

Karolina PETELA^{1*} and Andrzej SZŁĘK¹

ASSESSMENT OF PASSIVE COOLING IN RESIDENTIAL APPLICATION UNDER MODERATE CLIMATE CONDITIONS

OCENA MOŻLIWOŚCI STOSOWANIA PASYWNEGO CHŁODZENIA DO CELÓW DOMOWYCH W KLIMACIE UMIARKOWANYM

Abstract: In the face of environmental regulations, renewable energy systems are anticipated to become more attractive. Passive buildings may appear promising in terms of energy saving. The aim of the work is an investigation of energy effects of using radiative passive cooling. System analysed here bases on the radiative heat exchange with nocturnal sky. On every exposed surface, beyond the convection mechanism, a radiative heat exchange with the sky takes place. Analysis shows that passive cooling has a potential in cold production, however is sensitive to ambient conditions and that cold supply is inversely proportional to demands. Small value of average heat loss from the radiator makes the system independently unable to fulfil cooling demand, however may become an attractive, eco-friendly supplement to a conventional air-conditioner.

Keywords: passive cooling, nocturnal sky, solar collector, residential cooling

Introduction

The subject of limiting the use of non-renewable energy sources is still important, considering the continuous worldwide growth of electric energy usage and demand. One has to be aware of the tendency, that the use of electric energy increases in summer as well, which results from increasing use of air conditioning systems. Most commonly applied are compression chillers. However, in order to decrease the use of non-renewable energy, alternatives are needed. One of the possibilities is the application of commercially-spread sorption chillers. Their mode of operation is analogical to that of compression chiller, but the electric compressor is substituted by a colloquially named

¹ Faculty of Energy and Environmental Engineering, Institute of Thermal Technology, Silesian University of Technology, ul. Konarskiego 22, 44-100 Gliwice, Poland, phone: +48 32 237 10 31, +48 32 237 10 41, fax: +48 32 237 28 72, email: karolina.petela@polsl.pl, andrzej.szlek@polsl.pl

* Corresponding author: karolina.petela@polsl.pl

sorption compressor which comprises of absorber, generator driven by heat and a solution heat exchanger. Electric energy consumed by this type of system is used only to drive solution pumps inside the cycle. Nevertheless, to minimize the energy use by final consumers, passive architecture is becoming more and more popular and attractive. Those buildings are characterised by good insulating properties, installed systems of ventilation heat recuperation, use of internal gains and passive use of solar energy. Passive buildings are also equipped with cooling structures like ground heat exchanger or evaporative cooling. A system that is expected to have the potential of covering the cooling demand of an inhabitant is a device using the phenomenon of heat exchange with a radiating nocturnal sky. The idea proposed in this study is a theoretical investigation of possibilities of radiative cooling system application into a residential building in Nowy Sacz, Poland.

Radiative cooling phenomenon

Radiative cooling belongs to one of the heat dissipation techniques, where heat is transferred to a lower temperature sink. Radiative cooling exploits sky as the sink and the heat loss is conducted by long-wave radiation to the sky [1]. On every exposed surface, beyond the convection mechanism, a radiative heat exchange with the sky takes place. This phenomenon is most effective during night-time, when no solar gains appear and the nocturnal sky temperature as low as even -50°C . The efficiency of radiative cooling system is affected by cloud cover, wind and humidity of the air. That is the reason why those systems are most commonly installed in the desert climates, while the intensive ambient temperature drop in the night is additionally advantageous [2]. A scheme of heat exchange between a surface, ambient and nocturnal sky is presented in Fig. 1.

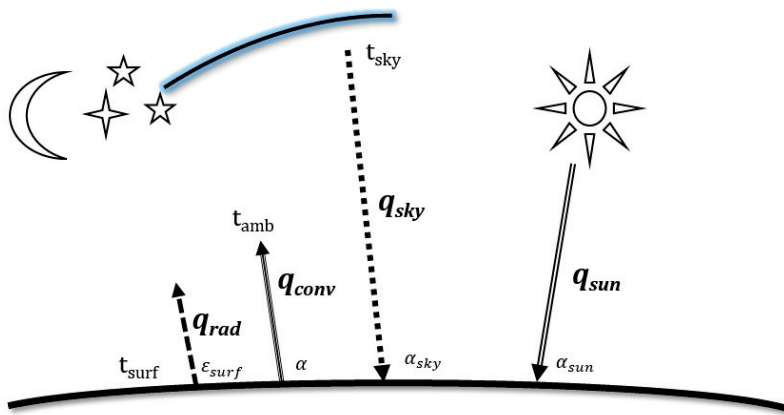


Fig. 1. Heat fluxes for a surface exposed to the sky; t_{surf} [$^{\circ}\text{C}$] – surface temperature, q_{rad} [W/m^2] – surface radiative heat flux, ϵ_{surf} – surface’s emissivity, q_{conv} [W/m^2] – convective heat flux, α [$\text{W}/(\text{m}^2\text{K})$] – convective heat transfer coefficient, q_{sky} [W/m^2] – sky radiative heat flux, α_{sky} – sky-absorptivity factor, t_{sky} [$^{\circ}\text{C}$] – radiative sky temperature, q_{sun} [W/m^2] – absorbed solar radiation heat flux rate, α_{sun} – solar-absorptivity factor

A heat loss from a surface that has been heated during the day will be higher, if the ambient temperature in the night is low. However, unlike the desert climate, the night-time temperature drop in the moderate climate locations does not occur in the hot few-days periods, with cooling demand. In the opposite, if the day is hot, it is highly probable, that the night will be warm as well. If the surface temperature of a radiator drops below the ambient temperature under those conditions, convection gains partially neutralize the radiant heat loss. This tendency is amplified with increasing wind speed. A parameter at which convective heat gains equal radiant heat loss is known as radiator's stagnation temperature [3]. To prevent the heat gains by a forced convection different types of wind screens have already been investigated. Predominantly discussed are: glazings transparent in the infra-red range or some open coverings (eg honeycomb-shaped) that limit the wind speed and general motion of the air above the radiator [3].

It is evidential in the available literature that the issues of radiative cooling were discussed already in 1980s [3]. Various types of radiators have been investigated since then. Erell and Etzion [4] analysed the prospects of using an unglazed flat plate solar collector for purposes of radiative cooling, while Dimoudi [5] performed an experimental study of performance on a roof component comprised of white painted pipes. Bagiorgas and Mihalakakou [1] proposed an experimental model of a roof radiator made of white painted folded aluminium tube used for space cooling in Greece. Farmahini-Farahani and Heidarinejad [6] used flat-plate radiators to pre-cool the air that was subsequently cooled in an evaporative cooling device. They assumed the presence of a water storage tank in the circuit, but Zhang and Niu [7] were to analyse the cooling performance of nocturnal radiative cooling combined with microencapsulated phase change material slurry storage. Al-Obaidi et al [8] reviewed the effect of using different paints and materials on radiative roof's operation.

The idea here proposed considers a performance analysis of a radiator in form of a flat plate collector with white painted pipes that is covering whole roof area of a residential building in moderate climate conditions.

Building under consideration

The radiative cooling system is expected to meet the cooling needs of a one-family detached house with four inhabitants. Construction of the building is assumed to be light-weight with insulated walls made of hollow brick. The building has two levels: ground floor and first floor. No basement or attic are considered. Ground floor is divided into 5 separate spaces: kitchen, bathroom, living room, office and hall. A staircase leads to the first floor, where 5 rooms are located: main bedroom, 2 children bedrooms, a bathroom and a wardrobe. Only the wardrobe is excluded from the group of cooled spaces. A scheme of house under consideration is presented in Fig. 2. Dimensions of every floor are 10×7 m. It is ideally assumed, that whole flat roof is covered with an unglazed flat plate collector playing the role of a passive radiator. The system is combined with a 3 m^3 storage tank.

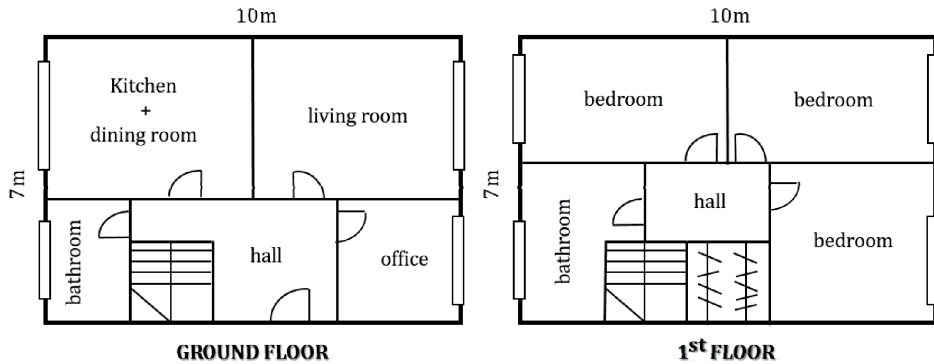


Fig. 2. A simplified scheme of rooms projection inside the project house

The project-building is situated in Nowy Sącz. It is a town located in southern Poland, in Lesser Poland Voivodship, and its coordinates are: $49^{\circ}37'26''\text{N}$ $20^{\circ}41'50''\text{E}$. Poland represents moderate climate conditions. However, being under eastern continental influences, summers tend to be hot. Consequently, more and more electric energy consumption in the summer results from installing the air conditioning systems. Nevertheless, it has to be emphasized that despite the occurrence of high daily temperatures, night temperature drop is not intense, cloud cover and air pollution appears regularly, which are the factors hindering an effective use of passive radiators.

Cooling demand calculation

For the purpose of this study, cooling demand was declared basing on a German standard VDI 2078, described in [9]. According to this norm cooling demand is amount of heat created by external and internal gains that should be removed from the space, to obtain steady state indoor temperature (26°C). To define the cooling demand, an energy balance has to be made. In Poland, residential buildings are air-conditioned only during summer, so the energy balance has been conducted for five typical cases of the months: May, June, July, August, and September. Each case reflects a simulation of an hour with the highest ambient temperature in the month. Meteorological data used for the calculations consider a typical meteorological year for station in Nowy Sącz [10].

Two types of heat gains are to be considered: external heat through walls, windows, doors etc., as well as internal heat gains from people, machines, lighting, and rooms' walls.

Before any kind of heat balance can be made, the partition properties have to be stated. All of partitions are projected to fulfil obligatory standards in terms of heat transfer coefficient. Table 1 presents types of partitions existing in project house.

Table 1

Heat transfer coefficients of the analyzed partitions

No.	Type of partition	Heat transfer coefficient [W/(m ² K)]
1	Outer wall	0.25
2	Inner wall	1
3	Floor on the ground	0.3
4	Ceiling	1
5	Roof	0.2
6	Window	1.3
7	Outer door	1.7
8	Inner door	5

While simulating a building with walls made of hollow brick, it is also ideally set, that it has a negligible thermal inertia. It means that heat transfer to the inner part of the room is not delayed by accumulative properties of the wall and enables to conduct calculations for real time ambient temperature.

Heat flux through walls and other opaque barriers depends on temperature difference and solar radiation – these phenomena have to be considered together, as solar radiation increases ambient temperature. It requires determination of an equivalent temperature difference $\Delta\vartheta_{eqv}$, taking into account the solar ambient temperature. Values of $\Delta\vartheta_{eqv}$ depends on hour of the day, wall's orientation and building's construction class. The values have been empirically determined and tabulated [9] for location $\sim 50^{\circ}\text{N}$, ambient temperature $t_{amb} = 24.5^{\circ}\text{C}$ and inner temperature $t_{in} = 22^{\circ}\text{C}$. It imposes a need of a correction, if the conditions differ and is given as $\Delta\vartheta'_{eqv}$:

$$\Delta\vartheta'_{eqv} = \Delta\vartheta_{eqv} + (t_{amb} - 24.5) + (22 - t_{in}) \quad (1)$$

Heat gains through outer walls can be calculated then from equation (2):

$$\dot{Q}_w = k \cdot A \cdot \Delta\vartheta'_{eqv} \quad (2)$$

where k is the heat transfer coefficient [W/(m²K)], while A is the surface area of the wall [m²].

Convective heat gains through windows are determined by the following equation:

$$\dot{Q}_T = k \cdot A \cdot (t_{amb} - t_{in}) \quad (3)$$

Solar radiation heat gains through windows have to be defined separately by equation:

$$\dot{Q}_S = A \cdot I \cdot b \cdot S_a \quad (4)$$

where I is total irradiation on a given surface [W/m²], b is a window transmission coefficient assumed as $b = 0.75$.

S_a is a solar heat-accumulation coefficient which was empirically defined for a specific type of construction, wall's orientation and specific hour of the day [9].

Internal heat gains are a sum of human heat gain, lighting heat gain and electric devices heat gain. Human heat gains are calculated accordingly to determination of occupants residing in one room. It is specified that sensible heat gain from one person equals 70 W. Heat gains from electric lighting may be defined with the help of equation:

$$\dot{Q}_B = N \cdot A_{room} \cdot \varphi \cdot S_i \quad (5)$$

where N stands for the power of installed lighting [W/m^2], A_{room} is the surface area of one room [m^2], φ is the lighting's coincidence factor assumed as 0.7, while S_i is a lamp heat-accumulation coefficient. According to literature [9] S_i may be introduced as 0.63. Depending on the amount of working electric devices in the room, heat gains are calculated from the equation:

$$\dot{Q}_B = \sum_i N_{mach_i} \cdot \varphi_i \quad (6)$$

where N_{mach_i} is heat gain of every device [W] and φ_i is a device's coincidence factor. Values of these factors are taken from Table 2. According to the standard it is simplified and assumed that the heat gains from domestic devices occurs all the time with a constant coincidence factor.

Table 2
Heat gains and coincidence factor of domestic electric devices

Electric device	N_{mach_i} [W]	φ_i [-]
Computer	90	0.5
Screen	50	0.5
Printer	10	0.2
Oven	1500	0.2
Fridge	300	1
Electric kettle	250	0.2
Washing machine	1500	0.2

In order to choose a representative day, cooling demand was calculated for 5 cases: May 18th at 15:00, June 7th at 12:00, July 5th at 12:00, August 8th at 15:00 and September 6th at 12:00.

These hours were not selected randomly, as each represents the warmest hour of every month according to meteorological data. Calculated cooling demand fits within the range 4–6 kW. It is visible on a chart presented in Fig. 3.

Although July 5th at 12:00 is the warmest hour in the analysed group, cooling demand for this point is slightly lower than for June 7th, August 5th and September 6th. It may be clarified with the chart presented in Fig. 4.

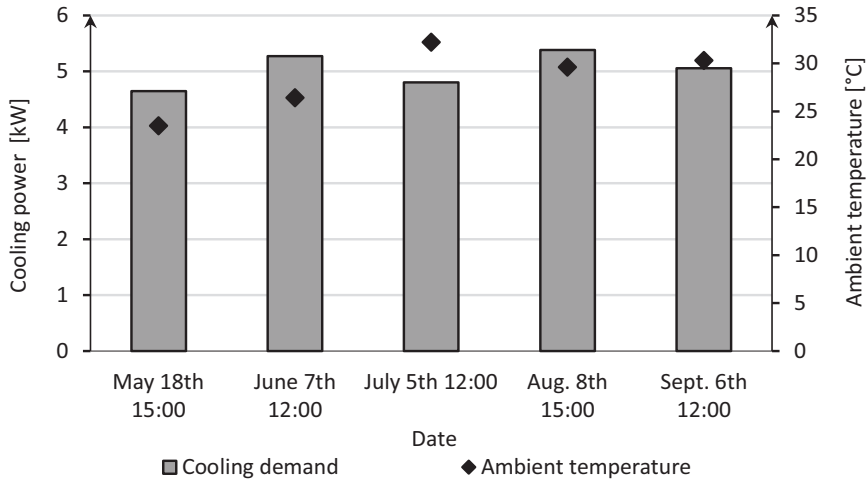


Fig. 3. Calculated cooling demand for each representative hour versus ambient temperature

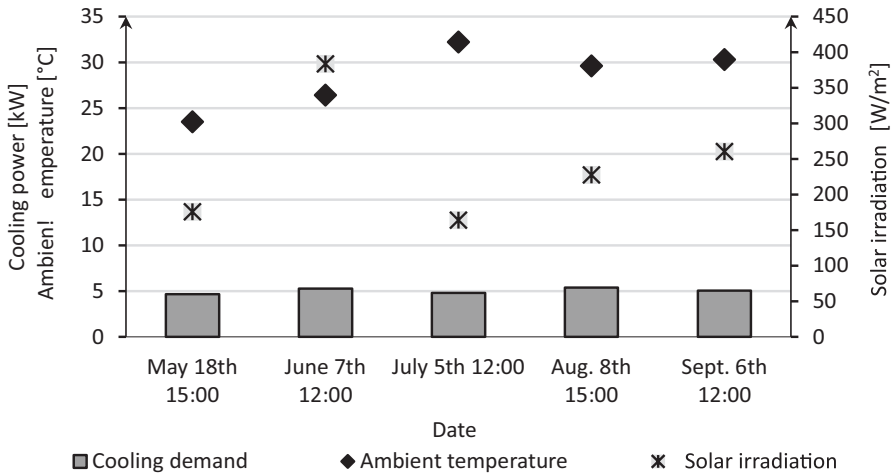


Fig. 4. Calculated cooling demand for each representative hour versus ambient temperature and solar irradiation on northern oriented vertical wall

It is evident that the solar irradiation on northern oriented walls, which represents a large part of the building’s gains is the lowest for the warmest hour, so the cooling demand may be smaller. However, value of northern solar irradiation on August 8th is lower than on June 7th and the cooling demands are higher. It arises from the difference of solar heat-accumulative coefficient for western walls at 15:00 and 12:00. On June 7th

at 12:00 $S_a = 0.17$, while already on August 8th at 15:00 $S_a = 0.53$. Even if the solar radiation on western walls is lower on August 8th at 15:00 (310.8 W/m^2) than on June 7th at 12:00 (436.48 W/m^2), the multiplication factor of accumulative coefficient is decisive. To omit any further ambiguity of this kind, it has been decided that the representative day will be July 5th as the hottest day of the year. Cooling power demand on July 5th at 12:00 equals 4.8 kW. Outdoor temperature is given as 32.2°C , while desired indoor temperature should be 26°C . It allows to calculate an indicator saying how much cooling power has to be delivered to chill one cubic meter of space by one Celsius degree. This indicator equals in the project case $2.11 \text{ W}/(\text{m}^3\text{K})$. Subsequently, cooling demand was determined for each hour between 7:00 and 19:00 on July 5th. This time range should correspond to the period when air-conditioning system is used. It was made, so that the amount of cooling energy needed for a whole day would be known. Following the VDI 2078 standard, the sum of cooling energy needed on July 5th equals 61.8 kWh. For an independent operation of passive cooling system, the value of daily cooling demand should be equal to the amount of cooling energy accumulated during the night activity. This study rests on evaluation of a night-radiator cooling possibilities to meet a daily cooling demand.

Application of radiative cooling system

To determine the possibilities of cooling power generation by heat transfer between passive radiator and radiating nocturnal sky, a heat balance has to be made. Meteorological data needed for the evaluation of the balance (ambient temperature – t_{amb} [$^\circ\text{C}$]; relative humidity – φ [%]; total solar irradiation – G [W/m^2], wind speed – w [m/s]; and wind direction) have been retrieved from meteorological data set for Nowy Sacz [10] for the night (21:00–6:00) preceding July 5th. Simulation was performed for various theoretical cloud cover conditions: for a clear sky and for 25%, 50%, 75%, and 100% of cloud cover. Passive radiator is an unglazed flat-plate collector having dimensions of the roof: $(10 \cdot 7) \text{ m}^2$. Collector is insulated from the roof and in a distance from other objects, no temperature drop according to the height of the roof has been considered. Collector's pipes are made of copper and covered with white paint of a low solar absorptivity factor. To calculate the convective heat transfer coefficient (α [$\text{W}/(\text{m}^2\text{K})$]), the collector is simulated as a flat surface. The night heat balance was conducted in a time step mode, assuming an hourly change of collector's surface temperature.

A heat flux from the surface of collector (\dot{q}_{loss} [W/m^2]) is described by equation:

$$\dot{q}_{loss} = \dot{q}_{conv} + \dot{q}_{sky_{rad}} - \dot{q}_{sun} \quad (7)$$

It takes into consideration convective loss (\dot{q}_{conv}), radiative loss from the surface (\dot{q}_{rad}), nocturnal sky radiative heat gains ($\dot{q}_{sky_{rad}}$), and solar radiation heat gains after sunrise (\dot{q}_{sun}).

Convective heat flux can be calculated from:

$$\dot{q}_{conv} = \alpha (t_{surf} - t_{amb}) \quad (8)$$

where t_{surf} [$^\circ\text{C}$] stands for surface temperature.

To simplify the calculations, during the simulation each hour represents a separate steady state. Surface temperature in the first hour of the calculation is equal to ambient temperature (21.4°C). An independent discussion could consider the evaluation of convective heat transfer coefficient. Various empirically obtained function for α are available in the literature, while the most common is the approach of Clark and Berdahl [5], where α depends only on the wind speed (w), and is equal to 3.5 W/m²K if $w < 1$ m/s, while if $w = 1-5$ m/s then $\alpha = 2.8 + 0.76 w$. Since in this study radiator is simplified to a flat surface of given dimension, convective heat transfer coefficient was calculated basing on determination of criterial numbers: Nusselt, Rayleigh or Reynolds for a parallel flow over flat plates, according to functions available in [11]. Coefficient α depends then on ambient temperature, surface temperature, wind speed and collector's dimensions.

Radiative heat flux from collector's surface is given by equation:

$$\dot{q}_{rad} = \varepsilon_{surf} \cdot \sigma \cdot T_{surf}^4 \quad (9)$$

where ε_{surf} is the surface emissivity and for a white paint equals 0.93 [12], while

$$\sigma = 5.67 \cdot 10^{-8} \frac{\text{W}}{\text{m}^2 \text{K}^4}$$

is the Stefan-Boltzmann constant.

The amount of sky radiation absorbed by the surface can be calculated using equation:

$$\dot{q}_{skyrad} = \alpha_{sky} \cdot \sigma \cdot T_{sky}^4 \quad (10)$$

Since the temperature of the source-sky is of the same order as surface temperature, it is assumed that sky radiation absorptivity equals surface emissivity. Sky temperature T_{sky} [K] can be found in the meteorological data set, but since the cloud cover conditions are not stated, it was calculated basing on an empirical function available in [4], presented in equation:

$$T_{sky} = \varepsilon_{sky}^{0.25} \cdot T_{amb} \quad (11)$$

It allowed to perform an individual investigation of cloud effect on passive system performance.

Sky emissivity (ε_{sky}) was a subject of many researches [13] and is an empirical function depending on air humidity. It was calculated from equation [4]:

$$\varepsilon_{sky} = 0.006 \cdot t_{dp} + 0.74 \quad (12)$$

Dew point temperature, t_{dp} , is a saturation temperature for vapour partial pressure.

Sky radiation heat gains may be increased by a cloudiness factor C , being an empirical function of cloudiness indicator, where $n = 0$ stands for a cloudless sky, and $n = 10$ for overcast sky [5]. Heat gains can be then obtained from following equation:

$$\dot{q}_{sky,rad} = C \cdot \varepsilon_{surf} \cdot \sigma \cdot \varepsilon_{sky} \cdot T_{amb}^4 \quad (13)$$

$$C = 1 + 0.0224 \cdot n - 0.0035 \cdot n^2 + 0.00028 \cdot n^3 \quad (14)$$

It is possible that around morning hours the surface of the collector will be heated by incoming solar radiation. Heat absorbed by the collector is evaluated with the use of equation (15). Solar absorptivity of the white paint equals $\alpha_{sun} = 0.2$:

$$\dot{q}_{sun} = \alpha_{sun} \cdot G \quad (15)$$

It is assumed that under hourly change of ambient conditions, the surface temperature of the collector is changing through heat transfer between collector's surface and nocturnal sky. More accurate calculations would need applying CFD calculations which are not a part of this paper. Therefore, some assumption had to be made connecting the surface temperature and storage tank installed. It is ideally assumed, that the heat lost from the surface during one hour at its initial temperature is equal to the change of internal energy (ΔE_u) of the storage tank, as shown in the equations below. It affects the change of the fluid at outlet of the tank. Since it is also assumed, that the heat transfer fluid flows very quickly in the radiator pipes, surface temperature after one hour of heat exchange equals the storage tank outlet temperature. It can be then an initial value for next hour of heat balance:

$$E_1 = \Delta E_u + E_2 \quad (16)$$

$$E_1 = Q_{loss} + E_2 \quad (17)$$

$$\rho \cdot V \cdot C_p \cdot t_1 = A_{surf} \cdot \dot{q}_{loss} \cdot \tau + \rho \cdot V \cdot C_p \cdot t_2 \quad (18)$$

where: ρ – density [kg/m^3]; V – tank volume [m^3]; C_p – specific heat of storage fluid [$\text{J}/(\text{kgK})$]; t_1 and t_2 fluid temperatures at the inlet and outlet of the tank, respectively.

If the project building was equipped with a solar absorption chiller, the installed storage tank capacity could be chosen according to the ratios available in [14], equal to 40 dm^3 per 1 m^2 of solar collector. It has been decided, that the radiative passive system will be connected with a 3 m^3 storage tank with cold water as a storage fluid.

Results of the radiative cooling simulation

The simulation proposed in this study enabled to define heat loss rate from the passive collector for every hour of the night. According to the foregoing calculation path and taking into account above-mentioned assumptions, a group of results has been obtained. Fig. 5 shows how the hourly heat loss profile could look like, if the sky was clear. The chart present a unit heat loss rate for 1 m^2 of radiator.

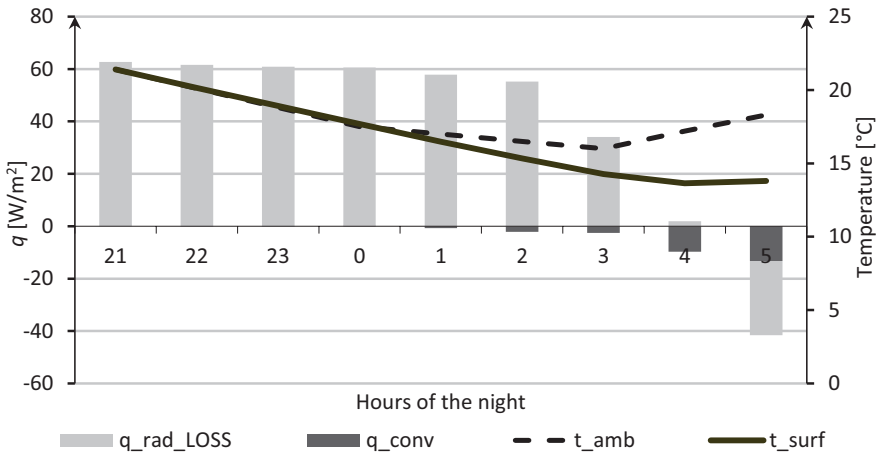


Fig. 5. A chart presenting the heat loss rate from the collector’s surface in comparison with its temperature and ambient temperature

The value of presented q_{rad_LOSS} equals radiative heat loss from the surface of collector after subtraction of nocturnal sky radiative gains and solar heat gains occurring at dawn. It is evident that by losing around 60 W/m^2 of heat, the surface temperature may fall well below the ambient temperature. If the night temperature drop was bigger, the heat loss rate could be much higher, what speaks against application of radiative cooling system in moderate climate. Together with the decrease below outdoor temperature, convective heat gains have to increase. The heat loss from the surface drops dynamically at 3:00. If not for the solar heat gains, the heat loss rate would decrease much slower. At 3:00 solar heat gain rate equals 19.1 W/m^2 , while at 4:00 and at 5:00 already 42.1 W/m^2 and 67.92 W/m^2 , respectively. Cooling power for every hour of the passive radiator operation can be obtained by multiplication of unit heat loss rate \dot{q}_{loss} and surface of collector ($A_{surf} = 70 \text{ m}^2$), results are presented in Table 3.

Table 3

Heat loss rates from the total surface of the passive collector

Hour	21:00	22:00	23:00	00:00	1:00	2:00	3:00	4:00	5:00
Heat loss rate [kW]	4.390	4.311	4.264	4.242	3.995	3.715	2.210	-0.540	-2.908

Calculated heat loss implies temperature drop inside the ideally assumed chilled water storage tank. If the cooling power generated by the passive radiator could be accumulated in a form of chilled water inside the ideally insulated storage tank, the sum of accumulated cooling energy, by losing heat from collector’s surface, would equal 23.68 kWh. To compare the values, the afore-mentioned cooling energy demand of the project building should equal 61.8 kWh. By the assumption of an average cooling power demand of 5 kW, the storage tank would be unloaded already after 4 hours and

44 minutes. Main conclusions coming from this comparison is that passive cooling is not sufficient for covering cooling demands in whole extent and system should be assisted for instance by compression chiller.

Second analysis concerns the impact of cloud cover on the performance of a radiative cooling system. According to equation (13) and (14), 4 stages of cloudiness have been analysed: 25%, 50%, 75% and 100%. A chart presented in Fig. 6. shows, how the heat loss rate changes with the increase of cloud cover indicator.

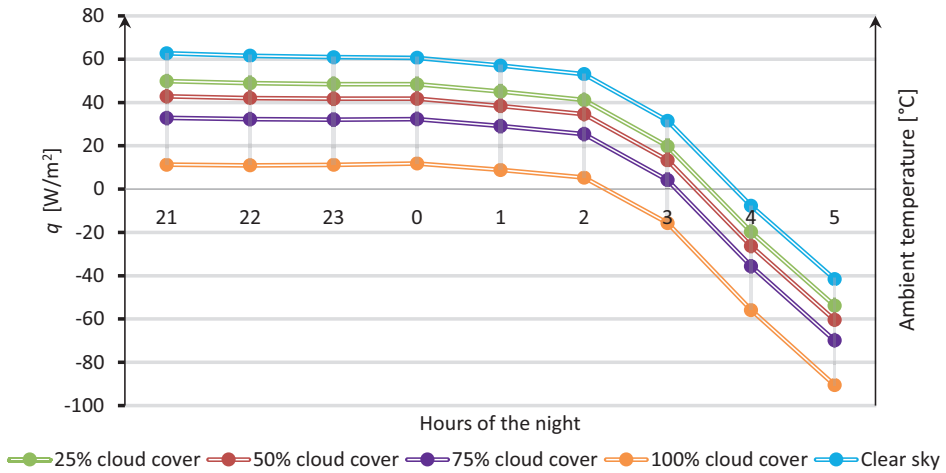


Fig. 6. Chart of a unit heat loss rate's profile during the night (July 4th/5th) under different cloud cover conditions

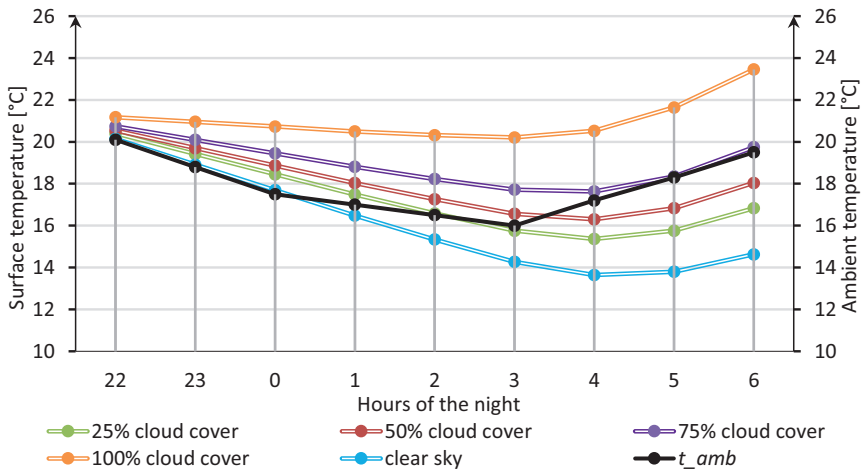


Fig. 7. Chart of the surface temperature change profile during the night (July 4th/5th) under different cloud cover conditions versus ambient temperature

It is visible that clouds are a factor which increases the sky radiation heat gains leading to the decrease of total unit heat loss rate from the surface. Additionally, theoretical change of surface temperature under the same cloud cover variation is shown in Fig. 7.

If the sky is overcast, or covered with clouds in 75%, the surface temperature would never fall below the ambient temperature. It is clear, that operating a radiative system under those conditions can be only efficient if ambient temperature at night is low (eg 10°C), while the following day is expected to be hot (eg 30°C). However, as mentioned before, this tendency is more typical for tropical and subtropical climates rather than for moderate climate locations.

It is valuable to compare the daily cooling energy demand and nightly passive cooling energy supply for the above discussed dates. Those values are presented in Fig. 8.

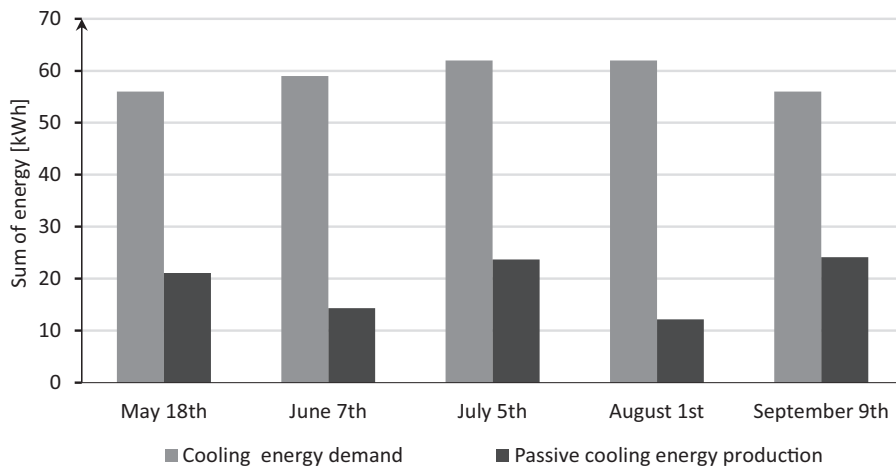


Fig. 8. Daily cooling energy demand and nightly cooling energy supply for given days

The chosen five days are the hottest of each month. Consequently, it influences the cooling energy demand and causes its value similar. However, solar heat gain is the highest for July and August, what results in the supreme cooling energy demand for these days. It could be suspected that the increment of average night temperature has a direct impact on the decrease of heat loss from surface. It is evident while comparing the cooling energy supply on colder nights of May and September with the cooling energy production on slightly warmer nights of June and August, as the nocturnal sky temperature is the lowest in this case in May and September. However, the example of night in July shows, that summer nights may have higher cooling energy supply under specific conditions. It is strictly bound with the surface initial temperature. For the night of July 4th/5th, according to the meteorological data, initial temperature was relatively high (21.4°C) resulting in high radiative heat loss from the surface, while nocturnal sky heat gain was comparable to other nights'. The gradual decrease of surface temperature shown in Fig. 5. enables to maintain a positive value of convective heat loss till

midnight. The initial surface temperatures at 21:00 on June 6th and August 1st are 15.5°C and 16.5°C, respectively. Already after one hour of radiator's operation, surface temperature falls below ambient temperature and the convective heat gains begin to increase, leading to stagnation. It proves the positive effects of desert climate. Surfaces warmed during the day accumulate part of heat and are able to lose more heat through radiation on a cold night. Therefore, the initial temperature of surface in simulation should take into account the fact of being heated during the day. Even more reasonable could be a 24-hours-simulation of change of surface temperature. Moreover, from a thermo-economic point of view, the investigation should be supplemented by the analysis of the cooling quality of temperature level at which heat is lost from the surface.

Conclusions

A theoretical analysis of possibilities, that installing a radiative cooling system into a residential building under moderate climate may bring, has been conducted. Cooling power demand of a project house on July 5th at 12:00 was defined as 4.8 kW, while total cooling energy foreseen for this day equaled 61.8 kWh. A passive flat plate collector installed on the 70 m² flat roof can generate a heat loss rate close to 60 W/m² during the night, if no clouds on the sky appear, and almost no wind occurs. It means a 4.2 kW heat loss from the whole surface of the radiator. Whole night of exploitation may bring a theoretical value of 23 kWh of cold accumulated in chilled water. If the storage tank had to be unloaded, it would be sufficient for 4.5 hours of utilization. Furthermore, if the sky was partially covered with clouds or overcast, the heat loss rate would gradually decrease, preventing the use of nocturnal radiator. It is visible, that even under idealised conditions, a radiative nocturnal cooling system is not sufficient to fulfil a daily cooling demand rate under moderate climate conditions for the project building. Nevertheless, it might be considered as an attractive, eco-friendly supplement to a conventional air-conditioner, what could require further investigation. It is worth mentioning, that in systems, where domestic hot water is generated by solar collectors, installation of a passive radiator would not be an obstacle, as it can be located on the northern roof slope. Application of nocturnal radiative cooling systems is believed to be more sensible in spaces, where the cooling power could be consumed simultaneously, like server rooms. Moreover, it could support food preservation processes, or the operation of cold stores leading to limitation of non-renewable energy consumption in the agricultural industry.

Acknowledgements

This research has been supported by the statutory research funds of the Faculty of Power and Environmental Engineering of the Silesian University of Technology.

References

- [1] Bagiorgas HS, Mihalakakou G. Experimental and theoretical investigation of a nocturnal radiator for space cooling. *Renew Energy*. 2008;33:1220-1227. DOI:10.1016/j.renene.2007.04.015.

- [2] Al-Nimr M, Tahat M, Al-Rashdan M. Night cold storage system enhanced by radiative cooling – a modified Australian cooling system. *Appl Therm Eng.* 1999;19:1013-1026. DOI:10.1016/S1359-4311(98)00103-3.
- [3] Santamouris M. *Advances in Passive Cooling.* New York: Earthscan publishing; 2007.
- [4] Erell E, Etzion Y. Radiative cooling of buildings with flat-plate solar collectors. *Build Environ.* 2000;35:297-305. DOI:10.1016/S0360-1323(99)00019-0.
- [5] Dimoudi A, Androutsopoulos A. The cooling performance of a radiator based roof component. *Sol Energy.* 2006;80:1039-1047. DOI:10.1016/j.solener.2005.06.017.
- [6] Farmahini-Farahani M, Heidarinejad G. Increasing effectiveness of evaporative cooling by pre-cooling using nocturnally stored water, *Appl Therm Eng.* 2012;38:117-123. DOI:10.1016/j.applthermaleng.2012.01.023.
- [7] Zhang S, Niu J. Cooling performance of nocturnal radiative cooling combined with microencapsulated phase change material (MPCM) slurry storage. *Energ Buildings.* 2012;54:122-130. DOI:10.1016/j.enbuild.2012.07.041.
- [8] Al-Obaidi K, Ismail M, Abdul Rahman A. M. Passive cooling techniques through reflective and radiative roofs in tropical houses in Southeast Asia: A literature review. *Frontiers Architectural Res.* 2014;3:283-297. DOI:10.1016/j.foar.2014.06.002.
- [9] Recknagel H. *Handbook for Heating and Air Conditioning Technology.* Wroclaw. Omni Scala publishing. 2008.
- [10] Meteorological data file for Nowy Sącz, available at: http://www.mir.gov.pl/budownictwo/rynek_budowlany_i_technika/Efektywnosc_energetyczna_budynkow/typowe_lata_meteorologiczne/ Strony/start.aspx.
- [11] Cengel YA. *Heat Transfer: A Practical Approach.* New York: McGraw-Hill publishing; 2002;373:468.
- [12] Solar absorptivity data tables. [Http://www.solarmirror.com/fom/fom-serve/cache/43.html](http://www.solarmirror.com/fom/fom-serve/cache/43.html).
- [13] Chen B, Kasher J, Maloney J, Girgis G. A, Clark D. Determination of the clear sky emissivity for use in cool storage roof and roof pond applications. *Proc ASES Annual Meeting.* 1991;1-5. http://www.ceen.unomaha.edu/solar/documents/SOL_26.pdf [accessed 05.03.2015].
- [14] Molero-Villar N, Cejudo-Lopez JM, Dominguez-Munoz F, Carrillo-Andres A. A comparison of solar absorption system configurations. *Sol Energy.* 2012;86:242-252. DOI:10.1016/j.solener.2011.09.

OCENA MOŻLIWOŚCI STOSOWANIA PASYWNEGO CHŁODZENIA DO CELÓW DOMOWYCH W KLIMACIE UMIARKOWANYM

Instytut Techniki Ciepłej, Wydział Inżynierii Środowiska i Energetyki Politechniki Śląskiej
Politechnika Śląska, Gliwice

Abstrakt: Wobec wymagań środowiskowych systemy wykorzystujące odnawialne źródła energii są coraz częściej stosowane i wdrażane również do rozwiązań budownictwa pasywnego. Celem pracy jest analiza efektów energetycznych wykorzystania pasywnego chłodzenia w budynku mieszkalnym. Podmiotem pracy jest system opierający swoje działanie na promienistej wymianie ciepła z nocnym nieboskłonem. Na powierzchni każdego ciała wyeksponowanej ku niebu, oprócz konwekcyjnej wymiany ciepła, odbywa się również radiacyjna wymiana ciepła z nieboskłonem. Analiza ukazuje potencjał chłodzenia pasywnego tego typu w produkcji chłodu, jednakże wskazuje na silną zależność systemu od warunków otoczenia oraz na fakt, że podaż chłodu jest odwrotnie proporcjonalna do zapotrzebowania. Niskie wartości strumieni strat ciepła z pasywnego radiatora sprawiają, że system nie może stanowić samodzielnego źródła produkcji chłodu, jednak może stać się atrakcyjnym, przyjaznym środowisku dodatkiem do konwencjonalnego układu klimatyzacyjnego.

Słowa kluczowe: chłodzenie pasywne, nocny nieboskłon, kolektor słoneczny, chłodzenie radiacyjne

