A LOW-ENERGY INNOVATIVE SYSTEM FOR SPACE COOLING

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ABSTRACT

A lightweight aluminium nocturnal radiator, painted with an appropriate paint, was established on the roof of the Department of Environmental & Natural Resources Management in Agrinio, in Western Greece. The dynamic thermal performance of the system during summer months was calculated using an accurate mathematical model, based on the heat transferred from the air circulating inside the radiator to the ambient air. Furthermore, an extensive validation process was carried out. Thus, the experimental air temperature values at the radiator's outlet were compared with the theoretical ones and a very good agreement was achieved. The validation procedure was extended for two different radiator's paints and it was found that there is a significant impact of the paints' emissivities on the system's efficiency.

Moreover, the more effective lightweight nocturnal radiator was used to provide space cooling or pre-cooling for the building of the University. Indoor air temperature values of the thermal zone connected with the radiator were compared with those of a similar zone without any cooling system and the results demonstrated a remarkable effectiveness of the system.

KEYWORDS

Passive cooling, radiative cooling, sky temperature, clear sky emissivity, metallic nocturnal radiator.

1. INTRODUCTION

Passive cooling of buildings can achieve remarkable thermal comfort during summer with a great reduction of cooling loads (Givoni, 1994; Cook, 1989). Heat dissipation techniques are based on the transfer of a buildings' excess heat to a lower temperature environmental sink, as the ambient air, water, ground and sky (Santamouris and Assimakopoulos, 1996). In case the sink is the sky, heat dissipation is carried out by long wave radiation from a building to the sky (radiative cooling) (Mihalakakou et al, 1998). The sky temperature is usually lower than the temperature of the most objects on the earth, so, any ordinary surface that "sees" the sky has a net long wave radiant loss (Givoni, 1982; Mostrell and Givoni, 1982). Despite this radiative loss, the surface doesn't obtain lower temperature throughout the 24-hour daily cycle, as the incoming solar radiation during the daytime is greater than the net long wave radiation loss. This daytime disadvantage of radiative cooling is being eliminated if the radiating surface is not the building envelope directly (as happens in so-called direct, or passive radiative cooling), but a specific radiator with a suitable heat transfer medium circulating through it at the appropriate time – nocturnal radiator (as in so-called hybrid radiative cooling) (Givoni, 1994; Cook, 1989).

In the present study, a lightweight nocturnal radiator is placed on the roof of the 10 m height building of the Environmental and Natural Resources Management Department of University of Patras, in Agrinio (Western Greece). As a first step of the construction of the metallic panel of the radiator, we use a folded long aluminium tube, painted with an appropriate white paint of high emissivity (Figure 1).

Figure 1: Metalic base with the folded aluminium tube.

Then the aforementioned tube is been pressed both to its up and down surfaces, in order to obtain a flat appearance - instead of the cylindrical shape – with a width of 1 cm in the flat sides. One end of the tube is sited at one corner of the rectangular panel formation, carrying a small fan, which – when it operates – drives air masses through the parallel tubes. The fan is connected to the power supply through a time starter, in order to operate only at nocturnal hours (10.00 pm to 06.00 am). The other end of the tube is adapted to an appropriate window opening. The use of silicon glue keeps the tube well fitted to the window opening. The air mass through the radiator loses heat by convection to the metallic surface above, becomes cooler and finally is transferred into the building's interior, in an office of the university building, driven by the fan.

The above mentioned procedure has been repeated once more for the construction of a similar radiator panel, but, this time, the paint used has a higher emissivity. Finally, two similar radiator panels are ready to be used and the only difference between them is in their optical properties.

The main objectives of the present paper is primarily to present, describe and analyse an experimental performance of a metallic nocturnal radiator placed on the roof of University of Ioannina, in Agrinio, Greece, and secondly to compare the experimental data with the ones calculated using a mathematical model able to simulate the thermal performance of the radiator. Moreover, the cooling potential of the nocturnal radiator was investigated during the present research as the radiator was connected to the building of University and it was used to cool a specific thermal zone. Thus, the feasibility and the effectiveness of the cooling system was investigated and analysed.

2. MODELING OF THE NOCTURNAL RADIATOR

The dynamic thermal performance of the nocturnal radiator has been calculated using an accurate mathematical model. Modelling of water and air based radiators for cooling purposes can be considered to be quite similar with that of water-based solar collectors (Santamouris and Assimakopoulos, 1996; Mihalakakou et al, 1998; Ito and Miura, 1989).

The convective heat exchange due to wind, q_{win}, between the radiator and the ambient air is given by the following expression (Santamouris and Assimakopoulos, 1996; Mihalakakou et al, 1998):

$$q_{win} = h_{win}(T_r - T_a) \tag{1}$$

where h_{win} is the convective heat transfer coefficient and T_r and T_a are the radiator and ambient absolute temperatures respectively (K).

The convective heat transfer coefficient h_{win} is a function of the wind velocity v and, in case of nocturnal radiator with no wind screen, is calculated from the following expressions:

$$h_{win} = 5.7 + 3.8v \quad if \;\; v < 4ms^{-1}$$

$$h_{win} = 7.3v^{0.8} \quad if \;\; v > 4ms^{-1}$$

The radiative heat flux of the radiator to the sky could be incorporated into the "ambient" heat flux q_{amb} of the radiator, which includes both convective and radiative heat flux and is calculated by the following formulation:

$$q_{amb} = h_e(T_r - T_{th}) \tag{2}$$

where h_e is an effective heat transfer coefficient and T_{th} is the threshold absolute temperature.

The effective heat transfer coefficient h_e can be calculated as follows:

$$h_e = h_{win} + h_{rad} \tag{3}$$

In the previous expression h_{rad} is the radiative heat transfer coefficient, given by the following expression:

$$h_{rad} = 0.000000227\tau\varepsilon T_a^2 \tag{4}$$

where τ is the infrared transmittance of the wind screen (if there isn't any wind screen, then: τ = 1), ε is the infrared emissivity of the radiator plate and T_α is the absolute ambient temperature (K)

Generally, threshold temperature or stagnation temperature, (T_{th} in K or θ_{th} in ºC), is the temperature a surface will drop to without any heat being added to the surface. In case of the nocturnal radiator is the minimum stagnation temperature the radiator can achieve, constrained to be no less than 1ºC below the ambient dew point temperature.

Threshold temperature θ_{th} (°C) can be calculated using the following formula:

$$\theta_{th} = \theta_a - 0.000000057\varepsilon(1 - \varepsilon_s)\frac{\theta_a^4}{h_e} \tag{5}$$

where θ_α is the ambient temperature (°C) and ε_s is the sky emissivity.

Sky emissivity ε_s was calculated using the clear sky emissivity ε_{cs}, with a correction factor c which is used in case that the sky isn't clear and there is cloudiness (Clark, 1981):

$$\varepsilon_c = c\varepsilon_{cs} \tag{6}$$

If sky is clear, then c=1. Many different long wave radiation parameterization methods have been used in order to calculate clear sky emissivity ε_{cs} (Brutsaert, 1975;

Berdahl and Martin, 1984). The Swinbank formula (Swinbank, 1963) was used in the present research, as ε_{cs} is dependent only from the ambient temperature:

$$\varepsilon_{cs} = 9.36.10^{-6} T_a^2 \tag{7}$$

The outlet temperature θ_{out} (^oC) of the flowing air (when the air exits the tube) can be calculated by the following equation (Ito and Miura, 1989):

$$\theta_{out} = \theta_{th} + (\theta_{in} - \theta_{th}) e^{-\frac{UA}{mc_p}} \tag{8}$$

where θ_{in} is the inlet temperature of the heat transfer fluid (^oC), U is the overall heat transfer coefficient between the fluid circulating under the radiator and the ambient air (W m^{-2} K^{-1}), A is the surface of the radiator plate (m^2), m is the mass flow rate (kg s^{-1}) and c_p is the specific heat of the fluid (J kg^{-1} K^{-1}).
For the specific characteristics of the radiator, Eq. (8) is transferred to the following formulation (Santamouris and Asimakopoulos, 1996):

$$\theta_{out} = \theta_{th} + (\theta_{in} - \theta_{th}) e^{-\frac{Uw}{1173zu}} \tag{9}$$

where w is the width of the radiator plate (m), z is the height of the duct (m) and u is the air velocity of the circulating air under the radiator plate (ms^{-1}). The overall heat transfer coefficient U can be calculated by the overall thermal resistance R as:

$$U = \frac{1}{R} \text{ (W}^{-1}\text{ m}^2\text{ K)}$$

The overall thermal resistance R is the combination of individual thermal resistances (Kreith and Bohn, 1986): $R = R_r + R_e + R_a$ [R_r is the resistance of the radiator plate, given by: $R_r = \frac{d}{k}$, where d is the thickness of the radiator plate (m) and k is the radiator's thermal conductivity (k=236 $Wm^{-1}K^{-1}$), R_e is the effective resistance: $R_e = \frac{1}{h_e}$ and R_a is the resistance between the circulating air and the radiator: $R_a = \frac{1}{h}$, where h is the corresponding heat transfer coefficient between the radiator and the circulating air under the radiator].
Finally, the overall heat transfer coefficient U can be calculated using the following expression:

$$U = \frac{1}{\frac{1}{h_e} + \frac{1}{h} + \frac{d}{k}}$$

The heat transfer coefficient h can be determined from the values of hydraulic diameter D_h and the air velocity u of the circulating air into the tube.
The hydraulic diameter D_h can be calculated as follows (Santamouris and Asimakopoulos, 1996):

$$D_h = 2w[z/(w+z)].$$

3. RESULTS AND DISCUSSION

During the validation process the obtained experimental data were compared with those calculated with the mathematical model presented in Chapter 2 (theoretical ones) and this process was carried out for two similar radiators, painted with paints of different emissivities, in order to investigate the impact of different materials' optical parameters on the system efficiency. This procedure was performed for a quite long experimental period, in order to achieve safe results. During these time intervals, temperature at the radiator's outlet was monitored continuously with a self-recorded sensor, placed at the exit of the tube. After the validation process, in order to estimate the cooling effect of the radiator with the most efficient paint, the indoor temperature of the office connected with the radiator was monitored, as well as the temperature of a neighbouring office with the same structural and size characteristics, without any cooling system in operation.

Figure 2a shows the temporal variation of hourly theoretical and experimental air temperature values at the outlet of the radiator painted with a paint of emissivity 0.71, for the hours of the radiator's operation (nocturnal hours). The hours in the horizontal axis of the diagram are the nocturnal hours only (from 10.00 pm to 06.00 am), so the first 9 hours on the axis, from 1 to 9 are the nocturnal hours of the first randomly selected day (from the 25th of August at 10.00 pm to the 26th of August at 06.00 am), the next 9 hours from 10 to 18 are the corresponding nocturnal hours of the second selected day etc. As shown, the calculated values perform well with the experimental ones for every time interval of this 10-days demonstration. Figure 2b also shows the temporal variation of hourly theoretical and experimental air temperature values at the outlet of the radiator painted with a paint of emissivity 0.93, for the hours of the radiator's operation (nocturnal hours). The time interval presented in the horizontal axis is a randomly selected 10-days period, where there is a great agreement between theoretical and experimental values and this agreement was achieved for the whole set of measured and theoretical data, as well. The comparisons of the experimental air temperature values at the radiator's outlet with the calculated ones, for both paints, for the whole time of the each radiator's operation (10 days) showed an excellent agreement in both occasions, while the linear regression analysis confirmed this observation (R^2 = 0.9648 and R^2 = 0.9766 for the paints of 0.71 and 0.93 emissivities, respectively).

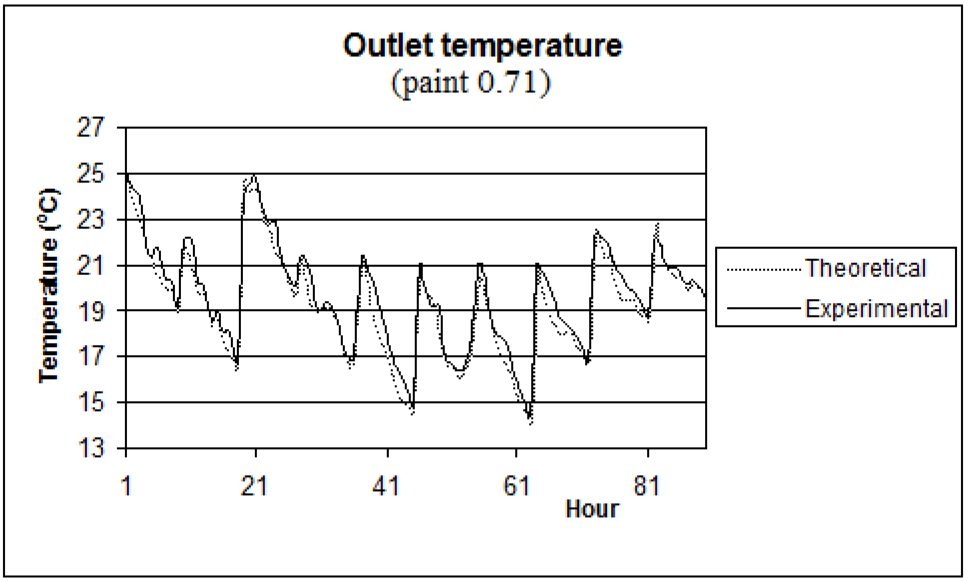

Figure 2a: Temporal variation of the measured (experimental) and calculated (theoretical) air temperature values at the radiator's exit for 10 selected days, when the radiator is painted with a paint of emissivity 0.71.

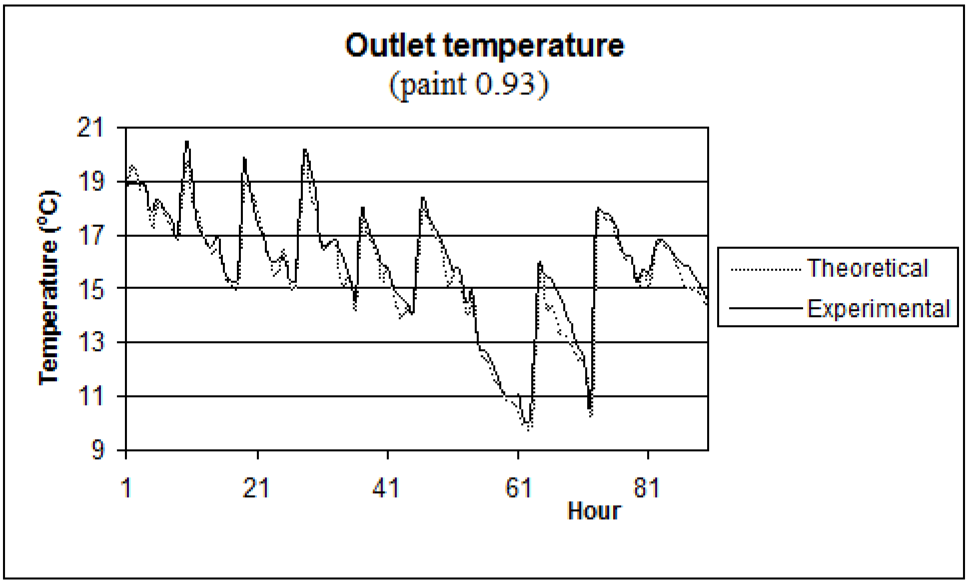

Figure 2b: Temporal variation of the measured (experimental) and calculated (theoretical) air temperature values at the radiator's exit for 10 selected days, when the radiator is painted with a paint of emissivity 0.93.

Tables 1a and 1b presents analytically the correlation coefficient values for the theoretical and experimental temperatures for each day of the experiment particularly, for both radiators. These high values (0.959278<CC<0.988345 for the paint with emissivity 0.71 and 0.957786<CC<0.986548 for the paint with emissivity 0.93) also demonstrate the strong agreement between them in both occasions.

Table 1a. Correlation coefficient of theoretical and experimental data (paint 0.71)

Day	**CC**
1	0.974169
2	0.977206
3	0.959278
4	0.959417
5	0.979823
6	0.977007
7	0.988345
8	0.959616
9	0.960218
10	0.967170

Table 1b. Correlation coefficient of theoretical and experimental data (paint 0.93)

Day	**CC**
1	0.958693
2	0.978610
3	0.977582
4	0.966045
5	0.986137
6	0.978473
7	0.967282
8	0.961923
9	0.986548
10	0.957786

From the presented results, two essential conclusions can be extracted:
a. the radiator with the paint of 0.71 emissivity is less "effective" in the achievement of lower temperatures than the radiator with the paint of 0.93 emissivity. As shown in Figures 2a and 2b, the differences between theoretical and experimental values at lower temperatures are greater for the radiator with the less emissive paint. So, for the 0.71 paint these differences are: 0.43, 0.41 and 0.42 ^oC at the 18th, 45th and 63rd hour respectively (Figure 2a), when for the paint 0.93 these differences are: 0.28, 0.20 and 0.29 ^oC at the 18th, 45th and 63rd hour respectively (Figure 2b).
b. there is a better agreement between theoretical and experimental temperatures for the radiator with the more emissive paint: R^2 = 0.9648 for the radiator with the 0.71 paint and R^2 = 0.9766 for the radiator with the 0.93 paint.
The above mentioned analysis shows that the emissivity of the radiator have significant influence on the system's efficiency.

Afterwards, the air cooled by the nocturnal radiator was transferred as ventilation into an office of the experimented building in order to provide space cooling. The radiator used is the one with the more effective paint (0.93). The indoor air temperature inside the ventilated office was monitored during the experimental period on a 24-hour basis. Moreover, the indoor air temperature of a neighboring office, with similar orientation, construction and internal gains elements, was measured on a 24-hour basis too, in order to estimate the system cooling capacity. During the experimental period there was not any cooling system operated in that second office.
Figure 3 shows the 24-hour temporal variation of the indoor air temperature values at both rooms, the room connected with the nocturnal radiator and that not equipped with any cooling system for 5 randomly selected days – 120 hours (from the 15th of September at 10.00 pm to the 20th of September at 10.00 pm).

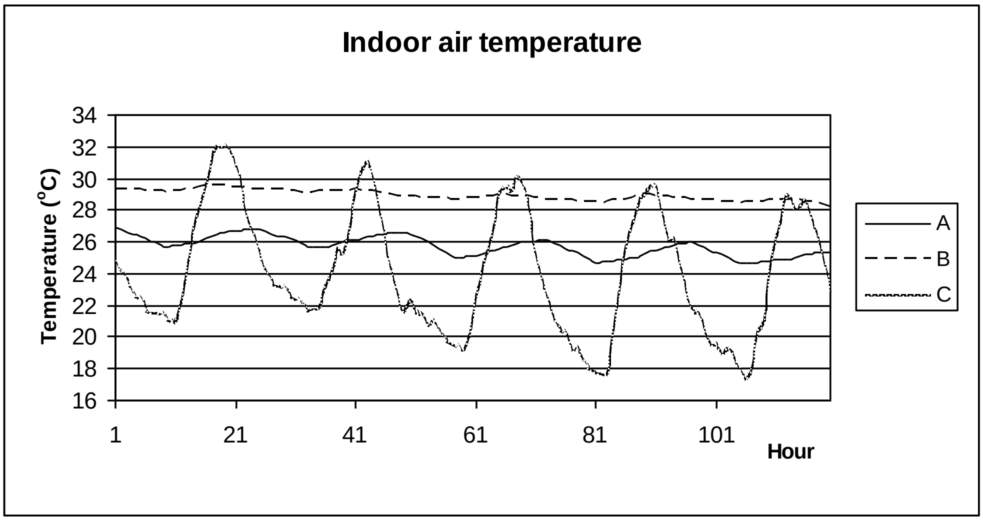

Figure 3: Temporal variation of the indoor air temperature values at both rooms, the room connected with the nocturnal radiator (A), and that not equipped with any cooling system (B) for 5 randomly selected days of the experimental time period (from the 15th of September at 10.00 pm to the 20th of September at 06.00 am), and ambient temperature for the same period (C).

As shown, the temperature difference between the two rooms fluctuated between 4 ºC at the late nocturnal hours, to 2.5 ºC at daily hours. This daily temperature difference is remarkable, despite the fact that the radiator doesn't operate and is due to the thermal inertia of the building. Especially, for every night this difference take its highest value at the last observation (at 06.00 am), when the fan still operates and the ambient and the radiator outlet temperature take its lowest values. At the presented 5-day period, this peak (at 06.00 am) difference increases as the temperature difference between the indoor temperature of the room without any cooling system and the radiator outlet temperature increases. This conclusion is clearly depicted in Table 2, which shows numerically these temperature differences at the "peak" hours (06.00 am).

Table 2. Indoor and outdoor temperature differences

Hour	**T_{room2}-T_{amb} (^oC)**	**T_{room2}-T_{room1} (^oC)**
9	7.69	3.41
33	7.42	3.36
57	9.07	3.53
81	10.60	3.79
105	10.46	3.75

T_{room1} = temperature of the room connected with the radiator (^oC)

T_{room2} = temperature of the room not equipped with any cooling system (^oC)

T_{amb} = ambient temperature (^oC)

Similar results were achieved for the whole experimental period. Thus, the nocturnal radiation system could be effectively used in order to remarkably reduce the energy consumption for space cooling.

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