface of the Earth exchanges thermal radiation with surrounding objects and with the atmosphere. The atmosphere emits thermal radiation with a spectral distribution close to that of a blackbody at a temperature equal to the dry bulb temperature of the air close to the ground, except in the spectral region 8–13 μm. The main component of a radiative cooling system is the radiating surface. The determinant parameter in evaluating the performance of a radiator is the sky-temperature depression, which is defined as the difference of the ambient air temperature and the "sky temperature" (i.e. the temperature of the blackbody having the same spectral distribution as the sky). The sky-temperature depression is

$$\mathrm{DT}_{\mathrm{sky}} = (1 - \varepsilon_{\mathrm{sky}}^{1/4})\, T_{\mathrm{a}}\,, \tag{1}$$

where T_a is the ambient temperature and ε_{sky} is the sky emissivity.

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Many correlations are reported in the literature for calculating the sky emissivity. In this paper, the Berdahl and Martin relation[6]

$$\varepsilon_{cs} = 0.711 + 0.56(T_{dp}/100) + 0.73(T_{dp}/100)^2 \tag{2}$$

is used, with the dew point given by[7]

$$T_{dp} = C_3[\ln(\mathrm{RH}) + C_1]/\{C_2 - [\ln(\mathrm{RH}) + C_1]\}\,, \tag{3}$$

$C_1 = C_2T_{dry}/(C_3 + T_{dry})$, $C_2 = 17.08085$, $C_3 = 234.175$, T_{dry} = ambient dry-bulb temperature (°C), and RH = relative humidity. With account for diurnal variation,[6] the instantaneous clear sky emissivity is

$$\Delta\varepsilon_d = 0.013\cos\{2\pi t/24\}\,, \tag{4}$$

where t = hour of the day. Equations (2) and (4) are valid for clear-sky conditions. Under cloudy skies, the sky emissivity is[6]

$$\varepsilon_S = \varepsilon_{CS}(1 + 0.0224n - 0.0035n^2 + 0.00028n^3)\,, \tag{5}$$

with n = total opaque cloud amount, which is 0 for a clear sky and 1 for an overcast sky.

2.2. *Radiator performance*

The operation of a simple horizontal, open loop, flat-plate air cooler exposed to the atmosphere at night was simulated. The cooler was a 2 m long rectangular air duct with a 1 m by 0.20 m flow section. The radiator was a 0.003 m thick stainless-steel plate with an emittance of 0.90 in the i.r. bandwidth. It was assumed that the cooler functioned only at night. The air velocity through the radiator was 2.5 m/sec. Simulations of this system covered with a wind screen were also carried out. The useful cooling energy and the air-outlet temperature provided by the radiator are calculated according to the methodology of Ito and Miura.[8]

2.3. *Cooling power of a radiator*

The net heat flux of a non-selective radiator at T_r is a linear function of an effective heat-transfer coefficient and a minimum threshold or stagnation temperature,

$$q_r = h_e(T_r - T_{th}) \tag{6}$$

with $h_e = h + 4\,\varepsilon_r\sigma\,T_a^3$ = effective heat-transfer coefficient, ε_r = emissivity of the radiator, $T_{th} = T_a - q_o/h_e$ = minimum temperature attained by the radiator, and h = convective heat-transfer coefficient due to the wind. This coefficient is[9]

$$h = 5.7 + 3.8V\,,\quad V < 4\ \mathrm{m/sec}\,, \tag{7}$$

$$h = 7.3V^{0.8}\,,\quad V > 4\ \mathrm{m/sec}\,, \tag{8}$$

for a radiator without wind screen and

$$h = 0.5 + 1.2\,V^{0.5} \tag{9}$$

for a radiator with wind screen. The net radiative power of a blackbody at the ambient temperature is

$$q_o = \sigma T_a^4 - q_s \tag{10}$$

with $q_s = \sigma\,\varepsilon_{sky}\,T_a^4$.

2.4. *Fluid temperature*

The air temperature at the outlet of the radiator is[8,10]

$$T_{fo} = T_{th} + (T_{fi} - T_{th}) \exp(- U_p A / m c_p) , \tag{11}$$

with T_{fi} = inlet temperature, A = radiator surface and

$$U_p = 1/[(1/h_e) + (1/h_f) + \delta/k_p] ,$$

where h_f = heat-transfer coefficient between the radiator and the air circulating in the duct, δ = thickness of the radiator plate and k_p = thermal conductivity of the radiator plate. Equation (11) is used to calculate the outlet temperature if the minimum threshold temperature T_{th} is known.

3. RESULTS

Our model has been used to calculate the sky-temperature depression and the outlet-air temperature for a flat-plate radiative air cooler using hourly values of the following weather data: ambient temperature, relative humidity, wind velocity, and opaque cloud cover provided by the National Observatory of Athens[11] (lat 37.58°N, long 23.43°E and altitude 107 m). The Observatory has been recording temperature data since 1857. An analysis of multiyear temperature data (1857–1990) indicates that no temperature trends can be defined.[12] We used data from 1977 to 1989. It should be noted that the opaque cloud cover was measured for the following hours (LST): 08:00, 11:00, 14:00, 17:00, and 20:00. Since the cooler operates at night when the opaque cloud cover is not estimated, this value has been assumed to be equal to the last observation of the day. Simulations have been performed for clear-sky (optimal) conditions and average sky conditions from June to September.

3.1. *Sky-temperature depression*

The histograms in Figs. 1 and 2 show the frequencies of sky-temperature depressions in Athens for the two conditions used. As explained in Sec. 2, this is the difference between the ambient air temperature and the "sky temperature". Figures 1 and 2 show that the frequency is a maximum around 16–18°C throughout the cooling season and for both weather conditions. Slightly higher frequencies occur in June because cloudiness and ambient relative humidity in Athens do not vary significantly throughout the cooling season. These values are 1–2° lower than the sky-temperature depression calculated using mean weather data.[5]

3.2. *Stagnation temperature*

While the sky-temperature depression quantifies the potential for radiative cooling as a function of weather data at a particular site, the stagnation temperature takes into account not only weather data but also the characteristics of the radiator. The stagnation temperature is the lowest temperature attainable by a radiator for specific weather and operational parameters and indicates the cooling efficiency. The stagnation temperature has been calculated for both average and optimal weather conditions. For both cases, the effect of the wind screen has also been determined.

The cumulative frequency distribution of the stagnation temperature is illustrated in Figs. 3 and 4. From these figures, the following may be observed: (i) the lowest stagnation temperatures and highest cooling potentials occur during September and June. The stagnation temperature is significantly higher in August. The highest values occur during July for both optimal and average conditions and for both covered and uncovered radiators. These findings are in accord with monthly variations of sky-temperature depressions during July and August, but not during June and September. The highest sky-temperature depression occurs in June and the lowest stagnation temperature occurs in September. These results show that calculations of sky-temperature depressions alone may not be sufficient for performance evaluations of a radiative cooling system. (ii) The contribution of the wind screen to reduction of the stagnation temperature seems to be significant. For optimal weather conditions, the stagnation temperature of the unprotected radiator ranges from 22°C in September to 27°C in July for 80% of the time. For the wind-screened radiators, it ranges from 19°C in September to 23°C in July. The stagnation-

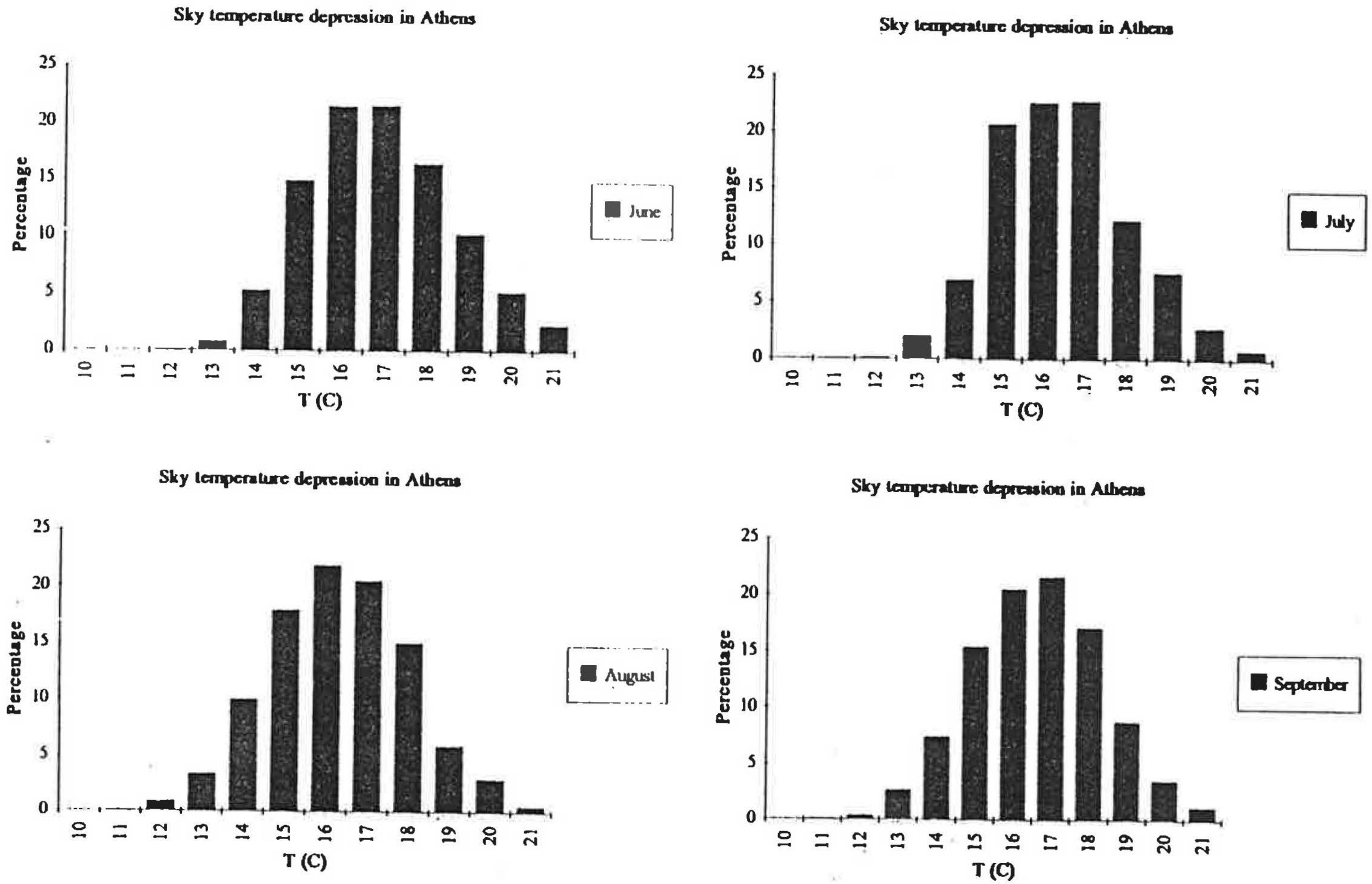

Fig. 1. Sky-temperature depression frequencies for optimal weather conditions.

depression frequencies for optimal weather conditions.

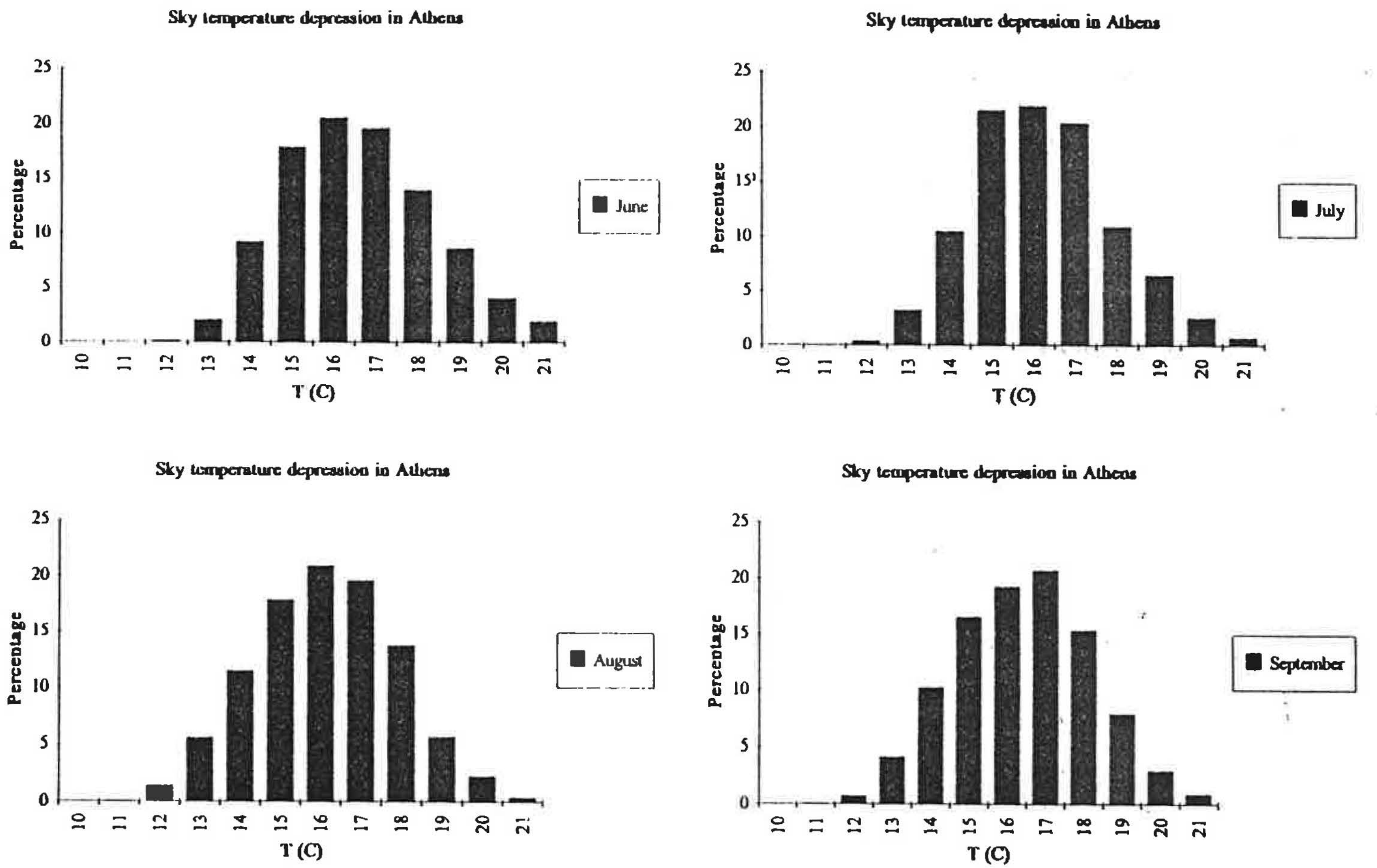

Fig. 2. Sky-temperature depression frequencies for average weather conditions.

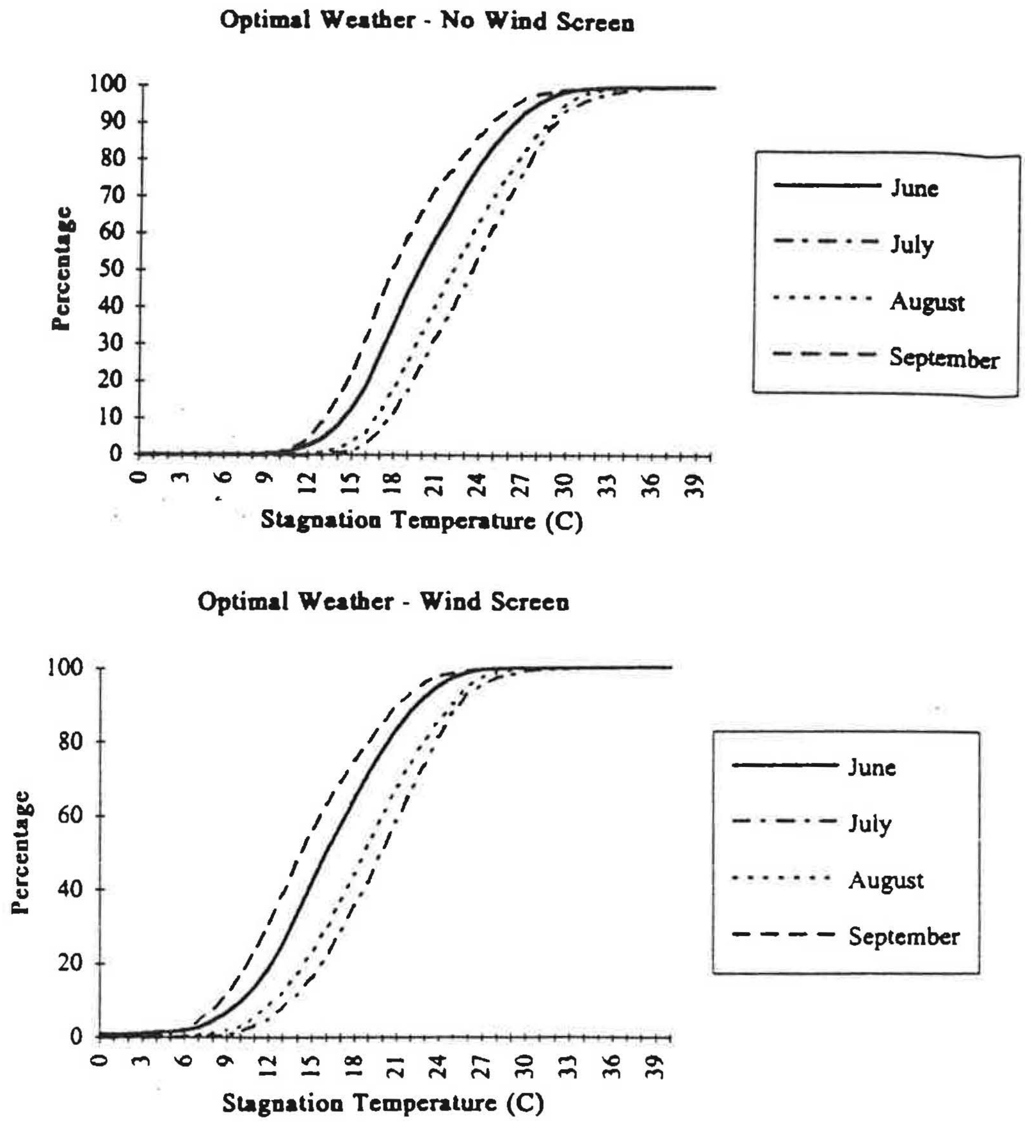

Fig. 3. Cumulative frequency distribution of the stagnation temperature without and with wind screen for optimal weather conditions.

temperature difference for average weather conditions is of the same magnitude. (iii) There is no significant difference in the stagnation temperatures between average and optimal weather conditions, which means that the probability of observing poor radiator performance due to cloudiness is rather small.

3.3. *Outlet temperature*

The outlet temperature has been calculated assuming that the temperature at the inlet of the radiator equals the ambient temperature. The cumulative frequency distributions for the outlet-air and ambient temperatures are illustrated in Figs. 5 and 6. For each data set results with and without windscreen are indicated. For both average- and clear-weather conditions, the average temperature drop is about 3°C because cloudiness in Greece during the cooling period is very small. As previously discussed, a stagnation-temperature difference of about 4°C occurs when a wind screen is used. But this difference becomes about 1°C when the temperature at the outlet of the radiator is calculated. These results confirm that the use of a wind screen for Athens should not be recommended because of its high cost and short life.[4]

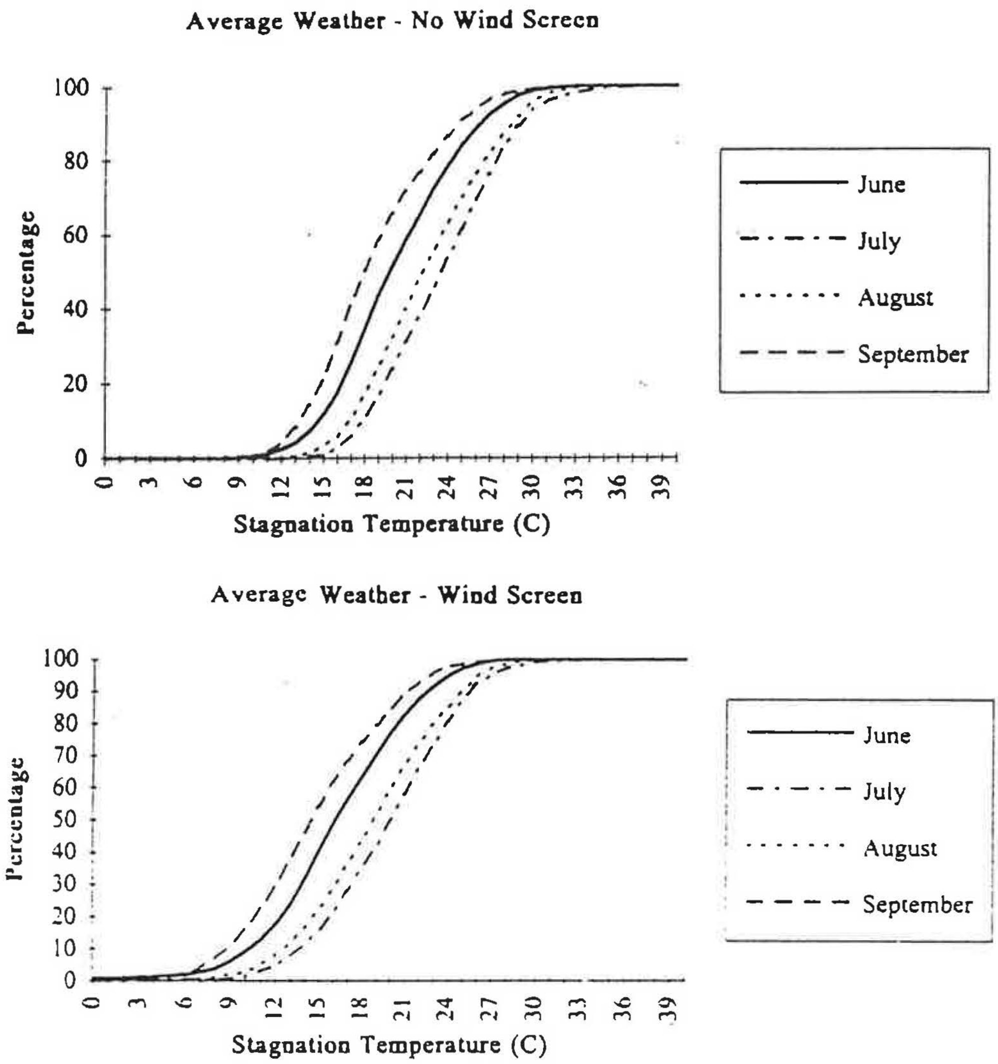

Fig. 4. Cumulative frequency distribution of the stagnation temperature without and with wind screen for average weather conditions.

4. CONCLUSIONS

The performance of a horizontal radiator, using hourly weather data at Athens for 12 years, shows that radiative cooling is useful in this location. The radiative cooling potential is characterised by the sky-temperature depression. The frequency distribution of the sky-temperature depression has a maximum around 16–17°C for every month of the cooling season. This value is close to values obtained for other areas with important radiative cooling potential.[6] Comparisons with previous assessments[4] and with mean weather data show that the use of hourly data predicts higher than actual values of sky-temperature depressions. Therefore performance evaluation of natural cooling techniques should be based on detailed climatic data.

The combined effects of climate and radiator characteristics on cooling potential are represented by the stagnation temperature. Calculations show that the stagnation temperature becomes a maximum in July and a minimum in September. It is significantly lower than the ambient temperature even during July and August, which are the warmest months in Greece.

The outlet-air temperature obtained, assuming that the air velocity under the surface of the radiator is 2.5 m/sec, shows that a wind screen has no significant effect and therefore the use of such devices is not recommended in Athens. The outlet temperature shows no significant differences for average and optimal weather conditions because cloudiness in Greece during the cooling season is rather low. In summary, our analysis has shown that the radiative cooling potential for Athens is useful.

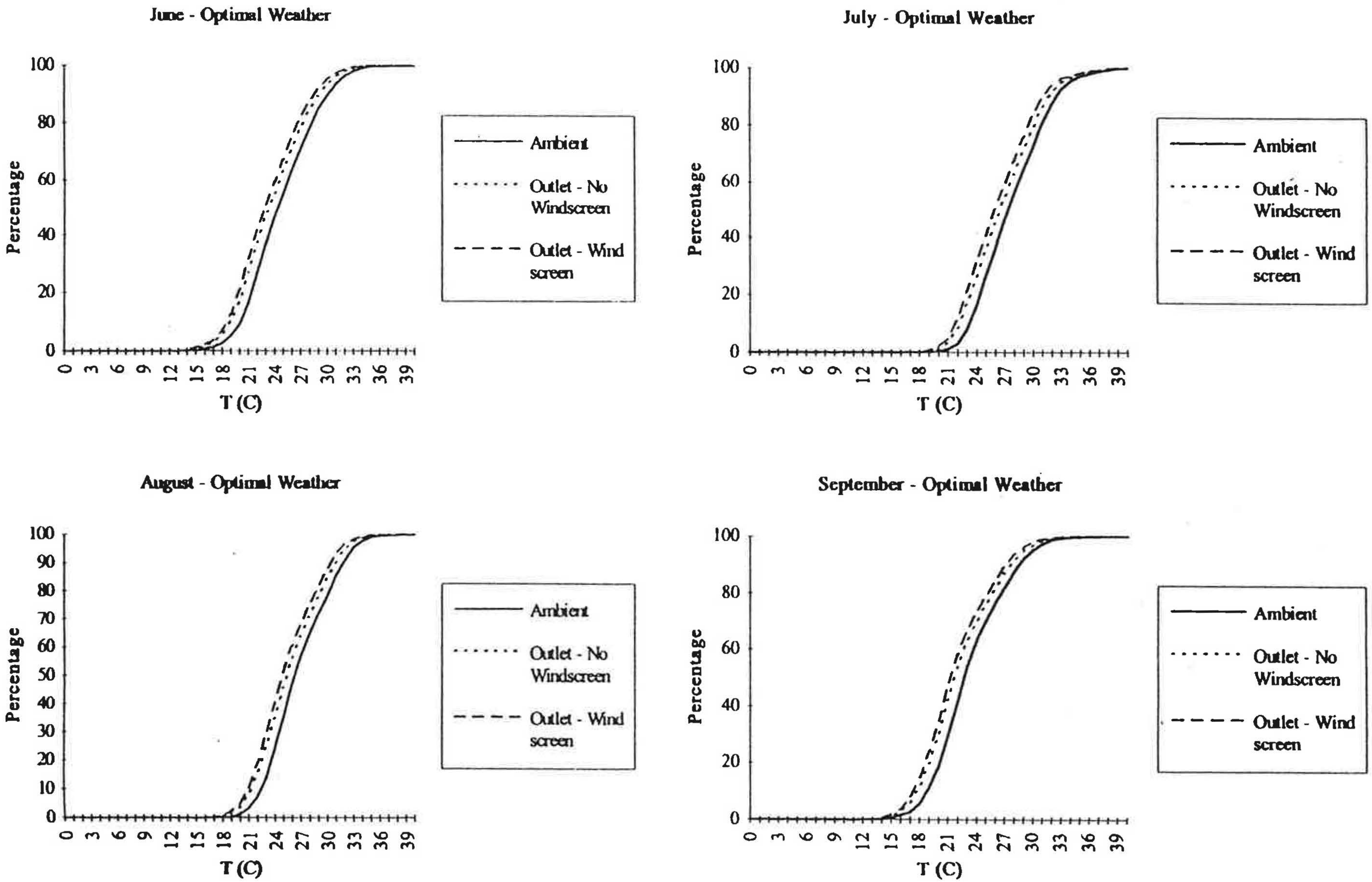

Fig. 5. Cumulative frequency distribution of the ambient and outlet-air temperatures with and without wind screen for optimal weather conditions.

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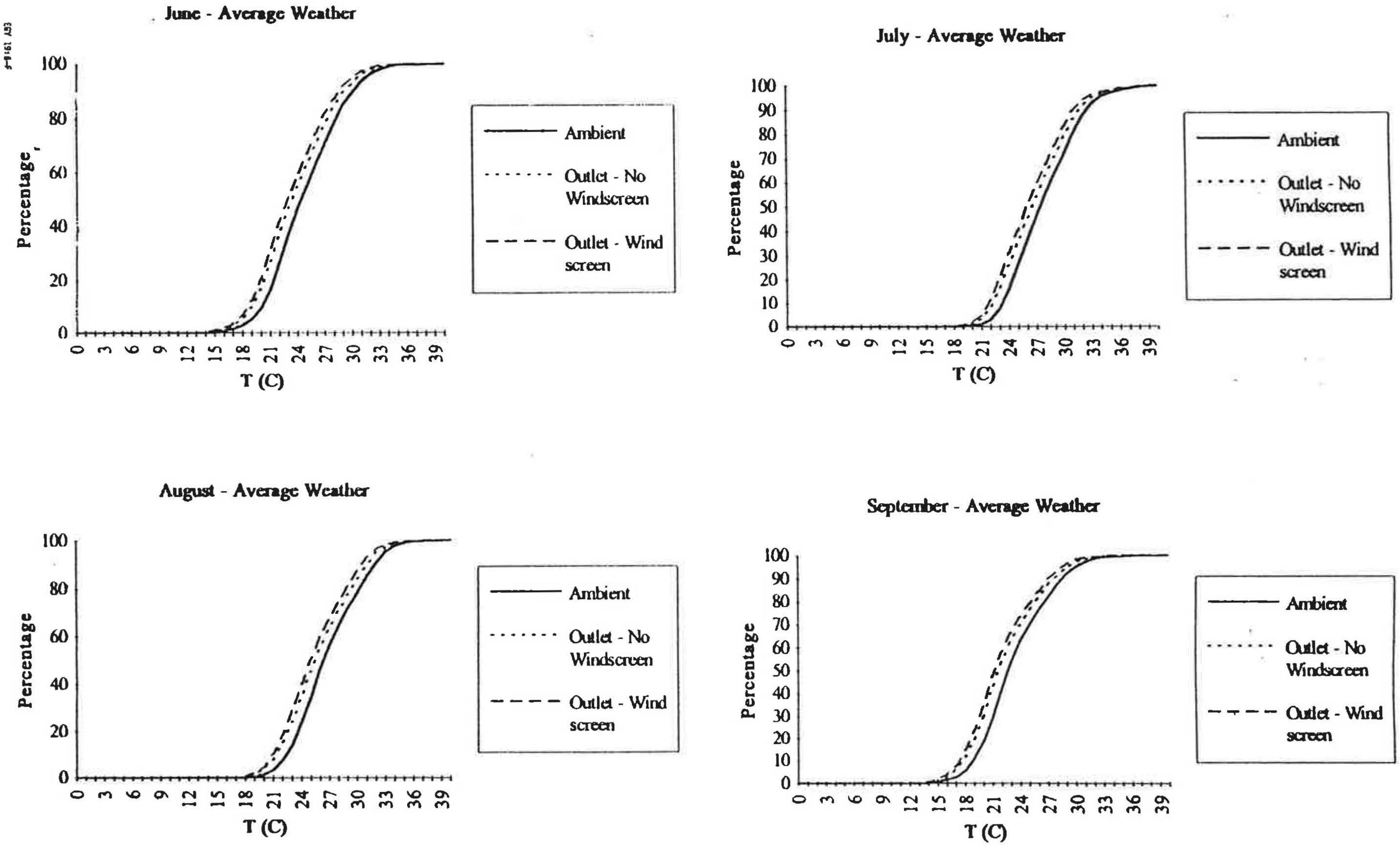

Fig. 6. Cumulative frequency distribution of the ambient and outlet-air temperatures with and without wind screen for average weather conditions.

REFERENCES

1. M. Santamouris, *Proc. of the Conf. on Passive Cooling*, C.E.C., DG XII, Ispra (1990).
2. T. C. Steemers, *Int. J. Sol. Energy* **10,** 5 (1991).
3. M. Santamouris and A. Argiriou, *Proc. of the 3rd European Conf. on Architecture*, C.E.C. DG XII, Florence (1994).
4. A. Argiriou, M. Santamouris, C. Balaras, and S. Jeter, *Int. J. Sol. Energy*, in press (1994).
5. M. Santamouris (ed.), *Horizontal Study on Passive Cooling*, C.E.C., Brussels (1990).
6. J. Cook (ed.), *Passive Cooling*, M.I.T. Press, Cambridge, MA (1989).
7. *Aspirations—Psychrometer Tafel*, Deutscher Wetterdienst, 5 Auflage, Vieweg, Braunschweig (1976).
8. S. Ito and N. Miura, *J. Sol. Energy Engng* **111,** 251 (1989).
9. M. Mostrel and B. Givoni, *Passive Sol. J.* **1,** 229 (1982).
10. J. A. Duffie and W. A. Beckman, *Solar Engineering of Thermal Processes*, Wiley, New York, NY (1980).
11. National Observatory of Athens, Climatological Bulletin, Athens, Hellas (1977–1989).
12. I. Tselepidaki, M. Santamouris, and D. Melitsiotis, *Sol. Energy* **50,** 197 (1993).

NOMENCLATURE

A = Radiator surface
$C_1 = (C_2 T_{dry})/(C_3 + T_{dry})$
C_2 = Constant = 17.08085
C_3 = Constant = 234.175
c_p = Specific heat at constant pressure
DT_{sky} = Sky-temperature depression
h = Convective heat-transfer coefficient
h_e = Effective heat-transfer coefficient
k_p = Thermal conductivity of the radiator plate
m = Mass-flow rate
n = Total opaque cloud amount
q_o = Net radiative power of a blackbody at T_a
q_r = Net heat flux of a non-selective radiator at T_r
q_s = Sky irradiance
RH = Relative humidity
t = Hour of day
T_{ad} = Adiabatic temperature
T_{dp} = Dew-point temperature
T_{dry} = Dry-bulb ambient temperature
U_p = Overall heat-transfer coefficient
V = Wind speed

Greek characters

$\Delta\varepsilon_d$ = Instantaneous clear-sky emissivity
ε_{cs} = Clear-sky emissivity
ε_s = Cloudy-sky emissivity
ε_{sky} = Sky emissivity
σ = Stefan–Boltzmann constant

Subscripts

a = Ambient
f = Fluid
i = Inlet condition
o = Outlet condition
r = Radiator
s = Sky