

## POTENTIAL OF RADIATIVE COOLING IN SOUTHERN EUROPE

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Research on passive and low energy architecture has been recently oriented towards passive techniques in order to satisfy the cooling needs of buildings. One of the techniques which has been considered is radiative cooling. The potential of radiative cooling mainly in the United States has already been evaluated. However, a similar attempt has not been made for the southern European countries, where the weather in summer is very hot and passive cooling applications could make a significant contribution to the achievement of thermal comfort in buildings.

This paper investigates the feasibility of applying radiative cooling techniques in southern European countries by presenting the results of the calculations of the sky temperature depression and of the performance of a typical radiative flat plate air cooler. These calculations were based on mean monthly weather data available for 28 southern European cities, covering a range of latitudes between 34° and 46°. Available data from some southeastern U.S. cities, have also been used. This allowed for a comparative study on the performance of radiative cooling systems between southern Europe and the southeastern United States. The results have shown that radiative cooling could be applied successfully in most south European locations.

KEY WORDS: Passive cooling, Radiative cooling.

### INTRODUCTION

Energy consumption for cooling in buildings has been increasing constantly during the last decade. Especially in southern European countries where the hot summer conditions extend the cooling period from May through September, the problem of achieving thermal comfort in buildings is of great importance. Actually, the cooling requirements have been primarily satisfied by active systems and the use of air conditioning systems (split units) have become highly popular. In Greece, for example, the imports of conventional air conditioning systems have increased by 900% from 1987 to 1990.

Energy economy and ecological constraints, however, have oriented the research in passive and low energy architecture towards the alternative solutions offered by passive and hybrid cooling techniques. These methods have been under investigation in the United States and in Israel for more than fifteen years. However, in Europe similar research has only been recently initiated with some research and development programs currently underway.<sup>1</sup>



The potential of radiative cooling has been evaluated for the United States and information is given in [3]. However, until now there have not been any extensive studies of radiative cooling for the southern European countries, other than a first estimate.<sup>4</sup>

This paper aims to satisfy the lack of information on the potential of radiative cooling around the northern part of the Mediterranean basin. The calculated data include the sky temperature depression, which is the parameter for determining the feasibility of radiative cooling systems and the mean daily useful cooling energy for a simple flat plate radiative air cooler. The cooled air can then be used to satisfy part of a building's cooling needs.

These calculations have been based on meteorological data from 28 southern European locations,<sup>5</sup> listed in Table 1. The efficiency of this radiative cooling technique in southern Europe is compared with the efficiency that it could have in some locations of the southeastern United States. For this reason the performance of the same flat plate air cooler has been evaluated using climatic data<sup>6</sup> for Atlanta (GA), Miami (FL), Charleston (NC) and Raleigh (SC).

## MODEL PRESENTATION

### *Sky Temperature Depression*

The atmosphere emits thermal radiation, except in the spectral region 8–13  $\mu\text{m}$ , with a spectral distribution very close to the one of a blackbody at a temperature equal to the dry bulb temperature of the air close to the ground. The thermal emission of the atmosphere is mainly due to the vibrational and rotational transitions of the asymmetrical molecules from which the earth's atmosphere is composed. These molecules are mainly water vapour, carbon dioxide, and ozone. The symmetrical molecules,  $\text{O}_2$  and  $\text{N}_2$ , which compose 99% of the atmosphere, are transparent to infrared radiation (beyond 3  $\mu\text{m}$ ).<sup>7</sup> Water vapour and carbon dioxide have a few transitions in the spectral region of 8–13  $\mu\text{m}$ . Consequently, for all practical reasons, the atmosphere can be considered transparent in this spectral region, and is usually called "atmospheric window".

The influence of the various atmospheric components to the thermal radiation of the atmosphere is as follows:<sup>7</sup>

- More than 90% of the total emitted radiation comes from the first 5 km in altitude. The contribution of each constituent to the total flux is 95.7% for  $\text{H}_2\text{O}$  + continuum, 2.8% for  $\text{CO}_2$  (including  $\text{CH}_4$  and  $\text{N}_2\text{O}$ ), and 1.5% for  $\text{O}_3$ .
- Ozone has a nearly constant peak of emission at 9.6  $\mu\text{m}$ , (near the centre of the atmospheric window), as it comes from the absorption in the stratosphere where its concentration is predominant.
- The atmospheric window is limited at about 14  $\mu\text{m}$  because of the emission of  $\text{CO}_2$ . The carbon dioxide concentration is practically constant and no significant variation of the emitted thermal energy has been observed from its variation, because the emission spectrum of  $\text{CO}_2$  is superimposed to the emission spectrum of water vapour.
- The contribution of all other elements of the atmosphere is very small.

If an object on the earth's surface emits thermal radiation within the range of

the atmospheric window, assuming that the atmospheric conditions are such that the atmospheric window is "open" (i.e. low relative humidity and clear sky), then its temperature decreases. A radiator performs better under clear sky conditions than under partly cloudy or average sky conditions. This was only to be expected since under clear sky conditions, the sky thermal radiation is low, enabling the radiator to emit more energy towards the low temperature heat sink than under average sky conditions. Increased amounts of clouds absorb and reemit the infrared radiation. As a result, it slows the rate of radiative cooling from the plate collector to the night sky.

The determinant parameter in evaluating the performance of a radiative cooling system is the sky temperature depression. This is defined as the difference of the ambient air temperature minus the "sky temperature" (i.e. the temperature of the blackbody having the same spectral distribution as the sky).

The sky temperature depression ( $DT_{sky}$ ) is calculated by the following relation:

$$DT_{sky} = (1 - \epsilon_{sky}^{1/4})T_a \quad (1)$$

where  $\epsilon_{sky}$  = sky emissivity, and  $T_a$  = ambient temperature.

Many correlations are reported in the literature for calculating the sky emissivity. As it has been explained already, the major part of the thermal radiation that the sky emits is due to the water vapour; for this reason, an expression of the sky emissivity as a function of a parameter related to the water content is required. In this paper, the Berdahl and Martin relationship has been used for the clear sky emissivity:<sup>8</sup>

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

where  $T_{dp}$  = dew point temperature, defined as:<sup>9</sup>

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

where  $C_1 = C_2 T_{dry}/(C_3 + T_{dry})$ ,  $C_2 = 17.08085$ ,  $C_3 = 234.175$ ,  $T_{dry}$  = ambient dry bulb temperature ( $^{\circ}\text{C}$ ), and  $RH$  = relative humidity  $0 \leq RH \leq 1$ .

The calculations for the instantaneous clear sky emissivities were estimated using the following expression which takes into account the diurnal variation:<sup>8</sup>

$$\Delta \epsilon_d = 0.013 \cos\{2\pi t/24\} \quad (4)$$

where  $t$  = hour of the day.

The values of sky emissivity obtained by equations (2) and (4) are clear sky emissivities. Under cloudy skies the sky emissivity ( $\epsilon_s$ ) can be calculated by the following relationship:<sup>8</sup>

$$\epsilon_s = \epsilon_{cs}(1 + 0.0224n - 0.0035n^2 + 0.00028n^3) \quad (5)$$

where  $n$  = total opaque cloud amount ( $0 \leq n \leq 1$ )

### Radiator Performance

The simulation of the operation of a typical flat plate air cooler was performed for an open loop radiative cooling system with an uncovered air collector whose surface is exposed to the atmosphere at night and cools the air that circulates through the system. The air cooler was assumed to be a horizontal 2 m long rectangular air duct. The dimensions of the flow section were 1 m by 0.20 m. The

radiator was considered to be a 0.1 of 0.90 in the I.R. bandwidth. It only during the night time with 2.5 m/sec. The radiator was assumed that sky radiation is less in the Simulations have been also carried screen in order to minimize conve

The useful cooling energy and the radiator are calculated based on the

### Cooling power of a radiator

The net heat flux ( $q_r$ ) of a nonselect as a linear function of an effective threshold temperature ( $T_{th}$ ), as fol

where  $h_e$  = effective heat transfe threshold temperature =  $T_s - \epsilon q_o/l$

The minimum threshold tempe the radiator. The convective heat velocity ( $V$ ) and it is calculated by

Radiator with no wind screen:

$$h = 5.7 \cdot$$

$$h = 7.3V$$

Radiator with wind screen:

$$h$$

The net radiative power of a bla given by:

where  $q_s$  = sky irradiance =  $\epsilon_{sky}\sigma T$

### Fluid temperature

The problem of calculating the through a one-dimensional path in in the same way as the case of a given by:

$$T_{fo} - T_{th} =$$

where  $T_{in}$  = inlet temperature of of the heat transfer fluid,  $U_p$  = c rate,  $c_p$  = specific heat at constant

Equation (11) calculates the ou that the minimum threshold temp temperature dependence of the t transfer fluid have also been take

The method used for calcula temperature of the air provided

radiator was considered to be a 0.003 m stainless steel plate, having an emittance of 0.90 in the I.R. bandwidth. It was assumed that the cooler was functioning only during the night time with an air velocity through the radiator set at 2.5 m/sec. The radiator was assumed to be horizontal, because it has been shown that sky radiation is less in the region of zenith than near the horizon.<sup>8</sup> Simulations have been also carried out for the same system covered with a wind screen in order to minimize convective losses.

The useful cooling energy and the outlet temperature of the air provided by the radiator are calculated based on the work by Ito and Miura:<sup>10</sup>

#### *Cooling power of a radiator*

The net heat flux ( $q_r$ ) of a nonselective radiator at temperature ( $T_r$ ) is calculated as a linear function of an effective heat transfer coefficient ( $h_e$ ) and a minimum threshold temperature ( $T_{th}$ ), as follows:

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

where  $h_e$  = effective heat transfer coefficient =  $h + 4\epsilon\sigma T_s^3$  and  $T_{th}$  = minimum threshold temperature =  $T_s - \epsilon q_o/h_e$ .

The minimum threshold temperature is the lowest temperature attainable by the radiator. The convective heat transfer coefficient ( $h$ ) is a function of the wind velocity ( $V$ ) and it is calculated by the following expressions:<sup>11</sup>

Radiator with no wind screen:

$$h = 5.7 + 3.8V \quad V \leq 4 \text{ m/s} \quad (7)$$

$$h = 7.3V^{0.8} \quad V > 4 \text{ m/s} \quad (8)$$

Radiator with wind screen:

$$h = 0.5 + 1.2V^{0.5} \quad (9)$$

The net radiative power of a blackbody ( $q_o$ ) at the ambient temperature ( $T_a$ ) is given by:

$$q_o = \sigma T_a^4 - q_s \quad (10)$$

where  $q_s$  = sky irradiance =  $\epsilon_{sky}\sigma T_a^4$

#### *Fluid temperature*

The problem of calculating the temperature of the heat transfer fluid flowing through a one-dimensional path in a radiator has been solved by Ito and Miura<sup>10</sup> in the same way as the case of a solar collector, Duffie and Beckman,<sup>12</sup> and is given by:

$$T_{fo} - T_{th} = (T_{fi} - T_{th}) \exp(-U_p A / mc_p) \quad (11)$$

where  $T_{fi}$  = inlet temperature of the heat transfer fluid,  $T_{fo}$  = outlet temperature of the heat transfer fluid,  $U_p$  = overall heat transfer coefficient,  $m$  = mass flow rate,  $c_p$  = specific heat at constant pressure, and  $A$  = surface of the radiator.

Equation (11) calculates the outlet temperature of the heat transfer fluid given that the minimum threshold temperature  $T_{th}$  is known. One should note that, the temperature dependence of the thermal properties of the radiator and the heat transfer fluid have also been taken into account in the numerical model.

The method used for calculating the useful cooling energy and the outlet temperature of the air provided by an uncovered radiator has been experimen-

tally tested by Ito and Miura.<sup>10</sup> The theoretical and experimental results were found in excellent agreement. The net radiative power obtained by the measurements was 40-60 W/m<sup>2</sup> on clear nights in the summer and 60-80 W/m<sup>2</sup> in the fall and winter. The average temperature of the energy storage tank on clear nights became 2-5°C below the ambient temperature.

RESULTS

The model presented in the previous section has been used to calculate the sky temperature depression and the outlet air temperature which was used to determine the useful cooling energy of a flat plate radiative air cooler for various

locations of southern simulations have been one for average sky c Tables 1 to 5 give t the cooling season (M reaches a given value which can be very applications at a give An example is give (May-September) fo for Ajaccio - France.

Table 2. Number of events for a given sky temperature depression in June

Table with 23 columns: LOCATION, WEATHER CONDITION, and 22 numbered columns (1-22) representing sky temperature depression (C). Rows list various locations like AJACCIO, ALMERIA, ANCONA, etc., with their weather conditions and event counts.

Table 3. Number of events

Table with 2 columns: LOCATION and WEATHER CONDITION. Rows list various locations like AJACCIO, ALMERIA, ANCONA, etc., with their weather conditions.







Table 5. Number of events for a given sky temperature depression in September

LOCATION	WEATHER CONDITION	SKY TEMPERATURE DEPRESSION (C)																					
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
AJACCIO, IT	OPTIMUM																3	2	2	2	2		
(LAT=41.93N)	AVERAGE										4	1	2	2	2	2	2	2	2	2	2		
ALMERIA, SP	OPTIMUM																						
(LAT=36.85N)	AVERAGE										4	2	2	2	2	1	2	3	1				
ANCONA, IT	OPTIMUM																						
(LAT=43.62N)	AVERAGE										5	3	3										
ATHENS, GR	OPTIMUM																						
(LAT=37.9N)	AVERAGE																5	2	3	1			
ATLANTA, USA	OPTIMUM																						
(LAT=33.65N)	AVERAGE						2	1	1	1	1	1	1	1	2						1	1	1
BARCELONA, SP	OPTIMUM																						
(LAT=41.38N)	AVERAGE										5	3	2	1									
BRINTISI, IT	OPTIMUM																						
(LAT=40.65N)	AVERAGE									2	3	2	3	1									
CACLIARI, IT	OPTIMUM																						
(LAT=39.23N)	AVERAGE																5	2	2				
CATANIA, IT	OPTIMUM																						
(LAT=37.47N)	AVERAGE																						
CHARLESTON, USA	OPTIMUM																						
(LAT=32.9N)	AVERAGE									2	1	1	1	1	1	1	1	1	1	2			
DUBROVNIC, YU	OPTIMUM																						
(LAT=42.63N)	AVERAGE																6	2	3				
GENOVA, IT	OPTIMUM																						
(LAT=44.40N)	AVERAGE																						
GIBRALTAR	OPTIMUM																						
(LAT=36.15N)	AVERAGE																						
IERAPETRA, GR	OPTIMUM																						
(LAT=35.00)	AVERAGE																4	3	2	2			
LIVORNO, IT	OPTIMUM																						
(LAT=43.55)	AVERAGE																4	3	2	2			
MARSEILLE, FR	OPTIMUM																						
(LAT=43.30N)	AVERAGE																						
NAIMI, USA	OPTIMUM																						
(LAT=25.8N)	AVERAGE																						
MILOS, GR	OPTIMUM																						
(LAT=36.44N)	AVERAGE																						
NAPOLI, IT	OPTIMUM																						
(LAT=40.68)	AVERAGE																4	3	3	1			
NICE, FR	OPTIMUM																						
(LAT=43.68N)	AVERAGE																						
NICOSIA, CY	OPTIMUM																						
(LAT=35.15N)	AVERAGE																						
PALMA, SP	OPTIMUM																						
(LAT=39.57N)	AVERAGE																						
PAPHOS, CY	OPTIMUM																						
(LAT=34.75N)	AVERAGE																						
PERPIGNANT, FR	OPTIMUM																						
(LAT=42.73N)	AVERAGE																						
RALEIGH, USA	OPTIMUM																						
(LAT=35.87N)	AVERAGE																						
ROMA, IT	OPTIMUM																						
(LAT=41.90N)	AVERAGE																						
SPLIT, YU	OPTIMUM																						
(LAT=43.52N)	AVERAGE																						
THESALONIKI, GR	OPTIMUM																						
(LAT=40.33N)	AVERAGE																						
TRESTE, IT	OPTIMUM																						
(LAT=45.65N)	AVERAGE																						
VALETTA, IT	OPTIMUM																						
(LAT=35.90N)	AVERAGE																						
VALENCIA, SP	OPTIMUM																						
(LAT=39.47N)	AVERAGE																						
VENEZIA, IT	OPTIMUM																						
(LAT=45.43N)	AVERAGE																						

cooling season among the four sites, has sky temperature depression values less than 10°C for a significant number of hours.

Figures 2-6 give the mean daily useful cooling energy provided per square meter of radiating surface, for the European locations. For each of the 28 southern European cities (from Table 1), the first row of numbers gives the values obtained for the optimum (clear) sky conditions, while the second row gives the values obtained for the average sky conditions. For each one of these conditions, there are two values. The first column corresponds to the case of an uncovered radiator and the second column to the case of a radiator covered with a wind screen. Each figure presents the results obtained for a typical day of each month in the colling season, May through September. The corresponding results for the 4 southeastern United States cities are given in Table 6.

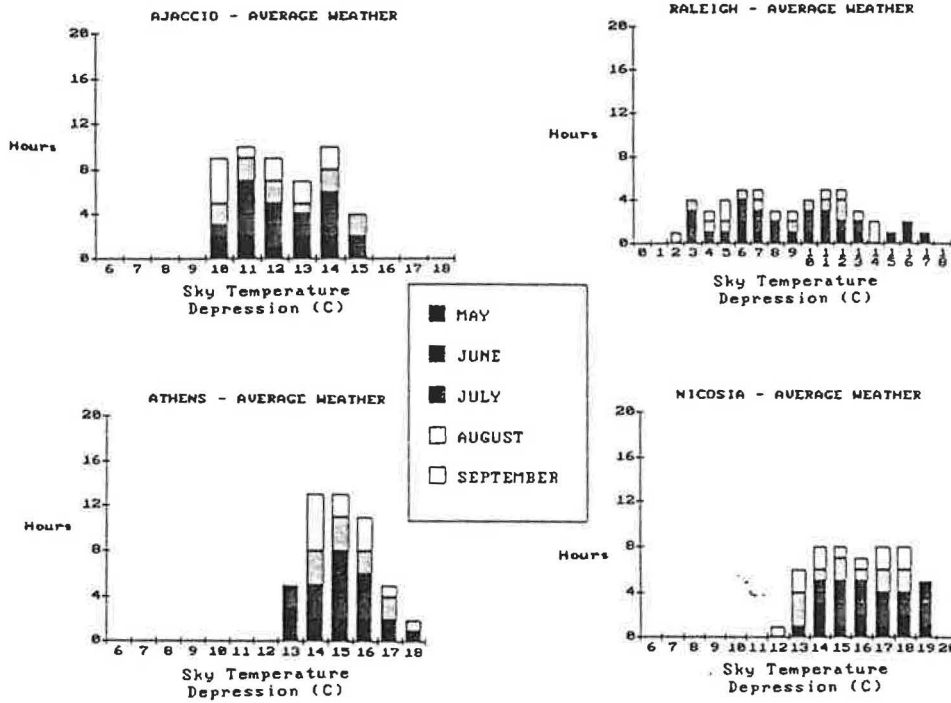


Figure 1 Variation of sky temperature depression at (a) Ajaccio, FR (b) Nicosia, CY (c) Athens, GR (d) Raleigh, NC, U.S.A.

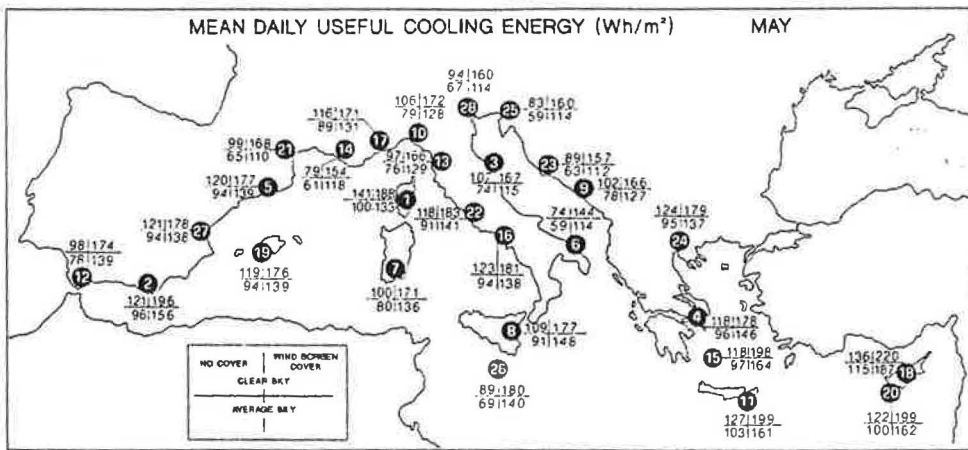


Figure 2 Mean daily useful cooling energy ( $\text{Wh/m}^2$ ) in May for a radiative cooler with a fluid velocity at 2.5 m/sec. Information for each numbered location are given in Table 1.

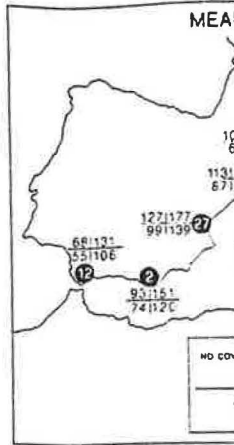


Figure 3 Mean daily useful cooling energy at 2.5 m/sec. Information for each numbered location are given in Table 1.

The mean daily useful cooling energy at the various southern locations is higher than the average sky conditions of the U.S. cities the corresponding sky conditions and the influence of the wind screen which are dominant in the month of May under the conditions 95% higher than the average sky conditions. On the other hand, at some locations

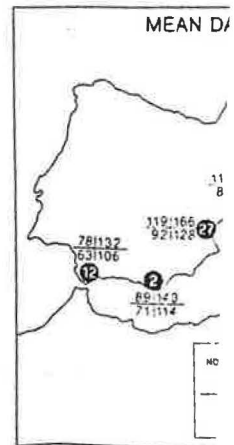


Figure 4 Mean daily useful cooling energy at 2.5 m/sec. Information for each numbered location are given in Table 1.

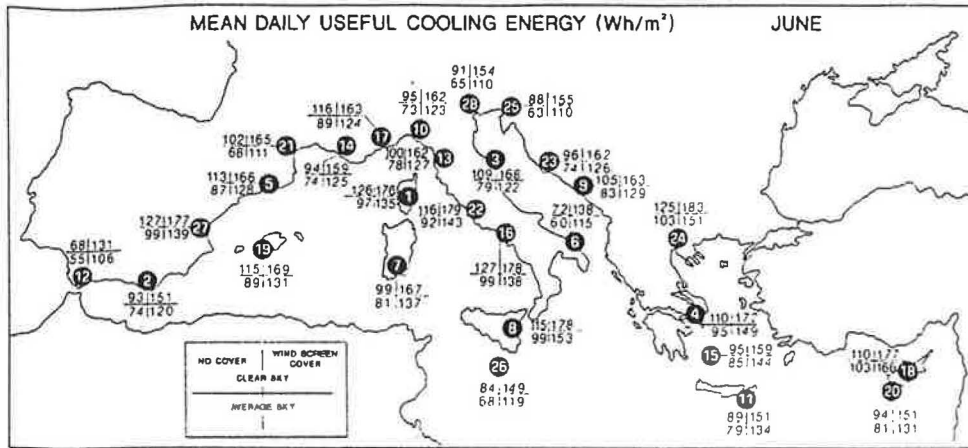


Figure 3 Mean daily useful cooling energy (Wh/m<sup>2</sup>) in June for a radiative cooler with a fluid velocity at 2.5 m/sec. Information for each numbered location are given in Table 1.

The mean daily useful cooling energy delivered by the flat plate radiative cooler at the various southern European cities, ranges between 55 and 208 Wh/m<sup>2</sup> for average sky conditions, and 68 to 220 Wh/m<sup>2</sup> for clear sky conditions. For the U.S. cities the corresponding values range between 41 to 136 Wh/m<sup>2</sup> for average sky conditions and 69 to 182 Wh/m<sup>2</sup> for clear sky conditions.

The influence of the wind screen can play an important role at some locations which are dominated by high wind speeds. For example, at Brindisi during the month of May under clear skies, the useful cooling energy of a covered radiator is 95% higher than the corresponding value of the uncovered radiator. On the other hand, at some locations where the wind speed is relatively low, the effect of a

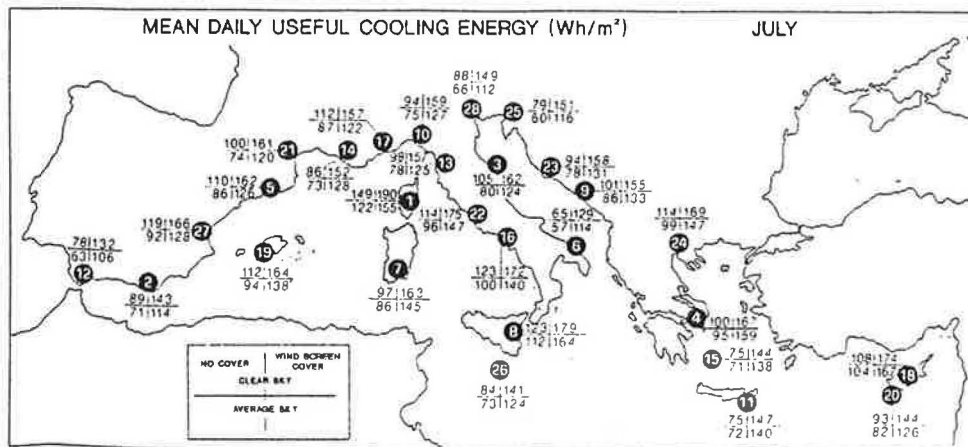


Figure 4 Mean daily useful cooling energy (Wh/m<sup>2</sup>) in July for a radiative cooler with a fluid velocity at 2.5 m/sec. Information for each numbered location are given in Table 1.

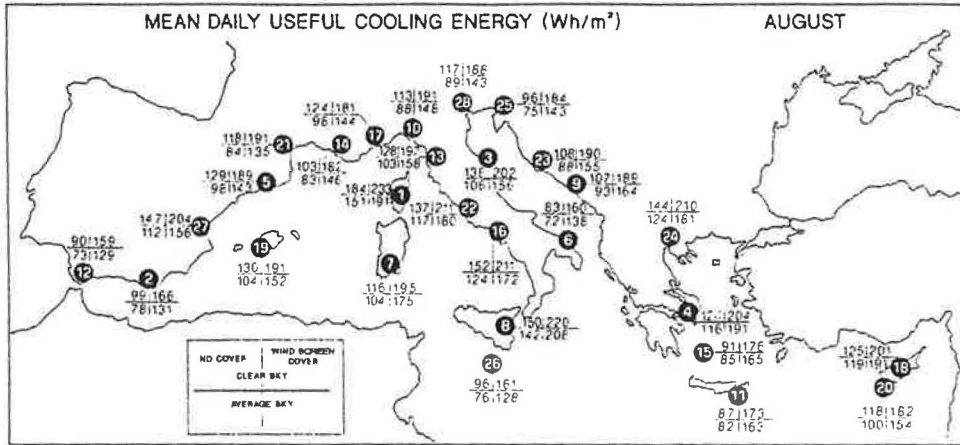


Figure 5 Mean daily useful cooling energy (Wh/m<sup>2</sup>) in August for a radiative cooler with a fluid velocity at 2.5 m/sec. Information for each numbered location are given in Table 1.

wind screen is less effective. For example, at Ajaccio the corresponding values differ only by 33%. A similar increase is also observed in August and September at the island of Milos - Greece, located at the Aegean sea. During these months the area is dominated by strong northern winds which influence greatly the performance of the system.

COMPARISON WITH EXPERIMENTAL DATA

Experimental data on the performance of radiative cooling components in southern Europe are very limited. Some results on specific aspects of radiative

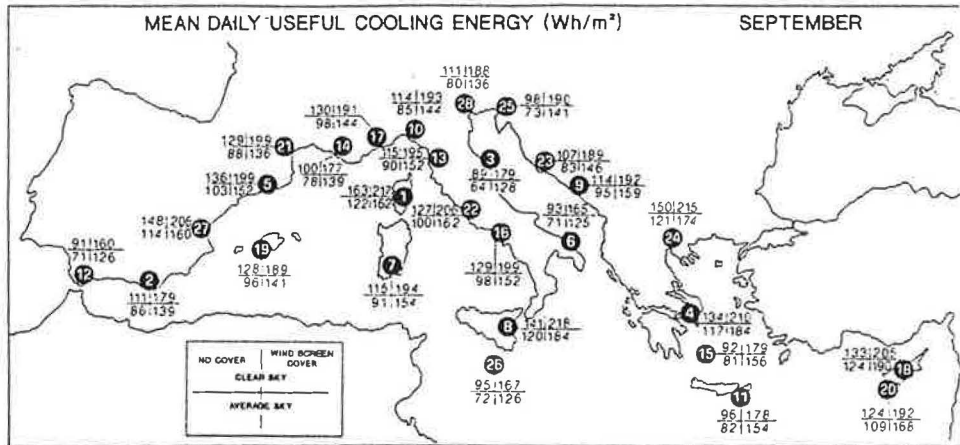


Figure 6 Mean daily useful cooling energy (Wh/m<sup>2</sup>) in September for a radiative cooler with a fluid velocity at 2.5 m/sec. Information for each numbered location are given in Table 1.

Table 6. Mea

LOCATION

ATLANTA, G

MIAMI, FL

CHARLESTON

RALEIGH, N

cooling have be  
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A comparison  
data in Europe  
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CONCLUSIONS

In this paper, th  
of a typical fla  
southeastern U.S  
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Accordingly,  
potential for the  
cities, it appears  
that radiative co

NOMENCLATURE

- A radiator
- C<sub>1</sub> (C<sub>2</sub>T<sub>dry</sub>)
- C<sub>2</sub> constant
- C<sub>3</sub> constant
- c<sub>p</sub> specific
- DT<sub>sky</sub> sky tem

**Table 6.** Mean daily useful cooling energy (Wh/m<sup>2</sup>) for Southern United States Cities

LOCATION	LAT	SKY CONDITION	MAY	COOLING ENERGY (Wh/m <sup>2</sup> )			
				NO SCREEN/WIND JUNE	SCREEN JULY	SCREEN AUGUST	SCREEN SEPTEMBER
ATLANTA, GA	33.5 N	Clear	103/182	77/131	75/124	95/155	98/167
		Average	77/136	51/87	45/74	60/99	69/117
MIAMI, FL	25.8 N	Clear	86/147	87/146	88/147	84/143	91/160
		Average	54/101	46/79	48/81	49/82	45/76
CHARLESTON, SC	33 N	Clear	91/160	69/120	83/141	86/143	90/152
		Average	60/107	41/72	45/76	49/81	56/96
RALEIGH, NC	35.9 N	Clear	104/178	81/134	78/127	97/155	103/170
		Average	74/128	54/89	49/79	60/97	72/119

cooling have been reported in the literature.<sup>13-20</sup> However, a complete experimental data set on the performance of metallic radiators is not available.

A comparison of the data used in the present study with available experimental data in Europe has shown a satisfactory agreement. As it was previously reported, the accuracy of the present method has also been experimentally verified by Ito and Miura,<sup>10</sup> while the overall method predicts results in close agreement with experimental data reported for northern American locations.<sup>6</sup>

The present work offers a simple and accurate method to predict the radiative cooling potential in southern Europe and therefore is very useful for building researchers and energy engineers. Further experimental work is necessary though in order to extend our knowledge on the topic.

## CONCLUSIONS

In this paper, the sky temperature depression and also the useful cooling energy of a typical flat plate radiative air cooler at 28 southern European and 4 southeastern U.S. cities was calculated. These parameters allow the estimation of the effectiveness and feasibility of radiative cooling applications.

Accordingly, one may conclude that southern Europe exhibits a promising potential for the use of radiative cooling. Compared with the results from the US cities, it appears that the weather conditions of the southeastern states are such that radiative cooling techniques will be of a very low efficiency if applied.

## 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 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 nonselective 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

$\Delta \epsilon_d$	instantaneous clear sky emissivity
$\epsilon_{cs}$	clear sky emissivity
$\epsilon_s$	cloudy sky emissivity
$\epsilon_{sky}$	sky emissivity
$\sigma$	Stefan-Boltzmann constant

#### Subscripts

a	ambient
f	fluid
i	inlet condition
o	outlet condition
r	radiator
s	sky

#### References

1. T. C. Steemers, *Int. J. Solar Energy*, **10**, 5 (1991).
2. Horizontal Study on Passive Cooling (M. Santamouris Editor, C.E.C. - BUILDING 2000 project, 1990), Chap. 1, pp. 1-7, organized by DG 12, EEC.
3. G. Clark, Proceedings of the International Passive and Hybrid Cooling Conference (A. Bowen, E. Clark, K. Labs editors, Miami Beach, 1981), pp. 682-714.
4. M. Santamouris, Proceedings of Passive Cooling Workshop (E. Aranovich, T. Steemers and O. Fernandes Editors, Ispra, 1990).
5. *Weather in the Mediterranean*, (Her Majesty's Stationery Office, 1964), Vol. II.
6. Climates of the States, Climatography of the United States No. 60-8, No. 60-9, No. 60-31, and No. 60-38, 1975.
7. X. Berger, B. Cubizolles and I. Donet, *Solar & Wind Technology* **5**, 353, (1988).
8. M. Martin, Passive Cooling (J. Cook Editor, M.I.T. Press, 1989), Chap. 4, pp. 138-196.
9. Aspirations - Psychrometer Tafel, Deutscher Wetterdienst, 5, Auflage, Vieweg Verlag, Braunschweig, 1976.
10. S. Ito and N. Miura, *Journal of Solar Energy Engineering*, **111**, 251, (1989).
11. M. Mostrel M. and B. Givoni, *Passive Solar Journal*, **1**, 229, (1982).
12. J. A. Duffie and W. A. Beckman, *Solar Engineering of Thermal Processes* (John Wiley and Sons, New York, 1980).

13. A. Fragoudaki
14. A. Addeo, *E Cimento*, **1**, 4
15. S. Catalanotti (1975).
16. S. Golli and (1981).
17. P. Grenier, *R*
18. X. Berger, *C*
19. B. Bartoli, *E Energy*, **3**, 26
20. G. Papadakis

13. A. Fragoudakis, G. Papadaki and S. Kyritsis, *Int. J. Solar Energy*, 7, 73, (1989).
14. A. Addeo, E. Monza, M. Peraldo, B. Bartoli, B. Coluzzi, V. Silvestrini and G. Troise, *Il Nuovo Cimento*, 1, 419, (1978).
15. S. Catalanotti, V. Cuomo, G. Piro, D. Ruggi, V. Silvestrini and G. Troise, *Solar Energy*, 17, 83, (1975).
16. S. Golli and Ph. Gremir, *Journal de Physique*, Collogue, Supplement to No. 1, 42, C1 431, (1981).
17. P. Grenier, *Revue de Physique Appliquee*, 14, 87, (1979).
18. X. Berger, C. Awanou and J. Bathiebo, *Int. J. of Ambient Energy*, 9, 155, (1988).
19. B. Bartoli, B. Catalomoti, B. Coluzzi, B. Cuomo, V. Silvestrini and G. Troise, *Applied Energy*, 3, 267, (1977).
20. G. Papadakis, G. Voulgaris, A. Fragoudakis and S. Kyritsis, *Int. J. Solar Energy*, 6, 279, (1988).