

RESEARCH ARTICLE



Heat transfer characteristics of nocturnal cooling system for clear sky climate of Western Maharashtra, India

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*Corresponding author.

Pramod V Mulik

Mechanical Engineering Dept, TKIET, Shivaji University, Warananagar, Kolhapur, India
pvmmech@tkietwarana.ac.in

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Pramod V Mulik^{1*}, Uday C Kapale², Gautam S Kamble¹

1 Mechanical Engineering Dept, TKIET, Shivaji University, Warananagar, Kolhapur, India

2 Department of Mechanical Engineering, Sanjay Ghodawat University, Atigre, Kolhapur, Maharashtra, India

Abstract

Objectives: To investigate heat transfer characteristics of nocturnal cooling system in the Indian climate with suitable material for radiator and proper coating to enhance the heat transfer. **Methodology:** A nighttime radiative cooling system with aluminum radiator in a dry area, is assessed both experimentally and theoretically. A theoretical model has been proposed to evaluate the radiative cooling potential for the dry type climate of Western Maharashtra, India. However, the end result can be improved by a nighttime cooling system based on nocturnal longwave radiation, which cools the air below ambient temperature. **Findings:** The experimental room temperature could be maintained at about 2- 4.5°C less than the outer room temperature. It is found that there is a huge scope in reducing the energy requirements for cooling purposes. Average radiative cooling rate of 55.84 - 71.88 W/m² is achieved for without coating and 72.30 - 80.99 W/m² for coating on the radiator. **Novelty:** In this study, we have designed and assessed the radiative cooling system which has a possibility of reducing the power using up for space cooling purposes. For achieving the building healthier and more comfortable, the building design is a key factor for saving energy and reducing emissions by considering nocturnal cooling design principles with suitable design and materials. The design of energy-efficient and sustainable buildings is important for the future.

Keywords: Radioactive cooling; nocturnal cooling; unglazed collector; flat-plate collector

1 Introduction

The phenomena of worldwide warming and the decrease of non-renewable energy sources have encouraged long-drawn-out consideration towards energy use. The home structure region represents a notable 40-50 % amount⁽¹⁾ of the EU's entire power utilization, which reflects on the significance of using an energy-aware conventional system in the house store. There are available records that reveal about the system studied for different climates of different countries but there is no record of a systematic study specifically for Indian climate. However, India has a large scope to use nocturnal

radiation system for space cooling in different cities. Very few researches are conducted to analyse the effects of the combined active and passive cooling methods on the enhancement of heat transfer rate, and hence it is an appropriate time to carry out work on this new approach. These facts motivate us to do research with greater emphasis on the different materials for radiators with a suitable coating to improve the heat transfer. It is required to heat the building during cold days and cool during hot days for comfortable living. As a large amount of primary energy is required for cooling, compared heating up, it is a matter of concern which depends upon the nation primary energy conversion factor. As far as energy efficiency is concerned natural cooling systems are useful but various circumstances make an effect on performance. For natural cooling, there is availability of three natural heat sinks. More heat is carried into these three heat sinks. Night-time convective cooling is the process of ventilating the building by using the atmosphere as a heat sink. When the temperature of night is cold then building structure gets cooled either by natural or conventional way causing the formation of colder thermal mass for the next day. This method has been investigated by various studies. Many factors make an influence on the effectiveness of this method. The building design is a key factor for saving energy and reducing emissions through passive solar heating and cooling design principles and using the correct materials and suitable design tools. Conventional cooling methods, that utilize an enormous amount of energy with a major carbon footprint in pollution, is replaced by daytime radiative cooling which is emerging as a passive and environmentally friendly cooling approach.⁽²⁾

The principal reasons for exploring options for proficient cooling systems are universal warming due to the emissions of carbon dioxide and hydro fluorocarbons used in refrigeration & air conditioning, exhaustion of non-renewable energy resources and demand for less polluting cooling systems. There is an urgent need of developing cooling systems which consumes less or non conventional energy source. However, we cannot fully replace conventional energy with non conventional or alternative form of energy but efforts should be made to reduce its consumption. In this regard, Radiative cooling can be the cost efficient technique to reduce the energy expence of cooling systems. It is a passive technique to exchange heat with large sink such as sky. These cooling techniques are mainly based on the principle of heat transfer by release of long-wave radiation from a more temperature body to a less body.⁽³⁾ This type of radiative heat transfer takes place considerably between earth and sky during night. This occurs due to the transmission of earths radiation to the sky during nights along with the emission of atmospheric radiation due to the water vapor, carbon dioxide and ozone layer there by reducing our reliance on fossil fuels required for conventional cooling methods.

A considerable research work in the area of nocturnal cooling system has been carried out of which a few are reviewed. The previous studies were concerned with non-conventional cooling, dissimilar radiator materials with experimental and numerical assessment. In⁽⁴⁾ studied the possibility of free cooling of house by nocturnal ventilation was examined by analyzing climatically information, irrespective of construction-particular parameters. Associate move toward for calculative degree-hours supported a variable house temperature inside an identical vary of thermal comfort was presented and functional to climatically information. In⁽⁵⁾ studied the current situation of continuous growth in energy price globally. Insulation of the exterior walls of the building is suggested as a strategy to increase the energy efficiency of constructions. In⁽⁶⁾ they declared that the suitability of free ventilation during night for cooling offices in modest climates like that of the United Kingdom was checked by describing plots of hot days climate information on the climatology chart for 3 locations inside the nation. In⁽⁷⁾ they reported that the house nocturnal ventilation may not provide effective cooling performance with intense solar radiation, conventional systems become essential in western France. A nighttime radiator is cool inside air below ambient temperature under nighttime radiation of the sky to increase the cooling performance of the system. In⁽⁸⁾ they explained a theoretical assessment of the probability of the cooling method to considerably decrease cooling requirements in houses in the North American atmosphere. The information represents the novel built-up cooling unit which uses nighttime radiation from a built-in heat exchange unit. They concluded that the performance of the system, system air temperature, radiator temperature and system airflow rate depend upon climate parameters such as outside temperature, cloudiness and wind speed. In⁽⁹⁾ conducted experiments on an aluminium nocturnal radiator. A radiator is colored with a suitable white shade. The metallic radiator was used for cooling the outdoor air below its primary temperature. The energetic thermal performance of the system throughout the days during hot months was calculated using numerical model, on the basis of heat transferred from the air moving inside the aluminum radiator to the ambient air. In study⁽¹⁰⁾ has examined that the sun radiates heat to earth in the day and at night the reverse happen warmer earth radiates heat to the cold night sky. Building roofs radiate heat at day and nighttime a rate of up to 75 W/m^2 . During the day, this is offset by solar radiation absorbed by the roofs of building, but at night, this heat loss has the ability to cool air as roofs can experience a temperature drop of 6°C to 20°C below ambient.⁽¹¹⁾ studied the mathematical modeling of a night sky radiation cooling system appropriate for a room located in the Namib Desert at Gobabeb, Namibia. The component of the system are radiator panels, a single water storage tank, room air-to-water natural convection heat exchangers or convectors, circulating pumps, interconnecting pipe work and temperature sensors and controls. In⁽¹²⁾ Longwave radiance of the sky during the hot season of 1979 at five cities in USA were measured. They monitored longwave radiations which are

more than 3 microns by a pyrgeometer. They found this radiation to be distributed in many spectral bands with five different zenith angles. They used a spectral radiometer for this purpose. In another study, they obtained various data related to the thermal radiance of clear skies in three U.S. cities and then they presented it in the terms of relation between sky emissivity and surface dew point temperature. In⁽¹³⁾ studied for clear and cloudy sky conditions the experimental search of the aluminum material and their surface color on nocturnal cooling was presented. MPCM slurry has been cited to be a good medium for the combined application of natural cooling technique in an HVAC system⁽¹⁴⁾. The outcomes showed the energy saving potential can reach 77% and 62% for low-rise buildings in Lanzhou and Urumqi, which show evidence of strong attractions for building energy conservation and emission decline.

The literature review shows that numerous investigations has been performed for heat transfer characteristics of a night cool nocturnal radiation system for space cooling at different climates and focused on the consumption of energy for building the environment and its effects on the surrounding. Also, there is no record of a systematic study of thermal analysis of the night cooling system in an Indian climate. The experimentation and mathematical models are presented in this research to expect the possible cooling for a clear sky climate of Western Maharashtra. The cooling requirements for the Western Indian climatic conditions are from month March to June. So, in this paper, we evaluate the radiative cooling potential in the clear sky condition in summer days. Also, the detailed study of the basic principle of nocturnal cooling system and the effect of wind speed, a velocity of mass flow rate on a rate of heat transfer and efficiency of the system is discussed.

2 Experimental setup of nocturnal cooling system

While conducting an experimental investigation on nocturnal cooling radiator, proper development of system and design of set up is of prime importance. The various research papers by different authors on night cooling systems are studied and with their reference CAD models are developed and the real experimental model is also built. Some considerations are taken into account while erecting the model. Necessary data for the construction is collected from journal papers and references. Detail modeling of all the parts is done by using PRO-E software.

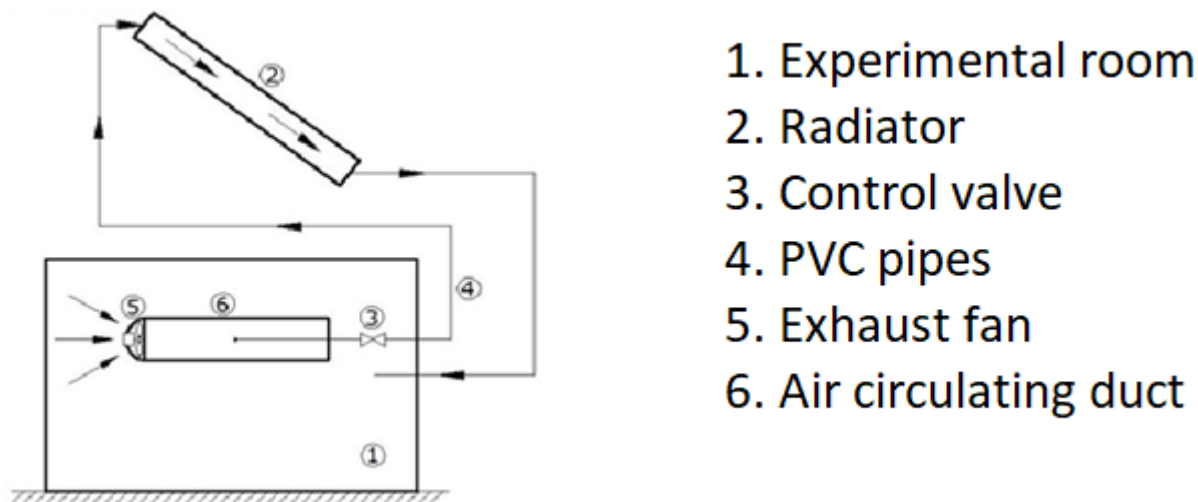


Fig 1. schematic diagram of nocturnal cooling system with flow circuit.

Figure 1 shows the actual experimental room with other components installed inside it.

Investigation of heat transfer characteristics of the nocturnal cooling system consisted of components are experimental Room model (3.5 m x 3.1 m x 3.4 m), two heat exchanger units (Radiator), air circulating duct, air circulation pipes from room to Radiator and exhaust fan.

As previously mentioned, it was decided that a radiator or heat exchanger unit dimensions should be 2 m x 2 m.⁽¹⁵⁾ But for easy handling and mounting purpose system two units of dimension 2 m x 1 m were fabricated as shown in Figure 2 (b). As the radiator (heat exchanger) has two aluminum sheet plates of 0.00044 m thickness separated by a gap of 0.1m. Nocturnal cooling is run by mostly the radiation of all surfaces towards the sky at night via thermal radiation change. The radiation between a radiating surface and the night sky can be described by approximating the surface as a black body. The sky longwave radiation

exchange is mainly a function of the sky effective temperature. In particular, radiative cooling is a result of heat loss by longwave radiation emission towards the sky, where the sky can be used as a heat sink for exterior surfaces of buildings.

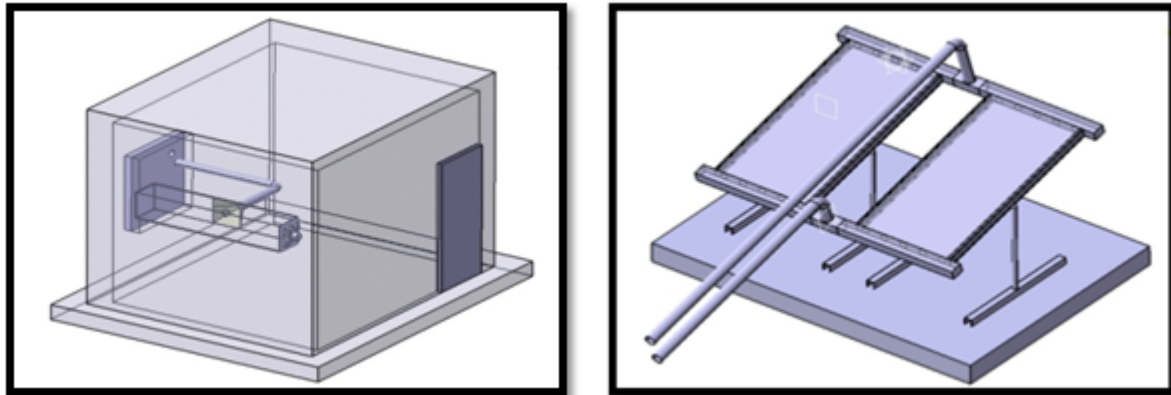


Fig 2. (a) & (b) PRO-E drawing of the actual experimental room and radiator

Radiative cooling is the largest at night darkness when the sky is clear. On a clear night, a building's external surface temperatures typically drop below the ambient temperature due to heat loss to the sky. The heat transfer takes place in the nocturnal cooling system due to all modes of heat transfer i.e. conduction, convection and radiation. Initially, air supplied by the exhaust fan enters in the radiator from the top inlet port and then it is distributed in the radiator uniformly. This hot air comes in contact with the surface of a radiator and gives its heat to the plate by the convection mode of heat transfer. Thus air is cooled on the loss of its heat energy. The temperature difference between the plate and air will create the driving force to take place heat transfer. To maintain the steady condition, the plate has to give transfer its heat to the surrounding air and sky. Therefore in plate heat is drawn out by conduction mode of heat transfer from all sides of a radiator. After this, heat is liberated to the surrounding flowing wind by the convection and the sky through electromagnetic waves form to the sky. Thus temperature difference takes the heat from circulating air to the atmosphere and cooled it which will be supplied back to the room. The entire process leads to maintain the required lower temperature in the room with the help of nocturnal cooling. Results show the correlation between the climate conditions, wind speed, velocity of the working fluid flow through the radiator and the getting temperature fall across the radiator and performance of the cooling rate.

3 Mathematical model of system

By using numerical equations the active thermal performance of the radiator has been determined. Entire heat extracted from radiator flowing air through the model is calculated by eq. (3.1),

$$Q_{total} = Q_{top} + Q_{bottom} \quad (3.1)$$

The change in internal energy of the circulating air in a radiator is useful the heat the airflow is calculated by eq. (3.2)

$$Q_{useful} = m \, c_p \, (T_{in} - T_{out}) \quad (3.2)$$

An Efficiency of the radiator system is the performance of the radiator unit is efficiency calculated by eq. (3.3).

$$\eta_{radiator} = \frac{Q_{useful}}{Q_{total}} \quad (3.3)$$

Heat loss from top & bottom surface to sky & surrounding is given by eq. (3.4) & (3.5),

$$Q_{top} = U_{top} \times A_{top} \times (T_{fin} - T_{atm}) \quad (3.4)$$

$$Q_{bottom} = U_{bottom} \times A_{bottom} \times (T_{fin} - T_{atm}) \quad (3.5)$$

Here, overall heat transfer coefficient from a top and bottom surface of a radiator are calculated by following equations (3.6) & (3.7),

$$U_{top} = \frac{1}{R_{totaltop}} \quad (3.6)$$

$$U_{bottom} = \frac{1}{R_{totalbottom}} \quad (3.7)$$

For the above equations (3.4) & (3.5) are required, to calculate total resistance offered for heat transfer from top surface & bottom surface are as shown in Figure 3, the total resistances offered from top and bottom surfaces are calculated by eq. (3.8) & (3.9)

$$R_{totaltop} = R_{top} + R_{cond. top} + R_{conv. air} \quad (3.8)$$

$$R_{total bottom} = R_{conv.wind bottom} + R_{cond. bottom} + R_{conv. air} \quad (3.9)$$

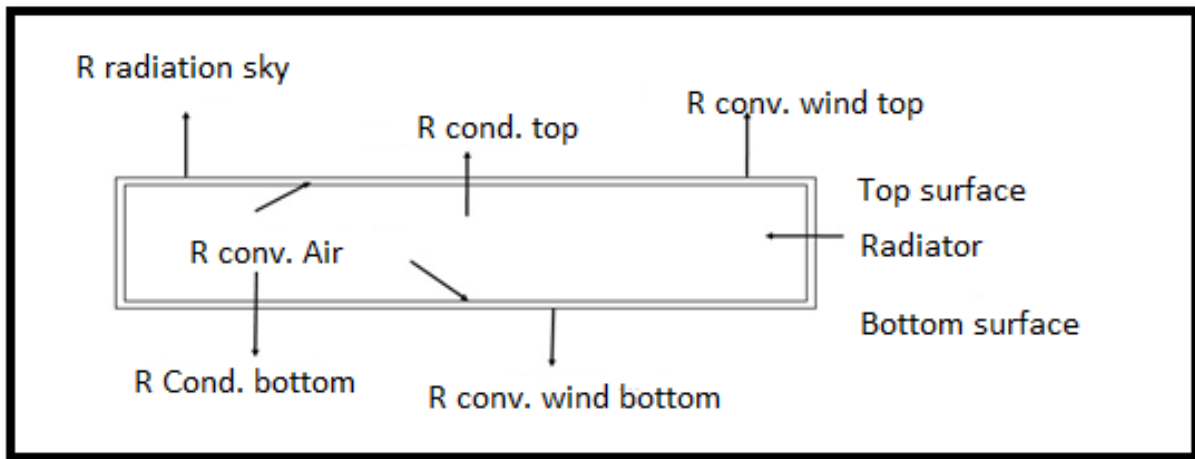


Fig 3. Different thermal resistances offered from radiator

3.1 Heat loss from the top surface of the radiator as follows

3.1.1. Convective heat loss from a surface to the surrounding air is calculated⁽⁸⁾ by eq. (3.10),

$$h_{conv.wind top} = 2.8 + 4.8 V_{wind} \quad (3.10)$$

Where, V_{wind} = Velocity of the surrounding air in (m/s)

3.1.2. Heat loss by radiation from the top surface to the sky is calculated⁽¹⁶⁾ by eq. (3.11),

$$h_{rad. sky} = \sigma * \epsilon * (T_r - T_{sky}) * (T_r + T_{sky}) * (T_r^2 + T_{sky}^2) / ((T_r - T_{sky})) \quad (3.11)$$

Where, σ = Stefan Boltzman constant = 5.670367×10^{-8} (W/m²·K⁴).

ϵ = Emissivity of the material of a radiator surface

T_r = Radiator surface average temp in (K)

T_{sky} = Sky temp in (K)

3.1.3 Sky temperature calculations,

- The effect of Cloudiness is denoted by 'CC'. The value of CC is varying from 0 to 1, according to the climate conditions. Take 0 for the clear climate and 1 for the fully cloudy climate.
- Air temperature T_{air} in (K).
- Dew point temperature i in Celsius is depending on the relative humidity in % and temperature of air i_{ni} . And it is calculated by eq. (3.12).

$$T_{dp} = T_{air} - \left(\frac{100 - RH}{5} \right) \quad (3.12)$$

Model for clear sky temperature:

At first, calculate the temperature of the clear sky^(17,18) using eq. (3.13)

$$T_{clear\ sky} = T_{air} (\epsilon_{clear})^{0.25} \quad (3.13)$$

Where, ϵ_{clear} =Emissivity of clear sky^(12,19)s calculated by the following eq. (3.14),

$$\epsilon_{clear} = 0.711 + 0.56 (T_{dp}/100) + 0.73 (T_{dp}/100)^2 \quad (3.14)$$

Now, the temperature of the sky is calculated by using the following eq. (3.15)

$$T_{sky} = (Ca)^{0.25} \times T_{clear\ sky} \quad (3.15)$$

Where, Ca= Constant factor is calculated by eq. (3.16)

$$Ca = 1 + 0.0224CC + (0.0035CC)^2 + (0.00028CC)^3 + \dots \quad (3.16)$$

Where, CC= Effect of cloudiness.

3.2. The R_{top} resistance is the combination of resistances offered from radiation sky and convective wind is given by eq. (3.17)

$$R_{top} = 1 / (h_{rad.sky} + h_{conv.top}) \quad (3.17)$$

Take values of $h_{rad. sky}$ & $h_{conv. top}$ are taken from eq. (3.10) & (3.11)

3.3. The $R_{cond. top}$ resistance due to the material of the radiator. Thermal resistance offered by a plate to conduction mode of heat transfer is calculated by eq. (3.18),

$$R_{cond. top} = \frac{tp}{Kp} \quad (3.18)$$

Where, tp = thickness of radiator plate in (m)

Kp = Thermal conductivity of radiator plate material in (W/mK).

3.4. The rate of heat transfer between a radiator plate and air being circulated is calculated using the below procedure.

Figure 1 (b) shows the radiator which is used for experimental work. The heat exchanger unit having dimensions are length L, width W and height H of radiator in meter. The inside air of a room is let in through the top end of a radiator and let out of the radiator again to the room. As this air passing near the radiator surface offered some resistance i.e. $R_{conv.air}$. The coefficient of convective heat transfer of circulating air is calculated using the below steps,

3.4.1 The airflow through the rectangular duct, hence the bulk mean temp is calculated by the following eq. (3.19),

$$T_{bm} = \frac{T_{in} + T_{out}}{2} \quad (3.19)$$

Where, T_{in} = Temperature of air at the inlet of a radiator system.

T_{out} = Temperature of air at the outlet of a radiator system.

3.4.2 The bulk mean temperature is required to select the properties of the air.

After calculating the bulk mean temperature, different properties of working fluid such as density, specific heat, thermal conductivity & dynamic viscosity are taken from a property table of air.

$$\rho = \text{Density of a working fluid, } \left(\frac{\text{kg}}{\text{m}^3}\right)$$

$$C_p = \text{Specific heat of working fluid, } \left(\frac{\text{J}}{\text{kg K}}\right)$$

$$K = \text{Thermal conductivity, } (\text{W} / \text{m K})$$

$$\mu = \text{Dynamic Viscosity, } \left(\frac{\text{Ns}}{\text{m}^2}\right)$$

3.4.3 The air passing through the rectangular duct, the characteristic length of duct⁽²⁰⁾ is required for calculate the Reynolds and Prandtl number calculated by the following eq. (3.20),

$$L = \frac{4 * W * H}{2 (W + H)} \quad (3.20)$$

Where, W = Width of radiator in m. H = Height of radiator in (m).

3.4.4 The air suck by the exhaust fan is given to the heat exchanger unit or radiator having with velocity. The output discharge of air through the pipe of the system is calculated by the following eq. (3.21),

$$Q = V_{air} \text{ at inlet } \times \text{ the cross – sectional area of the pipe} \quad (3.21)$$

Where, V_{air} = Velocity of air at the inlet of a system in (m /s).

Cross-sectional area of pipe = $\pi \times D^2 / 4$. D= Diameter of PVC pipe in (m).

3.4.5 Here, the velocity of air inside the radiator is required for calculating the Reynolds number which is essential for convective effect on the system is calculated by eq. (3.22),

$$V_r = \frac{Q}{W \times H} \quad (3.22)$$

Where, Q = Discharge of air through the system.

Cross-sectional area = W x H, W = Width of duct in m. H = height of duct in (m).

3.4.6 The Reynolds number is calculated by the following eq(3.23),

$$Re = (\rho \times V_r \times L) / \mu \quad (3.23)$$

3.4.7 Similarly, the Prandtl number⁽²¹⁾ is calculated by the following eq (3.24),

$$Pr = \mu \times Cp / K \quad (3.24)$$

3.4.8 The Nusselt number^(20,22) is calculated using the Dittus-Boelter's eq (3.25) as below,

$$Nu = 0.23Re^{0.8}Pr^{0.4} \text{ for } 3000 < Re < 10000 \quad (3.25)$$

3.4.9 Also, to calculate Nusselt number of convective air to determine the heat transfer coefficient by the following eq. (3.26),

$$h_{conv.air} = \frac{Nu K}{L} \quad (3.26)$$

3.4.10 Thermal resistance offered for convective heat transfer between circulating air & plate is given by eq. (3.27),

$$R_{conv. air} = \frac{1}{h_{conv.air}} \quad (3.27)$$

3.5 The following procedure is used to calculate all the resistances of the bottom surface which are mentioned in the above section.

3.5.1 The thermal resistance to the bottom surface is offered by wind to the convective mode of heat transfer. So the convective heat loss from a bottom surface to surrounding calculated by following eq. (3.28),

$$R_{conv.wind bottom} = \frac{1}{h_{conv.wind bottom}} \quad (3.28)$$

Where, $h_{conv.wind bottom}$ is calculated by eq. (3.29)

$$h_{conv.wind bottom} = 2.8 + 4.8 \times V_{wind bottom} \quad (3.29)$$

Here, $V_{wind bottom}$ = Velocity of wind at a bottom side of a radiator in (m/s).

3.5.2 The $R_{cond.bottom}$ resistance due to a material of the radiator. Thermal resistance offered by a plate to conduction mode of heat transfer is calculated by eq. (3.30),

$$R_{cond. bottom} = \frac{tp}{Kp} \quad (3.30)$$

3.5.3 Thermal resistance offered for convective heat transfer between circulating air & bottom side of the plate is given by eq. (3.31)

$$R_{conv. air} = \frac{1}{h_{conv. air}} \quad (3.31)$$

4 Results and Discussions

4.1 Night time evaluation of system with investigation and results

Depending upon the mathematical equations given in the above part, the nocturnal radiative cooling possible has been defined. The air temperature reduction depends on the intensity of radiation in the form of longwave, wind speed and the airflow through the aluminum radiator. It is observed that the reduction in temperature of the air increases step by step with the increase of the radiation's concentration i.e. clear sky condition. On the other hand, to changing airflow, a maximum value of air temperature falls can be found. For the high mass airflow rate, the cooling outcome can only just be realized on the air temperature drop. After installation of the nocturnal cooling system on the building roof, by using experimental set-up readings were taken in clear

sky hot day i. e. summer climate condition. The performance of nocturnal cooling should be checked in the clear sky climate condition of western Maharashtra in India. The performance of a nocturnal cooling system was determined by comparing the results obtained under the above basic conditions of climate.

The analysis of a nighttime cooling system has been done in month of May. A radiator with a total area of 4 (m²) of nocturnal cooling system was investigated. Domestic cooling load is dependent on individuals, electrical devices and independent from the climate. Comfort cooling depends on ambient temperature. The data contains parameters of each day such as ambient temperature, relative humidity, and wind speed, to estimate the radiative cooling possible. Using these factors sky temperature is calculated which will be used to calculate the cooling power. The performance of the system was tested by varying the mass flow rate of air which passes through the radiator. The mass flow rate was controlled by the control valve fixed between the duct and exchanger unit. The velocity of air at the inlet of the unit was kept as 3.6 (m/s) and 4.9 (m/s). The humidity and cloudiness are required for dew point and sky temperatures were taken from the metrological website of Government of India⁽²³⁾. The dew point and sky temperature are depending upon relative humidity and presence of the cloud in the sky.

Table 1. Measuring parameters 23 May 2017 like temperature, wind velocity, humidity etc.

Sr. No.	Time	Temp of radiator K	Temp. of air K	Velocity of wind m/s	Humidity RH	Cloudiness CC
1	5	305.50	306.60	4.917	52.00	0.00
2	6	303.80	305.40	4.917	57.00	0.00
3	7	302.60	304.00	4.917	59.00	0.00
4	8	301.70	303.30	4.917	65.00	0.00
5	9	301.00	302.60	4.917	72.00	0.00
6	10	300.10	301.70	4.470	72.00	0.00
7	11	299.00	300.60	4.470	74.00	0.05
8	12	297.90	299.50	3.576	77.00	0.11

After assessing different parameters as declared in Table 1, the study is to estimate the radiative cooling possible. The sky emissivity is calculated using eq. (3.14). It is found that the emissivity of the sky varies from 0.887 to 0.869 on the dated of 23 May. The sky temperature and ambient temperature difference is 9.26 to 10.89 (°C). Figure 4 shows the graph of time Vs variations in the temperature of a radiator, sky temperature and ambient temperature.

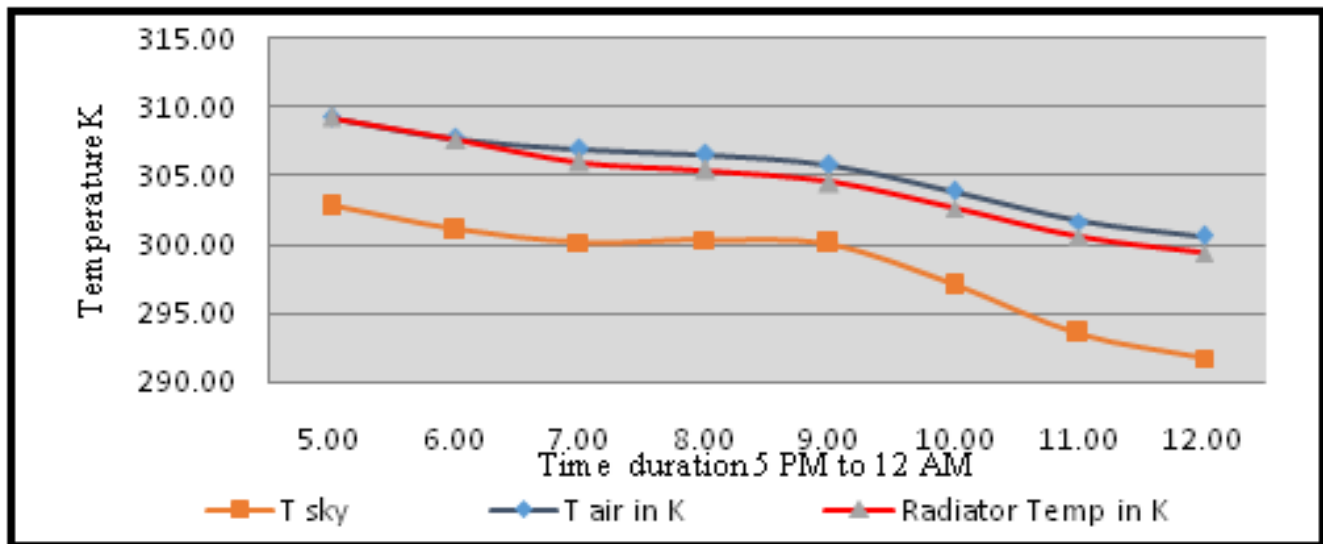


Fig 4. The time Vs air temp., radiator temp. & sky temp

The model's radiation heat loss coefficient is a depending emissivity and ambient temperature of the radiator; according to eq., (3.11), is an important factor for dissipating the excess heat from the top surface. In the cooling application, the radiator's emissivity is expected to be between 0.80 and 0.94. Figure 5 illustrates the slight effect of this parameter on the radiative

coefficient in different ambient temperatures, as determined throughout numerical calculations. It has been seen that the radiative heat transfer coefficient can be varied in the range 5.03 to 4.50 ($\text{W/m}^2 \text{ } ^\circ\text{K}$) for without coating of mass flow is 0.03 (kg/s). Also it can be varied in the range 5.20 to 4.74 ($\text{W/m}^2 \text{ } ^\circ\text{K}$) for without coating of mass flow is 0.05 (kg/s).

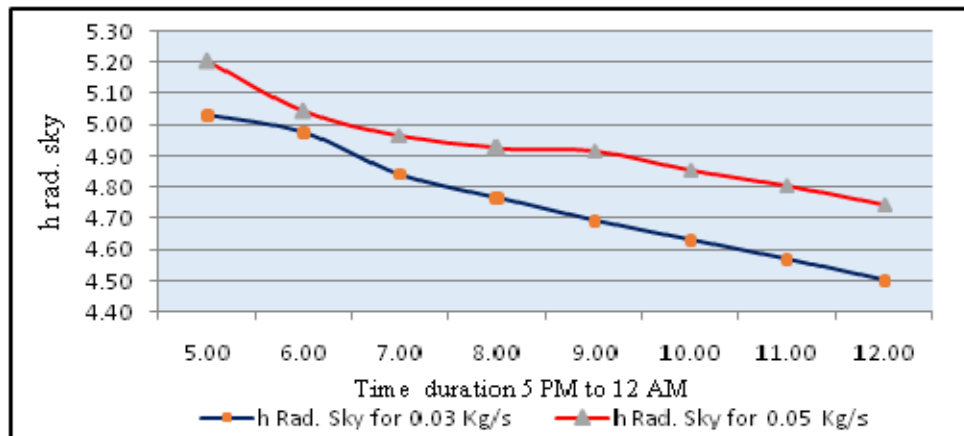


Fig 5. The time Vs $h_{rad.sky}$ (Without coating)

The above results indicated that if the mass flow rate is 0.05 (kg/s) then a radiator provides desirable value for the radiation sky heat transfer coefficient than that of the mass flow rate is 0.03 (kg/s) a radiator without coating. It is because, the higher mass flow rate increases the velocity of flow in the device and hence it increases the Reynolds number and consequently the rate heat transfer between the surface and air. Therefore, it can be concluded that the mass flow rate is 0.05 (kg/s) of air to a radiator without coating is more desirable to enhance heat transfer of the system.

The gap of ambient temperature, sky temperature and radiator temperature in the clear sky condition was bigger than that in the cloudy sky condition due to the effect of the protection of the infrared radiation. As per the working principle, as increases in mass flow rate reduces the temperature difference between the radiator inlet and outlet which results in increases in the mean surface temperature of the radiator. Therefore theoretically higher value of total heat loss should be recorded for clear sky condition with coating and higher mass flow rate through the system. It is found valid with observations made during the experimental work. In the present study, the values of net cooling ranged for without coating and coating on aluminium radiator is from 55.84 to 71.88 (W/m^2) and 72.30 to 80.99 (W/m^2) as the mass of airflow increases from 0.03 to 0.05 (kg/s) respectively for clear sky condition. It is shown in the Figure 6 .

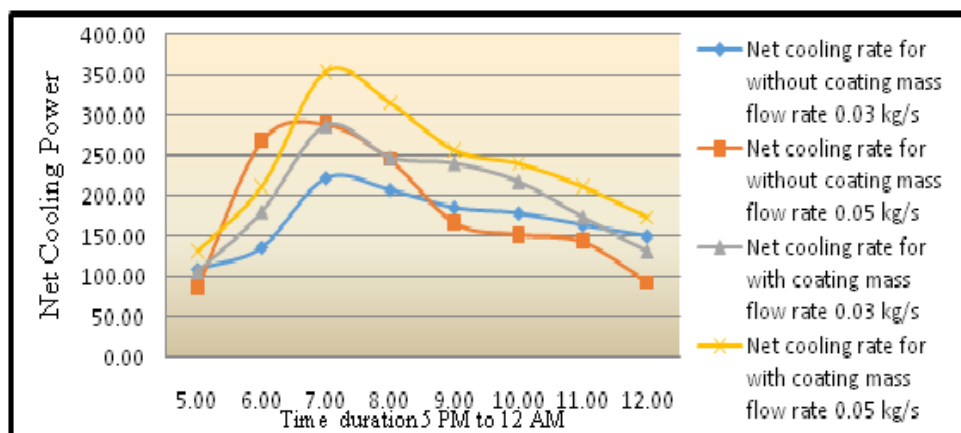


Fig 6. The time Vs net cooling power (With coating and without coating)

It follows a similar type of variation as that of all four different conditions i.e. initially it goes on increasing up to 7 pm and reaches its higher values because of higher temperature difference, wind speed, lower temperature of the sky and surface

temperature of the radiator. Then it starts decreasing, majorly due to the reduced gap between sky temperature and surface temperature. In the present study, similar types of results are found as cooling rates are between 70 – 80 (W/m^2) and 40 – 55 (W/m^2) for clear sky and cloudy sky condition respectively. The net cooling rate obtained in the experiments coincides with that reported by Dobson,⁽¹¹⁾ who recorded a 60.8 (W/m^2) of net cooling rate. Martin and Berdahl⁽¹⁹⁾ have recorded similar observation as this. In a clear sky night condition with surface temperature at 27°C achieved cooling rate is about 75 (W/m^2). In a cloudy sky humid climate with larger atmospheric moisture content, the rate of cooling falls to about 60 (W/m^2). Okoronkwo C. A et al⁽²⁴⁾ have obtained cooling power during the dry season was greater than that of the raining season. A maximum net power of about 52.5 (W/m^2) was obtained during the dry season while for the wet season the cooling power dropped to 37 (W/m^2).

The effect of increased wind convective heat exchange and the overall heat loss coefficient (U_{Top}) on the efficiency factor are expressed in eq. (3.10) and (3.6). The efficiency of the heat exchanger unit is determined by measuring parameters by using eq.(3.3). It observed that in Figure 7 the values of efficiency ranged for without coating aluminium radiator from 0.41 to 0.53 as the mass flow rate is 0.05 (kg/s) and 0.34 to 0.47 as the mass flow rate is 0.03 (kg/s). It was observed that increases in mass flow rate result in increases in the overall loss heat transfer coefficient, heat removal rate & efficiency.

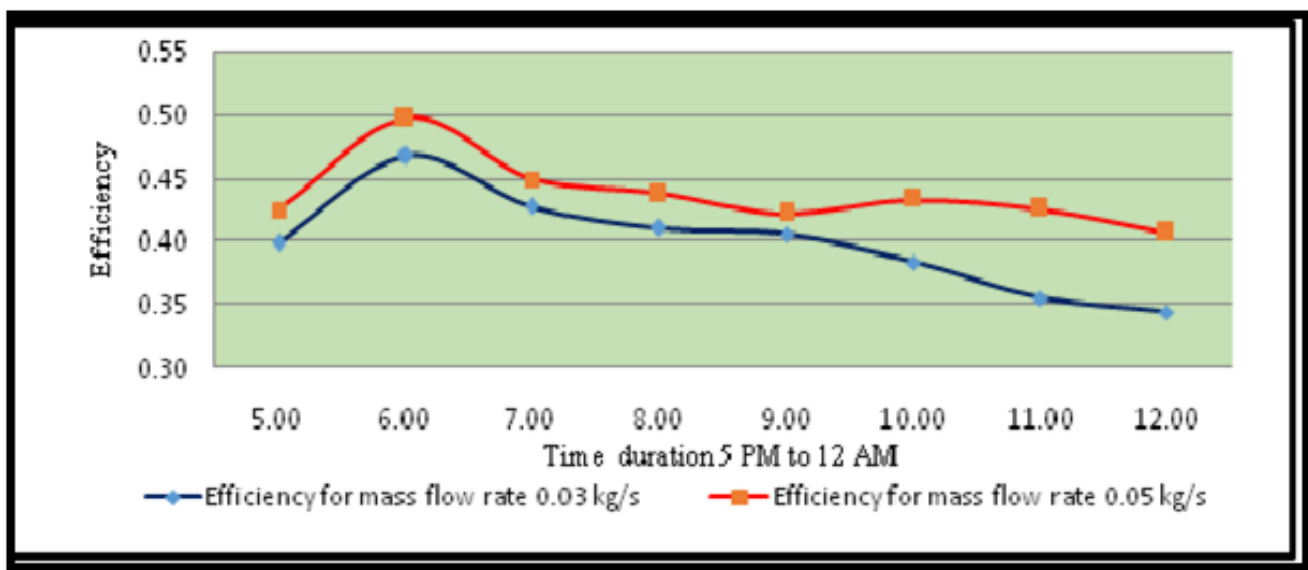


Fig 7. The time Vs efficiency(Without coating)

5 Conclusion

In conclusion, the accomplishment of the radiative cooling system for buildings in the tropical climate, especially in Maharashtra, is relatively encouraging. The following evidences were found during research work,

1. The active thermal performance of a light-weight metallic radiator covered by an aluminium sheet was calculated using the mathematical model. Finally, the thermal behavior at above mentioned location was prepared with a lightweight nocturnal cooling system and was analyzed using the empirical mathematical model. A review of nocturnal cooling system has been undertaken. From this study, the following are the observations in the clear sky conditions,

- The overall heat transfer coefficients are 12.73 – 10.74 ($\text{W/ m}^2\text{K}$) and 15.50 – 12.92 ($\text{W/ m}^2\text{K}$) during the night for two various mass flow rates of 0.03 and 0.05 (kg/s).
- The net cooling rate of night sky radiation system for with and without coating on aluminum of mass flow rate as 0.05 (kg/s) is near about 72.30 - 80.99 (W/m^2).
- The net cooling rate of night sky radiation system for with and without coating on aluminum of mass flow rate as 0.03 (kg/s) is near about 55.84 - 71.88 (W/m^2).
- The efficiency of the heat exchanger unit varies from 0.41 to 0.52 and 0.34 to 0.47 as the mass of airflow is 0.05 (kg/s) and 0.03 (kg/s) respectively.

2. Based on the experimental results obtained, air mass flow rate and sky climate condition are the principal factors which affected the performance of nocturnal cooling system. The Nusselt number, heat transfer coefficient and the recommended (turbulent) range of air mass flow rate which gives an appropriate outlet air temperature of the system.

3. The experimental room temperature could be maintained at about 2- 4.5(°C) less than other room temperature by using of nocturnal radiation cooling system.

4. The performance of the Aluminium material and surface coating with black coating a color has given better performance in clear sky summer climate. Also the performance of nocturnal cooling system was better in clear sky condition which was 30 – 40% more than that of results in cloudy condition.

5. There is a necessity to expand better consciousness of space cooling by nocturnal radiation since it has the potential of greatly dropping the consumption of electrical power and greenhouse gas emissions.

Nomenclatures.

Symbol	Name of symbol	Units
Q_{total}	Total heat loss from radiator	(W)
Q_{useful}	Change in internal energy of the circulating air in radiator	(W)
$\eta_{radiator}$	Efficiency of the radiator	
Q_{top} and Q_{bottom}	Heat loss from top and bottom surface	(W)
U_{top} and U_{bottom}	Heat transfer coefficient of top and bottom	(W/m ² K)
A_{top} and A_{bottom}	Area of top and bottom of Radiator	(m ²)
T_{fin}	Inlet temperature of air	(K)
T_{atm}	Ambient Temperature	(K)
$R_{totaltop}$ and $R_{total bottom}$	Total Resistances offered by radiator from top and bottom surface	(K/W)
$h_{conv.wind top}$ and $h_{conv.wind bottom}$	Convective heat transfer coefficient from top and bottom surface	(W/m ² k)
$V_{wind top}$ and $V_{wind bottom}$	Velocity of wind at top and bottom of surface	(m/s)
$h_{rad. sky}$	Radiative heat transfer coefficient	(*W/m ² k)
T_r	Radiator temperature	(K)
σ	Stefan Boltzman's constant	(W/m ² K ⁴)
T_{sky}	Temperature of sky	(K)
T_{dp}	Dew point temperature	(°C)
CC	Multiplying factor	
ϵ_{clear}	Emissivity of clear sky	
ϵ	Emissivity of radiator	
R_{top}	Thermal resistance offered for radiation mode of heat transfer and convective mode of heat transfer	
$R_{cond. top}$	Thermal resistance offered by material of plate	
$R_{conv. air}$	Thermal resistance offered from convective heat transfer of circulating air	
tp	Thickness of radiator	(M)
K_p	Thermal Conductivity of Radiator	(W/m ² k)
ρ	Density of Air	(Kg/m ³)
C_p	Specific Heat of Air	(J/kg K)
K	Thermal Conductivity of Air	(W/m K)
μ	Dynamic Viscosity of Air	(Ns/m ²)
T_{in}	Temperature of air at the inlet of system	(K)
T_{out}	Temperature of air at the outlet of system	(K)
T_{bm}	Bulk Mean Temperature of air	(K)
L	Characteristic length of Rectangular Duct	(m)
W	Width of Rectangular Duct	(m)
H	Height of Rectangular Duct	(m)
Q	Discharge of Air through pipe	(m ³ /s)
V_{air}	Velocity of air at the inlet of a system	(m/s)

Continued on next page

Table 2 continued

D	Diameter of Pipe	(m)
A	Cross sectional area of pipe	(m ²)
V_r	Velocity of air inside radiator	(m/s)
Nu	Nusselt number	
Re	Reynolds number	
Pr	Prandtl number	

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